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Semi-active Control of Civil Structures: Implementation Aspects

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Dottorato di Ricerca in Ingegneria Civile XV Ciclo

To my family

Description of the Ph.D. course

Settore:	Ingegneria
Sede Amministrativa non consortile:	Università degli Studi di Pavia
Durata del dottorato in anni:	3
Periodo formativo estero in mesi:	come previsto dal regolamento del Dottorato di Ricerca
Numero minimo di corsi	6

Il dottorato di ricerca in *Ingegneria Civile* è stato istituito presso la Facoltà di Ingegneria dell'Università degli Studi di Pavia nell'anno accademico 1994/95 (X Ciclo).
Il corso, sino alla sua ridefinizione in concomitanza con il XVI Ciclo, consente al dottorando di scegliere tra due curricula: idraulico o strutturale. Egli svolge la propria attività di ricerca rispettivamente presso il Dipartimento di Ingegneria Idraulica e Ambientale o quello di Meccanica Strutturale.
Durante i primi due anni sono previsti almeno sei corsi, seguiti dai rispettivi esami, che il dottorando è tenuto a sostenere. Il Collegio dei Docenti, composto da professori dei due Dipartimenti, organizza i corsi con lo scopo di fornire allo studente di dottorato opportunità di approfondimento in alcune delle discipline di base per entrambe le componenti, idraulica e strutturale. Corsi e seminari vengono tenuti da docenti di Università nazionali e estere.
Il Collegio dei Docenti, cui spetta la pianificazione della didattica, si è orientato ad attivare ad anni alterni corsi sui seguenti temi:

- Meccanica dei solidi e dei fluidi;
- Metodi numerici per la meccanica dei solidi e dei fluidi;
- Rischio strutturale e ambientale;
- Metodi sperimentali per la meccanica dei solidi e dei fluidi;
- Intelligenza artificiale.

A questi si aggiungono corsi specifici di indirizzo.
Al termine dei corsi del primo anno il Collegio dei Docenti assegna al dottorando un tema di ricerca da sviluppare, sotto forma di tesina, entro la fine del secondo anno; il tema, non necessariamente legato all'argomento della tesi finale, è di norma coerente con il curriculum, scelto dal dottorando (idraulico o strutturale).
All'inizio del secondo anno il dottorando discute con il Coordinatore l'argomento della tesi di dottorato, la cui assegnazione definitiva viene deliberata dal Collegio dei Docenti.
Alla fine di ogni anno i dottorandi devono presentare una relazione particolareggiata (scritta e orale) sull'attività svolta. Sulla base di tale relazione il Collegio dei Docenti, "previa valutazione della assiduità e dell'operosità dimostrata dall'iscritto", ne propone al Rettore l'esclusione dal corso o il passaggio all'anno successivo.
Il dottorando può svolgere attività di ricerca sia di tipo teorico che sperimentale, grazie ai laboratori di cui entrambi i Dipartimenti dispongono, nonché al Laboratorio Numerico di Ingegneria delle Infrastrutture.
Il "Laboratorio didattico sperimentale" del Dipartimento di Meccanica Strutturale dispone di:

1. una tavola vibrante che consente di effettuare prove dinamiche su prototipi strutturali;
2. opportuni sensori e un sistema di acquisizione dati per la misura della risposta strutturale;
3. strumentazione per la progettazione di sistemi di controllo attivo e loro verifica sperimentale;
4. strumentazione per la caratterizzazione dei materiali (macchina di prova universale biassiale), attraverso prove statiche e dinamiche.

Il laboratorio del Dipartimento di Ingegneria Idraulica e Ambientale dispone di:

1. un circuito in pressione che consente di effettuare simulazioni di moto vario;
2. un tunnel idrodinamico per lo studio di problemi di cavitazione;
3. canalette per lo studio delle correnti a pelo libero.

The Graduate School in *Civil Engineering* at Department of Structural Mechanics, University of Pavia, was established in 1994/95 (X Ciclo).
Two different type of curricula are available for students: hydraulic and structural engineering. The student works either at the Department of Structural Mechanics or at the Department of Hydraulics and Environmental Engineering.
Each student must select at least six courses with homeworks and final exams. The Teaching Council, compounded by professors coming from both departments, organise the courses with the aim to give the student the opportunity of improving his basic knowledge in the field of structural and hydraulic engineering. The courses and the seminars are held by national and international lecturers.
The following courses are organised every two years:

- Continuum Mechanics (Fluids and Solids);
- Computational Solids and Fluids Mechanics;
- Structural and Environmental Risk Analysis;
- Experimental Methods for Solids and Fluids Mechanics;
- Artificial Intelligence.

At the end of the first year, the Teaching Council assigns a homework covering one the above themes. This homework is not necessarily related to the final topic of the thesis.
During the second year the student will start his PhD thesis often discussing it with the Course Coordinator. The Teaching Council assigns the task.
At the end of each year the student has to present a detailed report regarding his research activity. The Teaching Council will decide, after a detailed revision of this report, to admit the student to the next year course.
A number of numerical and experimental laboratory can be used by all the students to improve their skill in civil engineering.
At the "Experimental and Teaching Laboratory" of the Department of Structural Mechanics the following equipment is available:

1. a shaking table for dynamic and control testing on small scale prototypes;
2. sensors and acquisition systems for measuring the structural response;
3. instrumentation for the design of active and semi-active control systems and their experimental validation;
4. instrumentation for material characterisation (bi–axial universal testing machine) with static and dynamic tests.

At the laboratory of the Department of Hydraulics and Environmental Engineering one has available:

1. a circuit in pressure able to model the unsteady motion of fluids;
2. a hydrodynamic tunnel to simulate cavitation problems;
3. a small scale model of channel.

Ph.D. theses

Dottorandi del X Ciclo:

Marco Battaini	Sistemi strutturali controllati: progettazione e affidabilità (Febbraio 1998).
Claudia Mariani	Problemi di ottimizzazione per strutture bidimensionali anisotrope (Febbraio 1998).
Antonella Negri	Stima delle perdite idrologiche nei bacini di drenaggio urbani (Febbraio 1999).

Dottorandi XI Ciclo:

Aurora Angela Pisano	Structural system identification: advanced approaches and applications (Febbraio 1999).
Carla Saltalippi	Preannuncio delle piene in tempo reale nei corsi d'acqua naturali (Febbraio 1999).
Eugenio Barbieri	Thermofluid dynamics and topology optimization of an active thermal insulation structure (Ottobre 1999).

Dottorandi XII Ciclo:

Massimiliano Barbolini	Dense snow avalanches: computational models, hazard mapping and related uncertainties (Ottobre 1999).

Paolo Espa	Moti atmosferici generati da forze di galleggiamento: simulazioni numeriche e studio su modello fisico (Ottobre 1999).
Lorenza Petrini	Shape memory alloys: modelling the martensitic phase behavior for structural engineering exploitation (Ottobre 1999).

Dottorandi XIV Ciclo:

Stefano Podestà	Risposta sismica di antichi edifici religiosi in muratura: sviluppo di nuovi modelli per l'analisi di vulnerabilità (Ottobre 2001).

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I was suggested to work similarly to the integral function: a little step per day, but every day. Effectively, this strategy has revealed as a very good one and, little by little, now I can say to have reached huge results.

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Connected with my PhD activities, during these years I had also the chance of being involved, with different grade of deepness, in many projects related to civil engineering open problems: this give me the opportunity to met new people and to work in a interdisciplinary environment. During these years I was involved into several European projects. I must mention at least those who were involved with me in the European project ACE[1] and CaSCo[2]: among the projects I took part, I followed these two ones very closely and I appreciated the fact of sharing knowledge with them and the ideas they transmitted to me. In particular, I would like to thank Claude Dumoulin, coordinator of ACE project, and Heino Fösterling, of Bosch-Rexroth.

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[1] *Active Control in Civil Engineering* - Project funded by the European Community, Brite-EuRam III Program project n^o BRPR-CT97-0402, contract n^o BE96-3334 (May 1, 1997 - April 30, 2000) with the following partners: Bouygues (co-ordinator), DERA, Newlands Technology, Johs.Holt, Joint Research Centre of EC, Mannessmann Rexroth, Techische Universität Dresden, Université Libre de Bruxelles, VSL.

[2] *Consistent Semiactive System Control* - Project funded by the European Community, Competitive and Sustainable Growth Program project n^o GRD1-1999-11095, contract n^o G1RD-CT 1999-00085 (February 1, 2000 - January 31, 2003) with the following partners: VCE Holding GmbH (co-ordinator), Mannessmann Rexroth, Institute of Structural Engineering, Joint Research Centre of EC, Université Libre de Bruxelles, Austrian Federal Railways, Micromega Dynamics.

Abstract

There is a continuous increasing interest from civil engineering community towards a new way of protecting structures against vibrations caused by wind, earthquake, traffic or human activities. Since the firsts attempts to face this new philosophy, the trend has lead to the formulation of the concept of "Structural Control". This wide term include in its meaning a lot of different kind of disciplines, techniques and methods connected them all by the common characteristic of aiming to mitigate vibrations into structures, both regarding displacements or accelerations reduction.

The subject has intrinsically an high degree of multidisciplinary, because it involves not only different type of engineers coming from geological, seismic, construction or mechanical side, but many others, such for example physics for the development of innovative materials, medics for the assessment of the maximum level of acceleration that can be supported by a person in a building without incoming in any trouble, electrics, electronics and informatics specialists, able to implement in a real-time system the control strategy and finally decision makers, whos, at the end, will chose one solution in spite of another one.

This strong need for a new way of protecting structures has lead to a increasing number of works in the field of structural control: PhD thesis, master thesis, research projects, networks, papers. This work inserts itself in this stream of studies. It makes a state of the art of what has already done by others and then propose some new solutions.

First of all, the definitions of the four types of control classes is given: passive, active, hybrid and semi-active control. This classification is fundamental and, as it is explained, it is not yet so well widely accepted. Then, the choice for the semi-active control strategies is explained: these kind of devices, that can be also called semi-passive, has an intermediate behaviour between the active and the passive ones and try to keep the best from each two ones and bypassing the limitations of both.

For this reason, and also because passive and active control has already been studied by others researchers, the semi-active control class has been chosen. However, because of its intermediate position between passive and active control, time

to time these other two classes are touched in a transversal way.

One example of an argument that is common to different classes of control is the comparison between collocated and non-collocated systems. After an original synthesis of the previous works on this theme, the advantage and disadvantage of both strategies are explained in details. An extensive section on numerical studies follows this more theoretical section and is divided in two parts: a probabilistic study trying to face out all the possible combinations of actuators and sensors in a system with increasing degree of non-collocation, and a case study on a 5 storey concrete building.

The development of a control strategy can be divided into several steps: first of all there is an idea on how (and why) to design a new control strategy, then a theory must be developed and a solid mathematical background must be established. At this point it is possible to design the control law and to perform numerical simulations. Once there is a well designed control law, an experimental campaign must be conducted. This is a fundamental step, because the established principle must be tested within a real application, where all the mechanical, electrical and structural limitations or problems can arise. The final step will be the application of the control strategy to a real structure.

The problem of semi-active control strategy implementation has been then considered deeply. An extensive state of the art in the field of semi-active control is given in appendixes (A). This choice is motivated by the fact that numerical studies and mathematical formulations are already being studied by several researchers, but practical implementation is still one of the most challenging aspects.

In the following chapters some numerical and experimental studies are presented jointly with characterisation tests on the semi-active devices that have been used. Some comparisons among different control strategies has been performed. A case study structure (a Single Degree of Freedom System) equipped with a magnetorheological damper has been considered and tested.

Some others experimental studies are then presented, some of them being conduceted in the frame of the European project CaSCo, which is presently still ongoing. A dynamic characterisation of the Baby-frame structure constructed at ELSA has been done as well, then it has been equipped with passive devices, too. Special attention has been given to the implementation aspects, so the controller program realisation, the acquisition system and the mechanical arrangements are explained in details.

The ending part of this work regards the conclusions that must be deduced by this work and some possible future directions for new researchers. In the opinion of the author, this last section is very important, because it permits to the future researchers to know what are the open problems still to solve and to face them in the best way.

Sommario esteso

Introduzione

Il controllo strutturale applicato al campo dell'ingegneria civile ha fatto registrare formidabili sviluppi negli ultimi decenni e si presenta oggi, agli occhi della ricerca, come una delle sfide più affascinanti e promettenti del prossimo futuro.

L'approccio tradizionale al problema di protezione sismica, che trova espressione nel "capacity design" (progetto per duttilità) prescritto dall'attuale normativa europea in materia (EuroCode8) consiste nell'assegnare alla struttura un'adeguata riserva di duttilità da spendersi, superato il campo di comportamento elastico, per realizzare quelle deformazioni anelastiche che da un lato prolungano la sopravvivenza della struttura evitandone il collasso di tipo fragile e dall'altro incrementano il coefficiente di smorzamento convenzionale della struttura dissipando energia. Lo svantaggio di un simile approccio sta nel fatto che in questo modo si accetta un danneggiamento permanente accumulato nella struttura, tale da richiedere una demolizione al termine dell'evento sismico o in ogni caso una pesante ristrutturazione. Appare anche evidente come tale concetto non possa in alcun modo essere esteso alla risoluzione di problemi d'attenuazione degli effetti fastidiosi per gli utenti di un edificio quando questo sia soggetto a vibrazioni indotte dal vento o da macchine operatrici.

Per contro, strategie di controllo strutturale, siano esse attive, semi-attive, passive o ibride, sono state concepite con il proposito di evitare o ridurre il ricorso a tali riserve di duttilità, preservando l'integrità dell'organismo strutturale. Tale suddivisione delle tecniche di controllo strutturale in quattro categorie si basa sulle seguenti definizioni:

Definizione 1 (Controllo Attivo) *Un sistema di controllo è attivo se uno o più attuatori applicano forze su una struttura secondo una definita legge di controllo e utilizzando per il loro funzionamento una sorgente d'energia esterna.*

Definizione 2 (Controllo Passivo) *Un sistema di controllo passivo è costituito da un dispositivo che modifica la rigidezza o lo smorzamento di una struttura senza* *richiedere una sorgente esterna d'energia e senza immettere energia nella struttura per il suo funzionamento.*

Definizione 3 (Controllo Ibrido) *Un sistema di controllo è ibrido se utilizza una combinazione di sistemi di controllo attivi e passivi.*

Definizione 4 (Controllo Semi-attivo) *Un sistema di controllo è semi-attivo se una sorgente esterna d'energia è utilizzata per modificare le proprietà meccaniche di un dispositivo.*

Mediante tali tecniche di controllo le caratteristiche dinamiche della struttura vengono quindi modificate in modo tale da ottenere una migliore risposta ad eccitazioni ambientali quali il sisma e il vento o ad eccitazioni di tipo antropico quali in traffico veicolare (specialmente per i ponti) o le macchine operatrici.

Storicamente i sistemi passivi sono stati i primi ad essere studiati e sperimentati, seguiti dai sistemi attivi e quindi da quelli ibridi. I sistemi passivi si sono invece affacciati sulla scena della ricerca più recentemente nella speranza di sviluppare dei dispositivi che unissero i vantaggi degli uni e degli altri. Per questo motivo essi sono attualmente un campo di ricerca molto aperto: tali dispositivi presentano tutti i vantaggi dei sistemi passivi unitamente all'adattabilità dei sistemi attivi.

Molti ricercatori hanno dedicato i loro sforzi ad uno studio delle leggi di controllo che possono essere utilizzate con tali dispositivi (Battaini et al. 1998; Nitsche 2001; Dyke and Spencer 1997; Soong and Spencer 2000; Housner et al. 1997; Hrovat et al. 1993; Marazzi and Magonette 2001; Marazzi et al. 2002), mentre i problemi di implementazione di tali leggi di controllo su dispositivi reali sono un tema di ricerca ancora molto aperto. Il presente lavoro si colloca quindi nella direzione di analizzare e risolvere alcuni dei problemi che si incontrano non appena si passi dalla fase di studio numerico-teorica alla realizzazione su dispositivi fisici di tali leggi.

Confronto tra sistemi collocati e non collocati

Prima di addentrarsi nelle tecniche di controllo semi-attive, è necessario rimanere ad un livello di analisi più elevato che prescinda dalla tecnica di controllo che poi si utilizzerà e operare un confronto preliminare tra sistemi collocati e non collocati, siano essi attivi, semi-attivi o ibridi. Un sistema è definito collocato quando l'algoritmo di controllo utilizza un segnale proveniente da un sensore in un punto della struttura per controllare un dispositivo di controllo agente nel medesimo punto. Tale definizione non ha niente a che fare con i sistemi di controllo centralizzati o decentralizzati, definiti invece rispettivamente come quei sistemi per i quali la legge di controllo è gestita da un unico elaboratore centrale o da diversi controllori distribuiti sulla struttura stessa.

I sistemi di controllo collocati hanno prestazioni superiori a quelli non collocati. La legge di controllo su cui si basano, infatti, tiene in considerazione la dinamica di tutta la struttura o di una sua parte, mentre i sistemi collocati, per definizione, basano la propria risposta solo sulle quantità misurate nella posizione specifica nella quale il dispositivo è inserito. Nonostante tale considerazione, per i sistemi lineari vengono spesso utilizzati sistemi collocati perché presentano proprietà molto vantaggiose derivanti dalla particolare forma della funzione di trasferimento associata. Infatti, come mostrato in figura (2.1), un sistema collocato presenta sempre l'alternanza di poli e zeri.

Viene quindi introdotto il problema dello spillover, fenomeno legato alle inevitabili schematizzazioni utilizzate nel formulare il modello matematico su sui si basa il progetto del controllore. Un breve paragrafo aiuta quindi a capire dal punto di vista fisico le origini dello spillover: sebbene tale problema possa poi, in fase d'implementazione, essere affrontato anche mediante l'opportuno uso di filtri, è fondamentale comprenderne bene le cause.

Nel caso ideale in cui non ci siano ritardi nella catena di misura, per un sistema collocato è più semplice evitare il problema dello spillover (figura (2.2)), mentre nei sistemi non collocati tali problema deve essere affrontato attentamente con opportune tecniche.

A questo punto il capitolo perde il carattere essenzialmente bibliografico per riportare alcuni studi originali sull'argomento. Dopo aver elencato dettagliatamente i pregi e i difetti del controllo collocato e non collocato, viene fornito uno studio relativo alle possibili combinazioni tra sensori e attuatori che si possono avere nel caso di controllo non collocato (il caso di controllo collocato ricade in questo studio come caso particolare in cui ogni attuatore funzioni in base al segnale proveniente dal sensore posto in sua corrispondenza). Già dalla tabella (2.1) si vede come, anche in presenza di un solo attuatore in posizione prefissata e per una struttura con pochi gradi di libertà, si abbiano comunque una molteplicità di possibili combinazioni per l'utilizzo dell'informazione proveniente dai sensori posti in corrispondenza di ciascun grado di libertà. Tali possibili combinazioni aumentano molto se l'attuatore può essere collocato in corrispondenza ad un qualunque grado di libertà della struttura (tabella (2.2)) per diventare una quantità molto elevata se sulla struttura possono essere posizionati un numero qualunque di attuatori ciascuno collegato ad un numero a piacere di sensori (tabella (2.3)).

Un esempio numerico elaborato "ad hoc" su un telaio in calcestruzzo armato di 5 piani conclude il capitolo. Le matrici di rigidezza e di massa della struttura sono ricavate a partire dai valori corrispondenti ai singoli piani. Si ricavano quindi i valori dei modi propri di vibrare e le corrispondenti forme modali (figura (2.8)). La matrice di smorzamento è ricavata mediante il metodo di Rayleigh assumendo uno smorzamento del primo e del terzo modo proprio di vibrare pari al 5% dello smorzamento critico. In questo modo è possibile ricavare anche i valori di smorzamento interpiano utilizzati in seguito per le simulazioni nel dominio del tempo.

Si considerano a questo punto tre casi di possibili schemi di implementazione relativi ai casi pratici che più plausibilmente possono verificarsi nella applicazioni:

1. presenza di un attuatore per piano collocato con il sensore posto al medesimo piano;

2. presenza di un attuatore per piano con legge di controllo basata sul sensore posto a tale piano più il sensore posto al piano superiore e inferiore;

3. presenza di un attuatore per piano utilizzante i sensori posti su tutti i piani.

Per effettuare un confronto nel dominio delle frequenze si sono riformulate le equazioni del moto nello spazio degli stati (state-space representation). I risultati (figura (2.9)) evidenziano come le prestazioni di un sistema di controllo aumentino sensibilmente passando dal caso collocato al caso non collocato con tre sensori. Al contrario, utilizzando uno schema che utilizzi tutti i sensori, non si hanno che lievi miglioramenti rispetto al caso in cui si utilizzino solo tre sensori. Tale comportamento rispecchia il fenomeno fisico tale per cui l'informazione proveniente dai sensori posti in posizione remota rispetto all'attuatore non migliora di molto la risposta del sistema dal momento che tali gradi di libertà sono anche poco controllabili da tale attuatore.

Per estendere il confronto tra sistemi collocati e non collocati anche al dominio del tempo (e poter in questo modo includere anche eventuali effetti non lineari, quali la presenza di un dispositivo di controllo semi-attivo), si è sviluppato un modello numerico (figura (2.10)) di un singolo piano di un edificio grazie al quale è possibile realizzare facilmente l'assemblaggio di un edificio multipiano eventualmente equipaggiato all'ultimo piano con uno smorzatore a massa accordata.

Le simulazioni hanno mostrato come la sensibilità di un attuatore al malfunzionamento di un sensore posto in posizione remota non porti a grossi problemi nel caso, ad esempio, di un controllore di tipo LQR.

Strategie di controllo semi-attivo e schemi di implementazione

Il concetto di controllo strutturale fu introdotto per la prima volta da (Yao 1972). Alcuni anni più tardi (Karnopp et al. 1974) propose l'idea di variare la forza in uno smorzatore a fluido controllando l'apertura di una valvola grazie alla quale le due camere del pistone sono messe in comunicazione: tale data viene assunta come la nascita del controllo semi-attivo. Karnopp aveva in mente come applicazione le sospensioni per autovetture, quindi un settore abbastanza lontano da quello delle strutture civili. La maggior parte delle strategie di controllo semi-attivo sono state, infatti, sviluppate nel campo dell'ingegneria meccanica e solo recentemente sono state prese in considerazione dalla comunità degli ingegneri civili.

Il presente lavoro si colloca su tale linea di "appropriazione" di tecniche sviluppate in diversi settori dell'ingegneria per applicarle al controllo di strutture civili. Dopo una carrellata di carattere essenzialmente bibliografico sulle varie tecniche di controllo semi-attivo maggiormente descritte in letteratura, si passa ad una formulazione originale di uno schema di implementazione per ciascuna tecnica.

La proposta di implementazione fornita ha carattere ancora generico, nel senso che non si va nel dettaglio dello specifico dispositivo che potrebbe realizzarla. Per ciascuna strategia di controllo viene formulato un diagramma di flusso in esplicazione dei passi principali che tale procedura dovrebbe eseguire, quindi si sviluppa tale diagramma di flusso mediante la scrittura di un programma in un linguaggio generico. Tale scrittura ha il vantaggio di contenere tutti i dettagli delle misure, dei calcoli e dei segnali di comando che la particolare procedura deve fornire senza però essere vincolata ad un particolare linguaggio di programmazione.

Nella parte finale del capitolo si formulano alcune considerazioni riguardo all'attuale stato di implementazione su strutture civili di tali tecniche. Da un esame del numero di edifici che, nel mondo, adottano dispositivi semi-attivi si evince subito come il loro numero sia estremamente limitato. Inoltre tali edifici hanno per lo più valore di "rappresentanza", nel senso che sono in qualche modo legati ad una volontà di dimostrare alla comunità degli ingegneri civili la bontà di tali tecniche. Si tratta infatti di progetti pilota e non di realizzazioni ordinarie.

Alcune ragioni di un uso ancora così limitato del controllo semi-attivo possono essere ricercate nei problemi tecnici ancora da risolvere, nella difficoltà di passare da strutture di laboratorio a strutture in scala reale, ma soprattutto nella mancanza di chiare indicazioni normative a riguardo. In particolare l'attuale normativa (sia in Europa che negli USA) risulta essere molto conservativa e di fatto penalizzante per le nuove tecnologie.

Implementazione delle strategie di controllo semi-attivo

La realizzazione su un controllore degli schemi d'implementazione descritti nella sezione precedente forma il corpo del presente capitolo. La realizzazione di una legge di controllo mediante un particolare controllore non può infatti prescindere dalla particolare configurazione hardware e software del controllore stesso: per questa ragione si dà una descrizione dettagliata di entrambi questi due aspetti.

L'hardware viene descritto dettagliatamente in quanto non si tratta di un controllore commerciale, ma appositamente sviluppato all'interno dell'ELSA (European Laboratory for Structural Assessment) per le esigenze del laboratorio, sia per quanto riguarda il metodo pseudodinamico che per le esigenze del controllo strutturale.

L'architettura (figura (4.1)) consiste in una scheda master (figura (4.2)) e una o più schede slave (figura (4.3)). Il collegamento tra le varie schede è realizzato mediante un bus passivo di tipo ISA (figura (4.4)). Le ragioni di tale architettura sono le seguenti:

- è la migliore soluzione in termini di modularità;
- è la migliore architettura per poter realizzare test pseudodinamici;
- è la configurazione ideale per poter implementare sia tecniche di controllo strutturale di tipo centralizzato che non centralizzato.

Il software ricalca l'architettura hardware e la sfrutta appieno. Esso è stato completamente realizzato su una piattaforma TNT Real-time Kernel, preferibile rispetto a WindowsNT[3] perché i semafori e le interruzioni di processo possono essere gestiti veramente in tempo reale con la priorità prescelta.

Lo schema del software è ben riassunto in figura (4.10). In tale figura si evidenzia come, sia sul master che sullo slave, siano presenti due processi, uno in background (bassa priorità) e uno in foreground (alta priorità). Al primo è delegata la gestione della tastiera, dell'aggiornamento dei parametri di controllo, all'aggiornamento della visualizzazione dello schermo, del disco rigido, della gestione della rete e della gestione della comunicazione con servizi remoti (per esempio sotto WindowsNT). Il foreground process invece opera il controllo vero e proprio: nel caso del master calcola ad esempio il passo successivo mediante il metodo pseudodinamico, mentre nel caso dello slave comanda l'attuatore oppure esegue l'algoritmo di controllo.

Gli scambi di dati tra master e slave avvengono grazie alla Dual-RAM, una speciale RAM a doppio accesso sviluppata anch'essa all'ELSA.

Appositi software sviluppati sotto WindowsNT quali l'acquisizione e il generatore di segnale completano i programmi di utilità a disposizione. Grazie alla DCOM Tecnology è inoltre possibile interfacciarsi al controllore anche con vari programmi commerciali quali CASTEM2000 o MATLAB.

Per poter implementare una legge di controllo è necessario avere a disposizione opportune funzioni in grado di filtrare i segnali provenienti dai vari sensori ed eliminarne le componenti indesiderate senza alterare le altre. Per questo motivo vengono descritti alcuni dei più comuni filtri utilizzati nel trattamento dei segnali digitali particolarmente utili in quanto relativamente semplici ed adatti ad

[3]Come descritto nel capitolo quarto, WindowsNT è un sistema intrinsecamente non deterministico, quindi non adatto per un suo utilizzo come sistema in tempo reale.

una reale implementazioni su controllori in tempo reale. Dopo una prima parte relativa alla formulazione matematica delle caratteristiche di tali filtri, si riporta il listato delle funzioni così come sono state sviluppate dall'autore per essere utilizzate all'interno del software di controllo.

Il capitolo si conclude con la descrizione della legge di controllo IFF (Integral Force Feedback) che è stata utilizzata nelle applicazioni che verranno descritte nel capitolo successivo.

Due casi di studio

È questo il capitolo che riassume tutta l'attività sperimentale a supporto delle analisi teoriche e numeriche riportate nella parte precedente della tesi e condotta durante la permanenza dell'autore presso il laboratorio ELSA. Centro di tutto il capitolo sono le due strutture a grande scala sulle quali si è operata la sperimentazione: il Baby-frame (figura (5.19)) e il ponte strallato (figura (5.25)), più una breve trattazione su uno studio preliminare realizzato su un modello ad un grado di libertà.

Per tale studio preliminare si è realizzato una struttura ad un solo grado di libertà (figura (5.15)), consistente in una massa montata su 4 isolatori in gomma e collegata mediante un dispositivo magnetoreologico al pavimento di reazione. Un eccitatore elettrodinamico (figura (5.16a)) imprime alla piastra un movimento verticale. Mediante un controllore è possibile agire sulla corrente di eccitazione del dispositivo magnetoreologico per poterne aumentare o diminuire il coefficiente di smorzamento. I risultati sono mostrati in figura (5.17) e (5.18). Si può notare come una corrente di 200 mA sia sufficiente per introdurre un grande smorzamento nella struttura.

Il Baby-frame consiste in una struttura mista acciaio-calcestruzzo di due campate e di tre piani d'altezza realizzata in scala 2/3 rispetto ad una struttura ordinaria. L'altezza di ciascun piano è quindi di 2 metri, l'altezza totale della struttura di 6 metri, la profondità di 2.4 metri e la larghezza di ciascuna campata di 4 metri. Tale edificio è stato caratterizzato in dinamica sia mediante un modello ad elementi finiti semplificato, sia mediante analisi modale sperimentale utilizzando come sorgente di eccitazione il martello strumentato. Tali dati sono stati poi utilizzati per tarare un nuovo modello numerico utilizzato per il progetto di uno smorzatore a massa accordata da applicarsi all'ultimo piano dell'edificio. Due tipi di smorzatori passivi sono quindi stati applicati all'interno di alcuni dei telai costituenti la struttura: dissipatori di tipo viscoso prodotti dalla Jarret (figura (5.22)) e dissipatori di tipo elasto-plastico sviluppati all'interno del progetto HYDE (figura (5.21)) per funzionare a taglio. Il Baby-frame è stato quindi sottoposto a diversi tipi di eccitazione per valutare le prestazioni delle diverse tecniche proposte: prove di rilascio elastico, test pseudo-dinamici per simulare sismi e prove di tipo ciclico. I risultati sono quindi stati confrontati e discussi.

Il secondo caso di studio riguarda una porzione di ponte strallato in fase di costruzione costituita da un'antenna vincolata al muro di reazione e da una semi-campata (cioè fino alla mezzeria) di 30 metri di lunghezza (5.24)). Tale struttura può quindi essere considerata in scala 1/10 rispetto ai più lunghi ponte strallati attualmente esistenti o, in alternativa, la realizzazione a scala reale di un ponte ordinario. Il ponte è sorretto da quattro coppie di stralli dei quali due, i più lunghi, sono collegati ad attuatori connessi all'impalcato. Nel caso del progetto ACE tali attuatori, di tipo idraulico, erano attivi mentre per il progetto CaSCo sono stati usati attuatori semi-attivi. Le due corrispondenti campagne di prove sperimentali hanno mostrato che l'utilizzo di dispositivi attivi riduce sensibilmente le vibrazioni indotte dall'eccitazione esterna sia per quanto riguarda l'impalcato che gli stralli. Lo smorzamento della struttura cresce dallo 0.4-0.5% al 10-12% (figure (5.28) e (5.29)). Il rovescio della medaglia è che l'energia necessaria (sotto forma di flusso idraulico d'olio) per il funzionamento di tali attuatori è ingente. Viceversa, nel caso degli attuatori semi-attivi, l'impiego d'energia è molto più modesto.

Preliminare a questi due paragrafi si colloca lo studio di caratterizzazione di alcuni dispositivi utilizzabili per implementare strategie di controllo semi-attive. Tale studio è stato possibile grazie al banco di caratterizzazione sviluppato all'interno del laboratorio ELSA e mostrato in figura (5.1). Grazie a tale apparecchiatura è possibile imporre agli estremi del dispositivo un qualunque spostamento alla velocità desiderata e registrare su un'apposita unità di acquisizione sia tali spostamenti che le rispettive forze. In questo modo si può poi tracciare diagrammi d'isteresi e calcolare le proprietà dissipative dei dispositivi.

Dapprima si sono caratterizzati i dispositivi magnetoreologici del tipo LORD RD-1005 mostrati in figura (A.3). Si sono eseguite prove a velocità crescente e con diversi livelli di corrente d'alimentazione. Come si può vedere bene in figura (5.2) dove vengono confrontati i risultati in assenza di corrente di alimentazione, i dispositivi si comportano in modo differente a bassa e alta velocità. A bassa velocità l'efficienza del dispositivo è più elevata che ad alta velocità. Nel caso invece in cui il liquido magnetoreologico sia polarizzato (corrente = 1 A), si ottiene la figura (5.3): l'efficienza rimane migliore a basse velocità, ma la transizione tra alta e bassa efficienza al variare della velocità è meno netta che non nel caso precedente.

Si passa quindi a descrivere e a caratterizzare il dissipatore variabile costituito da un pistoncino idraulico a valvola regolabile illustrato in figura (5.4). Il dispositivo è stato sottoposto a spostamenti ciclici di ampiezza ± 10 mm di frequenza 0.1, 0.5, 1 e 2 Hz. Si vede bene dalle figure (5.6(a)) e (5.6(b)) come un voltaggio di pilotaggio sulla servovalvola di valore compreso tra 1 e 10 V sia in pratica inefficace in quanto l'olio può scorrere senza attriti significativi attraverso il tubo di collegamento tra le due camere. Quando invece si forniscono valori di 0.1 o

0.2 V, il flusso attraverso la servovalvola diventa più difficoltoso e il dispositivo è quindi in grado di generare sensibili cicli dissipativi.

Per terzo si descrive in dettaglio un dispositivo idraulico semi-attivo sviluppato dalla Rexroth (gruppo Bosch) all'interno del progetto CaSCo. L'idea di base è mostrata in figura (5.7). Tale smorzatore deve essere utilizzato su un ponte strallato, quindi deve essere in grado di sopportare contemporaneamente carichi statici (dovuti al peso proprio del ponte, in questo caso dell'ordine dei 70 kN) e dinamici (dovuti al vento, al sisma o al traffico veicolare). Per questa ragione la maggior parte del cilindro viene usata per reggere il carico statico. Il dispositivo funziona utilizzando due pompe: una porta l'olio dal serbatoio alla camera dinamica fornendo una pressione differenziale di 100 bar. Tale pompa necessita di energia per funzionare. L'altra pompa è posizionata tra la camera dinamica e l'accumulatore di pressione e fornisce olio dalla pressione di 200 bar a 100 bar. Anche nel secondo caso ci sono quindi 100 bar di pressione differenziale, ma ora in posizione opposta che nel caso precedente. Questo vuol dire che il motore idraulico in questo caso genera corrente. Lo schema del dispositivo montato all'ancoraggio del ponte strallato è mostrato in figura (5.11).

Nella parte finale del capitolo vengono riportati alcuni degli eccellenti risultati ottenuti sul ponte strallato di figura (5.25) nell'ambito del progetto CaSCo. Le figure (5.32) e (5.34) mostrano, rispettivamente, il confronto tra gli spostamenti e le forze nel punto denominato P10 in figura (5.27) quando il ponte è soggetto ad una eccitazione con frequenze comprese tra 0.6 Hz e 1.3 Hz. Si vede bene come sia la riduzione degli spostamenti che la mitigazione della componente dinamica delle forze siano considerevoli. La tabella (5.3) riassume quindi i risultati di diverse prove condotte con guadagni d'anello differenti.

Conclusioni e futuri sviluppi

Nella parte finale della tesi vengono presentate alcune conclusioni, sottolineati gli aspetti originali del lavoro svolto ed enunciati i possibili sviluppi alla quale essa può portare mettendo in evidenza gli ottimi risultati ottenuti nella campagna di prove sperimentali.

Si evidenzia come lo studio sui sistemi collocati e non collocati risulti utile in vista dell'applicazione delle tecniche di controllo strutturale e come la descrizione dettagliata delle strategie semi-attive già studiate in letteratura permetta una loro implementazione più diretta. In particolare, la parte relativa alla programmazione all'interno di un controllore dei concetti elaborati nella parte precedente permette di operare il salto dalla teoria alla pratica applicativa.

Rimangono tuttavia, prima della piena applicazione delle tecniche descritte nel campo dell'ingegneria civile, alcune questioni da approfondire maggiormente, quali ad esempio l'individuazione di un caso reale che funga da progetto pilota per dimostrarne la validità anche ai non addetti ai lavori. Molto importante è anche l'impegno da dedicarsi alla disseminazione dei risultati ottenuti e allo sviluppo di dispositivi che integrino al loro interno anche la logica di controllo. Altro aspetto importante è infine il monitoraggio a lungo termine di strutture alle quali siano stati applicati i dispositivi studiati e soggetti ai vari carichi ambientali, quali vento, pioggia, irraggiamento solare e qualsiasi altro fattore a causa del quale il sistema di controllo potrebbe subire dei malfunzionamenti.

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ν	command value (in Volts) caming from a clipping control law
μ	friction coefficient
ξ_l	lower friction value
ξ_h	higher friction value
c_s	damper damping value
d_{rel}	relative displacement between two rigid body (equal to $z_b - z_t$)
d_{abs}	body mass absolute displacement
f	actual control force
f_c	required control force
$\mathbf{f}_{cf}$	vector of control forces
g	gain coefficient
g_n	equivalent to μg
k_s	spring stiffness
k_t	tire stiffness
m_b	body mass
m_t	tire mass
v_{rel}	relative velocity between two rigid bodies (equal to $\dot{z}_b - \dot{z}_t$)
$\mathbf{x}$	system state vector
z_b	body mass absolute displacement
z_t	tire mass absolute displacement
$\dot{z}_b$	body mass absolute velocity
$\dot{z}_t$	tire mass absolute velocity
$\boldsymbol{\mathcal{A}}$	status matrix
$\boldsymbol{\mathcal{B}}$	topological control matrix
$\boldsymbol{\mathcal{C}}$	output matrix
$\boldsymbol{\mathcal{D}}$	disturbance matrix
F_s	damper force

$\mathbf{P}$	matrix coming from solution of Lyapunov equation
$\mathbf{Q}$	positive definite matrix
V_{max}	maximum command value
$\Delta(.)$	variational operator
$\dot{\Delta}(.)$	derivative of the variational operator
$H[.]$	Heaviside step function
$P[.]$	operator used in modulated homogeneous friction strategy
$V(.)$	Lyapunov function

Chapter 1

Introduction

1.1 Preliminary considerations

Since ancient times the designers of civil engineering structures were responsible for buildings collapses: in the Code of Hammurabi[1], king of Mesopotamia during the XVIII century B.C., the builders were punished with the capital punishment in case the building felled down. Article 229 says[2]: "The builder has built a house for a man and his work is not strong and if the house he has built falls in and kills a householder, that builder shall be slain".

It was implicit in the law that the collapse should have happen for ordinary loads, let's say dead loads (caused for example by a large amount of people in the same room) or common dynamic loads (for example a strong by common wind). On the contrary, the collapse caused by extraordinary loads, such as, for example, earthquakes and hurricanes, were not taken into considerations, because they were considered out of control for the designer and the constructor and were retained as the manifestation of a "supernatural" event send by the gods for punishment or simply for taking joke of the human being.

Later on, even if the believe that these events were of supernatural origin progressively disappeared, the idea that the designer could not taken into account the effects of such loads still persisted. On one side earthquakes and hurricanes were considered completely unpredictable both it time occurrence and intensity, while on the other side there were no suitable techniques to reduce the risk of

[1] The Code of Hammurabi, from the name of the sovereign of the kingdom of Mesopotamia who wrote it, is a collection of laws performed on 51 columns on a stele discovered at the beginning of the XX century and now held at the Louvre Museums in Paris. It consists of a prologue and an epilogue celebrative of the king and of 282 articles regarding various aspects of the civil, penal and commercial law. It is one of the firsts examples of written laws.

[2] Translation was taken from (Foliente, 2000).

collapse.

For these reasons, design techniques developed mainly for taking into account the effects of gravity. The gravity loads are surely the primary loads that must be considered. These loads are always present and consequently must be resisted throughout the life of the building. Typically, the variation with time is slow compared to characteristic times of the structure. As a result, a static idealisation is quite appropriate. Furthermore, the magnitudes can be readily determined based upon self-weight and occupancy requirements. This combination of factors greatly simplifies building design under ordinary loads and was the main reason for the development of several static calculation techniques, such for example the graphical static technique.

During the years, better models or measures of the wind and earthquake intensity were developed as well the request for a better protection against these phenomena, which typically induce horizontal forces, instead of vertical ones. Because of the presence of well-established and well-known static methods for gravity loads, the first obvious choice was to try to extend those techniques to the horizontal loads. This was surely a great step forward, because it recognise that loads can be both vertical and horizontal. Even an approximate accounting for lateral effects will almost surely improve building survivability.

However, the equivalent horizontal loads caused by an earthquake or the wind was not the definitive method adopted by civil engineers to face out lateral displacements, because, in reality, the structure is more prone to move under some frequency content excitations than others. In different words, it has a dynamic behaviour. Moreover, an equivalent static analysis require that the structure remains in the elastic range. This last request is unfeasible mainly from an economical point of view, so the additional concept of dynamic analysis and capacity design were then introduced.

The capacity design consists in assigning a proper level of ductility reserve to the structural members. In this way the structure will remain elastic for the major part of its life (under ordinary loads), but it will go into the plastic state under exceptional lateral loads, in this way dissipating the input energy. In this case, even if there will be a permanent damage on the structural members, the building is designed in such a way that it will never collapse, so no loss of lives should happen. On the other side, however, the structure should be retrofitted after each strong event in which damage is generated into the structure, and this can be very expensive and time consuming.

This technique has also another drawback: it is not able to mitigate vibrations that didn't induce damage into the structure. This means that comfort aspect cannot be considered with this technique. So the problem of disturbing swaying of tall building caused by not very intense wind, for example, is not resolvable. It must be notice that, for very flexible structures such as long bridges or tall buildings, the comfort requirements can be more stringent than earthquake resistance ones.

For all these reasons the engineering community has evolved towards the concept of "Structural Control". This means that the structure is view as a dynamic system in which some properties, typically the stiffness or the damping, can be adjusted in such a way that the dynamic effect of the load on the building decreases under an acceptable level.

The natural frequency of the structure, its natural shapes and the corresponding damping values are changed in such a way that the dynamic forces coming from the natural loads are reduced. This can be done with a large variety of techniques that can be collects in four classes: passive, active, hybrid and semi-active. From an historical point of view, the first class of control techniques (passive) has extensively studied both from a theoretical and experimental side, and a lot of practical realisations have already implemented especially in USA, Japan, China, Italy and in many other countries. A great number of researchers is still working in this field for a second generation of passive devices, while the first generation (as for example rubber bearings) needs only that official designing and application rules will be given by the governments in order to permit their diffusion.

Active control techniques have been studied extensively from a theoretical, numerical and, more recently, also experimental point of view. They are surely the most effective one, but have also some disadvantages (as for example the need of a great power to operate) that have lead to a small number of applications until now.

Hybrid techniques are a combination of the first two techniques, so their development follows directly that of the first two. Their application is still limited, even if some tuned mass dampers equipped with a little active mass driver were patented and installed in some buildings in the Japanese area.

Semi-active techniques are, at present, the most studied ones, both from a theoretical, numerical and experimental point of view because of their excellent intermediate characteristics between those of active and passive techniques.

1.2 Definitions of passive, active, hybrid and semi-active systems

Before entering into the core of the discussion, a clarification must be made about the terminology used in this thesis. The following definitions of some key terms are provided:

Definition 1 (Active Control) *An active control system is one in which an external source powers one or many control actuators that apply forces to the structure in a prescribed manner.*

These forces can be used to either add or dissipate energy in the structure. In an active feedback control system, the signals sent to the control actuators are a function of the response of the system measured by physical sensors. Active control makes use of a wide variety of actuators, including active mass dampers, hybrid mass dampers and active tendons, which may employ hydraulic, pneumatic, electromagnetic or motor driven ball-screw actuation.

An essential feature of active control systems is that external power is used to effect the control action. This makes such systems vulnerable to power failure, which is always a possibility during, for instance, a strong earthquake.

Definition 2 (Passive Control) *A passive control system is made of an appended or embedded device that modify the stiffness or the damping of the structure in an appropriate manner without requiring an external power source to operate and feeding energy to the system.*

Passive control devices impart forces that are generated by the mutual displacement of the two connection points of the device inside the protected structure. Passive control may depend on the initial design of the structure, on the addition of viscoelastic material to the structure, on the use of impact dampers, or on the use of tuned mass dampers. Initial design may use a tapered distribution of mass and stiffness, or use techniques of base isolation, where the lowest floor is deliberately made very flexible, thereby reducing the transmission of forces into the upper stories. The energy of a passively controlled structural system cannot be increased by the passive controller devices.

Though seldom as effective as active control, passive control has three advantages:

1. it is usually relatively inexpensive;
2. it consumes no external energy;
3. it is inherently stable;

Definition 3 (Hybrid Control) *The common sense of the term "hybrid control" implies the combined use of active and passive control systems.*

A hybrid control system may use active control to supplement and improve the performance of a passive control scheme. Alternatively, passive control may be added to an active control scheme to decrease its energy requirements. For example, a structure equipped with distributed viscoelastic damping supplemented with an active mass damper on the top of the structure, or a base isolated structure with actuators actively controlled to enhance performance.

It should be noted that the only essential difference between an active and a hybrid control scheme is, in many cases, the amount of external energy used to implement control. Hybrid control schemes alleviate some of the limitations that exist for either a passive or an active control acting alone, thus leading to an improved solution.

A side benefit of hybrid control is that, in the case of a power failure, the passive component of the control still offers some degree of protection, unlike an active control system.

Definition 4 (Semi-active Control) *Semi-active controlled systems are a class of control systems for which energy is used to change the mechanical properties of the device.*

For this reason, usually semi-active controlled systems energy requirement are orders of magnitude smaller than typical active control systems. Typically, a battery power is sufficient to make them operative. Semi-active control devices do not add mechanical energy to the structural system, therefore bounded-input bounded-output stability is guaranteed, in the sense that no instability can occur[3]. Semi-active control devices are often viewed as controllable passive devices.

Preliminary studies indicate that appropriately implemented semi-active systems perform significantly better than passive devices and have the potential to achieve the performance of fully active systems, thus allowing for the possibility of effective response reduction during a wide array of dynamic loading conditions.

Examples of such devices includes variable-orifice fluid dampers, controllable friction devices, variable stiffness devices, semi-active impact dampers, adjustable tuned liquid dampers, and controllable fluid dampers (with electrorheological and magnetorheological fluids).

1.3 Motivations

Once upon ago, the only way to resist to lateral loads was to increase the "strength" of the structure: this was usually obtained by making more larger sections, so stiffer and more massive buildings. This method can be sometime a solution, and in the past has been used extensively also as a retrofitting technique after an earthquake.

However it is obvious that, with this method the inertial masses (and so the seismic forces) become greater and greater and moreover it is not able to solve some other problems has risen in these last years:

[3]However, the system overall energy can be increased by adding a passive system! This depends on the external excitation. The passively controlled system may have resonances that now coincide with the main excitation frequencies contrary to what happened before "appending" the passive device. In this case much more energy than in the unprotected case will go inside the system. This means that bad performances can be obtained if passive or even semi-active systems are improperly tuned. Passive or semi-active devices can be even dangerous because they may permit external excitation feed a system that would be (partially) isolated otherwise.

- **higher security level demand**: there is an increasing request for safer building, both for ordinary and for special ones, such as off-shore petroleum platforms, high buildings, nuclear power plants. For these types of structures, even a little damage can be synonymous of disaster;
- **higher performance request**: once the security has been guaranteed, the conventional structures are allowed to deform themselves over the elastic range, even if under a fixed maximum level. For special structure the maximum level can be smaller than the yielding point, such for example structures carrying special measurement devices like space structures, telescope, radar and so on. Also for more common building, however, the acceptable level of vibration and noise is being reduced;
- **better use of material**: this imply both a saving in money and in weight. Perhaps this second one is the more important, because it is crucial for satellite, high buildings and long bridges;
- **more slander structures**: there is the tendency of build structures that are more and more slander and so more prone to vibrate under harmonic loads and with less internal damping.

As already explained in the previous section, the capacity design has intrinsically no possibility to face off all these problems. On the other side structural control can offer a solution both from the point of view of the security and the comfort[4]. Since this field is becoming very huge, it is impossible to dealt with all the techniques has been developed in these years: this work has consequently been devoted to the most developing class of control devices, i.e. the semi-active ones.

The semi-active class of control devices is, in fact, the most promising one because it tries to joint the positive characteristics of passive devices:

- **low energy consumption**: the power needed for operating is very low, order of magnitude lower than with active system, because it is used only for modifying some properties of the device (the opening of a valve, for example);
- **no need for external power**: since the energy consumption is very low, there is no need for an external energy power supply (like a pumping system, for example), but a set of battery can be enough;
- **robustness**: even in the worst case (some failure in a sensor, for example) the device has intrinsically dissipative properties, it never insert energy into the controlled system;

[4]It must be notice, however, that structural control is not seen as an alternative to capacity design, but as an extension.

but it has also the advantages of active systems:

- **adaptability**: the possibility of adapting itself to different operating conditions, to different loading paths, to different levels of excitation;
- **optimality**: the control system is design in order to have optimal response characteristics on a wide spectrum of frequencies;
- **flexibility in tuning**: if the controlled structure changes it characteristics during the time, as for example on a tall building for ageing effects, the control device can be reprogrammed without any substitution;
- **monitoring**: the controlled structure will be instrumented with sensors. The data flow can be observed and stored for a continuous monitoring of the structure "health status";

In spite of these numerous advantages, there are still some problems to solve or some difficulties to be overcome before the semi-active control class of devices will be accepted worldwide as a feasible solution for vibration reduction. Some semi-active control lows are non-linear. This laws have to deal with saturation on the devices (caused by the fact that these devices can adapt themselves only in a predefined range) and with the fact that semi-active devices can only extract energy from the system and not to insert it. This means that some active control strategy as for example LQR or PID can be used only under some conditions and new techniques should be developed. This fact often leads to computational complexity. Moreover, the real implementations of these systems are still few in the world and experimental campaigns of large scale models are still too pour. Finally there is a lack of regulation regarding these new technology (but this is common to passive and active control, however) that penalises their use and favourites the adoption of traditional capacity design methods.

1.4 Objectives

The development of a control strategy can be divided in several step (figure 1.1): first of all there is an idea on how (and why) to design a new control strategy, then a theory must be developed and a solid mathematical background must be established. This can be done ex-novo or a useful theory can be borrowed from another scientific field. In this second case there are surely some adaptations to be considered, some conditions to be checked, some limitations to be posed, but usually the cross-fertilisation among different scientific sectors is very useful.

Before entering into the details of the control law design, the semi-active device must be constructed. This means that it must be designed, its behaviour must be simulated and finally an experimental campaign must be conducted in order to characterised it. This procedure is usually iterative: if some unexpected behaviours are shown, the device must be modify accordingly. Once the device has been manufactured, it is possible to design the control law. Numerical simulations must be conducted on the whole system (structure to be controlled and semi-active device) in order to have an idea of the final behaviour.

At this point, if the control law seams to be properly designed, an experimental campaign must be conducted. This is a fundamental step, because the established principle must be tested in a real application, with all the mechanical, electrical and structural limitations or problems than can arise in a practical case.

The final step will be the implementation of the control strategy in a real structure, with all the arrangements that will be necessary.

As already told in the previous paragraph, this thesis has been devoted to the semi-active class of control techniques and has been focused mainly on some implementation aspects. The all process of a new control system conception has been faced having in mind the typical problems usually arising in practical implantations. For this reason, as example, a deep comparison between collocated and non-collocated system has been conducted: it is important to know, in different operating condition and real environmental, which is the best choice from the practical point of view. Moreover, on a real control program to be implemented on a commercial semi-active device, there are a lot of securities checks, of saturation points, of transient conditions in switching the device on and off, that make a simple control program of a few lines a list of hundreds lines.

On the other side it may happen that some critical behaviour that can be seen with the theory and the simulations are never been reached in the specific real application: in this case a simplify design method can be derived in order to simplify the implementation process and the installation and calibration procedures. Complexity, when it is not needed, is dangerous because it can much more easily leads to errors and bad operating effects.

1.5 Adopted methods

The adopted working method has been to start from a literature study of what has already done in the field of semi-active control and to theoretically discuss and compare the different strategies. Then it was necessary to properly develop numerical tools to enable the simulation of dynamics structures to be controlled and to tests some of the above described control laws. Next step was to research a suitable implementation scheme for each control laws. Once the general implementation scheme of the chosen control law has been prepared, it has been programmed into a dedicated control hardware developed during these year at the ELSA laboratory. An experimental campaign has then followed on two different structures (a cable-stayed bridge and a three-storeys steel frame) in order to show, on one side, that

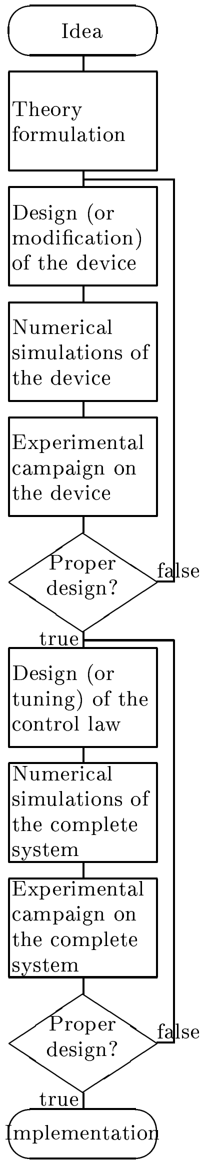


Figure 1.1: Flow chart of a control system conception

the adopted scheme is general and, on the other side, the effectiveness of the adopted choice in these two different situations.

1.6 Organisation of the thesis

The thesis can be divided into two parts. The fist part is more theoretical, with general considerations about different types of control schemes and strategies. The second part is more devoted to experimental evaluations of the previous explained concepts and to the related implementations aspects.

More in details, in the first chapter (this one) the necessary definitions of passive, active, semi-active and hybrid control are given. Proper motivations for this study are given and the common state-space representation of dynamic system is introduced to facilitate the comprehension of the following chapters.

In the second chapter a comparison between collocated and non-collocated systems is presented. A proper definition is given and the advantage and disadvantage of these two control schemes are presented. This chapter is transversal to the other ones because is not related to a particular control type (active, semi-active, passive or hybrid), but deals with the general philosophy that underlies the selection of the control type and the control law. Some numerical studies about a 5 storeys reinforced concrete building are presented and discussed both in the time and frequency domain.

The third chapter can be divided into two parts. The first one is related to the history of semi-active control and to a bibliographical review of the most common semi-active control strategies. The second part, developed by the author, deals with the implementations schemes to be adopted to follow the above described control law. The main steps are presented, even if in a schematic way, to allow coding them in C or Fortran. This chapter is concluded with some critical consideration concerning the few real full-scale implementations.

The forth chapter is devoted to the description of the implementation of the semi-active control strategies. The already mentioned flowcharts are developed in details: the main background concept about the adopted architecture, the hardware and the software needed, the connections to be prepared are described following the guidelines given by the work conducted in the lasts years at the ELSA laboratory. Once the general control frame has been described, the semi-active control software developed by the author is presented with details about the particular filtering techniques that has been adopted.

The next chapter (the fifth one) shows the results of two case studies conducted by the author at ELSA. A three-storey two-bays 2/3 scale mixed steel-concrete structure and a 1/10 cable-stayed bridge 30 meters long are being studied with different types of control laws (both passive and active) and different kinds of excitation to be compared with the semi-active strategies. The characterisation of the adopted devices is a key-issue for properly calibrating the control strategies. Numerical models are also necessary in order to perform numerical simulations.

Finally, in chapter six, conclusions and further foreseen developments are described.

Four appendixes conclude the thesis. The first one is devoted to a short review of the most common semi-active devices studied in literature. The second appendix presents some basic issues related to signal acquisition, analysis and treatment of the acquired signals. Sensors localisations and errors due to sensors and sampling frequencies are also considered. The third appendix contains some critical considerations about the concept and the evaluation of the original damping of a structure. This is a critical aspect when dealing with semi-active control systems studied to increase the structural damping. The last appendix is devoted to illustrate the comparison between the time and the frequency domain to properly analyse dynamical systems.

1.7 System representation

Let's us consider a linear time-invariant system given by the following equations:

$$\dot{\mathbf{z}}(t) = \boldsymbol{\mathcal{A}}\mathbf{z}(t) + \boldsymbol{\mathcal{B}}\mathbf{u}(t) \tag{1.7.1a}$$

$$\mathbf{y}(t) = \boldsymbol{\mathcal{C}}\mathbf{z}(t) + \boldsymbol{\mathcal{D}}\mathbf{u}(t) \tag{1.7.1b}$$

where $\mathbf{z}$ is the $2n$ dimensions vector of the state variables, $\mathbf{y}$ is the p dimensions vector of the measurable variables, $\mathbf{u}$ is the m dimensions vector of the controllable and forcing variables, $\boldsymbol{\mathcal{A}} \in \mathcal{R}^{2n\times 2n}$, $\boldsymbol{\mathcal{B}} \in \mathcal{R}^{2n\times m}$, $\boldsymbol{\mathcal{C}} \in \mathcal{R}^{p\times 2n}$ e $\boldsymbol{\mathcal{D}} \in \mathcal{R}^{p\times m}$.

This representation is called *State-Space Representation* and is commonly used in control engineering.

To understand what are matrixes $\boldsymbol{\mathcal{A}}$ and $\boldsymbol{\mathcal{B}}$ to be inserted into (1.7.1a), the equation of motion of a generic n degree of freedom system can be considered:

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{C}\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{F}(t) \tag{1.7.2}$$

where $\mathbf{M}$, $\mathbf{C}$ and $\mathbf{K}$ are square matrixes of dimension n, $\mathbf{x}$ are the inter-masses displacements (in the building case the inter-storeys displacements) and $\mathbf{F}(t)$ are the external forces[5].

The state space representation can be obtained from equation (1.7.2) by calling the displacement vector with $\mathbf{x}_1$ and the velocity vector with $\mathbf{x}_2$ (so $\dot{\mathbf{x}}_1$ is the

[5]These forces can be generated by an external excitation (as for example by an earthquake or a storm) or by a control action given by an actuator.

velocity vector and $\dot{\mathbf{x}}_2$ the acceleration vector). The system represented by (1.7.2) can be rewritten in the following form[6]:

$$\begin{cases} \dot{\mathbf{x}}_1 = \mathbf{x}_2 \\ \dot{\mathbf{x}}_2 = -\mathbf{M}^{-1}\mathbf{K}\mathbf{x}_1 - \mathbf{M}^{-1}\mathbf{C}\mathbf{x}_2 + \mathbf{M}^{-1}\mathbf{u} \end{cases} \tag{1.7.3}$$

or, with another notation:

$$\begin{bmatrix} \dot{\mathbf{x}}_1 \\ \dot{\mathbf{x}}_2 \end{bmatrix} = \begin{bmatrix} \mathbf{0} & \mathbf{I} \\ -\mathbf{M}^{-1}\mathbf{K} & -\mathbf{M}^{-1}\mathbf{C} \end{bmatrix} \begin{bmatrix} \mathbf{x}_1 \\ \mathbf{x}_2 \end{bmatrix} + \begin{bmatrix} \mathbf{0} \\ \mathbf{M}^{-1} \end{bmatrix} \mathbf{u} \tag{1.7.4}$$

Considering $\mathbf{x}$ and $\dot{\mathbf{x}}$ as the two interesting states, system (1.7.4) is the state-space representation of system (1.7.2) with $\boldsymbol{\mathcal{A}}$ and $\boldsymbol{\mathcal{B}}$ given by:

$$\boldsymbol{\mathcal{A}} = \begin{bmatrix} \mathbf{0} & \mathbf{I} \\ -\mathbf{M}^{-1}\mathbf{K} & -\mathbf{M}^{-1}\mathbf{C} \end{bmatrix} \tag{1.7.5}$$

$$\boldsymbol{\mathcal{B}} = \begin{bmatrix} \mathbf{0} \\ \mathbf{M}^{-1} \end{bmatrix} \tag{1.7.6}$$

The output (1.7.1b), or measurable variables, is a weighted sum of states and the control variables. Matrix $\boldsymbol{\mathcal{C}}$ is a weighting matrix that combines the states of the system (1.7.2), matrix $\boldsymbol{\mathcal{D}}$ is a weighting matrix that combines the control input and the sum of these two quantities gives the output $\mathbf{y}(t)$. The direct action of control variable $\mathbf{u}$ on the output vector $\mathbf{y}(t)$ is often negligible: matrix $\boldsymbol{\mathcal{D}}$ is often considered as zero in practical applications.

The transfer function $\mathbf{G}(\mathrm{s})$ related to system (1.7.2) can be very easily obtained with the following expression:

$$\mathbf{G}(s) = \boldsymbol{\mathcal{C}}(s\mathbf{I} - \boldsymbol{\mathcal{A}})^{-1}\boldsymbol{\mathcal{B}} + \boldsymbol{\mathcal{D}} \tag{1.7.7}$$

It must be recalled that this representation is not possible for non-linear system: speaking of a transfer function it has no meaning, in that case.

1.7.1 An example: a shear-type multi-storeys building

A shear-type n-storeys building is a good example of the use of the state-space representation in civil engineering. This type of buildings can be schematised as lumped masses at floors connected by springs and dashpots describing the columns behaviour under horizontal loadings (figure (1.2)). These assumptions are true if the mass of the columns is one order of magnitude less than the storey mass and if the floors are very rigid (and so they are not subject to significant deformations).

[6]The vector $\mathbf{F}(t)$ has been substituted with $\mathbf{u}$ to be in agreement with control community standard notations.

The stiffness and the damping are completely due to columns, walls, non-structural vertical elements. The identification of these values is usually a critical point, especially regarding damping (see appendix (C)).

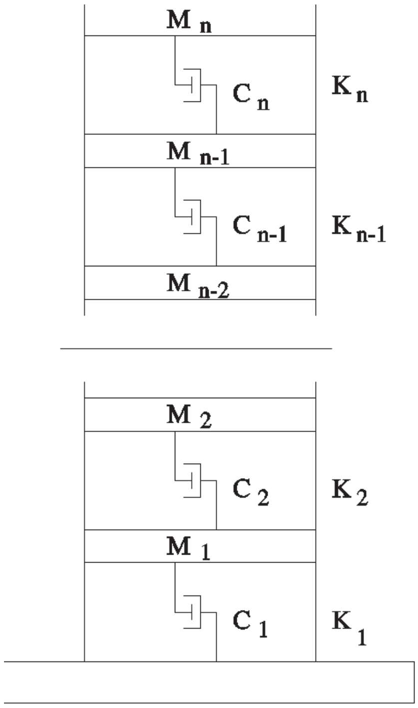


Figure 1.2: Simplified model of a multi-storeys building

In this case, the three matrixes **M**, **C** and **K** can be obtained as follows. The forces acting on each storey are depicted in figure (1.3). The inertial force due to the acceleration of the storey is proportional to the acceleration itself times the floor mass.

On both the lower and the upper side of the mass there are two forces acting: the viscous and the elastic forces. These forces are related to the inter-storey displacement and the inter-storey velocity.

Making the balance of all forces acting on each storey, the generic equation for the i^{th} storey can be obtained:

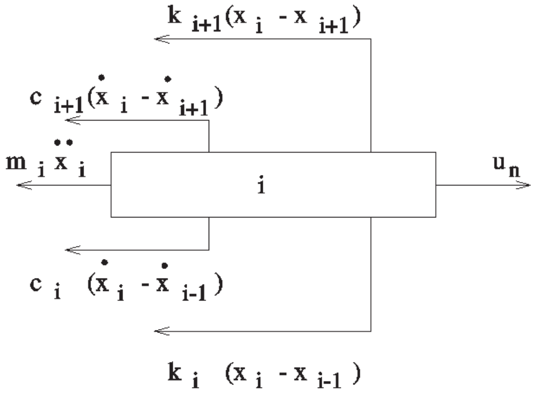


Figure 1.3: Forces acting on the ith storey

$$m_i\ddot{x}_n + (c_{i+1} + c_i)\dot{x}_i + (k_{i+1} + k_i)x_i - c_{i+1}\dot{x}_{i+1} \\ - c_i\dot{x}_{i-1} - k_{i+1}x_{i+1} - k_i x_{i-1} = u_i \quad (1.7.8)$$

where u_i is the generic control force acting on the i^{th} floor.

After some calculations, the **M**, **C** and **K** can be obtained from (1.7.8) as follows (the omitted terms are all zero):

$$\mathbf{M} = \begin{bmatrix} m_1 & & & & & \\ & m_2 & & & & \\ & & \ddots & & & \\ & & & m_i & & \\ & & & & \ddots & \\ & & & & & m_n \end{bmatrix}$$

$$\mathbf{C} = \begin{bmatrix} c_1 + c_2 & -c_2 & & & & & \\ -c_2 & c_2 + c_3 & -c_3 & & & & \\ & \ddots & \ddots & \ddots & & & \\ & & -c_i & c_i + c_{i+1} & -c_{i+i} & & \\ & & & \ddots & \ddots & \ddots & \\ & & & & -c_{n-1} & c_{n-1} + c_n & -c_n \\ & & & & & -c_n & c_n \end{bmatrix}$$

$$
\mathbf{K} = \begin{bmatrix}
k_1 + k_2 & -k_2 & & & & & \\
-k_2 & k_2 + k_3 & -k_3 & & & & \\
& \ddots & \ddots & \ddots & & & \\
& & -k_i & k_i + k_{i+1} & -k_{i+i} & & \\
& & & \ddots & \ddots & \ddots & \\
& & & & -k_{n-1} & k_{n-1} + k_n & -k_n \\
& & & & & -k_n & k_n
\end{bmatrix}
$$

It can be observed that $\mathbf{M}$ is a diagonal matrix, while $\mathbf{C}$ and $\mathbf{K}$ are tri-diagonal.

Matrix $\boldsymbol{\mathcal{A}}$ can be obtained following equation (1.7.5). Matrix $\boldsymbol{\mathcal{B}}$ has usually more than one column: some of them are usually related to the forcing force (as for example the forces induces by the ground acceleration caused by an earthquake or given by the wind acting on the façade of the building) and can be obtained with equation (1.7.6). The columns of $\boldsymbol{\mathcal{B}}$ related to the action of the control actuators must be evaluated following the idea that matrix $\boldsymbol{\mathcal{B}}$ will be a kind of topological operator telling where the actuators are presents. For example, if this multi-storeys building will be equipped with only one actuator on the top floor, the $\boldsymbol{\mathcal{B}}$ matrix will be zero everywhere except in correspondence of that degree of freedom. Matrix $\boldsymbol{\mathcal{D}}$, as already said, is usually set to zero.

As regarding matrix $\boldsymbol{\mathcal{C}}$, it must be constructed accordingly with the aims of the particular problem. One common choice is to use the identity matrix in order to have direct access to the states of the system: in this case each output is directly in correspondence with a state.

1.8 Comparisons among passive, active and semi-active control strategies

A brief comparison among passive, active and semi-active control strategies is given in the following paragraphs. The few works present in literature on this subject both in time and frequency domain will be taken into account. Some critical consideration about the opportunity to use one domain or the other one are presented in appendix (D).

1.8.1 Semi-active versus passive

Some properties of passive and semi-active systems will be discussed hereafter. A single degree of freedom system (SDOF), i.e. the classical dashpot, mass and spring oscillator, will be considered. The transmissibility of this system (i.e. the ratio between the maximum base force and the amplitude of the applied force) is given graphically in figure (1.4) for different damping values. On the abscissas axis there is the ratio β between the natural frequency of the system Ω and the frequency of the forcing force ω. It clearly appears that a higher value of damping is positive for frequencies lower than the critical value, but for high frequencies a very low value of damping is preferable.

This means that the insertion of passive dissipation devices into structures can sometimes lead to unwanted drawbacks, especially during transient response (Pinkaew and Fujino 2001). Usually an augmented level of damping means also higher level of forces at some structural connection. With a semi-active device it is possible to adjust damping in the most proper manner, for example using an on-off control law that switch the damping value from a high value to a minimum one. In the example of figure (1.4) a switching frequency corresponding at $\beta = 1.414$ can be chosen. Using more sophisticated control laws (as for example skyhook control or clipping control) performances comparable with active devices can be obtained.

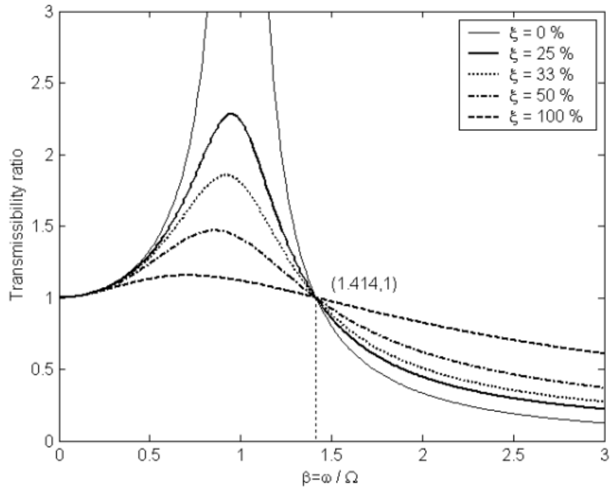


Figure 1.4: Transmissibility of a SDOF system with several value of supplemental damping

1.8.2 Semi-active versus active

Figure (1.5) summarised what just exposed. The grey region between the active and passive response curves is the theoretical possible one for a semi-active system (for more details about transfer function of semi-active systems see also (Pinkaew and Fujino 2001)).

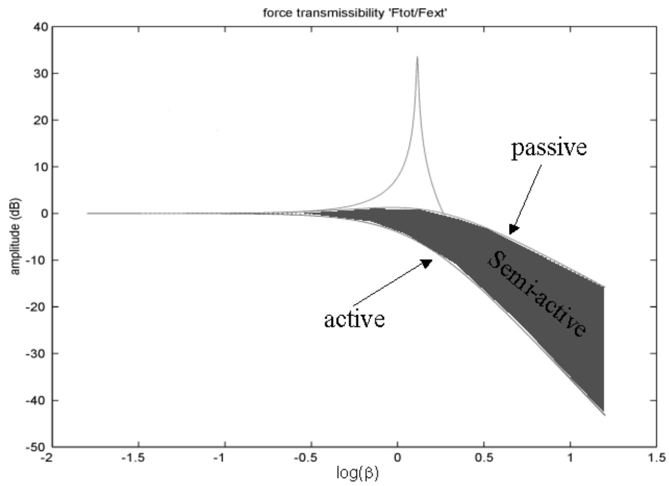


Figure 1.5: Transmissibility of a SDOF system with active, semi-active and passive control systems

This figure has the advantage of showing in a very simple manner how the semi-active control is better than a passive one and how it can approach the performance of the active systems. However, in real applications, not only the damping is the critical parameter to be modify, but also the forces that are present at the various storey of the structure, the displacements, the inter-storey drifts, the accelerations: it is usually necessary to try make a balance among all this constrains.

Chapter 2

Comparison between collocated and non-collocated systems

2.1 Introduction

Before entering the core of the comparison between collocated and non collocated systems, a clarification about terminology is necessary. What is the exact meaning of the statement "collocated system"?

2.1.1 Definition of collocated system

A system is collocated when the force generated by an actuator in a point of the structure is measured by a force sensor at the same location. This definition implicitly assumes that the transfer function that can be obtained is related to the same location of the structure. With this hypothesis, and in an ideal world in which the actuator and the sensor are connected exactly in the same location on the structure, this definition is correct.

Following the way paved by (Curtain and Zwart 1995), in a more mathematical language, Degryse and Mottelet (Degryse and Mottelet 2000) give a definition[1] based on the transfer function between the control matrix $\boldsymbol{\mathcal{B}}$ and the output matrix $\boldsymbol{\mathcal{C}}$ (see S (1.7) for the definition of the notations).

Definition 5 *When $\boldsymbol{\mathcal{C}} = \boldsymbol{\mathcal{B}}^T$ the sensors and the actuator are collocated.*

In the previous given definition it must be considered that operators $\boldsymbol{\mathcal{B}}$ and $\boldsymbol{\mathcal{C}}$ are defined up to some multiplicative constant. In fact the outputs of the measurement

[1](Degryse and Mottelet 2000) definition is more general, but in our case the simplification hereafter explained is sufficient.

device and the inputs of the actuators are voltages which are proportional to the physical quantities of interest. So normalization constants are everywhere. Moreover, multiplying $\boldsymbol{\mathcal{B}}$ by some constant does not change the system.

In case definition (5) is satisfied, the feedback law $\mathbf{u} = -\mathbf{k}\mathbf{y}$, with $\mathbf{k} > \mathbf{0}$, provides unconditional stability of the system (1.7.1). The system (1.7.1) transfer function is:

$$G(s) = \sum_{k>0} \frac{\boldsymbol{\mathcal{C}}\phi_k \boldsymbol{\mathcal{B}}\phi_k^T}{s^2 + \lambda_k^2} s \tag{2.1.1}$$

where ϕ_k are the eigenfunction of $\boldsymbol{\mathcal{A}}$ and λ_k^2 are its eigenpairs ($k > 0$).
The transfer function is demonstrated to be positive-real,

$$G(s) > 0 \qquad \text{for} \quad \Re(s) > 0 \tag{2.1.2}$$

and positive-real systems can be stabilised by strictly positive-real gain matrixes through negative feedback.

Besides this definition of collocations, in (Degryse and Mottelet 2000) a relaxation of the hypothesis is presented where unconditional stability is nevertheless maintained. In fact, it can be shown that, if the following inequalities are verified

$$\boldsymbol{\mathcal{C}}\phi_k \boldsymbol{\mathcal{B}}\phi_k^T > 0 \qquad k > 0 \tag{2.1.3}$$

then unconditional stability[2] can be proved even if $\boldsymbol{\mathcal{C}} \neq \boldsymbol{\mathcal{B}}^T$. It can be shown that (2.1.3) implies that the transfer function (2.1.1) is positive-real, but the necessity of placing the sensors and the actuators at the same location still remains, at least when $\boldsymbol{\mathcal{B}}$ and $\boldsymbol{\mathcal{C}}$ are bounded, as shown in (Degryse and Mottelet 2000). This means that actuators and sensors must still not have disjoint supports, even if it is possible to control an actuator using the input coming from a sensor placed on another actuator.

Another important point must be noted: if a time delay occurs, even if the collocation definition is respected, the transfer function is distorted. In practice, time delays that are much smaller than the system dynamic can be neglected.

So, the lying hypothesis that no time delay occurs in the measurement and command chain is always present in civil engineering controlled structures.

2.2 Centralised and non-centralised systems

Sometimes there is a little confusion between collocated and non-collocated systems with centralised and non-centralised system (improperly called also distributed and non-distributed systems). This confusion comes from the fact that a

[2] It not necessary that equation (2.1.3) has being satisfied for all k, i.e. for every mode.

collocated system will be quite probably also a non-centralised system and a non-collocated system is often a centralised system. But this is not always the case. The two concepts are different as explained hereafter.

Definition 6 *A system is considered centralised if the control systems is managed by a unique computer that takes the inputs from all sensors and give the command outputs to all actuators.*

The definition of non-centralised system is consequent:

Definition 7 *A system is considered non-centralised if the control system is managed by several computers that take the input from some specific sensors and give the command outputs to some actuators.*

With these definitions in mind, it clearly comes that collocation has nothing to do with centralisation: in some cases it my happen that a completely non-centralised system can be also a collocated system, but this is not compulsory. We can have, for example, a collocated system in which the control is performed on a central computer, or we can have structures controlled by several sets of non-collocated control systems. This last situation can be use to avoid problems related to failure of a portion of the whole control system, for example.

2.3 Linear and non-linear systems

Before entering into the details of the following paragraphs, it is important to notice that, in general, there are two classes of systems:

1. **linear systems** in which equations describing their dynamic behaviour are linear;
2. **non-linear systems** in which equations are non longer linear.

The first class is characterised by having the property of linearity, which implies that the superposition principle can be applied and proportional causes give proportional effects.

The second class is rather a non-class, in the sense that every system not belonging to the first one naturally fall into this second one. In this case no general rules can be assess, even if some particular sub-classes may still be found.

In civil engineering applications, especially in earthquake engineering, non-linear behaviour is very common because of inelastic deformations and damages. This is especially true if no protection is added to the structure. If a control system is designed to help the structure to resist to ambient loads, it can be designed in order to maintain the structure in a linear range. In this case the real behaviour of the system can be linearised around an equilibrium point and the definition of linear system still apply.

For this reasons, in the following, the only class of linear system will be taken into consideration, even if must be pointed out that the second class is receiving more and more attention, see for example (Barroso 1999).

2.4 Characteristic of collocated systems

2.4.1 Properties of the transfer function

According to (Preumont 1997), the following expression for the generic transfer function of a system can be written as a frequency response function as follows[3]:

$$G(\omega) \simeq \sum_{i=1}^{n} \frac{\phi_i \phi_i^T}{\mu_i(\omega_i^2 - \omega^2 + 2i\xi_i\omega_i\omega)} \tag{2.4.4}$$

where the sum extends to all modes, μ_i is the modal mass of mode i, ξ_i is the modal damping ratio, ω_i is the natural frequency of mode i and ϕ_i is the corresponding natural mode shape. $G_{lk}(\omega)$ express the complex amplitude of the structural response of degree of freedom l when the structure is exposed to a steady state harmonic excitation $e^{j\omega t}$ at degree of freedom k. For a limited frequency band, it is possible to select a sufficiently high index m by which to split (2.4.4) into two

[3]To be consistent with the previous given definition (§ 2.1.1) it must be notice that in the following:

- the transfer function G has been hereafter written in the Fourier domain ω instead than in the Laplace domain s: for this reason there is $G(\omega)$ instead of $G(s)$. This transformation can be done in this case very easily substituting s with $j\omega$ and s^2 with $-\omega^2$;
- the running index is i and no longer k;
- the natural frequencies of the system are hereafter called ω_i and not λ_k;
- in their analysis, (Degryse and Mottelet 2000) didn't considered the damping in the wave equation. So in the inverse of matrix $\boldsymbol{\mathcal{A}}$ the term

  $$\frac{1}{s^2 + \lambda_k^2}$$

  becomes

  $$\frac{1}{\omega_i^2 - \omega^2 + 2i\xi_i\omega_i\omega}$$

  where ξ_i is the damping associated to the i^{th} modes;
- $\boldsymbol{\mathcal{B}}$ is in this case $\begin{bmatrix} 0 \\ \phi^T/\mu \end{bmatrix}$ where μ_i are the normalised masses;
- $\boldsymbol{\mathcal{C}}$ is in this case $[0 \quad \phi]$.

parts: the first one will respond dynamically and the part with high frequency modes which responds statically:

$$G(\omega) \simeq \sum_{i=1}^{m} \frac{\phi_i \phi_i^T}{\mu_i(\omega_i^2 - \omega^2 + 2i\xi_i\omega_i\omega)} + \sum_{i=m+1}^{n} \frac{\phi_i \phi_i^T}{-\mu_i \omega_i^2} \tag{2.4.5}$$

or, in another form:

$$G(\omega) \simeq \sum_{i=1}^{m} \frac{\phi_i \phi_i^T}{\mu_i(\omega_i^2 - \omega^2 + 2i\xi_i\omega_i\omega)} + R - \sum_{i=1}^{m} \frac{\phi_i \phi_i^T}{-\mu_i \omega_i^2} \tag{2.4.6}$$

where the residual mode given by R is the static contribution of the high frequency modes to the flexibility matrix K^{-1}. It is independent of the frequency ω and introduces a feed-through component in the transfer matrix: part of the output is proportional to the input. Truncate the modal expansion of the transfer function without introducing a residual mode can lead to substantial errors in the calculation of the open-loop zeros and, as results, of the performance of the control system.

With this in mind, let's now consider the sum of the firsts k^{th} terms of equation (2.4.6):

$$G_{kk}(\omega) = \sum_{i=1}^{r} \frac{\phi_i^2(k)}{-\mu_i \omega_i^2} + \sum_{i=r+1}^{m} \frac{\phi_i^2(k)}{\mu_i(\omega_i^2 - \omega^2)} + R_{kk} \tag{2.4.7}$$

Equation (2.4.7) is the transfer function between the input and the output of the corresponding degree of freedom.

The behaviour of $G_{kk}(\omega)$ is represented in figure (2.1). The shape of this transfer function reflects the fact that the derivative of $G_{kk}(\omega)$ with respect of ω is always positive. This means that the function must be always increasing. It will goes from $-\infty$ to $+\infty$ passing 0 at the anti-resonance frequency. The amplitude of the transfer function goes to $\pm\infty$ at the resonance frequencies ω_i. A series of properties can be inferred:

- if there is no damping, control systems using collocated actuator and sensor pairs have alternated poles and zeros along the imaginary axis of the complex plane. In case of little damping (very common case when unacceptable vibration are observed), the poles and zeros are still alternated near the imaginary axis of the complex plane. This observation will be very useful to study the stability;

- A harmonic excitation at an anti-resonance frequency produce no response at the degree of freedom where the excitation is applied: as can be seen from figure (2.1), the transfer function has 0 value in these points, so nothing is

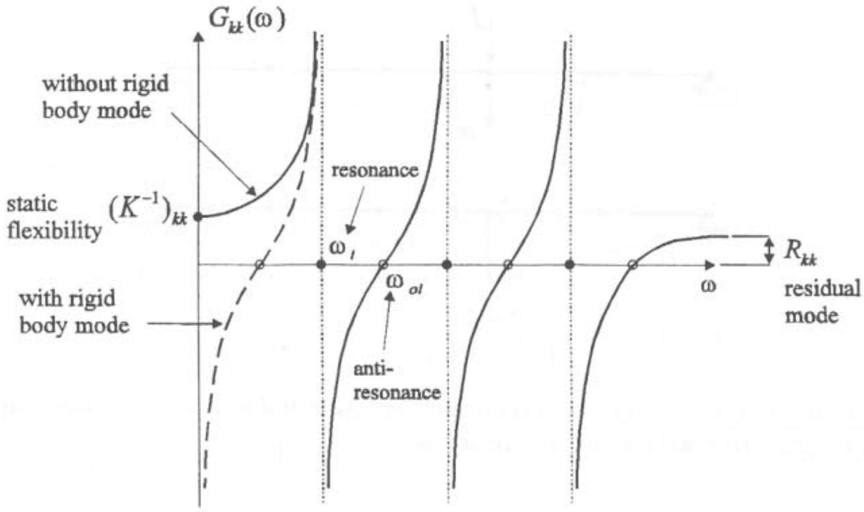


Figure 2.1: Transfer function of an undamped structure with collocated actuator and sensor (from Preumont)

transmitted. This means that, at anti-resonance frequency, the structure behaves as if an additional restrain has been added in that point.

- In contrast to the resonance frequencies, the anti-resonance frequencies depend on the actuator location, so if another diagonal term $G_{kk}(\omega)$ is considered, the anti-resonance frequencies will, in general, change.

2.5 The problem of spillover

Generally speaking, any structure can be viewed as a distributed parameter system with an infinite degree of freedom number. A physical system, in fact, has always infinite natural modal shapes and frequencies. Spillover is related to the discretisation and the approximation of a continuous system to a finite degree of freedom system. From an engineering and practical point of view, it is common practice to operate two kinds of reduction of the number of degrees of freedom:

- a continuous system is discretised in a finite number of degree of freedom (for example using some Finite Element Method or by simplified considerations on mass and stiffness concentration in some structural members);

- the discretisation of a system, especially if obtained with a FEM procedure, is often either too heavy to be treated by a control system or not economically feasible. Only the lowest and most meaningful frequencies are then taken into consideration.

These discretisations (the first step is compulsory if numerical solutions are searched, while the second one can be performed or not, accordingly with the complexity of the considered structure) lead to the consequence that the obtained model is able to describe only a part of the real system dynamic. The numerical model (often called the reduced model) will deal with the few dominant low frequency modes.

In this scenario, when dealing with flexible structures, there is the danger that the state feedback based on the reduced model destabilises the high frequency remaining modes (called also "residual modes") which are not included in the model of the structure. This can lead to a destabilisation of the reduced model. This phenomenon is called spillover.

2.5.1 Observation and control spillover

The sensor outputs are contaminated by the residual modes through the (so called) observation spillover and the feedback control excites these modes through the control spillover (see figure (2.2)).

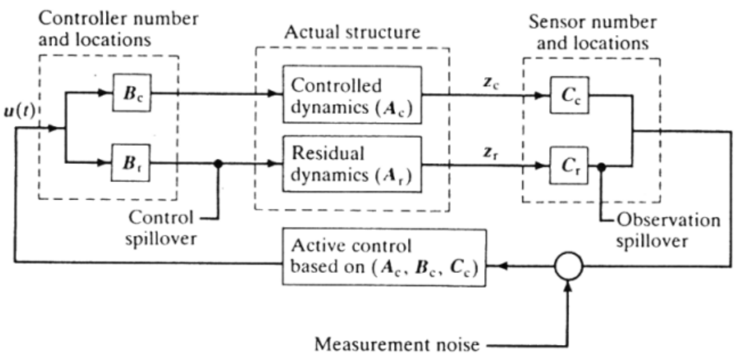


Figure 2.2: Control and osservation spillover (from Soong)

A pioneer work in spillover definition and study can be found in (Balas 1978), where the control and observation spillover due to residual (uncontrolled) modes are examined and the combined effect of control and observation spillover is shown to lead to potential instabilities in the closed-loop system. It is also proved that, if observation spillover is absent, control spillover itself cannot destabilised the system. In this case, control spillover causes unwanted excitation of the residual modes that can degrade the system response but cannot destabilised the system.

In his book, (Meirovitch 1990) arrives to the same conclusions. He says that, because the term given by control spillover has no effect on the eigenvalues of the close-loop system, it can be concluded that control spillover cannot destabilise the system, although it can cause some degradation in the system performance. So, only observation spillover is really dangerous.

He proposes also some solutions for reducing observation spillover. As a rule of thumb he affirms that it can be greatly reduced by using a large number of sensors[4]. The same author says that instability in the residual modes given by observation spillover can be often overcome by a small amount of inherent damping[5] in the structure or can be eliminated if the sensor signals are prefiltered[6] so as to screen out the contribution of the uncontrolled modes. However this last solution is not as simple as can seems, because it must be known in advance which are the uncontrolled modes. Direct output feedback control is proposed to avoid spillover, but in this proposal it not considered that time delay can occur that can still generate instability.

In his book, (Soong 1989) describes very well the two stages model reduction procedure whereby the distributed parameter system to be controlled is first reduced to a many degree of freedom system discretised in space, then to a reduced order system with only a small number of degree of freedom. Since the control law design is based on this reduced order system, spillover is always possible.

From the point of view of spillover reduction, he says that it should be better to locate controllers and sensors at or very near the zeros of the affected modes, but he admits that usually this is quite impossible. Because the controller is designed taking into consideration only the lowest modes, he proposed a method that penalised the highest and non modelled modes. Clearly this method can be effective only for spillover due to the second step of discretisation, because the designer must know in advance which modes is not considering.

A very useful example shows the amount of high frequency noises given by spillover in a practical example.

2.5.2 Mathematical formulation

In order to see the effect of spillovers (both observable and control ones), let us consider a discretised system given by equation

[4] Surely, in this case, a true analysis should be carried out. Moreover, sensors are usually not so expensive, but actuators are much more expensive: it is not feasible to put them everywhere!

[5] This, in general, may be true or not: in case of wind, structures can have negative damping!

[6] This may be very difficult if modes are very close and overlap each other!

$$\dot{\mathbf{z}}(t) = \boldsymbol{\mathcal{A}}\mathbf{z}(t) + \boldsymbol{\mathcal{B}}\mathbf{u}(t) \tag{2.5.8}$$

with the observation equation given by

$$\mathbf{y}(t) = \boldsymbol{\mathcal{C}}\mathbf{z}(t) \tag{2.5.9}$$

where, as mentioned in §(1.7), $\mathbf{z}(t)$ is the $2n$-dimensional state vector of the structural system with n degree of freedom, $\mathbf{u}(t)$ is the m-dimensional control vector and $\mathbf{y}(t)$ is the p-dimensional observation vector.

A reduced order model can be generated through aggregation or modal eigenfunction expansion techniques by retaining only the controlled modes of the system, giving

$$\dot{\mathbf{z}}_c(t) = \boldsymbol{\mathcal{A}}_c\mathbf{z}_c(t) + \boldsymbol{\mathcal{B}}_c\mathbf{u}(t) + \boldsymbol{\mathcal{E}}_c(t) \tag{2.5.10}$$

with the observation equation

$$\mathbf{y}(t) = \boldsymbol{\mathcal{C}}_c\mathbf{z}_c(t) + \boldsymbol{\mathcal{R}}_c(t) \tag{2.5.11}$$

In the above, $\mathbf{z}_c(t)$ is the controlled portion of the state vector $\mathbf{z}(t)$, whose dimension is in general much smaller. $\boldsymbol{\mathcal{E}}_c(t)$ and $\boldsymbol{\mathcal{R}}_c(t)$ are error terms introduced through the truncation process; they can be represented by

$$\boldsymbol{\mathcal{E}}_c(t) = \boldsymbol{\mathcal{A}}_{cr}\mathbf{z}_r(t) \tag{2.5.12}$$

and

$$\boldsymbol{\mathcal{R}}_c(t) = \boldsymbol{\mathcal{C}}_r\mathbf{z}_r(t) \tag{2.5.13}$$

where $\mathbf{z}_r(t)$ is the state vector associated with the residual (or uncontrolled) modes of the system (2.5.8). It is governed by

$$\dot{\mathbf{z}}_r(t) = \boldsymbol{\mathcal{A}}_r\mathbf{z}_r(t) + \boldsymbol{\mathcal{B}}_r\mathbf{u}(t) + \boldsymbol{\mathcal{E}}_r(t) \tag{2.5.14}$$

The error term $\boldsymbol{\mathcal{E}}_r(t)$ in the residual equation has the form

$$\boldsymbol{\mathcal{E}}_r(t) = \boldsymbol{\mathcal{A}}_{rc}\mathbf{z}_c(t) \tag{2.5.15}$$

The error term $\boldsymbol{\mathcal{E}}_c(t)$ in equation (2.5.10) represents the modelling error due to the model reduction process. The term $\boldsymbol{\mathcal{B}}_r(t)\mathbf{u}(t)$ in equation (2.5.14) shows the effect of control $\mathbf{u}(t)$ entering the residual subsystem, or control spillover, to the residual modes. The contamination of observation spillover in equation (2.5.11) with residual information $\boldsymbol{\mathcal{R}}_c(t)$ produces observation spillover. Thus, the controller imparts energy to the residual modes through the interaction term $\boldsymbol{\mathcal{B}}_r(t)$ and the resulting residual mode excitation is detected by the sensors through the term $\boldsymbol{\mathcal{R}}_c(t)$ for the control design, resulting in an escalating performance degradation.

These interactions are shown graphically in figure (2.2). It can be shown that spillovers can reduce stability margins of the actual structure and are at the heart of the control problem based on the reduced-order models.

Clearly, the magnitude of the control and observation spillover is a function of the model reduction process. It is also a function of controller and sensors locations and their effects on the residual modes. Spillover effect is important when the control design is carried out based on the reduce-order model assuming that $\boldsymbol{\mathcal{E}}_c(t) = \mathbf{0}$ and $\boldsymbol{\mathcal{R}}_c(t) = \mathbf{0}$, but is applied to the full-order system given by equations (2.5.8) and (2.5.9).

2.5.3 Physical interpretation of spillover

To give a more physical meaning to the above mentioned definition of spillover, a practical explanation of this effect is given hereafter with reference to figure (2.3).

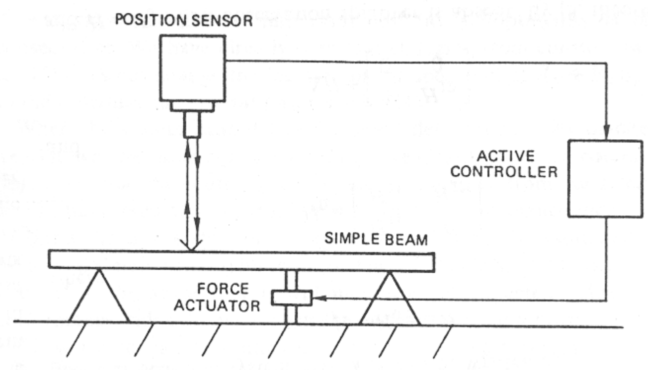


Figure 2.3: Control of a simply supported beam (from Balas)

A force actuator is placed under a simple supported beam and is controlled via an active controller. The input signal to the control algorithm is given by a position sensor (in this case a laser transducer sensor) placed somewhere on the beam but not in the same position of the actuator. So the beam is controlled in a point measuring the displacement in a different point (this is the typical non collocated architecture).

In an ideal world, the actuator will exactly follow the command signal that the controller sends it and the sensor will exactly measure the current displacement without any delay. The controller will execute the algorithm instantaneously and communication times among the parties will be zero.

Even in this case, however, instability can occur because of spillover phenomenon. The controlled beam, in fact, has its internal dynamic: this means that the waves generated by the actuator will propagates along the beam. A continuous beam has infinite natural modes but only a few number of them can be taken into account when designing the control law. This means that there will surely be some modes the control strategy has not been designed for. If the sensor is placed at a distance that is a multiple of the half wavelength of the displacement travelling wave of an uncontrolled beam mode, the resulting signal could be a displacement with an inverted sign with respect to the displacement at force actuator location.

This mode has not been properly taken into account in the design process of the non collocated control law, so it may happen that algorithm command the actuator to generate forces that excite the beam instead of stopping it.

This means that the dynamic behaviour of the beam must be considered in the controller design with highest possible definition. Obviously, in the controller process design it is not realistic to consider more than a few modes, usually the modes associated with the lower frequencies. This means that the controller can give improper command signal to the actuator due to the lack of the highest modes.

If there is no observation spillover, however, control spillover cannot destabilise the system, even if it can induce cyclic oscillations.

2.6 Advantage and disadvantage for collocated and non-collocated systems

In order to achieve an unbiased judgement about the opportunity of adopting a collocated or a non-collocated control system, the main characteristics of these two schemes will be discussed extensively hereafter. It is important to notice that no best solutions can be taken a priori, because the right choice depends on the particular needs and depends on the control systems for that particular case. Constrains are different from a situation to another and they must be evaluated by the control designer.

2.6.1 Robustness

As it was explained in the previous paragraphs, depending on the control law and the actuator dynamics, theoretically speaking a collocated control system is inherently stable. This is true only if the system is strictly collocated in the sense of definition (5) and if there are no delay in the measurement chain, the actuator is ideal, no saturation effects occur etc. In this case it was shown that the transfer function has poles and zeros alternating on the left side of the imaginary axis and the root locus gives trajectory always included in the left part of the complex plane. This means that any chosen gain of the control system insures stability, the remaining question being to chose the optimal gain value. Robustness of some collocated strategies is also related on the fact that, for particular choice of control laws (like for example Integral Force Feedback (IFF) and Direct Velocity Feedback (DVF)), they don't require the description of the mathematical model in order to run properly: this means that the control system can be designed and optimised with nominal values for the parameters involved and it will work even in non-nominal conditions. In case of non-nominal conditions the efficiency will probably decrease, but stability is related only on local considerations and is independent on the real behaviour of the structure, so only optimality is lost. Concluding, collocated systems can be considered inherently robust.

With reference to this last point, some authors deem that some common active control laws, like for example Integral Force Feedback, Direct Velocity Feedback etc. cannot be considered as active, because in these cases the actuator behaves exactly like a passive device, like a dashpot. This discussion is strictly related to the fact that collocated systems using the mentioned laws are guarantee to adsorb energy (and so to damp the structure) and cannot become unstable in a similar way of passive devices, providing additional damping. To solve this apparent contradiction, the definition of active control given in (§ (1.2)) must be recalled: devices using IFF or DVF control strategy need a large amount of energy (which usually does not go into the structure) to work, so they are surely active, even if, sometimes, the same effect could be obtained by an proper tuned passive damper.

The fact that only in ideal situation the absolute robustness for stability can be achieved must be attentively considered: the author himself had the opportunity of observing the unstable behaviour of a collocated IFF control system applied to a cable-stayed bridge under the framework of ACE[7] project when very little displacements are required to an hydraulic actuator with internal dry friction. Even if this instability led to a very little cyclic behaviour of the actuator (and so not dramatic from the point of view of safety), the energy consumption due to the continuous corrections of the actuator's position was high.

A non-collocated control system is always subject to spillover problems, either regarding observation spillover or control spillover, as it was mentioned earlier. This problem is inherent to the non-collocated control system itself, where a model of the structure to be controlled is needed[8] and, even in the best situation, it is a discretisation of a continuous structure. Moreover, one must consider that the root locus of a non-collocated system can pass from the left part of the complex

[7] Active Control in Civil Engineering, project funded by the European Community under the Industrial & Materials Technologies, Programme Brite Euram 3, Proposal N.BE96-3334, Contract N.BRPR-CT97-0402, (1998).

[8] It could be said that the model structure is the "connection" among actuators and sensors, so the model of the structure cannot be avoided!

plane to the right one (and viceversa), so the control system can be very sensible to any distortion from the nominal design situation.

These two problems (spillover and non-nominal working conditions) can however be addressed quite efficiently with modern robust control techniques. For example with μ-synthesis techniques a robust control design can be performed taking into account the unmodelled high dynamic and the possible variation of some parameters into bounded intervals (see for example (Marazzi 1997)). With this technique we accept that the nominal gain is no longer the very optimal one (so we loose a little in efficiency in the nominal situation), but in exchange we are mathematically sure that the system will remain at least stable in the range of variation of the parameters (and usually quite efficient, too).

Another important question is: what happen if a sensor or an actuator will fail? Could we maintain unconditional stability even in this case? This question is very important, but the answer is very complex.

In practice it may happen that a sensor fails. Two cases can be considered:

- the signal falls down to zero value;
- the signal assume a static value (for example to value 1 or any other saturated value).

In principle, a control system can be design to be able to bypass the signal coming from the bad sensor. In a collocated control system the corresponding actuator could be switched off, while in the non-collocated control strategy a new control algorithm avoiding that signal can be adopted.

A more difficult situation must be faced when the sensor is not responding or is delivering a wrong signal due to mismatch of gain, an offset, a noise corruption and so on. In this case it is very difficult, for the control system, to detect the sensor fault. So, once again, a system must be retained robust if the overall behaviour of the control system remain stable even in presence of fault and error.

One strategy for strengthening robustness could be the use of redundancy measurements (Noltingk 1996) (common practice in nuclear power plants, for example) in order to be able to accept that some sensors can give wrong values. Three measures can be taken for each quantity and the mode value can be assumed as the right one. In this case it is sufficient that only two sensors are measuring the right value. With three sensors it is also possible to control if any sensor has failed: if the mean value of two sensor is quite close to the value of the third, all is going well, on the contrary, one or more sensors has surely failed.

2.6.2 Performance

Collocated control system can be very efficient when they are used to provide supplementary damping to structures. From a theoretical point of view, however, they cannot reach the high performances of non-collocated systems. A non-collocated control system, as already mentioned, need to have a model of the structure to be properly designed: this means that the level of optimisation that can be reached is considerably higher than in a collocated case in which the control system doesn't take into account the global behaviour of the structure, but only the very local one. This is true especially in positioning control problems, where the structure in its whole must assume a given configuration.

2.6.3 Realisation aspects

It is very difficult to deal with realisation aspects if both the structure, where the control system will be applied, and the control system itself is not yet chosen. Since a very large number of variables are involved, no preference can be given a priori to a collocated or non collocated strategy from the point of view of practical realisation.

The author can only report his experience during these past years with particular reference to the ACE project. In that case, a collocated systems resulted to be very easy to implement, more than a non-collocated one. For programming the controller, in fact, it was not necessary to have a complete model of the cable-stayed bridge, but only a rough idea about the interesting frequencies involved. Obviously the range of the working forces was needed to properly design the actuators, and in this sense some information about the hosting structure was necessary, but no very precise model was requested during the control law design stage. On the contrary, when implementing a non collocated control strategy, a model of the structure should always be used.

2.6.4 Simplicity

Mounting the actuators and sensors in a real environment is always an important and expensive task.

With a non-collocated control strategy, both actuators and sensors must be put in communication with one or more computers where the control law is implemented. It may be a problem to install the transmission cables inside the structure, because proper location must be found in order to avoid vandalism or whether damages.

On the contrary, a collocated system can be very compact and complete in itself, in the sense that the actuator, the sensor and the processor can be integrated in a single device. In this case there is no longer need for cables connections and the mounting phase consists only in some mechanical operations, but no electronic connections must be arranged.

The amount of electric power needed by the device is more related on the type of control than on the adopted strategy:

- for semi-active control a low power electrical supply is sufficient;
- for active control system a large amount of electrical supply (or hydraulic) is needed.

This influence the realisation, so semi-active device are much more easy to install than active ones.

2.6.5 Economical aspects

The a priori comparison between collocated and non-collocated control systems costs is very difficult. The needed implementation time makes the difference. As already explained, usually non-collocated systems need a more deep design, so they should be more expensive. Moreover the production of collocated systems can be industrialised easily, so the unit price should decrease. However, for the moment, control of civil structures is still on a pioneer phase and only a few standard products are available on the market.

The cost problem is much more related to the issue of efficiency and robustness, than on the collocated or non-collocated structure. If we use very precise sensors and actuators, in fact, we can achieve better results, but at a higher cost. However precision of sensors must be compatible with the accuracy of the actuators and viceversa, obviously. For example in ACE project, very precise load cells were used to measure force in the active stay-cables. This was due to the fact that a high resolution in control was searched for, but if design constrain are not so stringent or forces variations are high, a less accurate load cell would be enough.

2.7 Numerical studies

The control problem of a shear-type frame structure subject to dynamic loads such as earthquakes or strong wind is here addressed from the point of view of a comparison between collocated and non-collocated control strategies. The very core of this comparison is the answer at these following questions:

1. which approach has the best performance (for example comparing the transfer function between the input at the base of the structure and the displacement at the last storey or the maximum inter-storey drift along the building)?
2. which is the most robust approach (for example when any sensor fails)?
3. which is probably the best compromise between this two possibilities?

This problem is very complex and is only partly addressed in the following because a complete study of this type could be a complete thesis on its own. Moreover, as it will be shown in the next paragraph, even in the extreme simple case of a 10 storey building with equal distribution of masses, stiffness and damping, the answer to this question is almost impossible to be found.

2.7.1 Problem formulation

The structure taken into consideration is a n storey building with equal distribution of masses, stiffness and damping at each storeys. Under the hypothesis that the floor are much more rigid than the columns, the shear-type model is assumed. As it was considered only the one dimension case, no torsion effect or any coupling between two directions excitation has be taken into consideration. Figure (2.4) sketch this situation.

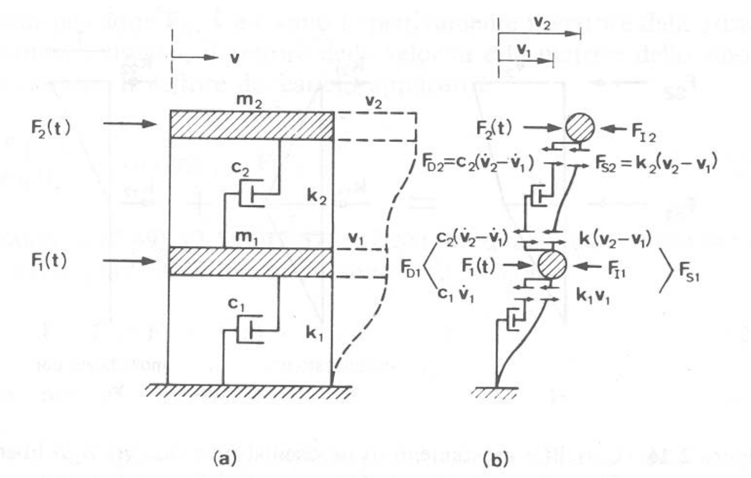


Figure 2.4: Shear type model (form Wakabayashi)

It was considered only the case of decentralised control, so each actuator works for its own. Even in the case when there is an actuator at each storey, they work independently each other. In the following we will call $A1$ the actuator placed at storey 1, $A2$ the actuator placed at storey 2 and so on.

There is only a sensor at each storey, giving displacement, velocity or acceleration and, in analogy with what defined for actuators, we will call $S1$ the sensor placed at floor 1 and so on.

2.7.2 Possible combinations

Preliminarily, the possibility of having a single actuator at any storey utilising a variable number of sensors was considered. This means that, by choosing the actuator placed at the first storey ($A1$), we have the possibility of controlling it by using the signal coming from a sensor placed on the first floor or from the sensor placed at the second floor or from the sensors placed at any floor of the building. We have also the possibility of feeding the control law with two signals coming from two different floors of the building and so on, every combination of sensors is possible.

For a 3 storeys building we have the following possibilities:

$$\{S1;\ S2;\ S3;\ S1+S2;\ S1+S3;\ S2+S3;\ S1+S2+S3\}$$

because the actuator can use one of the three signals coming from the different storeys or two of them or eventually all of them.

In this case the total number of combinations than can be found for a generic building is given by the following equation

$$N = \sum_{k=1}^{n} \binom{n}{k} \tag{2.7.16}$$

where n is the number of storeys and k is the number of sensors that the actuator can use simultaneously. In table (2.1) there is the results of equation (2.7.16) for some numbers of storeys.

$\boldsymbol{n}$	1	2	3	4	5	6	7	8	9	10	11	12
$\boldsymbol{N}$	1	3	7	15	31	63	127	255	511	1023	2047	4095

Table 2.1: Possible combinations N for a n storey building with an actuator in fixed position and a variable number of sensors

It can be seen easily from figure (2.5) that the number of combinations grows very fast, so the number of possibility to explore become extremely high as soon as the number of storeys increases. By the way, as it will be shown in the following, some of these possibilities are feasible but, in practical applications, not useful: why should we use a sensor at the last floor to control an actuator at the first floor?

The next step was to consider that the actuator could be at any storey of the structure. For the 3 storeys building already considered, the possibilities are then the followings:
$\{A1, S1;\ A1, S2;\ A1, S3;\ A1, S1+S2;\ A1, S1+S3;\ A1, S2+S3;\ A1, S1+S2+S3;$

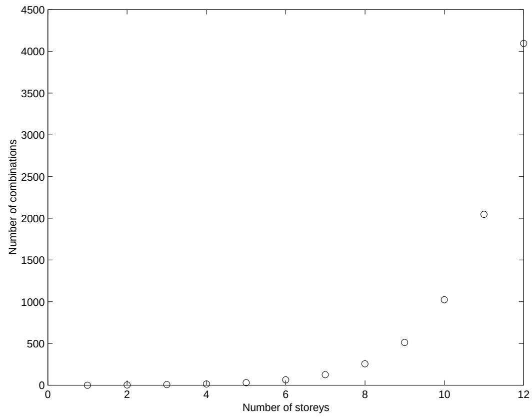


Figure 2.5: Possible combinations N versus number of storey n

$A2, S1$; $A2, S2$; $A2, S3$; $A2, S1+S2$; $A2, S1+S3$; $A2, S2+S3$; $A2, S1+S2+S3$; $A3, S1$; $A3, S2$; $A3, S3$; $A3, S1+S2$; $A3, S1+S3$; $A3, S2+S3$; $A3, S1+S2+S3\}$. The fact that the actuator can be at any floor of the building highly increases the number of possible combinations of the actuator and sensors positions, the total number being given by the following equation

$$N = n\sum_{k=1}^{n}\binom{n}{k} \qquad (2.7.17)$$

In this case the number of possible combinations is given in table (2.2).

$\boldsymbol{n}$	1	2	3	4	5	6	7	8	9	10	11	12
$\boldsymbol{N}$	1	6	21	60	155	378	889	2040	4599	10230	22517	49140

Table 2.2: Possible combinations N for a n storey building with an actuator placed somewhere and a variable number of sensors

The following step was to consider the fact that more than one actuator can simultaneously be present on the structure. In this case the equation that gives the total number of combinations becomes

$$N = \left[\sum_{k=1}^{n}\binom{n}{k}\right]^2 \qquad (2.7.18)$$

It can be noticed that the number of combinations becomes very soon practically untreatable. For a 3 storey building the combinations are:
$\{A1, S1;\ A1, S2;\ A1, S3;\ A1, S1+S2;\ A1, S1+S3;\ A1, S2+S3;\ A1, S1+S2+S3;\ A2, S1;\ A2, S2;\ A2, S3;\ A2, S1+S2;\ A2, S1+S3;\ A2, S2+S3;\ A2, S1+S2+S3;\ A3, S1;\ A3, S2;\ A3, S3;\ A3, S1+S2;\ A3, S1+S3;\ A3, S2+S3;\ A3, S1+S2+S3;\ A1+A2, S1;\ A1+A3, S1;\ A2+A3, S1;\ A1+A2, S2;\ A1+A3, S2;\ A2+A3, S2;\ A1+A2, S3;\ A1+A3, S3;\ A2+A3, S3;\ A1+A2, S1+S2;\ A1+A3, S1+S2;\ A2+A3, S1+S2;\ A1+A2, S1+S3;\ A1+A3, S1+S3;\ A2+A3, S1+S3;\ A1+A2, S2+S3;\ A1+A3, S2+S3;\ A2+A3, S2+S3;\ A1+A2, S1+S2+S3;\ A1+A3, S1+S2+S3;\ A2+A3, S1+S2+S3;\ A1+A2+A3, S1;\ A1+A2+A3, S2;\ A1+A2+A3, S3;\ A1+A2+A3, S1+S2;\ A1+A2+A3, S1+S3;\ A1+A2+A3, S2+S3;\ A1+A2+A3, S1+S2+S3\}$.
In table (2.3) there are reported the results only up to a building of 10 storeys. In figure (2.6) there is a graphical representation of the same table where the very fast grow can be clearly seen.

$\boldsymbol{n}$	1	2	3	4	5	6	7	8	9	10
$\boldsymbol{N}$	1	9	49	225	961	3969	16129	65025	261121	1046529

Table 2.3: Possible combinations N for a n storey building with a variable number of actuators and sensors

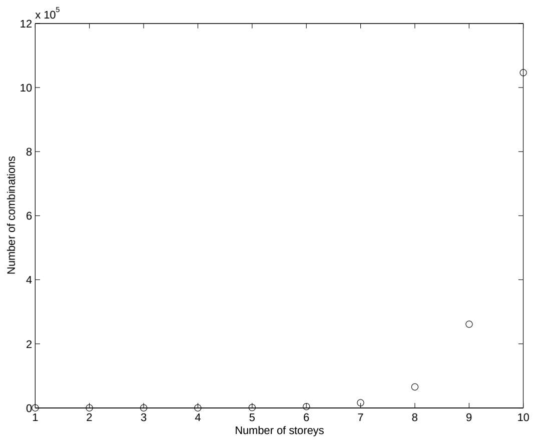


Figure 2.6: Possible combinations N versus number of storey n

This combinations are still not the very all ones, because a further possibility is to combine them (when possible) and so to have, for example $A1, S1$ and $A2 + A3, S1 + S2$, while such a combination as $A2 + A3, S1 + S2$ and $A2, S1$ is not possible.

2.7.3 A case study: 5 storey reinforced concrete building

Taking into account this preliminary study and wanting to treat the numerical comparison between collocated and non-collocated strategies in the case of a high number of storeys (at least greater then 3), only a restricted number of the previous combination sets were then taken into consideration. In particular the study was restricted to the case in which there is an actuator at each storeys that uses, for its feedback control law:

1. the only sensor collocated at the same storey;
2. the sensor collocated at the same storey and two other sensors, one at the upper floor and the other one at the lower floor;
3. the sensors at every storeys.

These three possibilities are quite realistic, because to use only a sensor for each storey is the cheapest way to achieve control and, in the meantime, is the case in which the sensors are collocated one by one with the actuators. The case of an actuator using the sensor placed on it own storey and the sensors immediately on the upper and lower one is a straight generalisation of the previous case. The case in which each actuator uses the information coming from all sensors is also interesting because, if the sensors are already placed on the structure, the additional cost and complexity can be compared with the potential benefits.

In this study these sensors are supposed to measure velocity and displacement, but the generalisation to acceleration sensors is straight. This assumption was considered because a LQR controller was designed and for this type of controller the full state knowledge is necessary.

2.7.3.1 Structural characteristics of the building

The floors of the building are very stiff in comparison with the columns, so a shear-type model is assumed. This means that the masses are all concentrated at the storeys, while damping and stiffness are only due to column deformation in shear. All floors have the same mass equal to 95538 kg, while the stiffness, in kN, are the following (from the lower to the upper storey):

160248	140086	140482	136014	136761

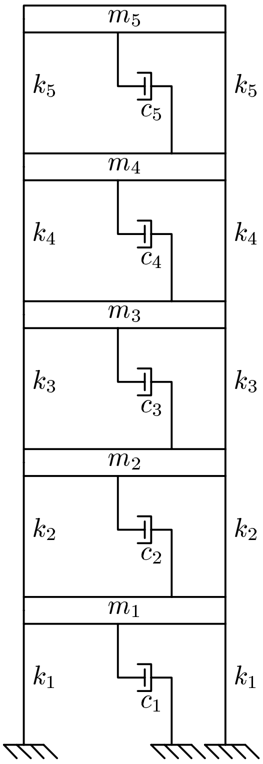


Figure 2.7: Sketch of the 5 storey reinforced concrete building

For the damping, no values are available. A Rayleigh damping (see Appendix (C.1.3)) was considered, assuming the damping of the first and third mode equal to 5%, a quite reasonable value for concrete structures.

This lead to the following inter-storey damping values (in kN/m, from the lower to the upper storey):

346.48	226.67	227.31	220.08	221.29

which correspond to a modal damping of:

0.05	0.0402	0.05	0.0592	0.0655

Figure (2.7) sketch the considered building.

Assembling the mass matrix **M** (in kg):

$$\mathbf{M} = \begin{pmatrix} 95538 & 0 & 0 & 0 & 0 \\ 0 & 95538 & 0 & 0 & 0 \\ 0 & 0 & 95538 & 0 & 0 \\ 0 & 0 & 0 & 95538 & 0 \\ 0 & 0 & 0 & 0 & 95538 \end{pmatrix} \tag{2.7.19}$$

and the matrix $\mathbf{K}$ (in kN/m) is:

$$\mathbf{K} = \begin{pmatrix} 300334 & -140086 & 0 & 0 & 0 \\ -140086 & 280568 & -140482 & 0 & 0 \\ 0 & -140482 & 276496 & -136014 & 0 \\ 0 & 0 & -136014 & 272775 & -136761 \\ 0 & 0 & 0 & -136761 & 136761 \end{pmatrix} \tag{2.7.20}$$

Assuming, as already said, a damping value of 5% on the first and third mode, with the Raleigh method the following damping matrix $\mathbf{C}$ (in Ns/m) can be obtained[9]:

$$\mathbf{C} = \begin{pmatrix} 573150 & -226670 & 0 & 0 & 0 \\ -226670 & 541170 & -227310 & 0 & 0 \\ 0 & -227310 & 534580 & -220080 & 0 \\ 0 & 0 & -220080 & 528560 & -221290 \\ 0 & 0 & 0 & -221290 & 308480 \end{pmatrix} \tag{2.7.21}$$

2.7.3.2 Natural frequencies and natural shapes

From the above matrixes, it is easy to calculate the 5 natural frequencies of the building (in Hz):

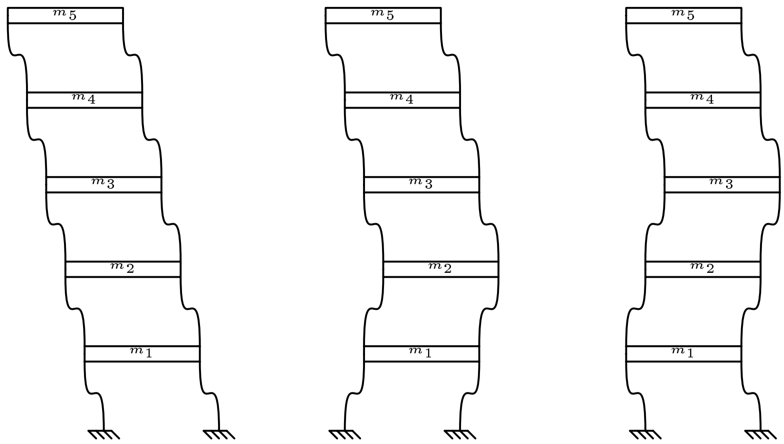
1.7715 5.1245 8.0646 10.2631 11.6678

and the corresponding modal shapes (each mode is normalized to 1):

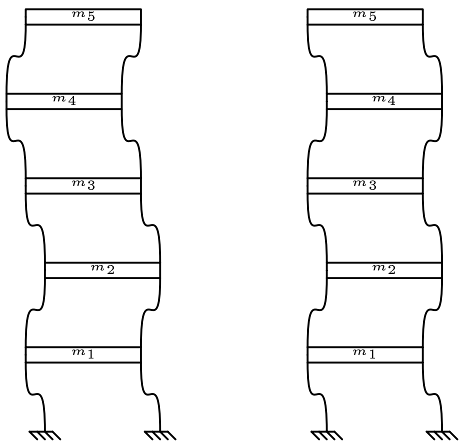
	mode 1	**mode 2**	**mode 3**	**mode 4**	**mode 5**
Storey 1	0.2539	0.6960	1.0000	0.9177	0.6557
Storey 2	0.5228	1.0000	0.3928	-0.6350	-0.9975
Storey 3	0.7469	0.5981	-0.8986	-0.3875	1.0000
Storey 4	0.9135	-0.2525	-0.6118	1.0000	-0.7120
Storey 5	1.0000	-0.9156	0.7708	-0.5250	0.2585

[9] We must recall that the Raleigh method, in a first step, provides the damping matrix that gives the requested damping on the first and the third mode and only in a second step the inter-storey damping values can be calculated. This second step must usually be performed in an approximate way, because, as in this case, the damping matrix can be a little different from the typical matrix configuration coming from a shear-type behaviour.

A sketch (not in scale) is shown in figure (2.8): the first mode gives the lowest inter-storey drift while the fifth one, on the opposite, generate the greatest inter-storey displacements.



(a) first mode shape (b) second mode shape (c) third mode shape



(d) fourth mode shape (e) fifth mode shape

Figure 2.8: Sketch of the mode shapes

This observation can be very useful, for example, if the semi-active device is designed to work as an inter-storey element, because its effectiveness is related to the relative displacement between a given storey and the upper or lower floor.

2.7.3.3 State-space representation of the 5 storeys building in the three cases

To compare the response of the structure in the three cases mentioned above, the following strategy has been used:

1. for the *one actuator - one sensor* case, the value of the mass, stiffness and damping for each storey was considered. The $\boldsymbol{\mathcal{A}}$ matrix in this case is:

$$\boldsymbol{\mathcal{A}} = \begin{pmatrix} 0 & 1 \\ M^{-1}K & M^{-1}C \end{pmatrix} \tag{2.7.22}$$

   and the $\boldsymbol{\mathcal{B}}$ matrix (a vector, in this case) is:

$$\boldsymbol{\mathcal{B}} = \begin{pmatrix} 0 \\ 1 \end{pmatrix} \tag{2.7.23}$$

2. for the *one actuator - three sensors* case, the submatrices $\mathbf{M}_s$, $\mathbf{K}_s$ and $\mathbf{C}_s$ (square matrices of order 3) were extracted from the $\mathbf{M}$, $\mathbf{K}$ and $\mathbf{C}$ respectively. The $\boldsymbol{\mathcal{A}}$ matrix is:

$$\boldsymbol{\mathcal{A}} = \begin{pmatrix} \mathbf{0}^{(3,3)} & \mathbf{I}^{(3,3)} \\ \mathbf{M}_s^{-1}\mathbf{K}_s & \mathbf{M}_s^{-1}\mathbf{C}_s \end{pmatrix} \tag{2.7.24}$$

   and the $\boldsymbol{\mathcal{B}}$ matrix is:

$$\boldsymbol{\mathcal{B}} = \begin{pmatrix} \mathbf{0}^{(3,1)} \\ \mathbf{1}^{(3,1)} \end{pmatrix} \tag{2.7.25}$$

   Some little adjustment must be done for the actuators at the first and last storey, because only two sensors could be taken into considerations in that case.

3. for the *one actuator - all sensor* case the $\mathbf{M}$, $\mathbf{K}$ and $\mathbf{C}$ were considered. Matrixes $\boldsymbol{\mathcal{A}}$ and $\boldsymbol{\mathcal{B}}$ become:

$$\boldsymbol{\mathcal{A}} = \begin{pmatrix} \mathbf{0}^{(n,n)} & \mathbf{I}^{(n,n)} \\ \mathbf{M}^{-1}\mathbf{K} & \mathbf{M}^{-1}\mathbf{C} \end{pmatrix} \tag{2.7.26}$$

$$\boldsymbol{\mathcal{B}} = \begin{pmatrix} \mathbf{0}^{(n,1)} \\ \mathbf{1}^{(n,1)} \end{pmatrix} \tag{2.7.27}$$

Using the above mentioned matrices, a LQR controller was design with $\mathbf{R} = 1$ and $\mathbf{Q}$ equal to the total energy[10] of the system (kinetic+elastic) and so equal to:

$$\mathbf{Q} = \begin{pmatrix} K & 0 \\ 0 & M \end{pmatrix} \tag{2.7.28}$$

for the *one actuator - one sensor* case: the implicit assumption is that each storey doesn't see what there is above it and considers to have the fixed ground under it.

For the *one actuator - three sensors* case the assumption is the same than in the previous case, but taking now the following block matrixes:

$$\mathbf{Q} = \begin{pmatrix} \mathbf{K}_s & \mathbf{0}^{(3,3)} \\ \mathbf{0}^{(3,3)} & \mathbf{M}_s \end{pmatrix} \tag{2.7.29}$$

Finally for the *one actuator - all sensors* case:

$$\mathbf{Q} = \begin{pmatrix} \mathbf{K} & \mathbf{0}^{(n,n)} \\ \mathbf{0}^{(n,n)} & \mathbf{M} \end{pmatrix} \tag{2.7.30}$$

In any case each actuator is design to work independently from the other ones, the only difference among the three cases being in the number of sensors it uses.

2.7.3.4 Comparisons in the frequency domain

The transfer function of the uncontrolled structure versus the controlled one (with a sensor for each actuator, with three sensors and with all sensors) was analysed. The results are shown in figure (2.9).

The dotted line represents the uncontrolled structure. The peaks relative to the five natural frequencies of the building are clearly shown: the sharpness of these peaks indicates that the natural damping is quite low. The dash-dot and the dash lines represent respectively the controlled structure in which each actuator uses one or three sensors. It is clear that the behaviour of the controlled structure is by far better than in the uncontrolled situation. The continuous line represents the case in which each actuator has full information. This case is much better than the one-sensors-only controlled one, but not relevant improvements are obtained in respect of the three-sensors case. This fact suggests that the information coming from storeys far away from the actuator is not important, so it may be not useful to have full state feedback. This can also be physically interpreted by observing

[10]This is a possible choice, but several others could be performed in this example. The important thing to take in mind is that the weighting function in the three cases must have a physical meaning in order to be able to compare these different controlling schemes. Here the meaning of assuming the $\mathbf{Q}$ matrix equal to the total energy of the considered storey is to show that, if the energy of a larger number of storeys is taken into consideration, the results can be much improved.

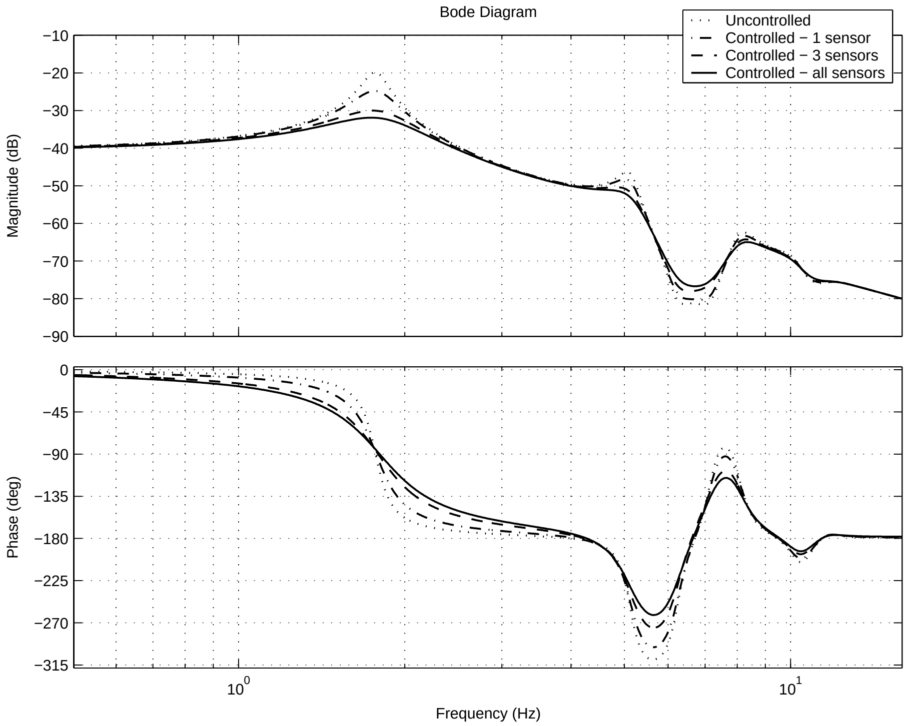


Figure 2.9: Comparison among trasfers functions between ground acceleration and displacement at the 5^{th} floor with different scheme of controlling action

that the controllability of a storey decrease with its distance from the actuated floor, so, in opposite, the vibration of a distant floor will not influence so much the movement of the controlled storey.

The comparison in the frequency domain is very useful if a linear behaviour is expected both from the building and the actuating system. It is very interesting, however, to see also if a sensor fails (when the actuator's control law was designed to taking into account such information), or if a saturation occurs on an actuator, or if the properties of the building change somewhere due to a damage and so on. In all these cases, a frequency analysis is impossible, because the linearity property is at the base of that method.

In case of non-linear effects, time domain analysis must be performed. These simulations must be done skilfully, because the input must necessary be a specific excitation time series and it may happen that it doesn't excite the building in a critical way.

2.7.3.5 Comparisons in the time domain

For this purpose, a Simulink© model was developed. In order to have a general scheme for testing a building with a chosen number of floors, the model is based on a physical balance of forces in which each storey is modelled independently from the others (see figure (2.10)).

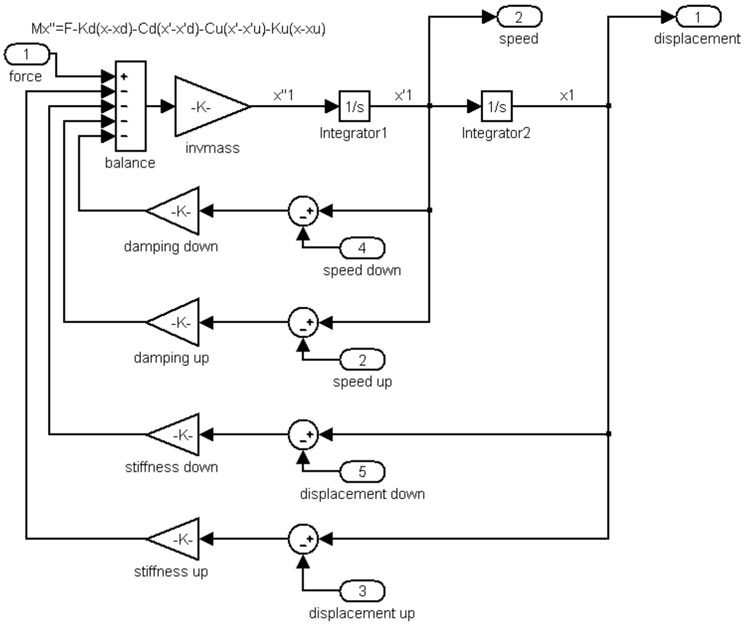


Figure 2.10: Simulink model of a single storey

The model of each storey forms a block with 5 inputs and two outputs. The five inputs are:

1. **force**: is the external force acting upon the storey. It can be an external force coming from wind, earthquake or other environmental loads or a control force caused by an actuator or the resultant of the sum of the two previous forces types;

2. **displacement up**: is the displacement of the upper storey. This signal, subtracted to the actual value of the displacement in the considered storey

and multiplied by the stiffness value of the columns connecting the actual storey with the upper one, enter in the force balance;

3. **displacement down**: is the displacement of the lower storey. Similarly to the previous case, it is subtracted to the actual value of the displacement in the considered storey and multiplied by the stiffness of the lower floor and enter in the force balance;

4. **speed up**: is the velocity of the upper storey. This signal, subtracted to the actual value of the velocity in the considered storey and multiplied by the damping value of the upper floor, enters in the force balance;

5. **speed down**: is the velocity of the lower storey. This signal also, subtracted to the actual value of the velocity in the considered storey and multiplied by the damping value of the lower floor, contributes to the force balance.

The two outputs are:

1. **displacement**: is the actual displacement of the storey;

2. **speed**: is the actual velocity of the storey.

A summation block subtract the damping and elastic forces coming from the upper and lower storey to the external force acting on the floor. The resulting signal is the inertial force of the storey. Dividing this signal by the mass value of the floor, the acceleration can be computed. The acceleration can be integrated once to have velocity or twice to have displacement.

With this basic block describing the behaviour of a storey, it is relatively easy to connect an arbitrary number of blocks to model a multi-storeys building. Once the global model of the building has been set up, the involved parameters (the stiffness and the damping of the upper and lower floor, the inverse value of the mass of the actual storey) for each floor must be set to the right values: this can be immediately done with an automatic procedure in MATLAB.

In this case study, five blocks were used to model the five storeys of the building. The resulting model can be seen in figure (2.11). At a first look it seems a complex model, but in reality it is very simple, it is only a matter of connecting the right outputs with the right inputs. Some scopes were added to the outputs, jointly with some blocks for exporting variable into the MATLAB workspace in order to be able to analyse the data after the simulation. The force signal, coming inside each storey block, is the sum of the earthquake excitation force and the control force given by an actuator. The response of the structure can be computed in the controlled or uncontrolled case simply by connecting or disconnecting the input. The earthquake signal comes directly from MATLAB workspace, so it is very easy to change and to perform simulations with different acceleration records.

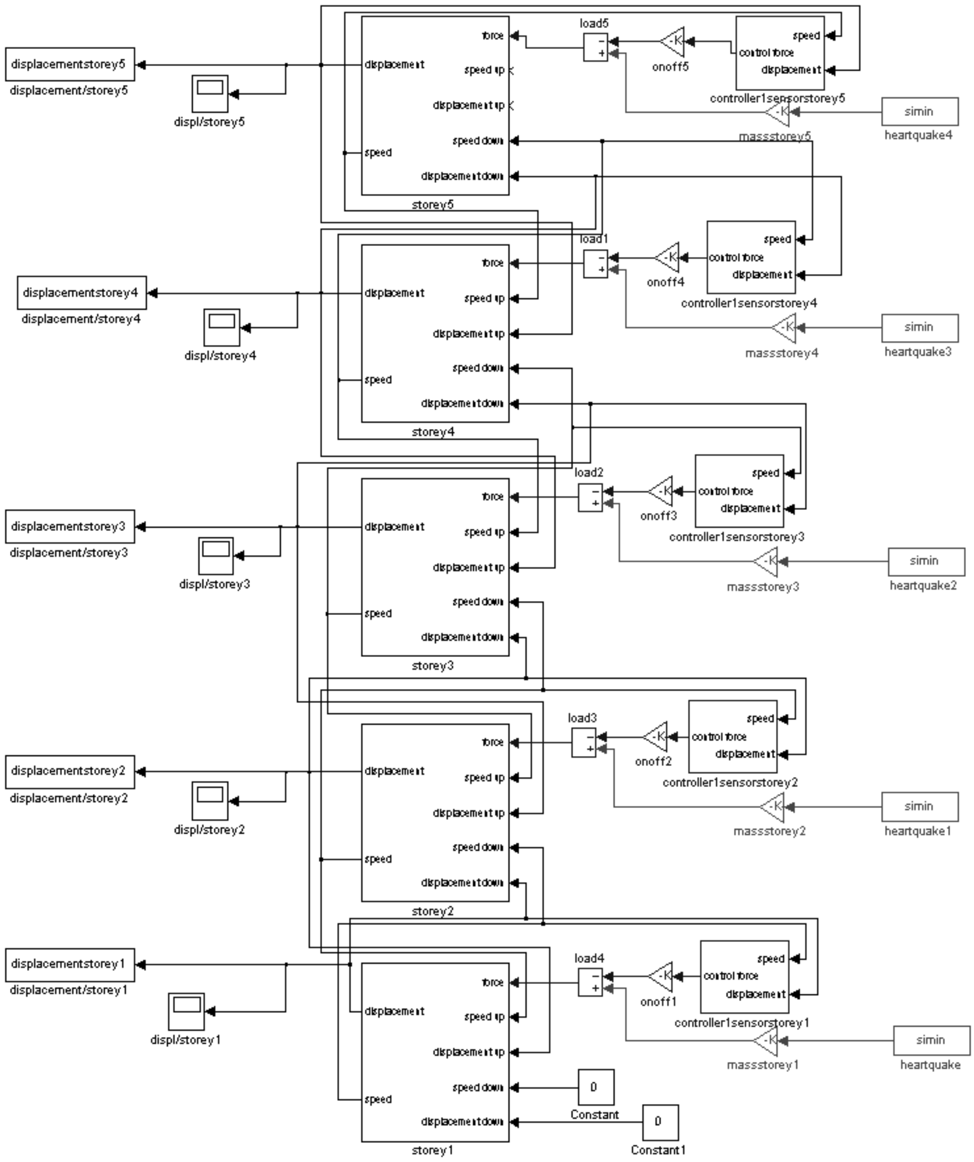


Figure 2.11: Simulink model of a 5 storey building

Regarding the first storey, it has only the displacement and the velocity coming from the upper floor, while the corresponding input from the lower floor can be left either free or set to a constant 0 value. The same observation refers to the last floor.

Coming to the actuators, two models can be taken into account:

- a *one sensor* actuator model (see figure (2.12)) with two inputs (the displacement and the velocity of the floor where actuator is acting) and one output (the force the actuator must apply on the storey);
- a *three sensors* actuator model (see figure (2.13)) with six inputs (the displacement and the velocity of the floor where actuator is acting plus the displacements and the velocities of the upper and lower storeys) and one output (the force the actuator must apply on the storey).

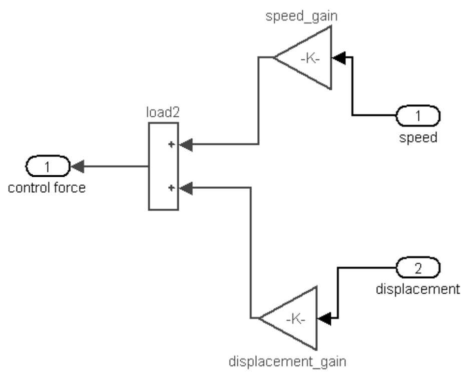


Figure 2.12: Simulink model of a one sensor actuator

As already mention in the previous sections, the failure of a sensor is critical if it refers to the controlled storey, while performance are not so badly affected if the failure refers to a sensor far away.

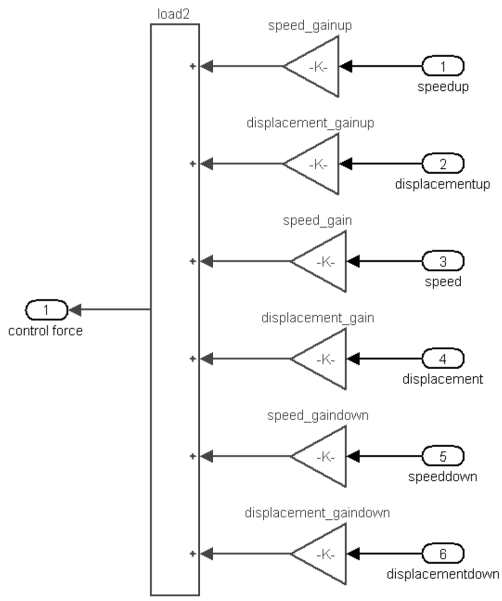


Figure 2.13: Simulink model of a three sensors actuator

Chapter 3

Semi-active control strategies and implementation schemes

3.1 Introduction

3.1.1 The idea

As mentioned in § (1.2), the basic idea of semi-active control is very simple: to change "on line" the characteristics of a passive dissipation device. This requires a minimum amount of energy to turn the mechanical component devoted to the changing behaviour of the system (a valve, for example, or a bolt friction connection). The main advantage is to joint the simplicity and reliability of a passive devices with the adaptability of the active systems.

3.1.2 The history

The concept of structural control was firstly introduced by (Yao 1972). Some years later the semi-active control concept was introduced for the first time by (Karnopp et al. 1974) that proposed to modify the force of a fluid damper controlling the opening of a valve. He had in mind automotive applications, so his target was to obtain a better isolation of the vehicle from the road roughness.

The first proposal for semi-active control of civil structures can be found in (Hrovat et al. 1993). In that work the concept of semi-active control was extended to civil buildings proposing a Tuned Mass Damper that was connected to the main structure with a semi-active viscous damper. The proposed device was a variable orifice damper.

Karnopp idea was based on an antilock braking system (ABS), which, of course, is a technology closely related to the same automotive field. In the case of ABS braking, the main concern is the avoidance of sticking in a frictional interface while, in the case of semi-active damping, the main aim is to dissipate energy as quickly as possible. The two objectives are different but they are very much interconnected because both deal with the problem of allowing a relative movement of the two parts. A sticking interface cannot dissipate energy, in fact. The optimal friction value is not constant, so the best device can adapt itself to these changes.

Since that time, a lot of studies were conducted both from control strategies and from implemented devices point of view. New hardware and software capabilities allow the design of more sophisticated control laws, while, on the other side, new materials such as magnetorheological liquids are now available to permit proper devices design. In the following sections the main control strategies present in literature will be described, while the most promising devices are illustrated in appendix (A).

3.2 Control strategies and algorithms for semi-active damping

Several approaches were proposed in literature (Dyke et al. 1996; Dyke and Spencer 1997; Carter 1998; François et al. 2000), some coming from an adaptation of active control laws, other ones coming from physical considerations.

With a semi-active control device, energy can only be dissipated (and so exerted from the system)[1]. This kind of devices could be seen as passive dampers with changing characteristics to be adjusted on line. For this reason, they cannot lead to an instable system, even in the worst case. Some of these algorithms will be explained hereafter.

3.2.1 Open loop control

In the open loop mode of operation no feedback in necessary. The control law is set a priori and no knowledge of the state variable is necessary. The damping characteristic of the devices can vary with continuity or by steps, depending on the operating conditions. Typically, the variable dampers work in a bi-state (on-off) manner. This means that the device can be considered as a common passive element in which the dissipative properties can be switch from one value to another. This is the simplest and cheapest way to implement a control law and can be used very usefully for vibration isolation of rotating machines. Such technique is used,

[1] In principle, the energy can be stored and used for some specific purpose, for example to give the necessary current to the controller for operate (Battaini et al. 1998; Battaini and Marioni 2001) or can even be pumped into the electric net. This kind of application, however, are still not state of the art (they are nowadays under development) and will be treated in the following chapters.

for example, for washing machines: one value of damping (high) is used when the drum is at low speed (i.e. during acceleration or deceleration), while a lower damping value is used for high speed (figure (3.1)). An automatic switch changes the damper's properties when a pre-set speed is reached.

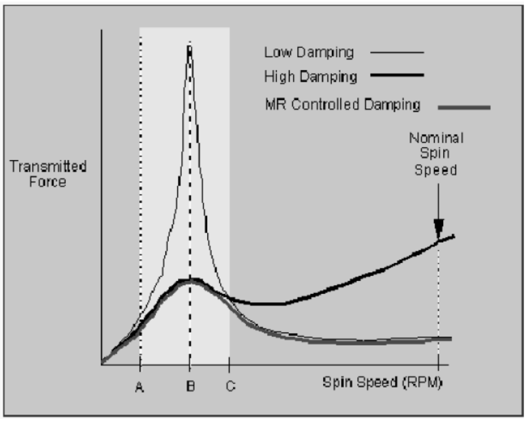


Figure 3.1: On-off open loop strategy

This type of control is useful when the system to be damped has very well known dynamic characteristics and in its loading conditions.

In some cases the different level of damping is chosen by the user, like in the case of semi-active suspension of vehicles.

3.2.2 On-off skyhook control

In on-off skyhook control, the damper is controlled by two damping values. This control law was developed in order to have the lower vibration amplitude on the body mass. Illustrated in (3.2), these are referred to as high-state and low-state damping.

The choice between a damper adjusted to its high state or to low state is made by the following control law. Depending on the product of the relative velocity v_{rel} across the suspension damper (figure (3.3)) and the absolute velocity $\dot{z}_b$ of the system body mass attached to that damper, a damping value is chosen. If the product is positive or zero, the damping value c_s of the damper device is adjusted to its high state; otherwise, c_s is set to the low state.

This concept is summarized by:

$$\begin{array}{ll} \dot{z}_b \times v_{rel} \geq 0 & c_s = high\ state \\ \dot{z}_b \times v_{rel} < 0 & c_s = low\ state \end{array} \tag{3.2.1}$$

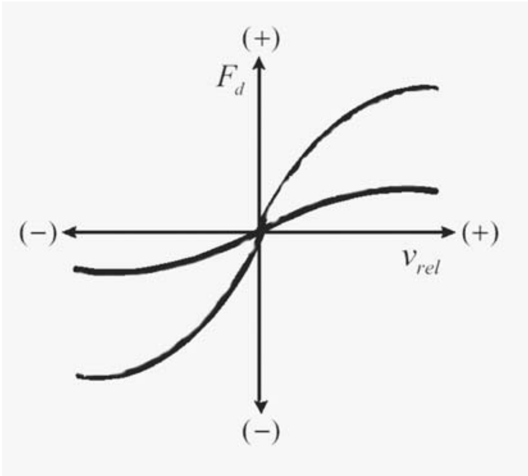


Figure 3.2: Typical semi-active damper curves

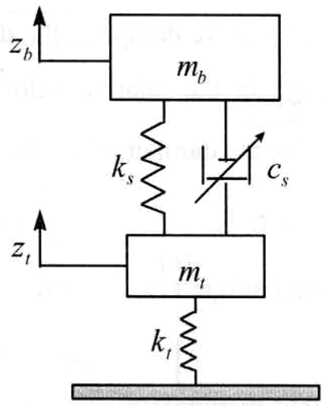


Figure 3.3: Suspension system model equipped with a variable damping device

where $v_{rel} = \dot{z}_b - \dot{z}_t$. The logic of the on-off skyhook control policy is as follows. When the relative velocity of the damper is positive, the force of the damper acts to pull down on the system body mass; when the relative velocity is negative, the force of the damper pushes up the mass. Thus, when the absolute velocity of the body mass is negative, it is travelling downwards and the maximum (high state) value of damping is required to push up the mass, while the minimum (low state) value of damping is required to continue pulling down on the mass.

However, if the absolute velocity of the body mass is positive and the mass is travelling upwards, the maximum (high state) damping value is required to pull down the mass, while the minimum (low state) damping value is required to further push the mass upwards (figure (3.4)).

3.2.2.1 Physical interpretation

In order to clarify the above-mentioned concepts, a physical interpretation of the equations (3.2.1) is given hereafter.

Without loosing in generality, $\dot{z}_b$ and $\dot{z}_t$ can be considered positive if the body and tire masses are moving upwards. It can be also observed that $v_{rel} < 0$ when the tire and body masses are approaching, while if $v_{rel} > 0$ the two masses are going away from each other.

The following four cases can so be found:

1. the two mass are moving away from each other and the tire mass is going upwards (product is positive): high damping is required in order to try to keep the body mass down;
2. the two mass are moving away from each other and the tire mass is going downwards (product is negative): low damping is required in order to try to reduce the pulling effect of the tire mass moving downwards. If the damping is high, the body mass would move downwards faster, in this case;
3. the two mass are approaching and the tire mass is going downwards (product is positive): high damping in order to keep the two masses as far as possible;
4. the two mass are approaching and the tire mass is going upwards (product is negative): low damping.

Concluding, a high damping value is used only while needed, the lowest possible damping value is used when damping is not needed.

The on-off skyhook semi-active policy emulates the ideal body displacement control configuration of a passive damper "hooked" between the body mass and the "sky," as shown in figure (3.5).

3.2.3 Continuous skyhook control

In continuous damping, a "high" and a "low" damping state can be defined as in the on-off damping control policy described previously. However, now, the damping values are not limited to these two states alone; they may exist at any value within the two states. As illustrated in figure (3.6), the high and low states serve

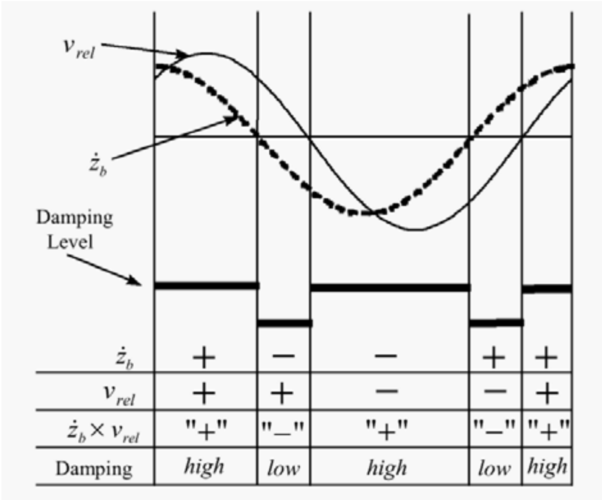


Figure 3.4: Semi-active damping with continuous skyhook control

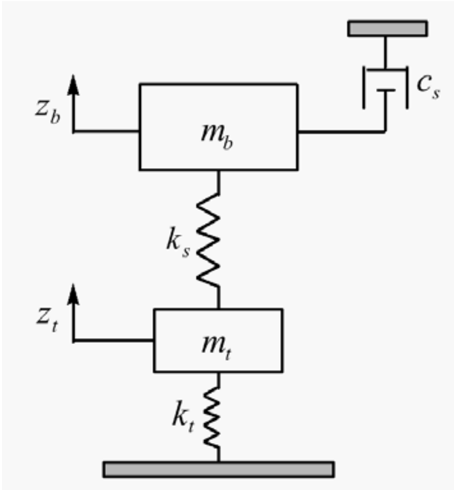


Figure 3.5: Passive damping representation of skyhook control

as the maximum and minimum damping values, with the intermediate (shaded) area as all possible damping values between the maximum and minimum.

Equations used for on-off skyhook control still apply, except for the definition of the high-state and low-state damping. In on-off skyhook control, the high and low

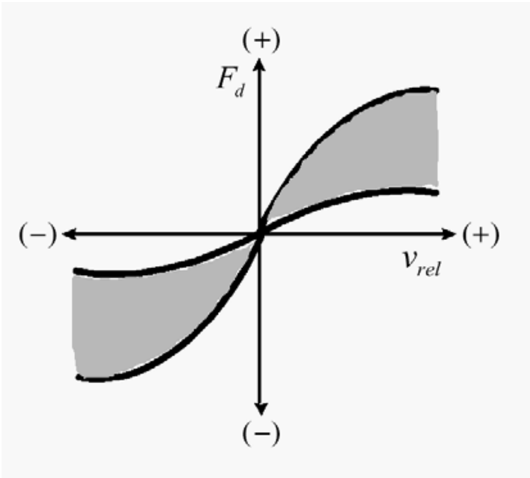


Figure 3.6: Semi-active damping with continuous skyhook control

states were defined as constant damping values. In continuous skyhook control, the low state remains defined by a constant damping value, while the high state is set equal to a constant gain value g multiplied by the absolute velocity of the system body attached to the damper, not to exceed the corresponding high and low state limits.

$$\begin{array}{ll} \dot{z}_b \times v_{rel} \geq 0 & c_s = \alpha \\ \dot{z}_b \times v_{rel} < 0 & c_s = \textit{low state} \end{array} \tag{3.2.2}$$

with $\alpha = \max\{\textit{low state}, \min[(g \times \dot{z}_b, \textit{high state}]\}$.

3.2.4 On-off groundhook control

In this case the determination of whether the damper is to be adjusted to its high state or its low state depends on the product of the relative velocity across the suspension damper and the absolute velocity of the vehicle tire mass attached to that damper.

Contrary to the skyhook control strategy, this control law was developed in order to have the lower vibration amplitude on the tire mass. The control law is the following:

- if the product of the relative damper and absolute tire velocities is negative or zero, the damper is adjusted to its high state;
- if this product is positive, the damper is adjusted to its low state.

Putting this into formulas it comes:

$$\begin{array}{ll} \dot{z}_b \times v_{rel} \leq 0 & c_s = high\ state \\ \dot{z}_b \times v_{rel} > 0 & c_s = low\ state \end{array} \tag{3.2.3}$$

The reasoning of the on-off groundhook control policy is similar to the on-off skyhook control policy, except that control is based on the unsprung mass. When the relative velocity of the damper is positive, the force of the damper acts to pull up on the tire mass; when the relative velocity is negative, the force of the damper pushes down on the tire mass. However, when the absolute velocity of the tire mass is negative, it is travelling downwards and the maximum (high state) value of damping is required to pull the mass, while the minimum (low state) value of damping is required to pushing down on the mass. But, if the absolute velocity of the tire mass is positive and the mass is travelling upwards, the maximum (high state) value of damping is required to push down the mass, while the minimum (low state) value of damping is required to further pull the mass upwards. The on-off groundhook semi-active policy emulates the ideal tire displacement control configuration of a passive damper "hooked" between the tire and the "ground," as shown in figure (3.7).

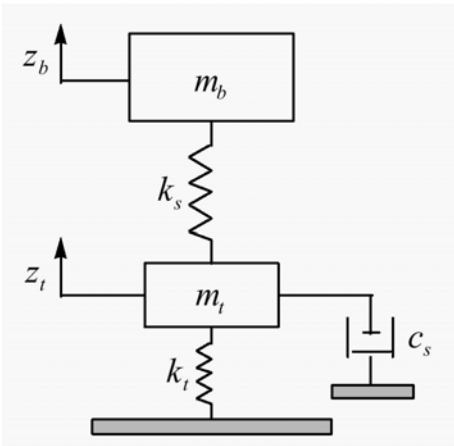


Figure 3.7: Passive damping representation of groundhook control

The continuous groundhook control strategy can be derived directly from the continuous skyhook control simply by changing the chosen condition on the damping value, so it will be omitted.

3.2.5 Clipping control

Under this denomination a class of controls is grouped. They common characteristic is the fact that these laws have a two stages architecture, i.e. the controller design can be divided in two parts.

The first step consists in designing an active control law assuming that an ideal active device is present. The second step design a clipping controller allowing the semi-active damper to develop the force that the active device would have exerted on the structure.

This means that, for the first step, every kind of control law design can be chosen (optimal control, integral force feedback control, H_2 or H_∞ control) because the second step is independent from it. To clip the active control law to a semi-active one the following rule is usually used: when the magnitude of the force F_d produced by the damper (that is the control force f in this case) is smaller than the required target force f_c and the two forces have the same sign, the voltage applied to the current driver is increased to the maximum level, so as to match the required control force; otherwise, the command voltage is set to zero. This strategy is usually called "clipped on-off" and with a formula appears as:

$$\nu = V_{\max} H[(f_c - f)f] \tag{3.2.4}$$

where ν is the command signal[2], $H[.]$ is the Heaviside step function, V_{max} is the maximum voltage applicable on the semi-active device to obtain the maximum damping and f and f_c are the measured and required control forces. Figure (3.8) illustrates graphically this strategy.

3.2.6 Direct Lyapunov control

This approach requires the use of a Lyapunov function $V(\mathbf{x})$, which must be compatible with the following conditions:

i. be a positive definite function of the states of the system, $\mathbf{x}$;

ii. have a negative time derivative for all trajectories from any initial state in the neighbourhood of the origin.

$$V(\mathbf{x}) = \frac{1}{2}\mathbf{x}^T\mathbf{P}\mathbf{x} \tag{3.2.5}$$

where $\mathbf{P}$ is a real, symmetric, positive definite matrix, found by solving the Lyapunov equation[3]

[2]This command signal can act on the excitation coil of a magnetorheological devices, on the positioning controller of an hydraulic device or the position of a valve closing an orifice.

[3]In the case of a linear system with state matrix $\mathbf{A}$, see also (1.7.1).

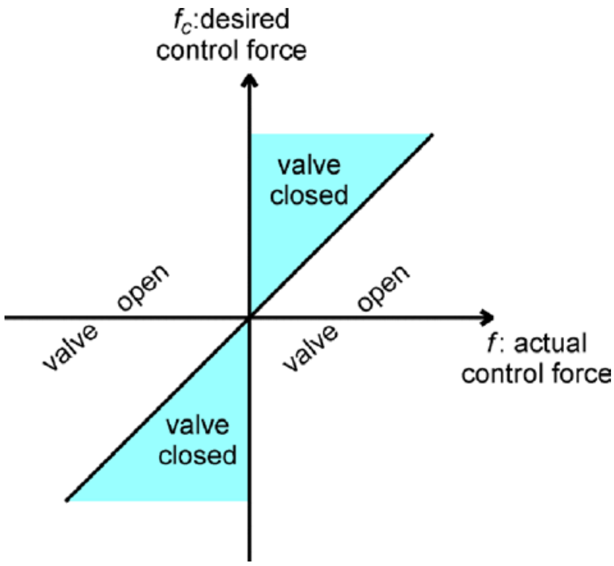


Figure 3.8: Clipping control strategy

$$\mathbf{A}^T\mathbf{P} + \mathbf{PA} = -\mathbf{Q} \tag{3.2.6}$$

with a positive definite $\mathbf{Q}$ matrix (ensuring $\dot{V}(\mathbf{x}) = -\frac{1}{2}\mathbf{x}^T\mathbf{Q}\mathbf{x} < 0$).
For a linear system with control forces $\mathbf{f}_{cf}$

$$\dot{\mathbf{x}} = \mathbf{Ax} + \mathbf{Bf}_{cf} \tag{3.2.7}$$

the derivative of the Lyapunov function becomes

$$\dot{V}(\mathbf{x}) = -\frac{1}{2}\mathbf{x}^T\mathbf{Q}\mathbf{x} + \mathbf{x}^T\mathbf{PBf}_{cf} \tag{3.2.8}$$

The second term containing $\mathbf{f}_{cf}$ can be directly affected by a change in the control voltage. The control law that will minimise $\dot{V}$ is

$$\nu_i = V_{\max}H(-\mathbf{x}^T\mathbf{P}B_i f_{cf_i}) \tag{3.2.9}$$

where $H(.)$ is the Heaviside step function, f_{cf_i} is the measured force produced by the i^{th} semi-active device and B_i is the corresponding column of $\mathbf{B}$.

3.2.7 Fuzzy logic control

Fuzzy logic control of semi-active dampers is another example of continuous control. The output of the controller determined by the fuzzy logic may exist anywhere between the high and low damper states. Fuzzy logic is used in a number of controllers because it does not require an accurate model of the system to be controlled. Fuzzy logic works by executing rules that correlate the controller inputs with the required outputs. These rules are typically created through the intuition or knowledge of the designer regarding the operation of the system being controlled. No matter what the system is, there are three basic steps that are characteristic to all fuzzy logic controllers. These steps include the fuzzification of the controller inputs, the execution of the rules of the controller, and the defuzzification of the output to a crisp value to be implemented by the controller. These steps can be briefly explained as follows:

- step one: fuzzification. This step is accomplished through the construction of a membership function for each of the inputs. This is the main point, because the possible shapes of these functions are infinite, though very often a triangular or trapezoidal-shape is used. Once chosen, the membership functions the input, read as a crisp value, is transformed into a fuzzy value by intersect each component of the membership function with this value. This must be done for all inputs of the controller.

- step two: execution of the rules. In order to create the rule-base of the controller, the membership function of the output must first be defined. Once these functions had been defined, a table of combination can be stated. In this table for each possible combination of two inputs a linguistic value of output is defined. These rules can be described as a series of “IF-THEN” statements.

- step three: defuzzification. These fuzzy output now go through the defuzzification process in which a single, or crisp, controller output value is obtained. Some common methods of defuzzification include the max or mean-max membership principles, the centroid method, and the weighted average method. To give an example, the weighted average method is described in the implementation paragraph. In this case the crisp output value is obtained as the sum of the product of each weighting function (to be chosen arbitrarily) with the maximum value of its respective membership value and dividing it by the sum of the weighting functions.

These three steps must be repeated for each input point to obtain continuous outputs.

3.2.8 Modulated homogeneous friction control

This control strategy was developed originally for a variable friction damper. In this approach, at every occurrence of local extremes in the deformation of the device (i.e. when the relative velocity between the ends of the semi-active device is zero), the normal force applied to the frictional interface is updated to a new value. At each local minimum or maximum in the deformation the normal force $N(t)$ is chosen to be proportional to the absolute value of the semi-active device deformation. The control law is written as:

$$N(t) = g\,|P[\Delta(t)]| \tag{3.2.10}$$

where g is a positive gain and the operator $P[.]$ (referred to the prior-local-peak operator) is defined as $P[\Delta(t)] = \Delta(t-s)$ where $s = \{\min x \geq 0 : \dot{\Delta}(t-x) = 0\}$ defining $\Delta(t-s)$ as the most recent local extreme in the deformation[4].

Because this algorithm was developed for variable friction devices, the following modifications are needed when applying it to other kind of semi-active devices (as, for example, magnetorheological or variable orifice devices):

1. there is often no need to check if the force is greater than the static friction, because some semi-active devices has no static friction;

2. a force feedback loop is used to induce the semi-active damper to produce approximately the frictional force corresponding to the required normal force. Thus, the goal is to generate a required control force with a magnitude

$$f_c = \mu g\,|P[\Delta(t)]| = g_n\,|P[\Delta(t)]| \tag{3.2.11}$$

   where the constant g_n has unit of stiffness.
   The resulting control law is

$$\nu = V_{\max} H(f_c - |f|) \tag{3.2.12}$$

where $H[.]$ is the Heaviside step function. An appropriate choice of g_n will keep the force f_c within the operating envelope of the semi-active damper most of the time, allowing the device force to closely approximate the required force.

3.2.9 Bang-bang control

Bang-bang control is used mainly for dissipative devices in order to increase their capabilities in dissipating energy with respect to a classical passive device. It provides a simple and yet often effective approach.

[4]This definition is quite ambiguous, as it will be explained in the implementation paragraph.

When the relative displacement and the relative velocity of the two ends of the damping device are in the same direction, bang-bang control act in the direction of increasing to a maximum value the friction forces into the device. In this way it works as a brake, thus allowing dissipating energy. On the other hand, when the relative displacement and the relative velocity of the ends of the device are in opposite direction, this control law decreases the friction forces to a minimum in order to make the device movement as easy as possible.

If the maximum value of the friction forces is obtained when the control signal is u_{max} and the minimum one when the control signal is u_{min}, the control law can be written as follows:

$$u(t) = u_{max} \qquad \text{if} \quad \text{sign}(x) = -\ \text{sign}(\dot{x}) \tag{3.2.13a}$$

$$u(t) = u_{min} \qquad \text{if} \quad \text{sign}(x) = \ \text{sign}\ (\dot{x}) \tag{3.2.13b}$$

The control parameter u_{min} should be set at a level as little as possible, to reduce the dissipative forces to a minimum, making the device telescope as much as possible. The control parameter u_{max} should be at the optimal value, i.e. a value which provides the maximum energy dissipation.

3.2.10 Instantaneous Optimal Control

In the instantaneous optimal control strategy the command signal $u(t)$ is determined by minimising the following time dependent objective function $J(t)$ at every time instant t for the entire duration of the excitation:

$$J(t) = q_d x^2(t) + q_f f^2(t) + r u^2(t) \tag{3.2.14}$$

in which x is the relative displacement. The normalised friction force f indirectly represents the amount of response acceleration and also serves as a measure of the transfer of induced force to the structure. The weighting coefficient q_d and q_f are non-negative and r is positive. They indicate the relative importance in the control objectives of relative displacement, response acceleration and control signal, respectively. The basic objective of the control is provide a device that dissipate the biggest amount of energy within an acceptable range and at the same time to minimise the transferred force.

Without entering into details, an explicit Newmark method can be used for implementation to solve numerically the involved equations.

3.3 Implementation schemes

3.3.1 Open loop control

This control strategy is very easy to implement, because no states knowledge is needed. In the case of a washing machine, for example, the control law can be implemented following these steps:

1. when the machine begins to spin-dry, the damping value is set to the maximum one and a time counter is reset to zero;

2. when the time counter reaches a pre-fixed value, i.e. after a fixed time, the damping value is switched to the minimum one.

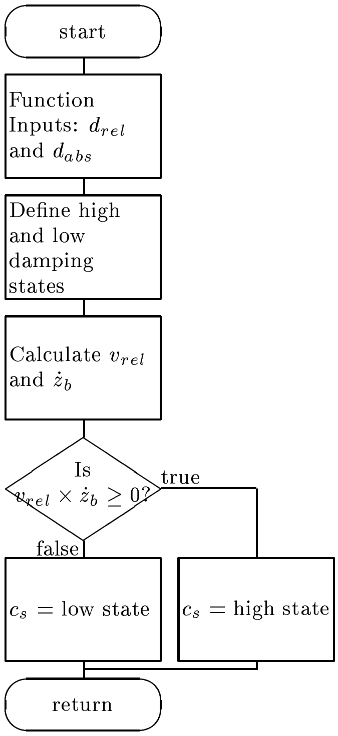


Figure 3.9: Flowchart of the on-off skyhook subroutine

The same procedure can be applied when the washing machine finishes the spin-dying phase and decreases the drum speed.

3.3.2 On-off skyhook control

The flowchart summarising the subroutine applying the on-off skyhook policy of damping control is shown in figure (3.9).

It comes immediately evident that to implement this control strategy is very simple: it needs only to measure two quantities (d_{rel} and d_{abs}) and then to act on some device switching the dissipator from one state to another. High and low damping are those given by the dissipation device when, for example, a valve is completely open or closed, so they cannot be modify once the device has been designed. The only change that the control law can induce into the device is to switch from one value to another.

In algorithm (1) the quantity $\xi_l \neq 0$ and $\xi_h \neq 0$ are respectively the low damping state of the device and the high damping state. With the Lord© magnetorheological device, for example, the first state is reached when the supply current is $1A$, while the second one is achieved for $0A$ current. The quantity Δt is the time step of acquisition. The algorithm checks at the beginning that the needed parameter are meaningful, then execute the procedure.

Algorithm 1 On-Off Skyhook Control Strategy

Require: $\Delta t > 0$, $\xi_h \neq 0$, $\xi_l \neq 0$ and $\xi_h \geq \xi_l$
  $d^v_{rel} \Leftarrow \bar{d}_{rel}$
  $d^v_{abs} \Leftarrow \bar{d}_{abs}$
  repeat
    acquire d^n_{rel}
    acquire d^n_{abs}
    $v_{rel} \Leftarrow (d^n_{rel} - d^v_{rel})/\Delta t$
    $\dot{z}_b \Leftarrow (d^n_{abs} - d^v_{abs})/\Delta t$
    if $v_{rel} \times \dot{z}_b \geq 0$ **then**
      $c_s \Leftarrow \xi_h$
    else
      $c_s \Leftarrow \xi_l$
    end if
    $d^v_{rel} \Leftarrow d^n_{rel}$
    $d^v_{abs} \Leftarrow d^n_{abs}$
  until stop condition

In the algorithm, $\bar{d}_{rel}$ and $\bar{d}_{abs}$ are the starting values for d_{rel} and d_{abs}, the apex ^{n} and ^{v} stay for ^{new} and ^{old}: the old values came from the previous calculation, while the new values are the just calculated ones.

It must be noticed that this procedure is reduced at bone: in real implementations it is necessary to take into account also the fact that, under some conditions, for example when $v_{rel} \times \dot{z}_b \approx 0$, the control action can continuously switch from the first to the second state. This can be energy consuming and can also lead to the damage of the device. To avoid this effect it is necessary to insert a relay behaviour: the switching command doesn't trigger exactly when the product $v_{rel} \times \dot{z}_b$ becomes zero, but a slightly later, so avoiding an oscillatory switching if one of the two quantities v_{rel} or $\dot{z}_b$ is moving around the zero.

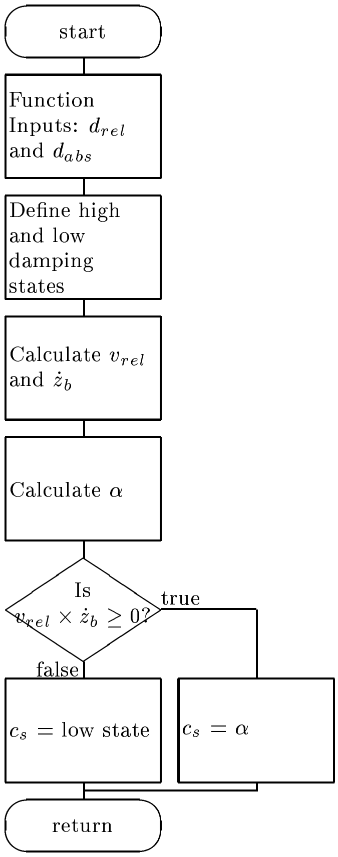


Figure 3.10: Flowchart of the continuous skyhook subroutine

3.3.3 Continuous skyhook control

The flowchart summarising the subroutine applying the continuous skyhook policy of damping control is shown in figure (3.10).

It must be notice that, like in figure (3.9), also figure (3.10) represents one single step of control: the continuous skyhook subroutine is called at each calculation step. This fact can lead to some optimisations, as it is shown in the corresponding algorithm: for example, it is not necessary to define high and low damping states at each step, but define them once for all when the semi-active control algorithm is started.

The continuous skyhook algorithm implementation is much more complex than the on-off one at least for three main reasons:

1. the coefficient α must be calculated at each step;

2. the correlation between the α coefficient and the command signal to the semi-active device could be highly non-linear;

3. the semi-active device must be complex enough to allow little modifications of its properties.

The first point is not very important if the calculation power is high enough: in this case the algorithm will be very fast even with some more complex calculation.

The second point can be also solved if a characterisation testing campaign is conducted on the semi-active device. In this way it is possible to define a calibration curve correlating, for example, the required value of damping with the necessary amount of voltage needed.

The last point means that only some special kind of device can perform this control strategy, as for example the MR devices, because response time is only some milliseconds. In case of mechanical arrangements, it is much easier to switch between two values (for example clamped and unclamped) than to pass through all the intermediate states.

The flow-chart of figure (3.10) is then translated into algorithm (2), where Δt, ξ_h, ξ_l, d_{rel}, d_{abs} have the same meaning than before, while g is a fixed gain chosen at the beginning of the algorithm.

3.3.4 On-off groundhook control

The flowchart summarising the subroutine applying the on-off groundhook policy is shown in figure (3.11). It looks very similar to figure (3.9), except for the chosen condition on the device damping value.

The flowchart of figure (3.11) is then translated into algorithm (3).

Algorithm 2 Continuous Skyhook Control Strategy

Require: $\Delta t > 0$, $\xi_h \neq 0$, $\xi_l \neq 0$, $\xi_h \geq \xi_l$ and g
  $d^v_{rel} \Leftarrow \bar{d}_{rel}$
  $d^v_{abs} \Leftarrow \bar{d}_{abs}$
  repeat
    acquire d^n_{rel}
    acquire d^n_{abs}
    $v_{rel} \Leftarrow (d^n_{rel} - d^v_{rel})/\Delta t$
    $\dot{z}_b \Leftarrow (d^n_{abs} - d^v_{abs})/\Delta t$
    if $v_{rel} \times \dot{z}_b \geq 0$ **then**
      $\alpha \Leftarrow \max\{\xi_l, \min[(g \times \dot{z}_b, \xi_h]\}$
      $c_s \Leftarrow \alpha$
    else
      $c_s \Leftarrow \xi_l$
    end if
    $d^v_{rel} \Leftarrow d^n_{rel}$
    $d^v_{abs} \Leftarrow d^n_{abs}$
  until stop condition

Algorithm 3 On-Off Goundhook Control Strategy

Require: $\Delta t > 0$, $\xi_h \neq 0$, $\xi_l \neq 0$ and $\xi_h \geq \xi_l$
  $d^v_{rel} \Leftarrow \bar{d}_{rel}$
  $d^v_{abs} \Leftarrow \bar{d}_{abs}$
  repeat
    acquire d^n_{rel}
    acquire d^n_{abs}
    $v_{rel} \Leftarrow (d^n_{rel} - d^v_{rel})/\Delta t$
    $\dot{z}_b \Leftarrow (d^n_{abs} - d^v_{abs})/\Delta t$
    if $v_{rel} \times \dot{z}_b \leq 0$ **then**
      $c_s \Leftarrow \xi_h$
    else
      $c_s \Leftarrow \xi_l$
    end if
    $d^v_{rel} \Leftarrow d^n_{rel}$
    $d^v_{abs} \Leftarrow d^n_{abs}$
  until stop condition

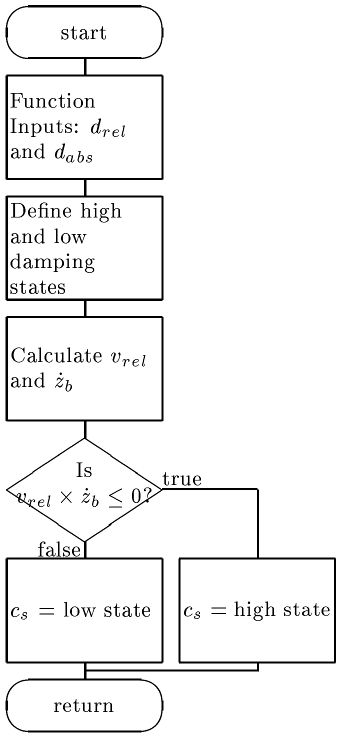


Figure 3.11: Flowchart of the on-off groundhook subroutine

3.3.5 Clipping control

From an implementation point of view, this control strategy seams to be the most direct one because it can take advantage of the great amount of experimental and practical studies that had been conducted on active control strategies. The clipping control can be viewed, in fact, as a control strategy in which the actuator can operate only resisting forces and not acting directly on the structure. So the active control law applies when the device is subjected to forces and turns into a constant value when it should act. The corresponding flowchart is shown in figure (3.12), where ν is that defined in (3.2.4).

Only some little modifications can lead to a continuous clipping control strategy. When the active control law would require a negative damping (i.e. it would require that the device acts on the structure) there is nothing better to do, i.e. set the device at its minimal resistance force. For the grey region on figure (3.8) a

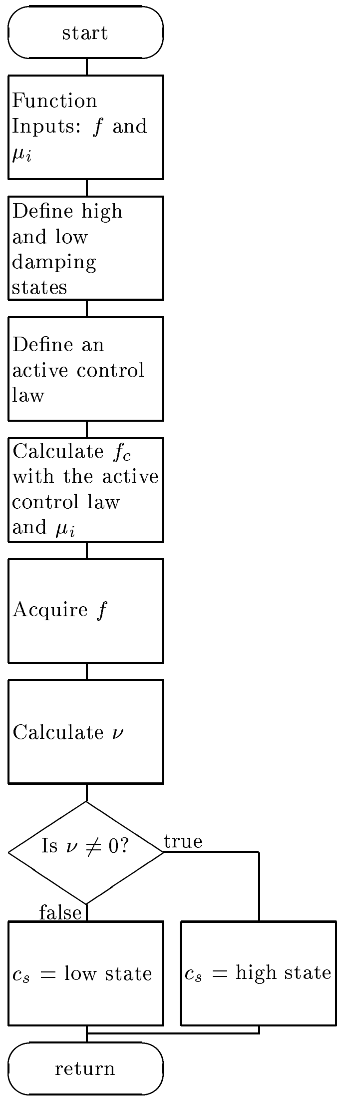


Figure 3.12: Flowchart of clipping control strategy subroutine

continuous value of the damping coefficient can be achieved if a proportional valve is actuated on the semi-active device or if a magnetorheological fluid is subjected to a magnetic field proportional to the active law required force. A control must be designed in this case to manage saturation on the semi-active device: a too high value of command voltage to the magnetic coil can lead to damages.

The flowchart is then detailed into algorithm (4) for both the on-off and the continuous clipping control strategy.

The clipping control strategy is more a class of strategies than a specific technique because every type of active control law can be used. For this reason it is very difficult to compare it with the other above-mentioned approach. The computing time required is just a little more than the one corresponding to the active implementation, because only some more "IF-THEN" statements must be evaluated. A general critic to this class of methods, however, is that there is no way of assessing if a good active control law can give good results also when turned into a clipped one. Because of the inherent nonlinearity, the clipping strategy cannot be assessed with a frequency domain technique, but time domain analysis must always be performed with several types of excitations and with different intensity levels (see appendix (D), however). The Heaviside Step Function is implemented in algorithm (5).

Algorithm 4 Clipping Control Strategy

Require: V_{max}, f_{max}, $\xi_h \neq 0$, $\xi_l \neq 0$ and $\xi_h \geq \xi_l$
Require: an active control law with the corresponding measured variables μ_i
Require: the Heaviside Step Function $H[.]$

```
repeat
  acquire f
  acquire μ_i
  calculate f_c accordingly with the active control law and μ_i
  ν ⇐ V_max H[(f_c − f)f]
  if ν ≠ 0 then
    if on-off control then
      c_s ⇐ ξ_h
    else
      if f_c ≥ f_max then
        c_s ⇐ ξ_h
      else
        c_s proportional to f_c
      end if
    end if
  else
    c_s ⇐ ξ_l
  end if
until stop condition
```

Algorithm 5 Heaviside Step Function

Require: f_1, f_2
  if $(f_1 - f_2)f_2 > 0$ **then**
    $H[.] \Leftarrow 1$
  else
    $H[.] \Leftarrow 0$
  end if

3.3.6 Direct Lyapunov control

As mentioned before, the Heaviside Step Function is very easy to implement (see algorithm (5)). The real task for implementing the Direct Lyapunov control method is then to obtain all the values that must be feed into the Heaviside Step Function. In fact:

- $\mathbf{x}$ must be known at each step: enough sensors must be placed on the structure to be controlled;
- $\mathbf{P}$ is calculated once for all at the beginning: Lyapunov equation must be solve once only, even if this is usually not easy;
- B_i is constant for a given device;
- f_{cf_i} must be measured at each step.

The most critical value to be obtained is the matrix $\mathbf{P}$, because the effectiveness of the final algorithm depends on it.

3.3.7 Fuzzy logic control

The flowchart summarising the subroutine applying the fuzzy logic policy to a damping control is shown in figure (3.13).

In more details it is translated into algorithm (6). At each step, a fuzzification-defazzification takes place. This may result in a long calculation time, so researchers have placed great care to the design of dedicated chip for fast online calculation (Faravelli and Rossi 2002).

In figure (3.14(a)) there is an example of Input Membership Function with the three linguistic variables N (acronym for Negative), P (Positive) and Z (Zero). Figure (3.14(b)) shows an Example of Output Membership Function where the linguistic output variables are defined as follows[5]: S (Small), MS (Medium Small), M (Medium), ML (Medium Large), L (Large). Obviously, the Rule Table can

[5]This definition is similar to that given for clothes size: XXXS $\leftarrow$ XS, S, M, L, XL $\rightarrow$ XXXL.

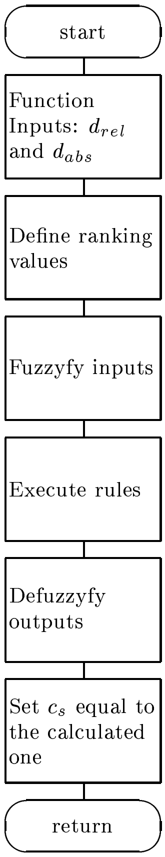


Figure 3.13: Flowchart of the fuzzy logic subroutine

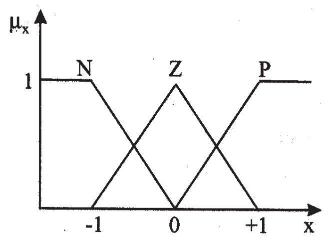
$x_2 \setminus x_1$	**N**	**Z**	**P**
N	S	MS	S
Z	M	M	M
P	L	ML	L

Table 3.1: Example of a Rule Table for a fuzzy controller

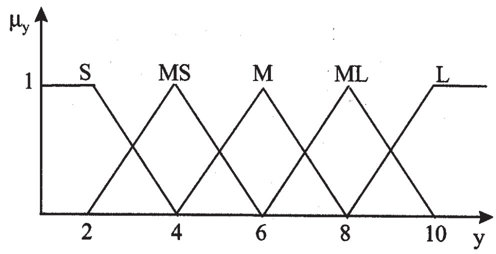
be described only after the Output Membership Function has been defined. An example of Rule Table is shown in table (3.1).

Algorithm 6 Fuzzy Logic Control Strategy

Require: $\Delta t > 0$, $c_s \neq 0$
Require: Input Membership Function, Rule Table, Output Membership Function
$d^v_{rel} \Leftarrow \bar{d}_{rel}$
$d^v_{abs} \Leftarrow \bar{d}_{abs}$
repeat
acquire d^n_{rel}
acquire d^n_{abs}
$v_{rel} \Leftarrow (d^n_{rel} - d^v_{rel})/\Delta t$
$\dot{z}_b \Leftarrow (d^n_{abs} - d^v_{abs})/\Delta t$
$\mathbf{v}^{fuzzy}_{rel} \Leftarrow$ intersect v_{rel} with Input Membership Function
$\dot{\mathbf{z}}^{fuzzy}_b \Leftarrow$ intersect $\dot{z}_b$ with Input Membership Function
$\mathbf{c}_s \Leftarrow$ execute the Rule Table for each inputs combination
combine the fuzzy values $\mathbf{c}_s$
$c_s \Leftarrow$ convert $\mathbf{c}_s$ to a crisp output value
$d^v_{rel} \Leftarrow d^n_{rel}$
$d^v_{abs} \Leftarrow d^n_{abs}$
until stop condition



(a) Input Membership Function



(b) Output Membership Function

Figure 3.14: Example of Input and Output Membership Function for a fuzzy controller

The main advantage of fuzzy control, i.e. the fact that it is based on verbal rules (and so very close to common sense practise), is also its main drawback. In fact it is impossible to obtain automatically the optimal solution or to check mathematically if the elaborated solution is stable or not. So simulations and experimental tests must be conducted to assess the control law.

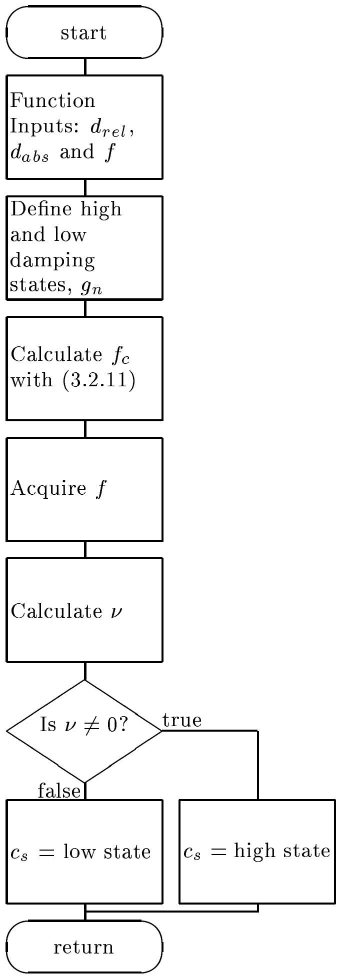


Figure 3.15: Flowchart of modulated homogeneous friction control strategy subroutine

3.3.8 Modulated homogeneous friction control

The implementation scheme is similar to the one described for the clipping control strategy. The main difference is that here the required control force is not calculated with an active control algorithm, but with equation (3.2.11).

The flowchart is shown in figure (3.15). The real point is to implement the operator $P[.]$ because its given definition is appropriated if the device behaviour is similar to a smooth "go and return" around the zero position: only in this case it makes sense to speak about "recent local extreme in the deformation". When deformation record is a very noisy signal, it must be filtered in order to have a more smooth behaviour.

Algorithm (7) describes more in detail what is already shown in figure (3.15).

Algorithm 7 Modulated Homogeneous Friction Control Strategy

Require: $\Delta t > 0$, V_{max}, g_n, $\xi_h \neq 0$, $\xi_l \neq 0$ and $\xi_h \geq \xi_l$
Require: the Heaviside Step Function $H[.]$
  $d^v_{rel} \Leftarrow \bar{d}_{rel}$
  repeat
    $v^n_{rel} \Leftarrow (d^n_{rel} - d^v_{rel})/\Delta t$
    if $(v^v_{rel} > 0$ And $v^n_{rel} < 0)$ Or $(v^v_{rel} < 0$ And $v^n_{rel} > 0)$ **then**
      $d_{max} \Leftarrow d^n_{rel}$
    else
      no change in d_{max}
    end if
    acquire f
    $f_c \Leftarrow g_n|P[d_{max}]|$
    $\nu \Leftarrow V_{\max} H[(f_c - |f|)$
    if $\nu \neq 0$ **then**
      $c_s \Leftarrow \xi_h$
    else
      $c_s \Leftarrow \xi_l$
    end if
    $d^v_{rel} \Leftarrow d^n_{rel}$
    $v^v_{rel} \Leftarrow v^n_{rel}$
  until stop condition

3.3.9 Bang-bang control

The bang-bang control strategy is easy to implement in real-time to control operations since the control signal switches between two values and only the relative displacement needs to be measured by a sensor and feed back to the control signal. The function $\text{sign}(x)$ can be obtained by the relative displacement signal and does not need to measure the velocity.

3.4 Some considerations about actual implementations

3.4.1 Practical implementations

At present time, only three buildings incorporate semi-active devices: two are in Japan and one is in USA (Soong and Spencer 2000). However, many other structures are planned to be equipped with semi-active devices. One example of future building will be given in the following paragraph.

Kajima Research Laboratory The first building equipped with semi-active devices is the Kajima Research Laboratory. It was built in 1990 by Kajima Corporation for its research centre in Tokyo (figure (3.16(a))). It is a three-storeys building with three Active Variable Stiffness systems with on-off behaviour (for this reason they are semi-active and not purely passive). Each floor has one actuator and some sensors (figure (3.16(b))): the information is passed to a digital computer where the control strategy is implemented. This is a typical example of centralised and non-collocated control scheme.

The building, during the last 12 years, has already tolerated three little earthquakes (November 1990, November 1991 and February 1992) able to activate the system: these were the best tests for the control system. The records of the response has shown that the behaviour of the protected structure is much better than the simulated not protected one.

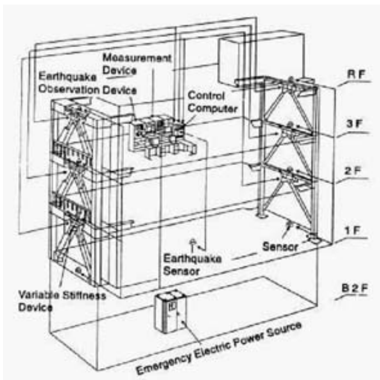
Walnut Creek Bridge The second application, of semi-active control techniques to real civil structures is the Walnut Creek Bridge (figure (3.17(a))) on Highway I-35 in Oklahoma, USA. Some controllable viscous fluid dampers were installed under the deck in order to dissipate the vibrational energy induced by the traffic.

This system was proposed by Prof. Petten (Patten et al. 1996; Petten et al. 1997) from the University of Oklahoma. The system was monitored for two years (Petten et al. 1999) with very promising results. The device principle is very simple (figure 3.17(b)) and was design for a on-off behaviour. A common battery for trucks at 12 V is sufficient to drive the switching valve.

Kajima Shizouka Building In 1998, Kajima Corporation has built the Kajima Shizouka Building. It has 5 storeys used as offices. The lower 4 storeys are equipped each with 2 semi-active devices; so 8 devices in total are present on the structure.

These semi-active devices can generate damping coefficient in a continuous range between 200 and 1000 Ns/mm. The value is set accordingly to a control

(a) External view

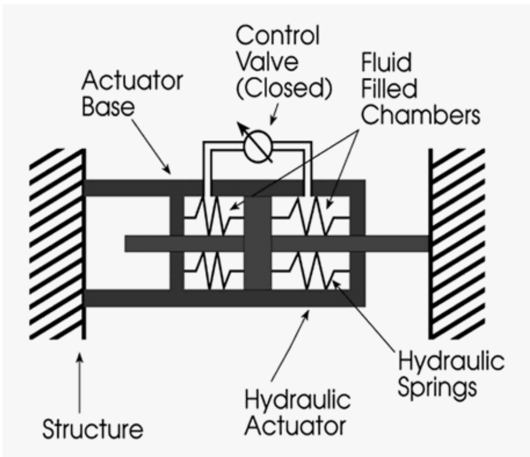


(b) Scheme of the control apparatus (sensors and actuators)

Figure 3.16: The Kajima Research Laboratory building

algorithm based on the Linear Quadratic Regulator (LQR) control theory (Marazzi 1997). Different performance can be obtained by changing the force-deformation loop of the variable semi-active damper (Kurino and Kobori 1998).

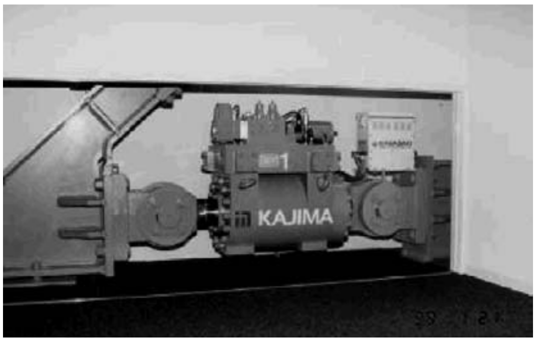
(a) Bridge view



(b) Scheme of the semi-active (SAVA) device

Figure 3.17: The Walnut Creek Bridge

(a) External view



(b) Scheme of the semi-active device

Figure 3.18: The Kajima Shizouka building

Nippon TV Network Corporation New Headquarters The Nippon TV Network Corporation New Headquarters are now under construction in Tokyo. The structure (figure (3.19(a))) is a 192.8 m height 32 storey steel structure founded on piles. The first 17 storeys are occupied by a steel trusses composing megaframe structures. These open spaces of 4 storeys each will accommodate broadcasting and film studios. The structure is supported in the longitudinal direction by two super steel braces. The megaframe is composed by unbounded diagonal braces filled with concrete to prevent local buckling. The megaframe is attached to the structure through link beams to absorb energy during the earthquake. The top part of the structure is equipped with oil dampers to increase damping during earthquakes and wind storms.

In addition to all these passive devices, at the top-storey the building is equipped with semi-active tuned-mass dampers (figure (3.19(b))) to decrease vibrations of the building during a windstorm.

3.4.2 Open problems

Before entering the next chapter of this thesis, is worthwhile to answer to a rather obvious question: what are the open problems in semi-active control of civil structures? There should be some hidden reasons for the few applications of this new technology.

3.4.2.1 Time delay

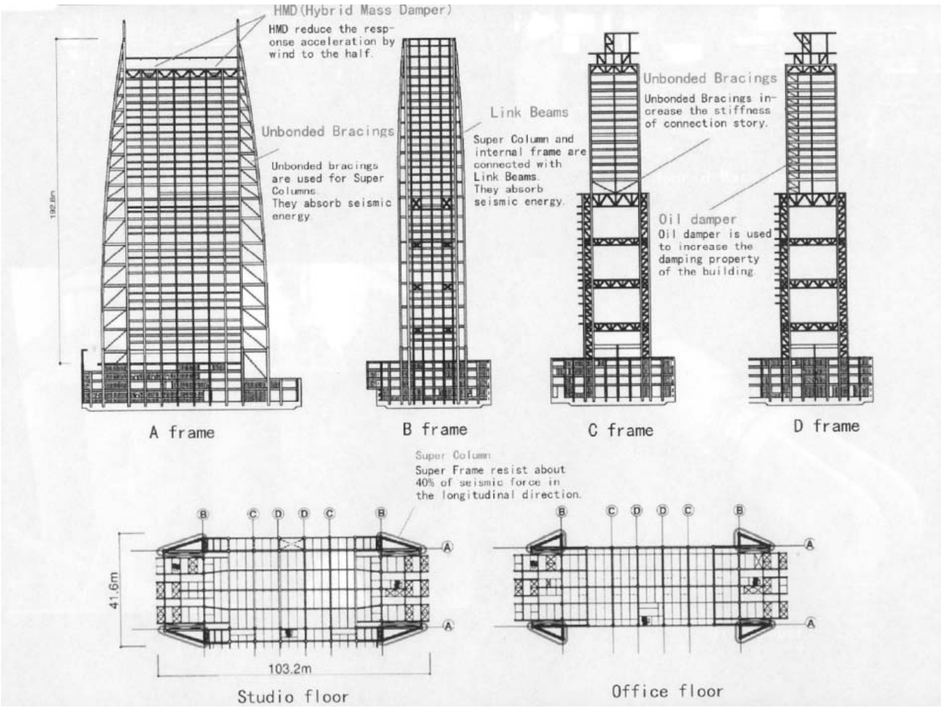
The problem of time delay was very recently addressed (Breitung and Casciati 2000) and studied. In a simplified approach, time delay is considered to be absent or very small, but, in a real word, even with the increasing computational power of today and faster computers, time delay is always present. Sources of time delay could be, for example:

- electronic time leg among sensors and actuators (due also to control computational time);
- inertia in actuator moving (for example for pneumatic, hydraulic and mechanical actuators);
- physical misalignment between actuators and sensors;
- compressibility of operating fluid;
- flexibility of control device;

The study of time delay in semi-active control is not so easy since semi-active control laws are inherently non-linear, so classical approach cannot be used.

In a classical approach, a finite time delay is described by the following mathematical model (see (Marro 1992) for more details):

$$F(s) = \frac{P(s)}{Q(s)}\, e^{-t_0 s} \tag{3.4.15}$$



(a) Scheme of the structural design and control devices

(b) Hybrid mass damper

Figure 3.19: The Nippon TV Network Corporation New Headquarters

where $P(s)$ and $Q(s)$ are polynomial in s and $e^{-t_0 s}$ is the transfer function of a finite time delay t_0. This transfer function, however, can be used only when the system is linear (or it is considered to be linear around the working point). Because of non-linearity of semi-active control laws, it makes no sense, in this case, to speak of transfer function of a system, so this approach must be abandoned. In (Breitung and Casciati 2000) the problem is studied in the time domain. Fixed a control law for a SDOF system, the only varying parameter remains the amount of time delay. Several simulation are done with increasing value of this parameter until unacceptable oscillation are reached.

3.4.2.2 Large scale problems

To move from laboratory to real live structures require the addressing to the additional problems related to the different scale and impact of the building (Baratta and Casciati 2000). These problems are sometimes technical, but in most of the cases are "cultural", in the sense that they involve more the habit, the juridical, and the political sphere of the human behaviour.

Leaving the latter ones for the next paragraph, the main possible technical problems are itemised hereafter:

- **high costs**: even if using semi-active devices may results in relevant money sawing during the lifetime of the structure both for the point of view of maintenance and of the added safety and comfort, the initial costs are often considered too high. So, this problem could be moved into the "cultural" problems, because it is more apparent then real;

- **difficult design**: as this chapter has shown, a lot of different control laws can be used, some of them being very simple to implement but difficult to optimise, others being optimal but more complexes. Also in this case, however, no real problems exist;

- **scaling actuators**: it is not always easy to scale small devices into bigger ones, it is not simply a matter of multiplying the dimensions by a scaling factor!;

- **limited market**: this problem could also be referred to the following paragraph, because the market has surely the potential to become enormously large. For the moment, however, the very low demand is a negative factor for private companies that have to develop the devices. Is relatively easy to undertake research studies because no directly income must come out, but for an industry is fundamental to sell the many devices to recover the development costs.

- **need for research**: theoretical and experimental studies are still needed in this field. Some semi-active control laws need to be more deeply investigated mainly because of the inherent non-linearity. Numerical methods for the calibration of the needed parameters and for non-linear simulation of the overall behaviour of the structure must be refined. On the other side, experimental validation of large scale devices and implementation is still needed.

3.4.2.3 Regulations and policy

The over mentioned questions had been clarified. Nevertheless it is difficult to understand why this promising class of techniques is still so little used in real scale applications.

Without exhausting the answer, some problems that have been identified are listed hereafter.

- Generally speaking, there is a lack in the standards related to earthquake protection of civil structure and, more in general, in vibration mitigation. This seems to be the case in Europe, but also in USA. Traditional earthquake mitigation techniques are encouraged within Eurocodes[6], while a little opening is admitted for passive devices.

  In Italy it is nevertheless allowed to use alternative techniques, but in this case a special demand must be submitted to the Public Ministry with a detailed technical report. This causes usually a construction time dilatation of the order of months (sometimes a couple of years) and so this possibility can be used only for very special structures.

- Although these new technologies seems to offer strong potential, it is not clear yet whether its performance can actually meet the market demands.

- Noise and vibration standards throughout Europe are not yet uniform. A European norm will help to accelerate the application of the new technology. Large civil engineering laboratory as ELSA will surely have an increasing and leading role in this direction.

[6] most of them being still in a draft stage, not yet operational and not referring to structural control.

Chapter 4

Implementation of the semi-active control strategies

4.1 Introduction

The implementation schemes of each semi-active control strategy have been described in the previous chapter. This has given the possibility to pass from a theoretical formulation to a more practical one. In this chapter, the realisation of some control strategies on a real case test structure will be described including hardware, software and any other tools that it is necessary for achieving the control goals.

4.2 Hardware

In order to perform dynamic and pseudo-dynamic tests[1], a dedicated hardware was necessary. Commercial software is usually very closed in order to avoid the possibility of copying the source program or to modify. Specific products for laboratory testing can be found on the market, but they are difficult to be freely modified and adapted to the various needs of a big laboratory in which it is common practice to face a lot of different structural problems and assessment approaches. In any case, it is impossible to find something really optimised for performing

[1]The PseudoDynamic test method consists in performing seismic tests on a real scale structure imposing displacements and measuring the corresponding forces accordingly with a fed accelerogram. The test is usually much more slower than in the case of a real earthquake. It has to be conducted in an extreme precise way in order to reproduce the real dynamic of the building. In the meantime the control process must be very quick and must record the important data for the understanding of the structural behaviour. For more details about the PseudoDynamic method, a good reference is (Negro and Magonette 1998).

pseudo-dynamic tests, so resulting in expensive and not efficient instruments. Just to give an example, substructuring techniques require real time control software that are guarantied to be efficient in any testing phase.

For all these reasons, the ELSA laboratory has chosen to develop its control system "in house", i.e. to design the proper hardware and the necessary software with the all needed interfaces from and towards the main analysis and processing software.

4.2.1 Architecture

The hardware consists of three main parts: the master card, the slave cards (they are usually more than one) and the passive bus connection (figure (4.1)).

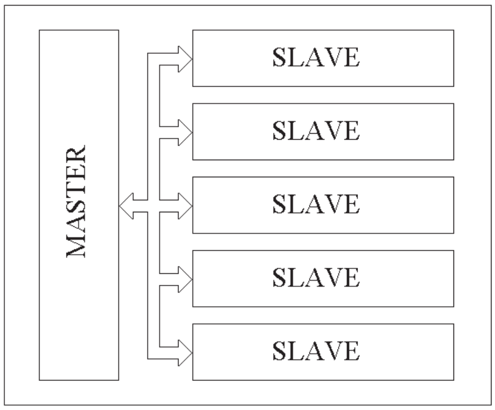


Figure 4.1: Scheme of controller architecture

4.2.1.1 Master

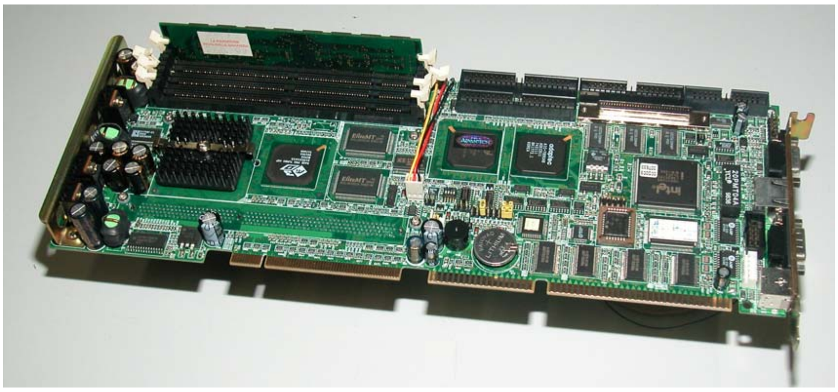
The master card (figure(4.2)) contains the kernel of the pseudo-dynamic algorithm. For this reason it is equipped with a fast processor (Pentium class) and enough memory in order to store the necessary data.

The master communicates with the slave cards through the passive ISA bus.

4.2.1.2 Slave

The slave card is shown in figure (4.3). It consists of a board with three main components: a PC104 central processing unit card, a digital input/output card

(a) front side



(b) back side

Figure 4.2: Photo of the master card

and an analogue input/output card. The board is also equipped with the ELSA developed Dual-RAM.

4.2.1.3 Passive bus

The ISA passive bus connects the master and slaves cards. In the current configuration it can connect up to one master card and seven slaves cards, but in principle

Figure 4.3: Photo of the slave card with components

the limitation on the number of cards that can be connected with a passive bus is 16. Figure (4.4) illustrates an example of an assembled controller with power supply and adjunct tools like floppy drive.

The assembled controller must then be put into a rack and the peripherals (floppy drive, LCD screen and so on) connected to it. It is possible to reset either the master CPU and the SLAVES CPUs separately.

4.2.1.4 Why to use this architecture?

After this very short description of the controller architecture, an explanation of these choices is given hereafter. This architecture was chosen for three mainly reasons:

- it is the best solution in terms of modularity and generality: the hardware and software configuration is always the same for each individual test excepted from the number of slaves cards used. In this way it is easier to implement the test and to train technicians to conduct the tests;

- it is the best architecture to perform pseudo-dynamic tests. A typical configuration of an actuating system on a reinforced concrete structure of three storeys, for example, consists of two actuators at each storey. It is necessary that all actuators are well coordinated by a central computer that has the duty to calculate the required target displacement accordingly with its internal pseudo-dynamic algorithm and the measured forces at each actuators. Once the target signal is sent to the slave, it is its duty to reach the required displacement;

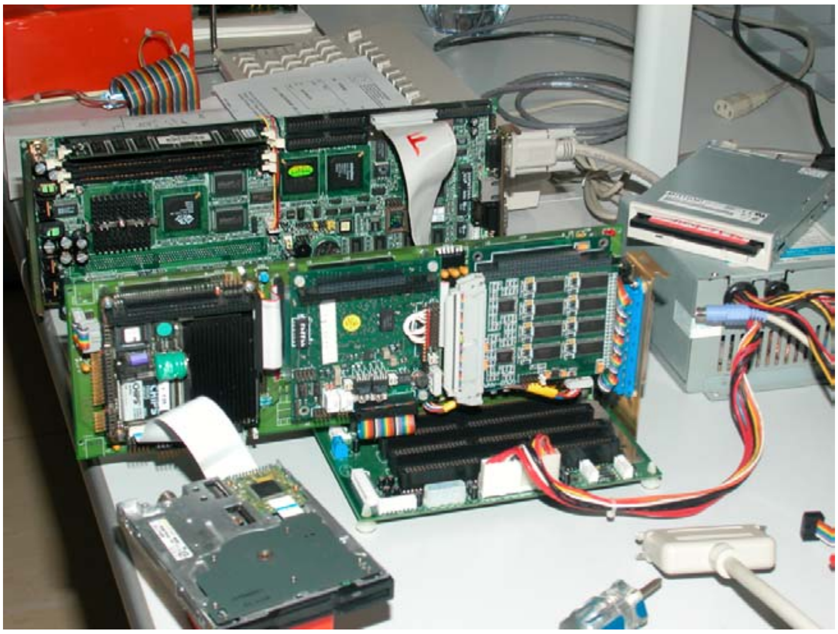


Figure 4.4: Photo of the passive bus equipped with a master card and a slave card

- it is the ideal configuration for implementing both centralised and decentralised control strategies in dynamic structural control. When a centralised strategy is desired, the master executes the control algorithm and sends the target signal to the slaves. They have the only duty to reach that displacement, for example via a PID control law. When a decentralised control strategy is requested, the master has the only duty to synchronise the slaves and to save the internal control values of the used algorithm variables, while the slaves execute their own algorithms.

Moreover, as it will be shown in the following paragraphs, this architecture is well suited for a general software implementation.

4.2.2 Connection box

The connection box is a special box in which all the measured signal coming from the sensors on the structure can be easily pick up and visualised. Strictly speaking, it is something independent from the control hardware, because it is used as interface between the controller and the hydraulic actuators used al ELSA, for debagging purposes and for an easier signal acquisition.

The signal coming from the Heidenhain displacement sensor internal to the hydraulic actuator is conditioned by the connection box and then can enter to the controller. The power supply of the actuator instrumentation is given as well. Moreover, when the test structure is complex and a lot of actuators and sensors are mounted on it, it can be crucial to be able to control, channel by channel, all signals coming into the controller.

4.2.3 Master card description

The master card is equipped with:

- CPU Pentium III - 800 MHz;
- 256 MB DRAM
- RT clock + Watch-dog;
- LPT1, COM1, COM2;
- AT-Keyboard;
- Floppy Disk Interface;
- AT-IDE Hard Disk Interface;
- VGA/LCD Video Interface.

This is the typical configuration, but it can be changed easily if it is necessary for one specific test.

4.2.4 Slave card description

The slave card scheme equipped with modules is shown figure (4.5).

The card had an ISA connector used to connect the card with the passive bus (and so to all the slaves cards and the master card). It has also three ports, one for connecting a screen, one for connecting the keyboard and the third on for connecting the control connection box (figure (4.6)). Three modules are connected to the slave card with a PC104 ISA connector: a PC104 Central Processing Unit (CPU) module (figure (4.7)), a PC104 analogue Input/Output (I/O) module (figure (4.8)) and a PC104 digital Input/Output (I/O) module (figure (4.9)). On the board there is also the Dual-RAM (it can be seen on the right side in figure (4.6). The CPU module is equipped with:

- CPU 486 DX4 - 100 MHz;
- 16 MB DRAM;

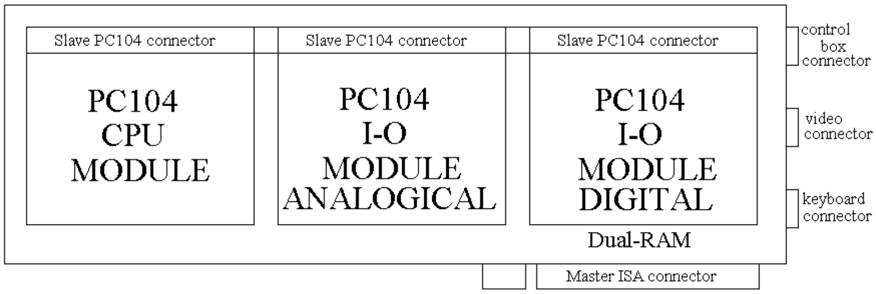


Figure 4.5: Scheme of a slave card

- solid state disk (2 MB);
- RT clock + Watch-dog;
- LPT1, COM1, COM2;
- AT-Keyboard;
- Floppy Disk Interface;
- AT-IDE Hard Disk Interface;
- Network Module Interface;
- VGA/LCD Video Interface.

The analogue board I/O module has the following channels inputs:

- 16 ADC - 16 bit;
- 2 DAC - 16 bit;
- 24 Digital IN/OUT;
- 3 Counter-Timer;
- Ref. and ± 15V.

The digital board I/O module has the following channels inputs:

- Heidenhain;
- 2x Temposonic;

- 2x Servo-valves (Vout);
- Watch-dog (relay);
- 8x Digital IN/OUT.

The Dual-RAM is equipped with the Dual-Port RAM (8k x 16 bit + Hard Disk). The Dual-Port RAM (or simply Dual-RAM) is a special ram memory developed at ELSA with the purpose of making faster the exchanging data process. This memory can be access from two sides in the same time. In this case, the master card can write data on the slave Dual-Ram while the slave is reading it. Special care must be taken in order to avoid improper cancellation of data. A specific dialog protocol has been developed to manage this data flux.

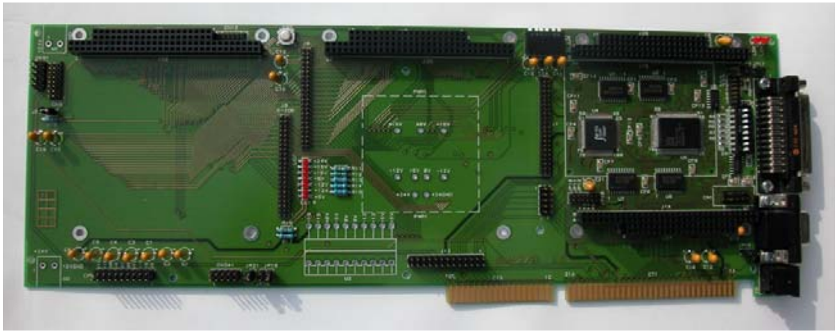
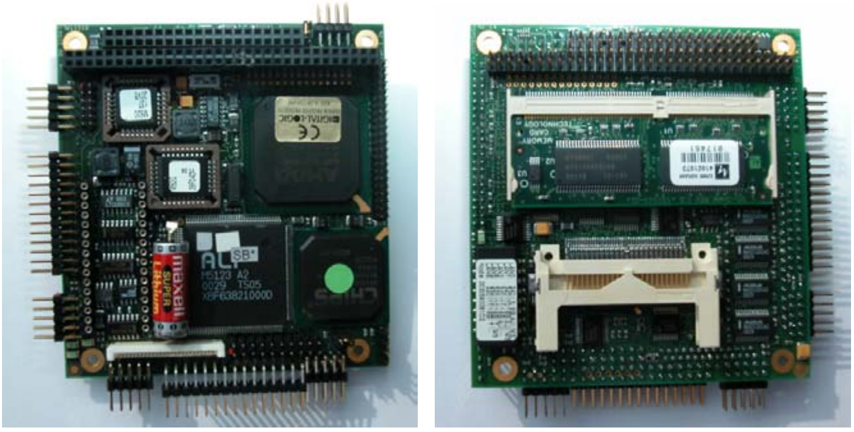


Figure 4.6: Slave card without modules (the Dual-RAM is visible on the left side)

4.3 The main control software

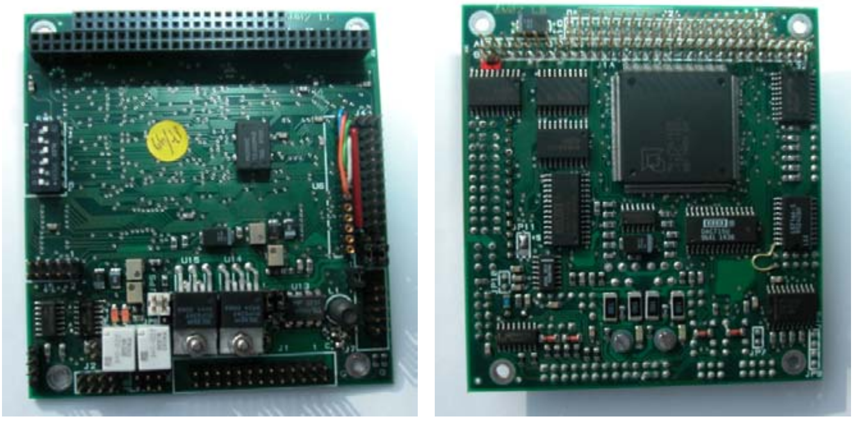
The first request for the main control software was to be as general as possible, in the sense that, for laboratory reasons, it must be structured in such a way that can be adapted easily and quickly to every building to be tested. All the measured values are available in the control loop. The user has the possibility to select the ones necessary for his control algorithm.

The main control program has been written in C++ under the TNT real-time kernel platform. This special operating system is dedicated to real-time applications. Real time control cannot be achieved with the MS-DOS operating system. MS-DOS and Microsoft Windows NT platform are not conceived to manage real-time processes implying switches of control tasks with a sampling rate around 1 kHz.



(a) front (b) back

Figure 4.7: Slave card CPU module



(a) front (b) back

Figure 4.8: Slave card analogical I/O module

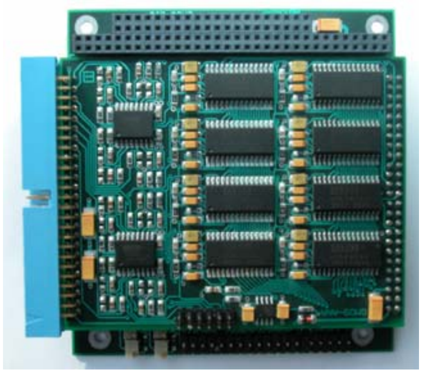


Figure 4.9: Slave card digital I/O module

4.3.1 Why TNT and not NT?

The Real-Time Kernel TNT has been selected because it has two main advantages:

- **reduced and fixed latency time**. When an interrupt arrive, the latency time is fixed and very short. This doesn't happen on NT platform, because this system is not really real-time and a variable latency time is observed because it is impossible to obtain absolute priority.
- **more easy kernel**. The NT platform is programmed at two levels: users level and kernel level. Users level is easy to manage but kernel level is very complicate and requires very skilful programmers. TNT kernel is simpler and more reliable. With TNT kernel, priority can be assigned easily and are guaranteed to be respected (this property is called "deterministic kernel").

The advanced C++ program language is currently used to program ELSA applications under TNT.

4.3.2 The TNT Software

The control software reflects the architecture of the hardware: there is one master program that communicates with several slaves programs. For the sake of simplicity, the following description will be made for only one slave, but it can be easily generalised to several slaves.

Both the master and the slave programs originate two mainly process: the background process and the foreground process. The first one is devoted to manage several services that are used during control:

- the keyboard;

- the uploading of control parameters;
- the displays refresh;
- the hard disk management;
- the LAN connection;
- the remote services (under NT platform).

Since these services are not strictly necessary (the display refresh, for example, can be delayed a little bit, if it is needed), these processes have a lower priority than those in the foreground process.

The foreground process is the core of the control software: it performs at a fixed sample rate the data acquisition and the computation of the control variables. For this reason it must have absolute priority on the background processes, because obviously delays cannot be accepted in the control algorithm. Having in mind figure (4.10), the foreground process can be described in details as follows:

- when an external interrupt occur (i.e. every 1 ms, given by an external clock) the master sets all slave internal interrupts flags in the Dual-RAM and then wait for the slave operation;
- this generates an internal interrupt in the slave that starts its foreground process;
- the slave reads the I/O module and writes there the I/O value on the Dual-RAM, then wait for the master operation;
- in the case of a semi-active collocated control strategy, the master performs a monitoring function and simply coordinates the slaves[2]. The monitoring function is used also to simulate spikes and other error on the sensors in order to assess the control algorithm robustness;
- the slave calculates the new target displacement;
- the slave execute the control algorithm[3], a semi-active control law for semi-active control;
- the slave write the command signal to be passed to the actuator in the I/O module and return into the background process;
- in the meantime the master reset all slave internal interrupt flags to be ready for the next step and return into the background process.

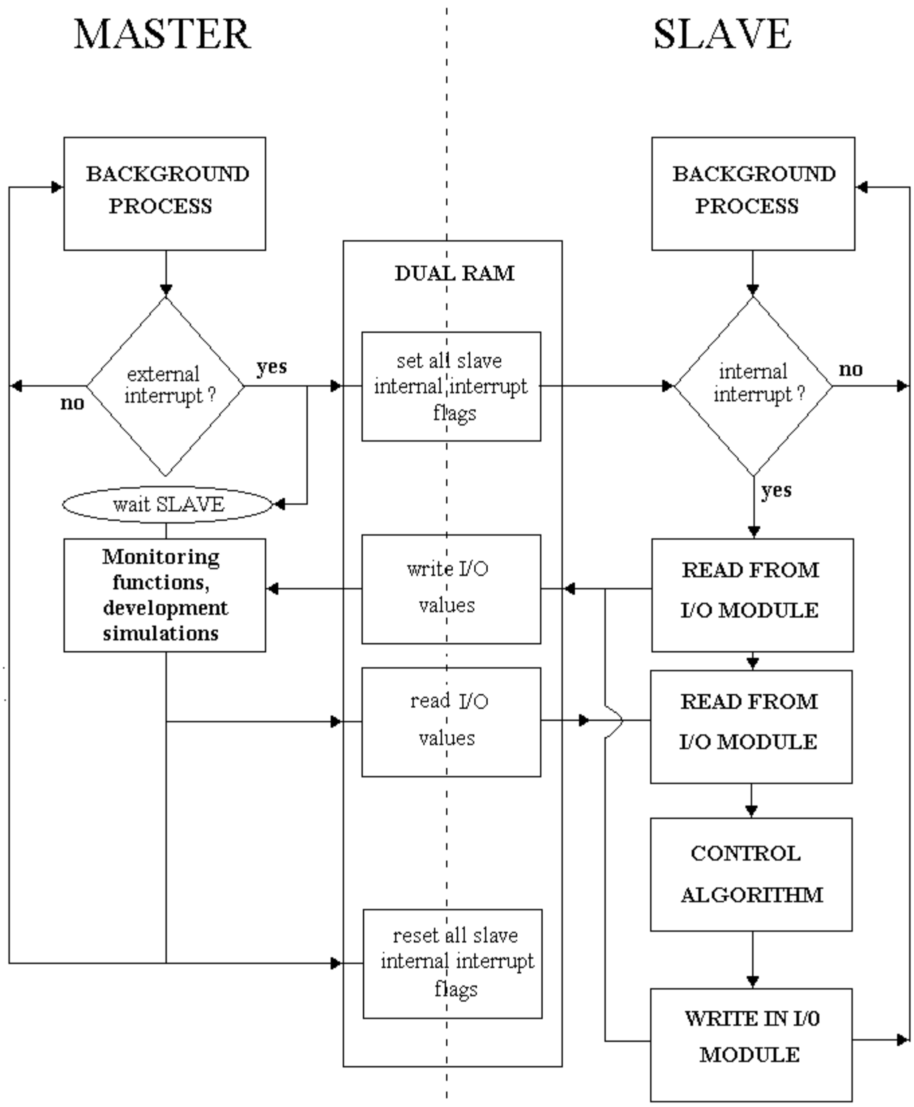


Figure 4.10: Flowchart of the TNT software

[2]In the case of the pseudo-dynamic algorithm the master calculates the new target displacement accordingly with its defined algorithm. This value is calculated taking into account the

Just to give an order of magnitude, the foreground process takes usually 300 μs, while the background process uses the remaining time (700 μs) to arrive to 1 ms.

It must be notice that this complex architecture is ideal for development and debagging of controllers and devices: if the algorithm has to be implemented into and industrial controller, only the slave process is then necessary.

4.3.3 The NT Software

As it will explain in the following paragraph, there is the possibility to directly connect the TNT software with NT, i.e. it is possible to command a TNT program from a NT interface. Two interfaces were developed up to now: the acquisition program and the generator program.

4.3.3.1 The acquisition program

The acquisition control panel is shown in figure (4.11). Within this interface is possible to give all the test characteristics in order to have proper description of the measured signals.

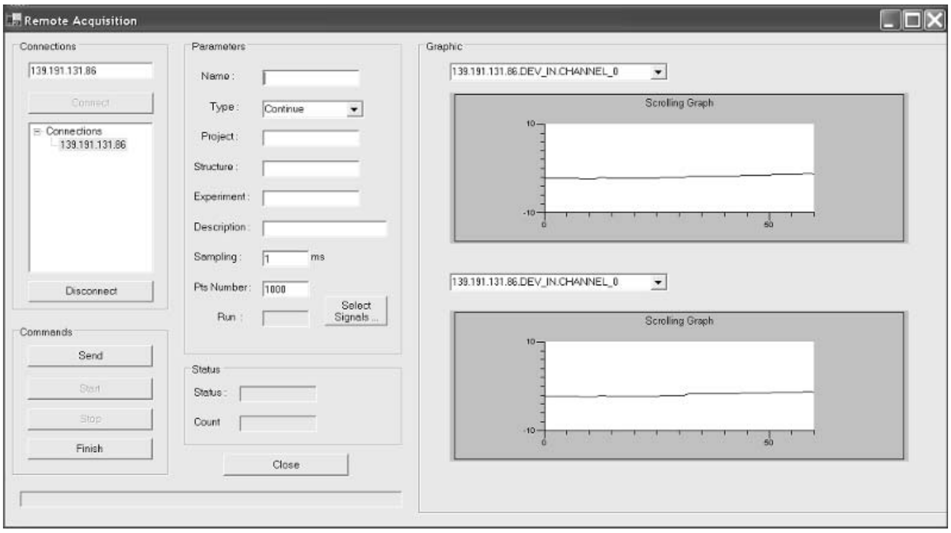


Figure 4.11: External acquisition control panel

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next acceleration point, the masses of the structures and the measured restoring forces.

[3]a positioning PID for the pseudo-dynamic tests.

The various control panel fields must be properly filled: the computer address where the acquisition will take place, the name of the project under which the test is being conducted, the description of the test, the sampling time, the number of points that are desired to be acquired and so on must be given. Then, it is possible to chose which variables must be acquired.

Once all these data has been inserted, it is possible to send this information to the TNT acquisition system and to start the acquisition. An external trigger can allow the start-pause of the acquisition without interrupting the actual acquisition.

4.3.3.2 The generator program

The generator control panel (figure (4.12)) is devoted to send to the TNT software a user-defined signal to be used as a reference signal into the control algorithm or as a command signal for a force generator device (for example a shaker).

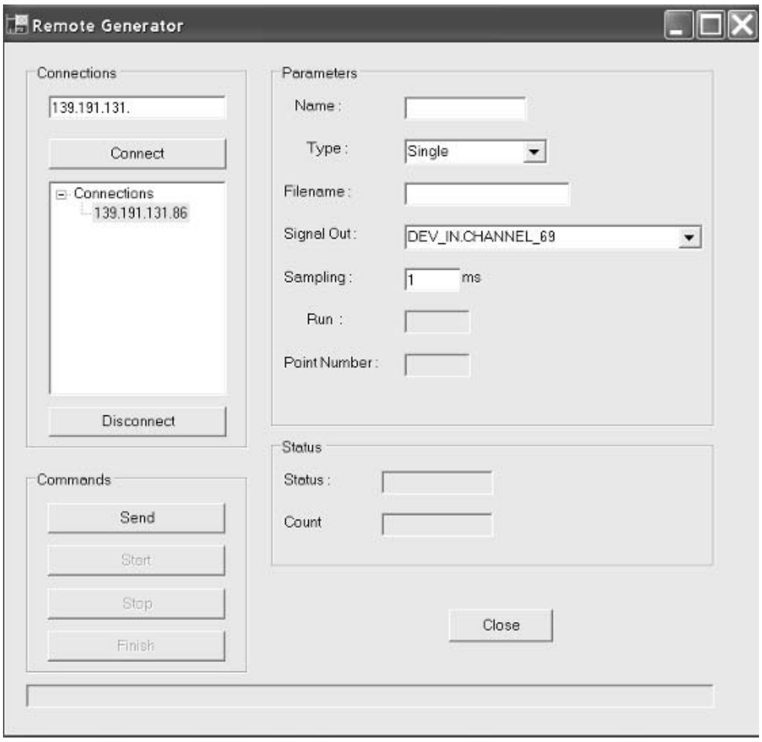


Figure 4.12: Generator control panel

In this case the signal filename, the sampling time, the address of the computer where the TNT generator software is present must be given. An important option is this: it is possible to chose that the generator signal is being repeated infinite times. In this way, for example, a test with a white noise excitation-forcing signal can be done with an excitation signal file rather short.

The signal can be generated with any software. The important point is that it must be saved is double precision format. It is possible to generate up to 4 generators signal simultaneously, so multiple excitation tests can be conducted.

4.3.4 Software tools

The software in support to the main controller program is included under this group. These tools were developed in order to better integrate the controller with commercial common used software and for making the data storage faster, more reliable and more consistent.

4.3.4.1 Exchanging variable

Internal variables are used both inside the master software and the slave software. The variables used by the master are, for each slaves connected to it, include:

- monitoring and measurement values;
- algorithm parameters and control variables.

The slave uses the following variables:

- measurement values;
- algorithm parameters and control variables.

A background process provides the necessary exchange of some parameters between the master and the slaves (alarm setting values, gains, ...).

From the NT workstation it is possible to modify one or more of these parameters: this can be very useful especially when working in remote mode. A similar exchange background process operates between NT and the controller (master + slaves). The exchange program scans all variable and, if one has been modified, it uploads the new value. Since in this play there are more than two players (NT system, master, slaves) a smart procedure has been developed in order to avoid infinite cyclic values modifications.

4.3.4.2 DCOM Technology

This technology permits to exchange data among several applications. It consists in a standard developed by Microsoft Corporation to permit easy interface among different applications.

In the case of ELSA laboratory, this technology is very useful because permits to use several commercial programs used very often by researchers. In figure (4.13) there is a scheme of these softwares all connected together via the DCOM standard.

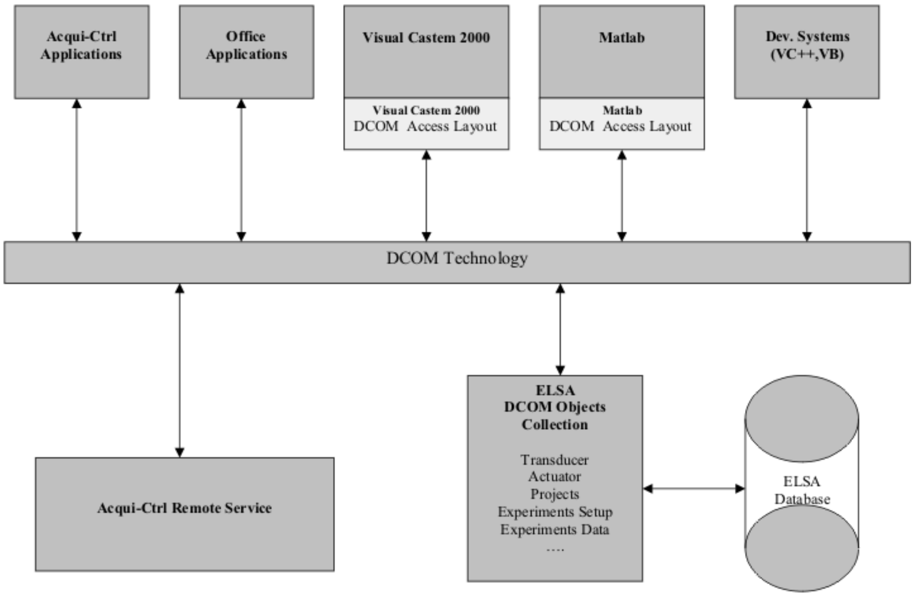


Figure 4.13: DCOM technology

Database The database can be directly connected with the acquisition via the DCOM technology. Once the test description and specification has being given, it is possible to directly send the data into the database under the proper project and test directory. This option is now under development at ELSA and will be available in a few months.

MATLAB and CASTEM2000 DCOM technology permits to directly interface between MATLAB and the control algorithm. It is possible to pass a reference variable to the controller directly from a MATLAB or CASTEM2000 routine. This has paved the way to sub-structuring techniques in which only one portion of the whole structure is being tested in the laboratory (usually the non-linear part or a part for which the behaviour is not well known), while the remaining part is numerically modelled (usually the linear or known part).

4.4 Implementation of the semiactive control software

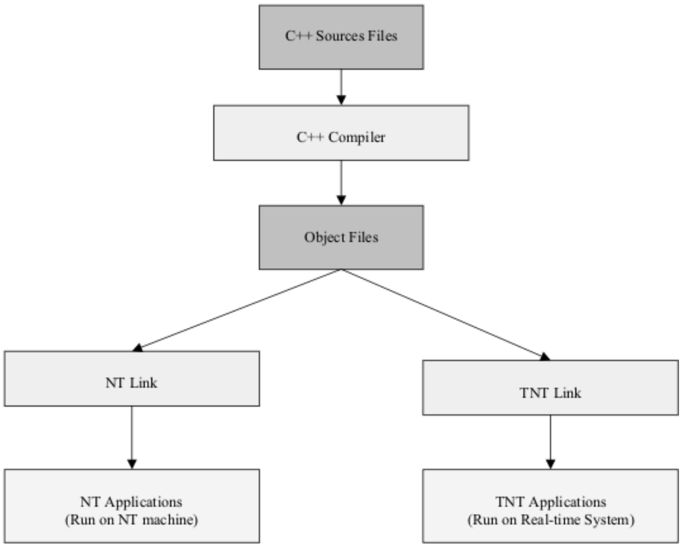


Figure 4.14: Software implementation scheme

4.4.1 Filter functions

Before proceeding into the control algorithm, the signals coming from the sensors distributed on the structure must be filtered to suppress both noise and specific undesired components (in particular 50 Hz and static offset). There are a multitude of methods for filters synthesis (Smith 1999): the ones used in this thesis will be presented hereafter.

The choice of the most proper filter for the specific application should be performed satisfying to the following questions:

1. Which shape should have the Bode plot of the transfer function between the input and the output[4]?

2. What is the acceptable time lag between the input and the output signals?

[4]Signal attenuation is considered to be significant when an amplitude ratio less than -3dB (approximately 0.7 as a ratio) is achieved.

Or: What it the maximum phase-shift that can be tolerate for a specific excitation frequency?

3. What is the complexity that can be achieved by the filter? Is that filter compatible with the computing power of the processor that calculates the filter?

4. Is the filter easy to implement and stable with respect to any kind of input?

5. Can it be used on-line?

6. How many points it needs to operate? How many bytes of memory need to store the data necessary for operatining?

Accordingly to the present work, fast on-line filters can be implemented with sufficient bandwidth attenuation to separate continuous and dynamic signals components. For offset compensation, no particular attention has been given to phase-shift, because in that case this parameter is of minor importance.

4.4.1.1 The Transfer Function Method

Using this method, the filter is designed in the continuous Laplace domain with ordinary MATLAB toolboxes, or equivalent tools. Poles and zeros of the transfer function are freely chosen and the filter response (both in amplitude and phase) is then evaluated. Once the filter has been established, it must be converted into the discrete z-domain to be implemented as a digital filter in software subroutine (Smith 1999).

In an ideal world the goal would be to have an amplitude response that can be arbitrary chosen without influencing the phase diagram. This is difficult and good performance in terms of amplitude reduction can be not so good in terms of phase shift. So a balance between these two requests must be reached. This means that it is not possible to make a low-pass filter with a roll-off very sharp without sensible degradation of the phase. It depends on the application if the phase and the inherent input/output delay is important or not.

Several transfer functions were evaluated in the continuous Laplace domain and then converted into z-domain. For example, considering the following transfer function with 2 real poles:

$$F(s) = \frac{s}{(1+\tau_1 s)(1+\tau_2 s)} \tag{4.4.1}$$

where $1/\tau_1 = \omega_1 = 2\pi f_1$ and $1/\tau_2 = \omega_2 = 2\pi f_2$, it comes, with $f_1 = 0.1$ Hz and $f_2 = 20$ Hz:

$$F(z) = \frac{0.058852z^2 - 0.058852}{z^2 - 1.8811z + 0.88121} \tag{4.4.2}$$

Considering a continuous transfer function with a two pairs of complex poles as

$$F(s) = \frac{s}{(s^2 + 2 \cdot 0.707\omega_1^2)(s^2 + 2 \cdot 0.707\omega_2^2)} \tag{4.4.3}$$

the corresponding discrete transfer function is:

$$F(z) = \frac{0.0036242z^4 - 0.0072484z^2 + 0.0036242}{z^4 - 3.8221z^3 + 5.4808z^2 - 3.4954z + 0.83666} \tag{4.4.4}$$

If a little overshoot is allowed, the transfer function (4.4.3) can be slightly modified by changing the 0.707 factor at the denominator into 0.6. This will lead to a sharper knee in correspondence to the poles. In this case the modified transfer function is:

$$F(s) = \frac{s}{(s^2 + 2 \cdot 0.6\omega_1^2)(s^2 + 2 \cdot 0.6\omega_2^2)} \tag{4.4.5}$$

and the corresponding discrete transfer function becomes:

$$F(z) = \frac{0.0036696z^4 - 0.0073391z^2 + 0.0036696}{z^4 - 3.8449z^3 + 5.5495z^2 - 3.5642z + 0.85964} \tag{4.4.6}$$

A comparison among these transfer functions is shown in figure (4.15)

Multiplying equation (4.4.6) by $1/0.85964$ at the numerator and the numerator, it becomes:

$$F(z) = \frac{0.0042687z^4 - 0.0085375z^2 + 0.0042687}{1.1633z^4 - 4.4727z^3 + 6.4556z^2 - 4.1462z + 1} \tag{4.4.7}$$

Recalling the fact that

$$\mathcal{Z}[f(k-1)] = \frac{1}{z}F(z) \tag{4.4.8}$$

and

$$\mathcal{Z}[f(k+1)] = zF(z) - zf(0) \tag{4.4.9}$$

where $\mathcal{Z}$ is the z-transform operator, and, under the hypothesis that $f(0) = 0$ and $Y(z) = X(z) \times F(z)$, the following digital form for the filter (4.4.7) can be obtained:

$$\begin{aligned} y(k) = {} & 4.1462y(k-1) - 6.4556y(k-2) + 4.4727y(k-3) - 1.1633y(k-4) \\ & + 0.0042687x(k) - 0.0085375x(k-2) + 0.0042687x(k-4) \end{aligned} \tag{4.4.10}$$

where $y(k)$ is the actual output value, $y(k-n)$ is the output value n steps ago, $x(k)$ is the actual input and $x(k-n)$ is the input n steps ago. This means that the latest 4 values of the input and the output must be stored to be able to calculate the filter.

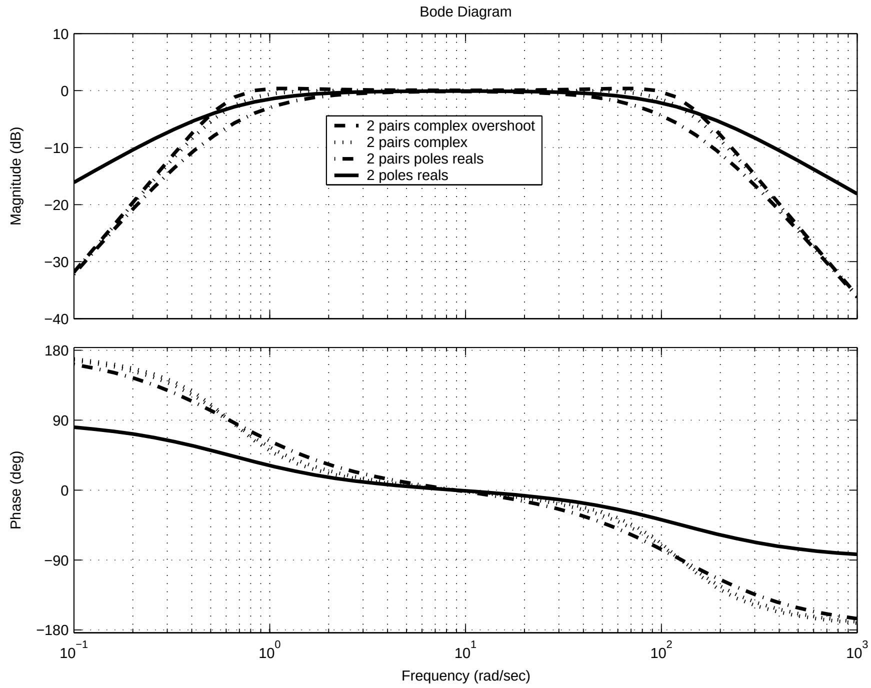


Figure 4.15: Comparison among different filters

The Transfer Function Method Subroutine as it was programmed in language C++ in the case of a double pair of complex poles is presented in algorithm (8). The variables `g_dAlgoParam4`, `g_dAlgoParam5`, `g_dAlgoParam6`, `g_dAlgoParam7`, `g_dAlgoParam8`, `g_dAlgoParam9`, `g_dAlgoParam10` correspond, respectively, to the coefficients of terms $y(k-1)$, $y(k-2)$, $y(k-3)$, $y(k-4)$, $x(k)$, $x(k-2)$, $x(k-4)$ of equation (4.4.10). With this subroutine it is so possible to implement any other filter derived from equation (4.4.3).

4.4.1.2 The Moving Average Method

Introduction The moving average is the most common filter in digital signal processing (Smith 1999), mainly because it is the easiest digital filter to be understood and used. In spite of its simplicity, the moving average filter is optimal for a common task: reducing random noise while retaining a sharp step response. This makes it the premier filter for time domain encoded signals. However, the moving average is the worst filter for frequency domain encoded signal, with little ability

Algorithm 8 Transfer Function Method Programmed Subroutine

```
float fFiltroBanda(float fx)
{
      double fy;
      //2 poli complessi sovraelongazione
      fy  = - (float)g_dAlgoParam4 * fy4
            + (float)g_dAlgoParam5 * fy3
            - (float)g_dAlgoParam6 * fy2
            + (float)g_dAlgoParam7 * fy1
            + (float)g_dAlgoParam8 * fx
            - (float)g_dAlgoParam9 * fx2
            + (float)g_dAlgoParam10 * fx4;
      fy4 = fy3;
      fy3 = fy2;
      fy2 = fy1;
      fy1 = (float)fy;
      fx4 = fx3;
      fx3 = fx2;
      fx2 = fx1;
      fx1 = fx;
      return (float)fy;
}
```

to separate one band of frequencies from another.

Implementation by convolution As the name implies, the moving average filter operates by averaging a number of points from the input signal to produce each point in the output signal. In an equation form, this is written:

$$\bar{y}_i = \frac{1}{N} \sum_{j=0}^{N-1} x_{i+j} \tag{4.4.11}$$

where x is the input signal, $\bar{y}$ is the output signal, and N is the number of points in the average. It can be notice that this formulation will shift the output value $\bar{y}$ of N points with respect to the input signal x, thus introducing a time delay. This delay will be as greater as the number of considered point increase.

As an alternative, if an odd number of points are chosen, the group of points from the input signal can be chosen symmetrically around the output point:

$$\bar{y}_i = \frac{1}{N} \sum_{j=-\gamma}^{\gamma} x_{i+j} \tag{4.4.12}$$

where $\gamma = (N-1)/2$. In this way time delays can be avoided. You should recognize that the moving average filter is a convolution using a very simple filter kernel. For example, a 5 point filter has the filter kernel: $\ldots$ 0, 0, 1/5, 1/5, 1/5, 1/5, 1/5, 0, 0 $\ldots$, that is, the moving average filter is a convolution of the input signal with a rectangular pulse having an area of one.

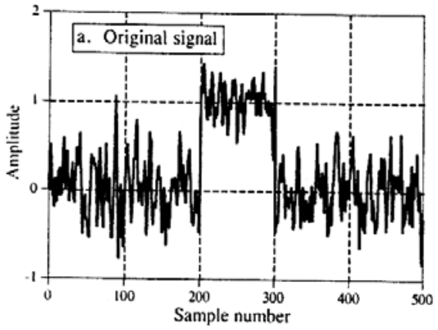
In both cases, however, there is the important practical drawback: the output value at point i is function of the input values at points i, $i+1$, $\ldots$, $i+N-1$ for equation (4.4.11) and $-(N-1)/2$, $\ldots$, $(N-1)/2$ for equation (4.4.12). This is not feasible for on-line filtering, but can be implemented only off-line, when the signal x is available in all its acquisition points. For this reason, the formulation above has not been considered anymore. An alternative formulation will be given afterwards.

Noise reduction vs. step response The moving average filter is very good for many applications and it is optimal for a common problem: reducing random white noise while keeping the sharpest step response.

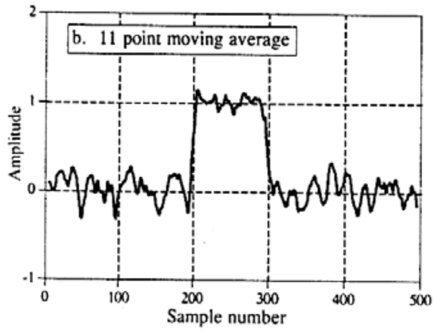
Figure (4.16) shows an example of how the off-line formulation works (the results for the on-line formulation are similars). The signal in (4.16(a)) is a pulse buried in random noise. In (4.16(b)) and (4.16(c)), the smoothing action of the moving average filter decreases the amplitude of the random noise (good), but also reduces the sharpness of the edges (bad). Of all the possible linear filters that could be used, the moving average produces the lowest noise for a given edge sharpness.

The amount of noise reduction is equal to the square root of the number of points in the average. For example, a 100 points moving average filter reduces the noise by a factor of 10.

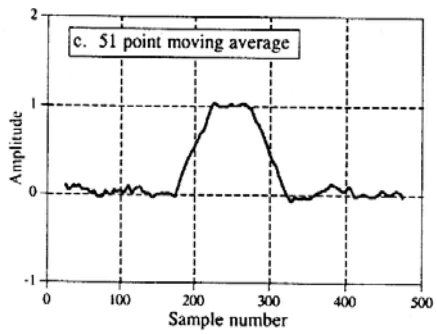
To understand why the moving average if the best solution, a filter with a fixed edge sharpness to be designed can be considered. For example, the edge sharpness is fixed by specifying that there are eleven points in the rise of the step response: this requires that the filter kernel have eleven points. The optimisation question is: how should be chosen the eleven values in the filter kernel to minimize the noise on the output signal? Since the considered noise to reduce is random, none of the input points is special; each is just as noisy as its neighbour. Therefore, it is useless to give preferential treatment to anyone of the input points by assigning it a larger coefficient in the filter kernel. The lowest noise is obtained when all the input samples are treated equally, i.e., if the moving average filter is used.



(a) original signal



(b) 11 points moving average



(c) 51 points moving average

Figure 4.16: Examples of moving average filters (off-line formulation)

Frequency response Figure (4.17) shows the frequency response of the moving average filter. It is mathematically described by the Fourier transform of the rectangular pulse given by:

$$H[f] = \frac{\sin(\pi f M)}{M \sin(\pi f)} \tag{4.4.13}$$

The roll-off is very slow and the stop band attenuation is ghastly. Clearly, the moving average filter cannot separate one band of frequencies from another.

In general, good performance in the time domain results in poor performance in the frequency domain, and viceversa. In short, the moving average is an good smoothing filter (the action in the time domain), but an bad low-pass filter (the action in the frequency domain). However the performance depends also on the specific application it is used for: if the problem is to obtain the quasi-static value of the signal, this filter gives good results.

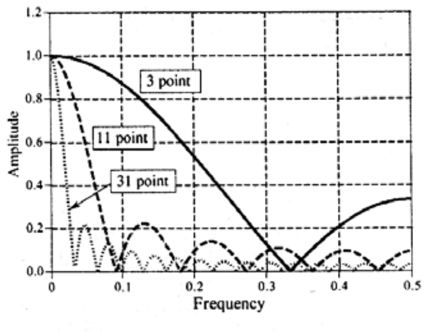


Figure 4.17: Some moving average filters in the frequency domain

Recursive formulation Equations (4.4.11) or (4.4.12) can be elaborate in order to obtain a recursive formula suitable to be implemented in an on-line filter. As a first step, equation (4.4.11) is rewritten in order to have a formulation that needs only old values of x for calculating $\bar{y}$:

$$\bar{y}_i = \frac{1}{N} \sum_{j=i-N+1}^{i} x_j \tag{4.4.14}$$

Then, the same equation is for the time instant $i-1$:

$$\bar{y}_{i-1} = \frac{1}{N} \sum_{j=i-N}^{i-1} x_j \tag{4.4.15}$$

By subtracting (4.4.15) to (4.4.12) it can be obtained:

$$\bar{y}_i = \frac{1}{N}[x_i - x_{i-N}] + \bar{y}_{i-1} \tag{4.4.16}$$

This is a recursive solution in which the actual calculated value depends on a previously calculated one. This formulation is also cheaper then the previous ones, because at each step it needs only one addition, one subtraction and a multiplication for a constant. In practice, the moving window of N values at each step sums the new value, subtracts the value that fall out of the window and uses the mean value as calculated in the previous step. The described method has two main drawbacks:

- the filtering cannot be initiated reliably until N measurements have been made;
- it is necessary to store the value x_{i-N} which, depending on the way the algorithm is coded, may require up to N storage locations.

This technique places equal emphasis on all data points. Thus a value in the past will have the same influence as a more current measurement when calculating the filtered signal. This may be a desirable feature when the mean value of the measurement is almost constant, but not when the signal has a drift.

Practical implementation From a practical point of view, equation (4.4.16) can be implemented into an on-line procedure. A sketch of the program flowchart is illustrated in figure (4.18).

This procedure is described with more details into algorithm (9).

Algorithm 9 Moving Average Method Subroutine

Require: N
  if $i < N$ **then**
    acquire x
    $x_{tot} \Leftarrow x_{tot} + x$
    $\hat{y} \Leftarrow x_{tot}/(i+1)$
    $\bar{y}_{old} \Leftarrow \hat{y}$
    $x_b[i] \Leftarrow x$
    $i \Leftarrow i + 1$
  else
    $\bar{y} \Leftarrow \bar{y}_{old} + 1/N * (x - x_b[0])$
    for $k = 0$ to $N - 2$ **do**
      $x_b[k] \Leftarrow x_b[k+1]$
    end for
    $x_b[N-1] \Leftarrow x$
    $\bar{y}_{old} = \bar{y}$
  end if

It can be seen clearly that for the first N times in which the subroutine is called, no $\bar{y}$ value is given back as output, because the filter is in an initialisation phase. After this initialisation phase, the filter becomes really operative and the mean value of the previous measured N points is returned at each call.

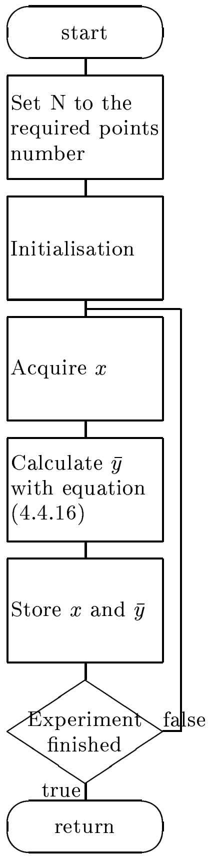


Figure 4.18: Flowchart of the moving average method subroutine

Offset compensation The above-described method is usually used for noise reduction. High frequencies are cut down, so only low frequencies remain. The higher is the number of utilised points, the lower is the cutting frequency (see figure (4.17). If the signal has an offset value, a moving average filter will preserve it. If there is a very low frequency component, it will be recovered only if a sufficient large number of points will be used in the filter.

These properties can be usefully used to create a filter that is able to suppress a quasi-static offset or a very slow drift in the original signal. The above described algorithm can be in fact modified in order to subtract the obtained smooth value to the input signal. In this way the output signal will contain only the dynamics components of the original one.

This property is fundamental in civil engineering applications where static and quasi-static actions must be filtered in order to have only the effects of the dynamical loads (earthquake, wind or traffic) on the structure. This is the case, for example, of cable-stayed bridges controlled by active tendons (Marazzi et al. 2000; Aupérin et al. 2001). The tension in each cable can be divided into two contributions: one related to self-weight and permanent loads, the other one related to traffic or wind deck excitation. During the design phase, a proper tension value is chosen for the first loading condition. As regards the dynamic part of the tension, control techniques can be used to mitigate its effect, but the static component of this force must not be considered by the control algorithm.

Programming the on-line filter into the real-time controller The insertion of the filter into the controller routine implies a higher level of complexity than to run it in a off-line sequential program because of the variables exchange among the various routines. The following aspect has been taken into consideration:

- there should be the possibility of starting the filter function before switching on the control low;
- there should be the possibility of observing on the screen the filtered function before switching on the controller in order to check if everything is working well;
- there should be the possibility to interactively change the number of used points while the program is running. This means that:
  - the necessary variable must be cleared and a proper amount of memory should be allocated;
  - the filter must be re-initialised;
  - the controlled must be stopped until the filter has been initialised again.

The Moving Average Method Subroutine as it was programmed in language C++ is presented in algorithm (10).

4.4.1.3 The Exponential Function Moving Average Method

Introduction In dynamic systems, the most current values tend to reflect better the state of the process. A filter that places more emphasis on the most recent data would therefore be more useful. Considering again the case of the cable-stayed bridge described in the previous paragraph, it can be observed that quasi-static

Algorithm 10 Moving Average Method Programmed Subroutine

```
float fFiltroMobile(float fx)
{
      float fy;
      fauxn = fauxv + finvn * (fx - fxb[0]);
      for (int k = 0; k < n - int(1); k++)
      {
            fxb[k] = fxb[k+1];
      }
      fxb[n-int(1)]=fx;
      fy=fx-fauxn;
      fauxv=fauxn;
      fmedia=fauxn;
      return fy;
}
```

forces on the cables can be generated by the alternation of day and night. When the sun heats them, they can be subject to lower forces values than when they are not heated or viceversa. If the mean is calculated with a large number of acquisition points or if the time interval between one measure and the other is long, the real dynamic component of the force could be better catch by giving more importance to the most recent value.

Recursive formulation Recalling the equation (4.4.12), the same expression of the mean but for one more additional point can be written as:

$$\bar{y}_{i+1} = \frac{1}{N} \sum_{j=i-N+1}^{i+1} x_j \tag{4.4.17}$$

Since $\sum_{j=i-N+1}^{i} x_j = n\bar{y}_i$, therefore:

$$\bar{y}_{i+1} = 1/(n+1)\,[y_{i+1} + n\bar{y}_i] \tag{4.4.18}$$

By shifting the time index back one time-step, the following expression for x_i is obtained:

$$\bar{y}_i = \left(\frac{1}{n+1}\right) x_i + \left(\frac{n}{n+1}\right) \bar{y}_{i-1} \tag{4.4.19}$$

To simplify the notation the factor $\alpha = n/(n+1)$ is introduced, so the (4.4.19) becomes:

$$\bar{y}_i = \alpha x_i + (1 - \alpha)\bar{y}_{i-1} \tag{4.4.20}$$

It must be notice that, similarly to the moving average filter, the amount of operation is really low and consists in a subtraction, an addition and two multiplications. One difference with the previous method is that in equation (4.4.20) there is no longer need for the storage of the N old values, since only the actual value x_i and the previous mean $\bar{y}_{i-1}$ are used.

The value of the filter constant, α, dictates the degree of filtering, i.e. how strong the filtering action will be. Since $n \geq 0$, this means that $0 \leq \alpha < 1$. When a large number of points are being considered, $\alpha \to 1$, and $\bar{y}_i \to \bar{y}_{i+1}$. This means that the degree of filtering is so great that the measurement does not play a part in the calculation of the average. On the other extreme, if $n \to 0$, then $\bar{y}_i \to x_i$, which means that no filtering is being performed.

If this filter is applied to an incoming series of data, the relation among the actual value of the mean and one old one is given by:

$$\bar{y}_i = \alpha^k \bar{y}_{i-k} + \alpha^{k-1}(1-\alpha)x_{i-k+1} + \cdots + \alpha(1-\alpha)x_{i-1} + (1+\alpha)x_i \tag{4.4.21}$$

It can be noticed that the contribution of older values of x_i are weighted by increasing powers of α. Since $\alpha < 1$, the contribution of the older value old x_i becomes progressively smaller.

Equivalence between the Exponential Function Moving Average Method and the 1^{st} order low-pass filter Consider the Laplace transfer function of a first-order low-pass filter, with time constant τ_f:

$$\frac{\bar{x}(s)}{x(s)} = \frac{1}{1 + \tau_f s} \tag{4.4.22}$$

which relates the filtered signal $\bar{x}(s)$ to the measurement signal $x(s)$. This has the following time domain equivalent[5]:

$$\tau_f \frac{d\bar{x}(t)}{dt} + \bar{x}(t) = x(t) \tag{4.4.23}$$

This differential equation can be discretised using the approximation:

$$\frac{d\bar{x}(t)}{dt} \approx \frac{\bar{x}_k - \bar{x}_{k-1}}{T_s} \tag{4.4.24}$$

[5]This differential equation can also be used to describe the input and the output behaviour of an electrical RC-circuit.

where T_s is the interval between each measurement, i.e. the sampling interval. Thus the differential equation representing the first-order low-pass filter is converted to:

$$\tau_f \frac{\bar{x}_k - \bar{x}_{k-1}}{T_s} + \bar{x}_k = x_k \tag{4.4.25}$$

Simplification and re-arrangement gives:

$$\bar{x}_k = \left(\frac{\tau_f}{\tau_f + T_s}\right)\bar{x}_{k-1} + \left(\frac{T_f}{\tau_f + T_s}\right)x_k \tag{4.4.26}$$

By letting:

$$\alpha = \left(\frac{\tau_f}{\tau_f + T_s}\right) \tag{4.4.27}$$

it can be seen that

$$(1-\alpha) = \left(\frac{T_f}{\tau_f + T_s}\right) \tag{4.4.28}$$

and then

$$\bar{x}_k = \alpha\bar{x}_{k-1} + (1-\alpha)x_k \tag{4.4.29}$$

which is identical to the exponentially weighted moving average filter (4.4.20).

The relation between n and τ_f is very simple: $\tau_f = nT_s$. With this relation, criteria given for first-order low-pass filter choice of parameters can be used also for the exponentially weighted moving average.

The Exponential Function Moving Average Method Subroutine as it was programmed in language C++ is presented in algorithm (11), where `fcomplalpa` is $1-\alpha$.

Algorithm 11 Exponential Function Moving Average Method Programmed Subroutine

```
float fFiltroMobilePesato(float fx)
{
      float fy;
            fauxn = falfa * fauxv + fcomplalfa * fx;
            fy=fx-fauxn;
            fauxv=fauxn;
      return fy;
}
```

4.4.2 The control algorithm software

4.4.2.1 The IFF strategy

The direct velocity feedback ($F = -g\dot{u}$) is an example of "energy absorbing" control: this control law is guaranteed to remove energy from the structure. When using a displacement actuator collocated with a force sensor, the (positive) Integral Force Feedback

$$u = g \int T dt \tag{4.4.30}$$

also belongs to this class, because the power flow from the control system is always negative: $\mathcal{W} = -T\dot{u} = -gT^2 < 0$ ($\mathcal{W}$ is the power flow into the structure, T is the measured force, u is the imposed displacement, $\dot{u}$ the velocity and g the control gain). This control law applies to nonlinear structures (Bossens et al. 2001; Marazzi et al. 2000); all the states that are controllable and observable are asymptotically stable for any value of g (infinite gain margin) (Preumont 1997).

The IFF control strategy was also selected here because of its effectiveness demonstrated in previous projects on active control of cable-stayed bridge (Aupérin et al. 2001): in that case, even if the non-linearity of the cable-deck system were very high, it has proven to be stable and well suited to damp externally induced vibrations.

The Integral Force Feedback Subroutine as it was programmed in language C++ is presented in algorithm (12). The implemented procedure has also a safety check on the maximum command signal that can be sent to the actuators in order to prevent their saturation.

4.4.2.2 The main control algorithm program

Without entering into details relative to the refresh of the parameters values on the screen and on the internal name of the various variable, the only general meaning of the program will be explained.

The main control algorithm is principally devoted to the management of the overall control process. It can be divided into 4 parts:

1. declaration of all the needed variables;
2. initialisation of the filters;
3. choice of the filter among the three types previously described;
4. switch on/off of the control algorithm and call of the IFF subroutine.

This Main Algorithm as it was programmed in language C++ is presented in algorithm (13) in which subroutine `flowfilter` was added in order a low pass filter on the displacement command value.

Algorithm 12 Integral Force Feedback Subroutine

```
void fIFF01(float fforceV, float *fvelV, float *fdispV)
{
      // converts the force from Volts to Newtons
      fforceN = ffattN * fforceV;
      // speed output force feedback
      fvelmps = (float)g_dAlgoParam2 * fforceN;
      // speed including centering filter
      fvelmps = fvelmps - (float)g_dAlgoParam10 * folddispm;
      // converts the velocity from meters/second to Volts
      *fvelV = ffattvel * fvelmps;
      // integrates the velocity and sums the old displacement
      fdispm = fvelmps * fdelta  + folddispm;
      // converts the displacement from meters to a Volts
      *fdispV = ffattD * fdispm;
      // security on the max voltage to be sent to actuators
      if (*fdispV > fvoltmax)
      {
            *fdispV = fvoltmax;
      }
      if (*fdispV < - fvoltmax)
      {
            *fdispV = - fvoltmax;
      }
      folddispm = fdispm;
}
```

4.4.3 Debugging phase

The main program and all the subroutines were intensively tested on the controller. A signal composed by the sum of various frequencies with different amplitudes and phases was applied with the generator described in § (4.3.3.2) and the coming output were recorded. The analysis of these data showed the effectiveness of the filter functions and of the IFF control strategy.

The computation time for filters was also observed on an oscilloscope in order to use the maximum number of points for mean calculation without affecting the overall control process duration.

Algorithm 13 Main Algorithm Control Program

```
#include "conio.h"
#include "math.h"
#include "stdio.h"

// variable initialisation //
int   n;
float finvn;
float fdelta=(float) 0.001,fgain=(float)2,fpi=(float) 3.14;
float ffreq=(float) 50;
float falfa = 0;
float fcomplalfa =0;
static float folddispm=0;
static float fy1,fy2,fy3,fy4,fx1,fx2,fx3,fx4;
static float *fxb = NULL;
static float fauxn,fauxv,fxtot,fmedia;
static float fxcompold;
float ffattvel = (float) 20;  // 1 m/s --> 20 V
float ffattD = (float) 333;   // 10 mm --> 3.3 V, so 1 m --> 333 V
float ffattN = (float) 25000; // 1 V --> 25 kN, so   1 V --> 25000 N
float fxcylcentr= (float) 0;
float g_dAlgoParam12finto = (float) 0.01, g_dAlgoParam13finto;
float fforceN,fdispV,fdispm,fvelV,fvelmps;
float fvoltmax= (float) 8.33;  // this corresponds to 25 mm
float fxcomp,fcorrettivo,fxcompV;
static int i=0;
float ff1,ff2,fnu;
float fspostactive,fstroke;
```

```
void Algorithm()
{
      float fResult = 0;
      static int nOld = 0;
      static int nAlgoOld = 0;

      if (g_dAlgoParam4 != nOld || g_dAlgoParam3 != nAlgoOld)
      {
            nOld = n = (int)g_dAlgoParam4;
            nAlgoOld = (int)g_dAlgoParam3;
            i = 0;
      }

      if (i == 0)
      {
            if (fxb != NULL) delete fxb;
            fxb = new float[n];

            fxtot = (float) 0;
            // deve essere ]0,1[
            falfa = (float) n / (float) (n + 1);
            fcomplalfa = (float) (1 - falfa);
            finvn = float(1) / float(n);
      }

      switch((int)g_dAlgoParam3)
      {
      case 1: // filter based on the                          //
              // Moving Average Method Exponentially Weighted //
            if (i<n)
            {
                  fxtot = g_fForce1+fxtot;
                  fauxv = (float) fxtot / (float) (i + int (1));
            }
            else
            {
                  fResult = fFiltroMobilePesato(g_fForce1);
            }
            break;

      case 2: // filter based on the Moving Average Method //
```

```
            if (i<n)
            {
                  fxtot = g_fForce1+fxtot;
                  fauxv = (float) fxtot / (float) (i + int (1));
                  fmedia=fauxv;
                  fxb[i] = g_fForce1;
            }
            else
            {
                  fResult = fFiltroMobile(g_fForce1);
            }
            break;

      case 3: // filter based on the Transfer Function Method //
            fResult = fFiltroBanda(g_fForce1);
            break;
      }
      i++;

      *g_pfInternalAlgoOutput3 = fResult;
      g_fDac2 = fResult;

      if (g_cRunPause == C_RUN)
      {
            switch((int)g_dAlgoParam1)
            {
                  case 1: // the control is inserted
                  fIFF01(fResult,&fvelV,&fdispV);
                  fcorrettivo = flowfilter(g_fForce2);
                  // g_fDac1 is the velocity command signal
                  g_fDac1=fvelV;
                  // g_fDac2 is the displacement command signal
                  g_fDac2=fdispV+fcorrettivo;
                  break;
            }
      }
}

float flowfilter(float fxcylV)
{
      // (float)g_dAlgoParam11 = switch
```

```
    // (float)g_dAlgoParam12 = omega3
    float fy;
    fxcomp = (float)g_dAlgoParam11 * (float)g_dAlgoParam12finto *
    * (fxcylcentr - (float)fxcylV/ffattD) * fdelta + fxcompold;

    // security on the max offset
    if (fxcomp > 0.01)
    {
        fxcomp = (float) 0.01;
    }
    if (fxcomp < - 0.01)
    {
        fxcomp = - (float) 0.01;
    }

    fxcompold = fxcomp * (float)g_dAlgoParam11;
    fxcompV = fxcompold*ffattD;
    fy = fxcompV;
    return fy;
}
```

Chapter 5

Two experimental case studies

5.1 Introduction

Two experimental case studies are presented in this chapter together with the related developments, such as the necessary characterisation of different kind of devices to be applied to these two structures, the ancillary simulations that has been conducted to investigate the expected behaviour of the systems (both controlled and uncontrolled) and the main results coming from the testing campaign.

5.2 Devices characterisation

Devices characterisation is a preliminary but very important step. It is necessary to obtain meaningful data to be used for the numerical simulations of the device and for properly modelling the device behaviour when it is inserted into the structure to be protected against vibrations. Semi-active devices are inherently non-linear, so several tests are needed to properly capture their principal characteristics.

A special testing equipment has been developed at ELSA to properly identify the components and the materials (figure (5.1)). It is composed by an hydraulic actuator with maximum force 150 kN (15 tons) and maximum stroke 20 cm that can be controlled either in force or displacement. The required signal set value is given to the control algorithm that follows this signal as the reference one. The needed signal can be generated with any program that permits exportation of data in binary double precision format. A special option permits the undefined repetition of the files in order to easily perform fatigue testing.

If the reference signal is the displacement (this is the most common case for devices characterisation), the corresponding force is measured at the same time. This allows plotting Force vs. Displacement graphics from which the energy dissipation

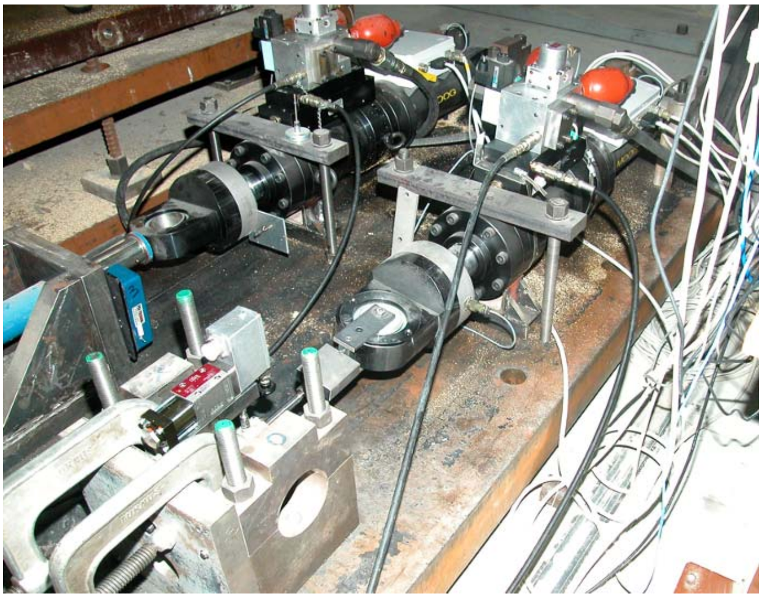


Figure 5.1: Experimental set-up for devices and components characterisation

or non-linear behaviour can be detached. Since the control program is very general, the entire internal variables used by the controller can be recorded, but this option is mainly used for debugging purposes, because for device characterisation these data are not usually needed.

5.2.1 Magnetorheological devices

As a first choice, the LORD RD-1005 semi-active devices (see appendix (A.3)) were chosen because they can be find easily on the market and are common used in automotive applications. These devices are based on a magnetorheological (MR) fluid that can change its viscosity very quickly by applying a current inducing a magnetic field. A complete review of this technology can be found in (François et al. 2000). In order to proper modelling the semi-active device behaviour, an experimental campaign has been performed at ELSA. Using a general testing machine developed in the laboratory to test damping devices, several cyclic tests were conducted with different speeds and three current supplies. The imposed displacement was a sinusoid with initial maximum amplitude of ±10 mm, then slowly decreasing to zero value.

In principle, MR devices are expected to have increasing hysteretic cycles at increasing speeds and currents supply. However it is well known that this behaviour is neither linear nor simple to describe. Figure (5.2) shows the characterization curves of the device at different speeds and without current supply. A λ value of 300 (the slowest) corresponds to an excitation frequency of 0.0067 Hz while a λ value of 10 (the fastest) corresponds to 0.2 Hz.

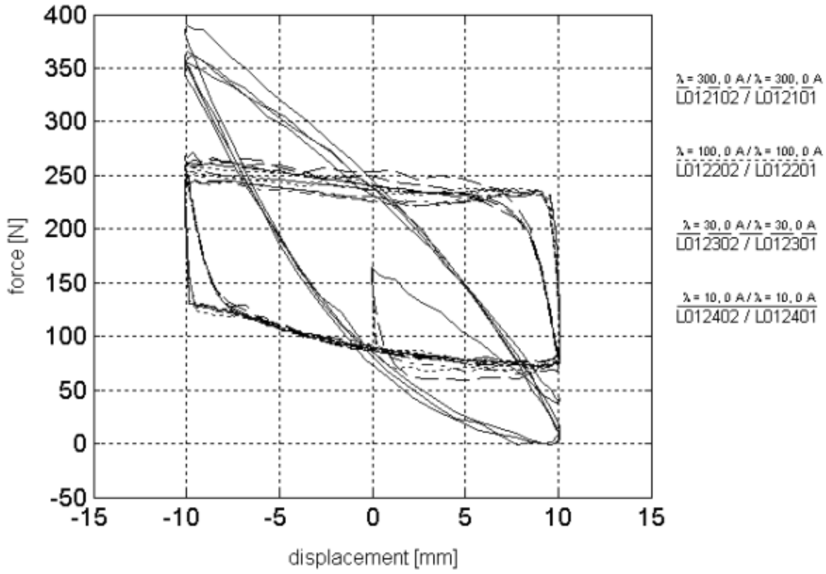


Figure 5.2: LORD (type RD-1005) characterisation (current = 0 A)

As clearly shown, the MR fluids behave in a different way at low and high speed. At low speed efficiency of the dissipation cycle is extremely high: this is due to the fact that MR chains are constituted more rapidly than the disrupting due to sliding. At a sufficient speed the classical viscous behaviour is obtained.

Figure (5.3) shows the characterisation curves with a current supply of 1 A. Also in this case the efficiency of the device is better at low speeds, but now the transition between the two behaviours is more graduated than in the previous case. The shape of the curves for $\lambda = 300$ and for $\lambda = 10$ are very different.

Unfortunately, in a real application (for example earthquake mitigation or wind induced vibration reduction) it is easier to fall in the high range of speeds, i.e. in the range where dissipation is lower. However the hysteretic cycle are still good enough to ensure good dissipation behaviour.

A very open problem for this kind of devices is sedimentation: if the MR liquid remains inactive for several days, the solid part deposits on the bottom of the

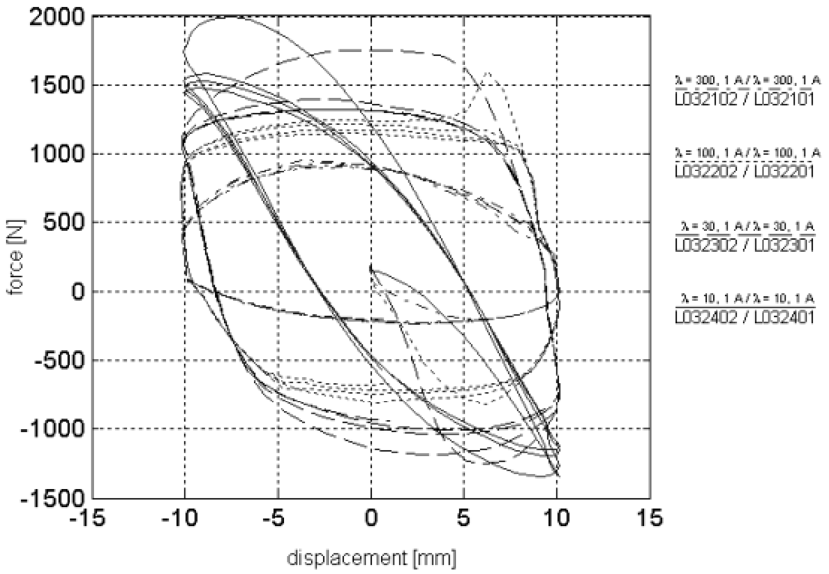


Figure 5.3: LORD (type RD-1005) characterisation (current = 1 A)

device, thus altering completely the MR device behaviour. Several researchers are trying to solve this problem using specials foams, but at present no definitive solution has yet been given.

5.2.2 Variable orifice damping device

5.2.2.1 Description of the considered device

Variable valve devices are currently considered among the most promising devices for semi-active control laws implementations in civil engineering. They are quite simple of construction (they are very similar to passive viscous dampers, which are a mature technology) and very reliable.

The working principle is simple: the oil passes from one chamber of the piston to the other thought a variable orifice placed on a by-pass (see figure (5.4)). This kind of devices must present a very law internal friction, but this is not a problem, because some oil should pass from one chamber to the other (inter-chamber leakage) to allow energy dissipation also when the by-pass is closed and to prevent its disrupting. The opening of the valve can be adjusted via an electronic command.

If the valve is completely closed, the oil passes from one chamber to the other thought the small gap due to the difference in diameter of the rod head and the internal diameter of the device and the device provides the maximum damping coefficient. If the valve is completely open, the damping can virtually be reduced to zero if the by-pass is large enough. In the intermediate positions the device produces a specific damping dissipation.

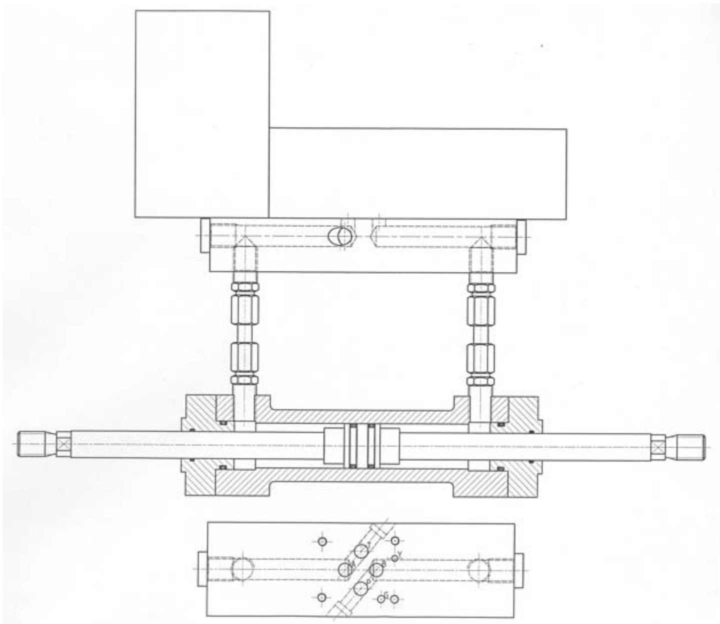


Figure 5.4: Drawing of the variable valve damping device

A variable damping device was developed at ELSA (figure (5.4)) to be finally mounted on a semi-active Tuned Mass Damper. A proper characterisation was needed to evaluate the damping and stiffness properties under different working conditions (stroke of the rod, type of excitation, opening of the valve). Since the opening of the valve is not directly correlated to the damping coefficient, the relation between these two quantities is needed. Figure (5.5) shows the variable valve device mounted on the characterisation machine.

A very accurate load cell ($\pm$5000N) is put at the interface between the semi-active device and the head of the testing actuator. The normally used load cell was too big for this application: its end-scale is 150 kN, so it is very noisy if a 5-10 kN range is measured. The second end of the device is fixed on a plate. The displacement is measured either by a Temposonic transducer (magnetic) and a Heidenhein one (optical). The function generator described in (4.3.3.2) for the controller can be used here also to provide any kind of signal.

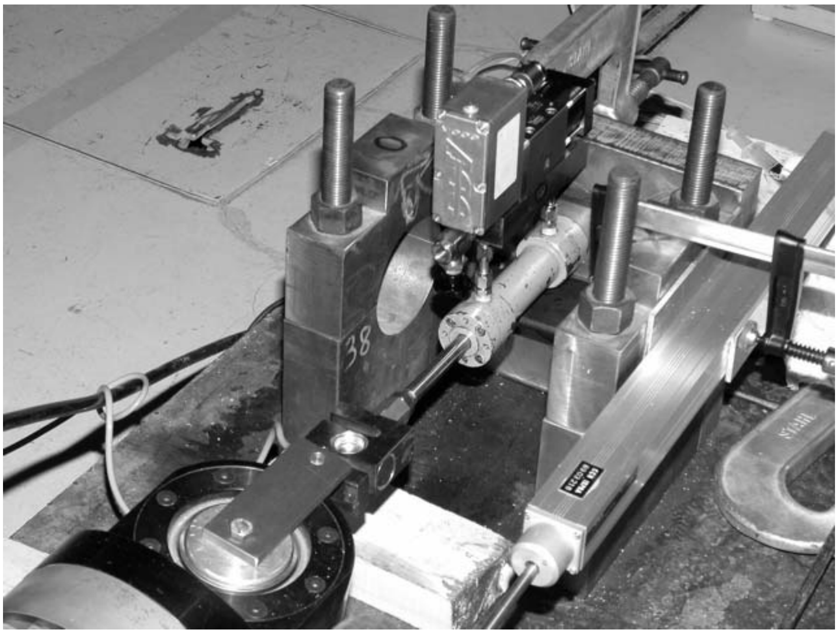


Figure 5.5: Variable valve damping device mounted on the characterisation machine

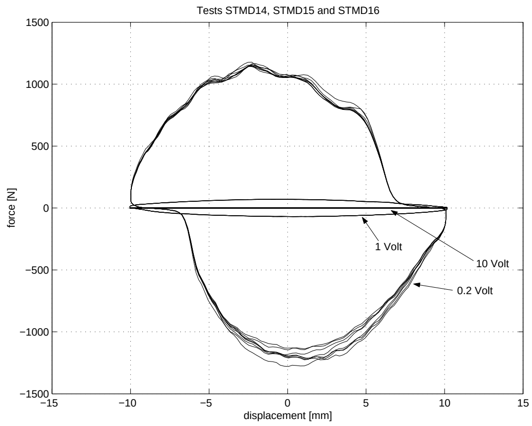
All the care was taken in avoiding any play among the connections: an ad hoc special fixing was designed. Nevertheless a little play cannot be avoided together with some insignificant disturbance in the measured displacement due to the flexibility of the S shaped load cell. The electronic valve is controlled “by hand” with a voltage generator connected to it.

5.2.2.2 Characterisation results

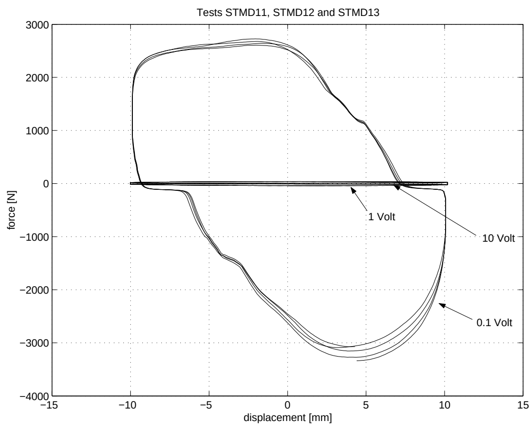
The main results are here summarised very briefly. During the testing campaign the device was subject to sinusoidal movements of $\pm$ 10 cm at frequency of 0.1, 1 and 2 Hz and with different positions of the orifice valve: completely open, completely closed and some other positions in between these two extremes.

Figure (5.6) shows some dissipative cycles given by the device for different exciting frequency and opening of the valve.

It can be clearly seen that a supplied voltage between 1 and 10 V doesn't produce appreciable different results: the valve is almost completely open and the oil, at the tested frequency, can flow without resistance inside the by-pass. This



(a) test performed at 2 Hz



(b) test performed at 1 Hz

Figure 5.6: Force vs. Displacement dissipative cycles at different exciting frequency and opening of the electric valve

happens because the valve was quite over-dimensioned in size. In this case the damping given by the device is very low. When the applied signal is 0.1 or 0.2 V, the oil flow through the valve is more difficult and the force grows. In this case the dissipative cycle is quite big.

The described behaviour can be used in developing a semi-active tuned mass damper in which the damping device can be controlled with an on-off strategy with very simple hardware equipment. In the case of continuous control strategy, the damping factor can be chosen to be any value between these two extreme values (as for example in the clipped optimal control), (Pinkaew and Fujino 2001) have demonstrated a substantial improvement of the steady-state amplitude reduction in the structure response with a semi-active tuned mass damper compared to a conventional tuned mass damper.

5.2.3 Hydraulic semi-active devices

5.2.3.1 Description of the considered device

Wind, earthquakes and traffic insert energy into the structures: this energy must be taken out in some way. Conventional designed buildings dissipate at some particular locations: the energy consumption is then strictly related to damage. Passively protected structures dissipate energy into special devices, the dissipator, in which energy is transformed into heat, so preserving the remaining part of the structure. In either this two cases the environmental energy flowed into the structure is lost.

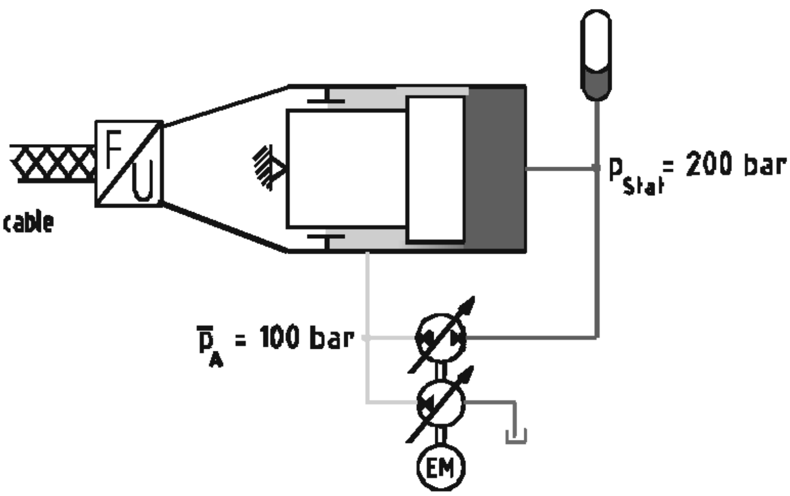


Figure 5.7: Scheme of the semi-active hydraulic system developed by Rexroth

A very promising concept in structural control envisages energy-transformation systems, the vibration energy is transformed in a more or less useable energy.

A transformation of the vibration energy into hydraulic, mechanical or electrical form avoids the heat problems due to conventional energy dissipation and allows the storage and use of this recovered energy. The usable energy is always smaller than the extracted vibration energy because energy transformation implies some losses and the control electronic and mechanic need some energy. However, especially for larger amounts of induced vibration energy acting over a long time, this concept is much more useful than purely dissipating systems.

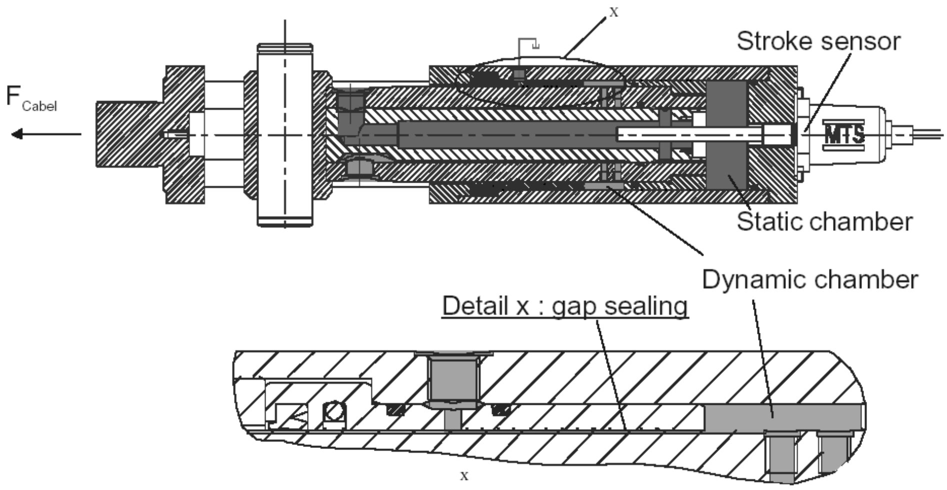


Figure 5.8: Drawing of the semi-active hydraulic actuator

Following these concepts, a semi-active hydraulic device[1] has been developed by Rexroth (Bosch Group). Figure (5.7) describes a scheme of this particular hydraulic system.

The connection pattern is the opposite of the usual one in hydraulic application, the backside been attached to the front one and viceversa. This scheme was found to be the best among some different other alternatives (see figure (5.7)).

This type of damper can be used for cable-stayed bridges, so it has to carry big static loads, whereas the dynamic forces are comparable small (in the order

[1]Following the definition given in § (1.2), this device should not be considered as semi-active, because in fact it is fully active. Because of the energy recuperation and of the reduced energy consumption, however, some authors call it "semi-active". In any case, a semi-active control law can be implemented with these actuators, so they were inserted in this section.

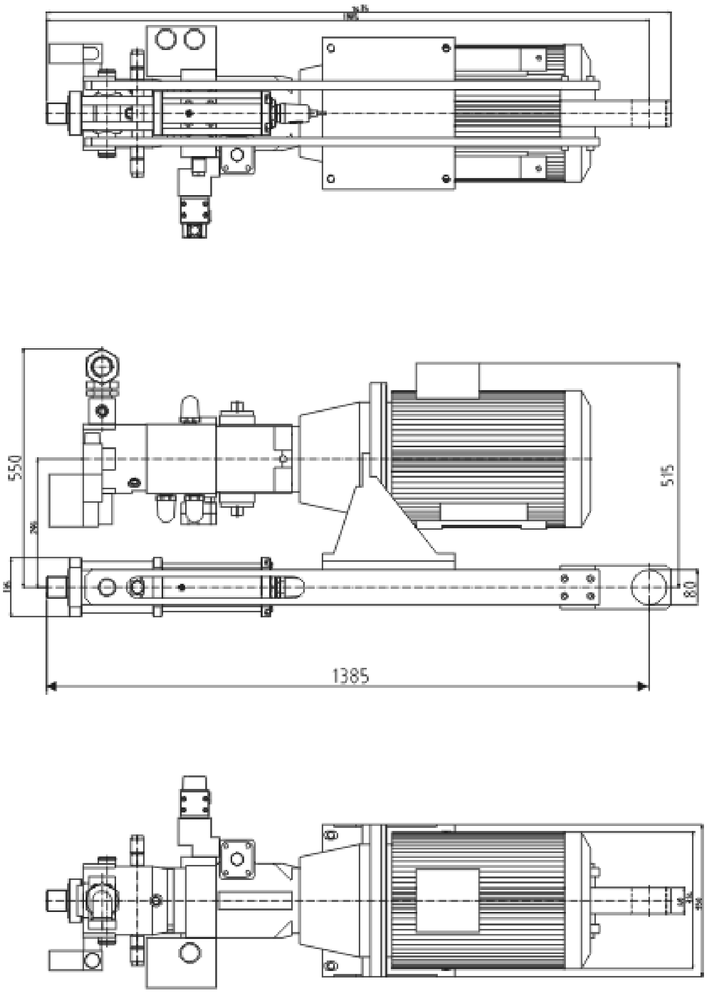


Figure 5.9: Semi-active main cable actuator for stayed-cable bridge mock-up

of magnitude of 10% of the static load). For this reason, the biggest part of the cylinder area is used to carry the static load.

The hydraulic circuit shown in figure (5.7) presents the static chamber of the actuator pressurised with about 200 bar[2] whereas the average pressure of dynamic

[2]Having in mind both the size of the actuator and the lifetime of the pump, a reasonable pressure level to drive the hydraulic actuators is about 200-230 bar.

piston diameter	80 mm
rod diameter	70 mm
stroke	60 mm
static force	105 kN @ 210 bar
dynamic force	12 kN @ 105 bar
operating pressure	210 bar
static testing pressure	350 bar

Table 5.1: Technical data of the semi-active hydraulic device developed for CaSCo designed by Rexroth

chamber is 100 bars only. This allows a “pressure stroke” of 200 bar (from 100 bar to 0 bar and from 100 bar to 200 bar during control). Using a double rod type actuator for the dynamic part of the cylinder both chambers has equal pressures (in average) and therefore the pressure forces are normally compensated.

Two pumps are used: one feeds oil from tank to the dynamic chamber at an average differential pressure of 100 bar; this pump needs energy input. The other pump is placed between the dynamic chamber and the static pressure accumulator, feeding oil at 200 bar to the dynamic level at 100 bar. Also in this case, there is a 100 bar acting differential pressure, but now in opposite direction. This means that mechanical energy is generated by this hydraulic motor.

If vibration forces act on the hydraulic damper, the additional energy input minus the frictional losses has to be compensated by the electric motor which works as a generator: the recovered energy can be feed into the electrical net.

Design details of the hydraulic actuators are shown in the assembly drawing of figure (5.8).

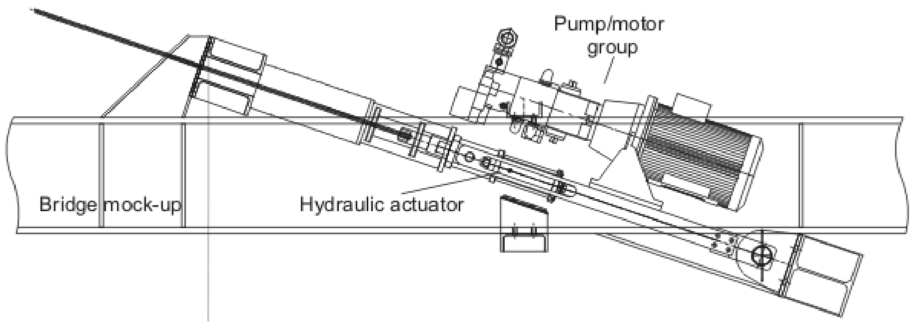


Figure 5.10: Installation of the semi-active actuator at the end of the stay

The hydraulic actuator has been integrated with a motor (figure (5.9)): doing so the installation into the CaSCo mock-up is much more easy (figure (5.10)).

In both figures the hydraulic actuator is just a small part of the whole device. The accumulator is not yet integrated into the device because in this developing stage it was better to have easy access to it at the ground floor level.

The technical data regarding the described devices can be found in table (5.1), while a picture of the system integrated into the cable-stayed bridge under testing is shown in figure (5.11). Reduced friction implies the opportunity to use pressure signals instead of force signals for the feedback of the IFF control law.

Figure (5.11) shows the semi-active device placed at the anchorage of the longest cable of the CaSCo cable-stayed bridge mock-up, tested at ELSA.

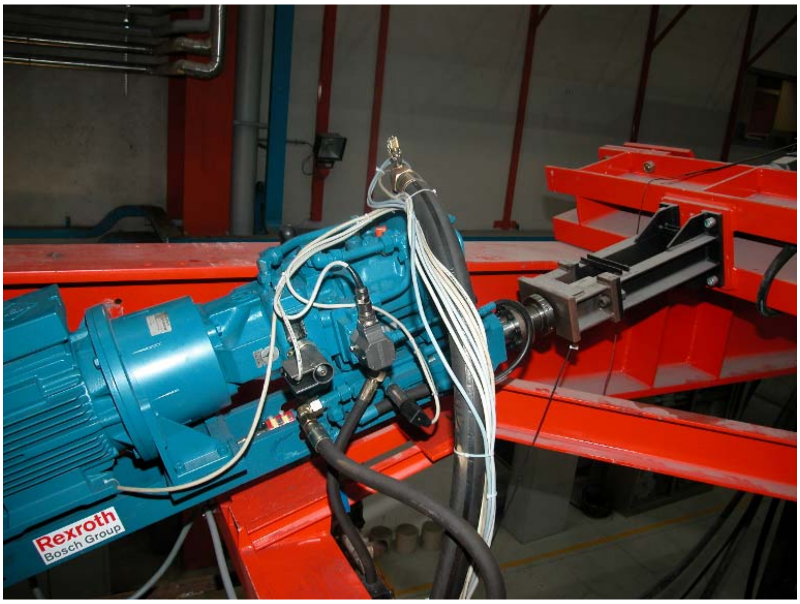


Figure 5.11: The semi-active device placed inside the CaSCo cable-stayed bridge mock-up

5.2.3.2 Characterisation results

Some characterisation results are reported in figure (5.12). Different parameters values have been tested. The amplitude response can be widely chosen by changing these parameters, while the phase shift response is almost insensible to these changes.

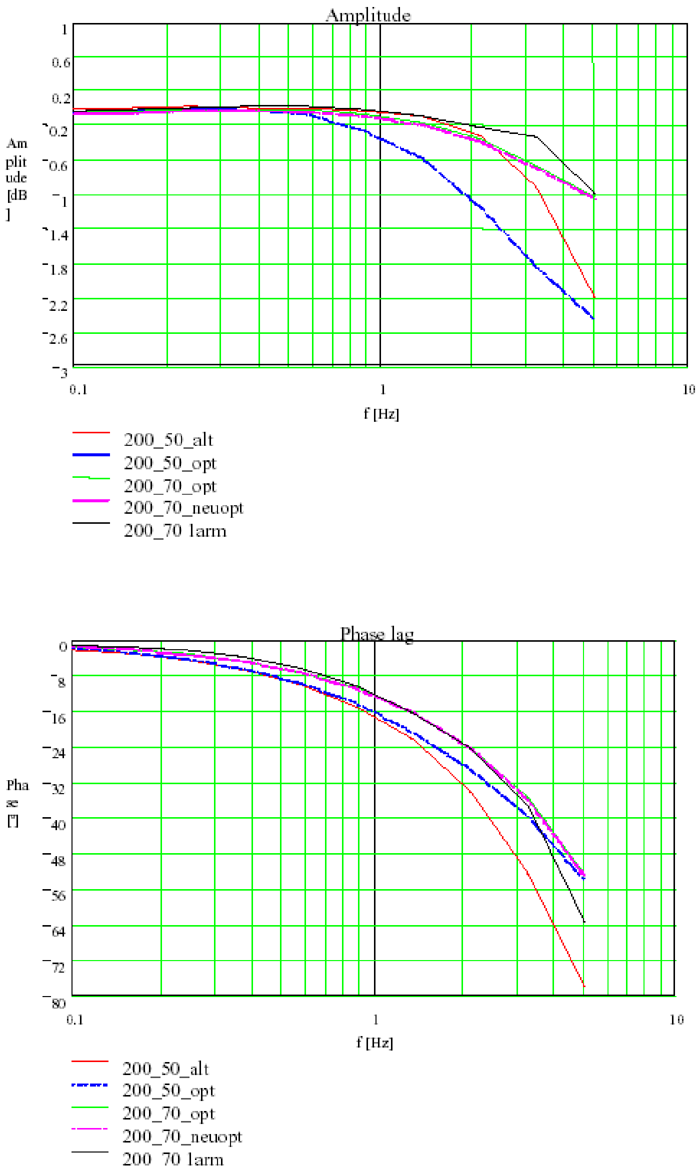


Figure 5.12: Characterisation results about the semi-active devices developed by Rexroth

5.3 Preliminary study on a SDOF system damped with MR devices

5.3.1 Introduction

A preliminary study about the effectiveness of the MR devices in vibrations damping has been conducted at ELSA.

The purpose of this preliminary study was:

- to evaluate the semi-active implementation of the IFF active control law for a ground isolation problem;
- to evaluate the Lord Corporation MR damper as a semi-active device.

A ground isolation problem consists in reducing the forces transmitted to the ground by an excitated moving mass through a supporting structure. The supporting structure behaves like a suspension, i.e.

1. the excitation force is well filtered beyond the corner frequency of the suspension (the force transmitted to the ground is low);
2. at the suspension resonance frequency, the force transmitted to the ground is more or less amplified according to the quality factor of the suspension.

In this experiment, the goal of the semi-active implementation of the IFF control law is to reduce the amplification at resonance while maintaining the original suspension properties beyond the corner frequency.

Even in a very simple case like a Single Degree of Freedom (SDOF) mass-spring-dashpot oscillator, several implementation aspects must be taken into account because they can heavily influence the response of the real system. Play among connections, non-linearity of the springs, torsional movements of the mass are examples of unwanted behaviours to be taken into consideration.

5.3.2 Physical model

The physical model under investigation is sketched in figure (5.13).

The mass is moving vertically and is connected to the ground with a spring and a dashpot with fixed characteristics plus another variable dashpot.

5.3.3 Design of the mock-up

In the practical configuration (figure (5.14)) a heavy squared plate is lifted upon four air-spring bearings at its corner. These special bearings have a variable stiffness accordingly with the inflation pressure. They have only little damping. A

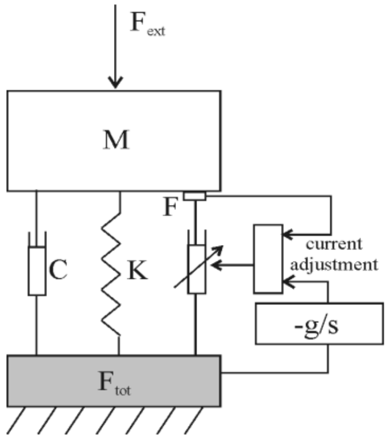


Figure 5.13: Sketch of the SDOF model equipped with a variable damping device

MR device (manufactured by Lord Corporation) is fixed in the middle of the plate. Four circular load cells were added under each air-spring to proper monitor the force flux thought the bearings, while a smaller load cell was added in series with the MR device to measure the force acting on the dissipator. Because the clipped IFF control law was used to optimally tune the damping properties of the device, only this last load cell was necessary from a control point of view, the other four being placed only for diagnostic purposes.

One real problem was to avoid lateral movements, because air-spring are not able to bear lateral forces. This aspect was partially solved by putting security stop-bars at some distance on each side of the plate, but torsional eigen-frequency of the plate has to be avoided in any case in order to prevent baring damages.

The frequency of the system was around 2.5 Hz. Table (5.2) summarised the mock-up characteristics.

5.3.4 Experimental model

The experimental model is showed in figure (5.15).

The SDOF system equipped with an electric shaker and a moving mass of 15 or 30 kg can be commanded by a signal generator to perform sinusoidal excitation or bandwidth noise. It could be controlled via a signal generator. Figure (5.16) shows the shaker installed on top of the set-up, the MR device locked between the mass, the load cell measuring its force and the air-springs installed at a corner of the thick iron slab. A LVDT transducer and a circular load cell mounted between the two ends of each air-spring isolator provide the vertical displacement and the

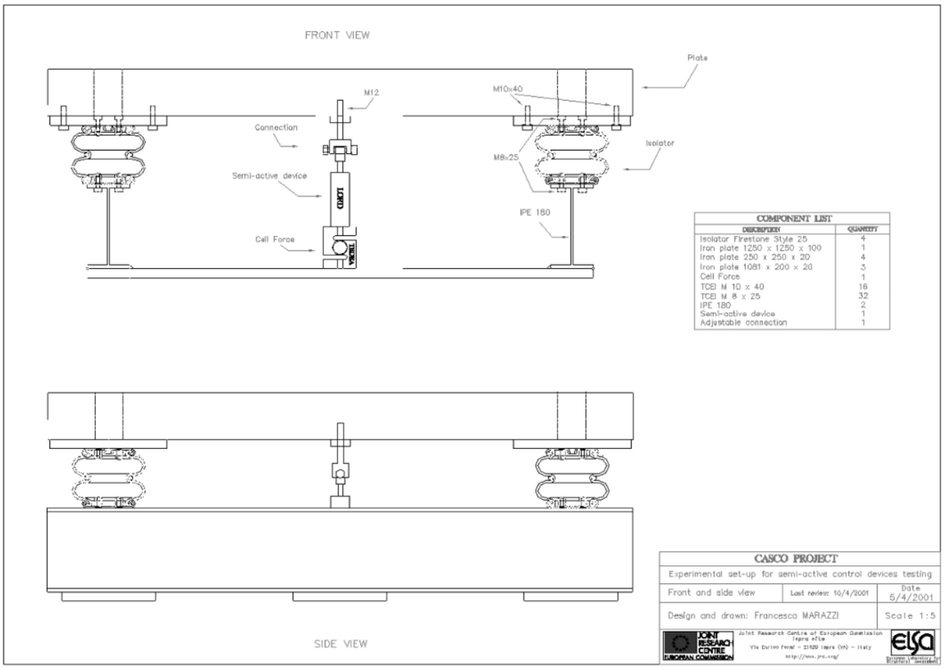


Figure 5.14: Drawing of the SDOF mock-up

force of the four corner of the slab. The sum of these four forces with the one on the damper, is the total force transmitted to the ground.

5.3.5 Tests and results

A testing campaign using sweep sine sinusoidal excitation waves forms were conducted to proper evaluate the performance of the semi-active MR device. Several tests with different constant level of applied current on the device were preliminary conducted. The same test was run also applying an IFF clipped control strategy (with two different level of maximum current admitted).

The results are summarised in figure (5.17) and (5.18).

Figure (5.17) shows the suspension transmission ($F_{ground}/F_{excitation}$) for several level at current excitations to the MR damper. The suspension exhibits amplification by a factor 10 at the resonance. A current of 200 mA is enough to introduce a large damping in the structure. The MR damper was thus quite oversized for this application. This results completely agree with the theory of passively damped structures (see figure (1.4)): a high damping level is good when

weight of the suspended mass	1700 kg
Firestone Airmount Isolator type 25	total stiffness 325 kN/m
MR damper	Lord Corp model RD-1005
shaker moving mass accelerometer	Analog Device ADXL05 (500 mV/g)
shaker model	APS Model 400
shaker moving mass	30 kg
supporting load cells	Tedlea Huntleigh, Model 620-5000 N/V
Lord damper load cell	1000 N/V
mass vertical displacement LVDT	5×10^{-3} m/V
control hardware	dSPACE DS1103

Table 5.2: Mock-up technical data

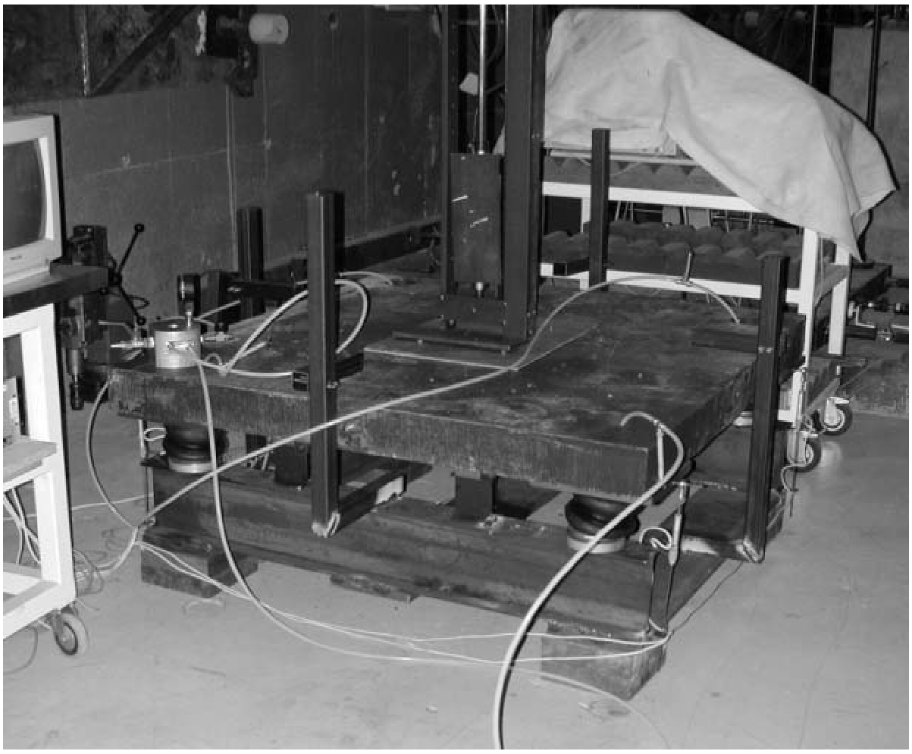
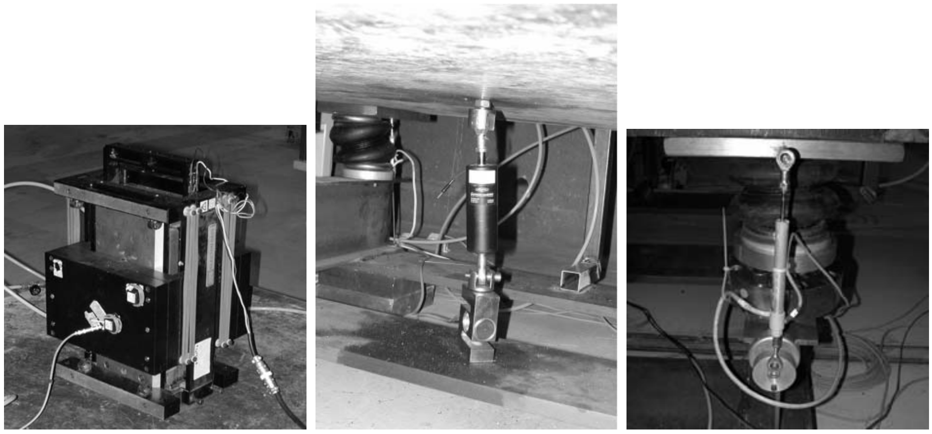


Figure 5.15: SDOF experimental mock-up



(a) shaker (b) MR device (c) air spring

Figure 5.16: Particulars of the SDOF mock-up

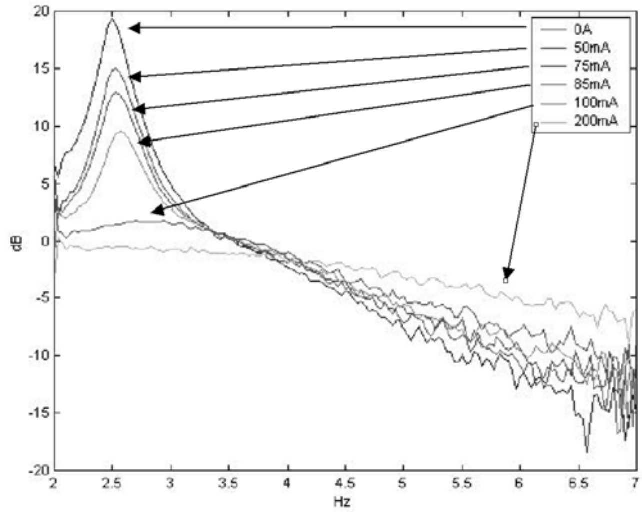


Figure 5.17: Transfer function between the exciting force $F_{excitation}$ and the sum of the transmission forces F_{ground}: passive behaviour (constant currents)

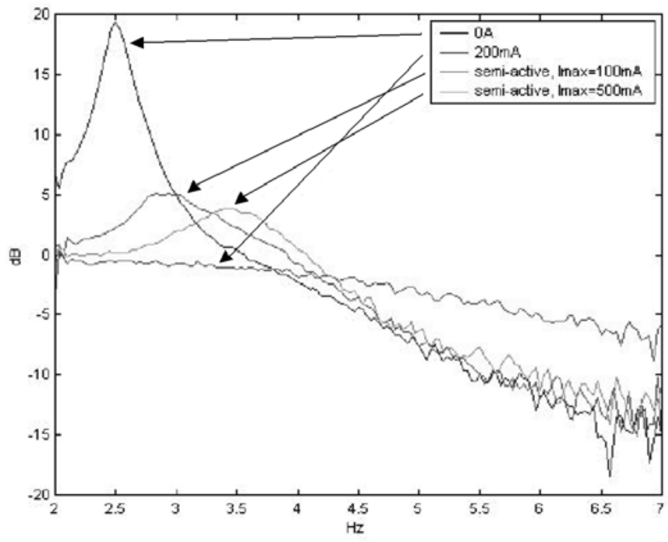


Figure 5.18: Transfer function between the exciting force $F_{excitation}$ and the sum of the transmission forces F_{ground}: passive and semi-active behaviour

the main aim is the reduction of resonance peak response. A high damping leads to a bad behaviour at high frequencies: high frequencies are better cut off with low damping.

Figure (5.18) shows the suspension transmission ($F_{ground}/F_{excitation}$) for the free suspension ($I_{damper} = 0$ A), for a constant current in the MR damper (I=200 mA) and for two semi-active control tests. For the semi-active control, only the max current in the damper differs.

Figure (5.18) shows that the semi-active control provide encouraging results. The transfer function peak is reduced from 20 to 5 ($I_{max} = 100$ mA) and to 20 to 4 ($I_{max} = 500$ mA). In the meantime, the high frequency behaviour is excellent: the transfer function can be compared to one obtained with very law damping. Nevertheless it is clear that the semi-active control strategy did not work as expected from the theory. The peak response should decrease with increasing values of I_{max}. On the contrary, this phenomenon cannot be observed, but figure (5.18) shows that a shift towards the high frequencies for increasing values of I_{max} is obtained.

The reason can be searched in some problems occurred during tests. These problems were probably connected with the transient phenomenon due to filtering (high-pass and low-pass) of the load cells signals:

1. the calculation of the needed forces in the active control law (using an integrator) requires to remove the load cells bias: this leads to a little time-delay;
2. the dynamic forces involved in the experiment are much less important than the static forces, the sensitivity of the load cells was not well adapted to the ADC converters of the DSP board.

Further investigations to completely explain these last results are still ongoing. These assumptions should also be verified with proper simulations.

5.4 The Baby-Frame test structure

5.4.1 Description

The Baby-Frame structure (figure (5.19)) consists in a steel frame with floors constituted by sheet metal and concrete properly connected.

Figure 5.19: The Baby-Frame structure in front of the Reaction Wall at ELSA (without energy dissipators)

It has three storeys and two bays. The scale is 2/3 of a real structure, so the inter-storey high is 2 meters. This structure has been extensively tested with dynamical and pseudodynamic techniques. Natural frequencies, modal shapes and damping values were obtained. Without entering into details[3], it is sufficient to say that a shear-type model was assumed and the mass, stiffness and damping matrixes were defined as follows:

$$\mathbf{M} = \begin{pmatrix} 5000 & 0 & 0 \\ 0 & 5000 & 0 \\ 0 & 0 & 5000 \end{pmatrix} \tag{5.4.1}$$

$$\mathbf{C} = \begin{pmatrix} 6592 & 6888 & 0 \\ 6888 & -3496 & -3392 \\ 0 & -3392 & 3392 \end{pmatrix} \tag{5.4.2}$$

$$\mathbf{K} = \begin{pmatrix} 47321 & -28941 & 0 \\ -28941 & 36190 & -12462 \\ 0 & -12462 & 12462 \end{pmatrix} \tag{5.4.3}$$

where $\mathbf{M}$ is in kg, $\mathbf{C}$ is in Ns/m and $\mathbf{K}$ is in kN/m. Matrix $\mathbf{C}$ has a negative value on the main diagonal. This result comes from the particular used identification technique based on modal identification and denotes that some more investigations and analysis should be conducted on this method.

The natural frequencies are: 3.0889 Hz for the first mode, 10.4835 Hz for the second one and 19.1510 Hz for the third one. The participant mass for each mode can be calculated as 14091 kg for the first mode (corresponding to 93.9421% of the total mass), 620 kg for the second one (4.1354% of the total mass) and 288 kg for the third one (1.9225% of the total mass). Clearly the first mode dominates the dynamic behaviour of the structure. Damping values are respectively 0.43%, 0.2% and 0.21% of the critical one. These very low values are quite typical for steel structures.

5.4.2 Design of a Tuned Mass Damper (TMD)

In order to design a vibration mitigation system, it can be useful to reduce the three degrees of freedom (3DOF) model to a single degree of freedom (SDOF) system. Taking into account the first natural frequency and the corresponding modal mass, the following value are obtained:

$$m = 14091 \ \text{kg} \qquad k = 5307700 \ \text{N/m} \qquad c = 4589.2 \ \text{Ns/m}$$

In this way the natural frequency is equal to the first natural frequency of the original structure.

[3]A description of the adopted techniques developed at ELSA can be found in (Molina et al. 1999).

A tuned mass damper (TMD) can be now design using this simplified model. TMDs can be very effective if precisely tuned on the resonance frequency they want to reduce. Exact tuning in the damping value is not so critical, indeed. The ratio μ between the mass of the original system and of the TMD should be chosen in the range of 100/1. In this particular case μ is set equal to 0.008. This means that the mass of the TMD should be in the order of 113 kg. Accordingly with (Bechmann and Weber 1996), the optimal value for damping and stiffness are obtained:

$$\xi = 5.41\% \qquad k = 41790 \ \ \mathrm{N/m}$$

With these optimal values a reduction of more than 60% of the resonance peak is obtained. Figure (5.20) shows the comparison between the frequency response of the SDOF system and the simulated response of the same structure with the TMD installed. The effectiveness of the device is well illustrated.

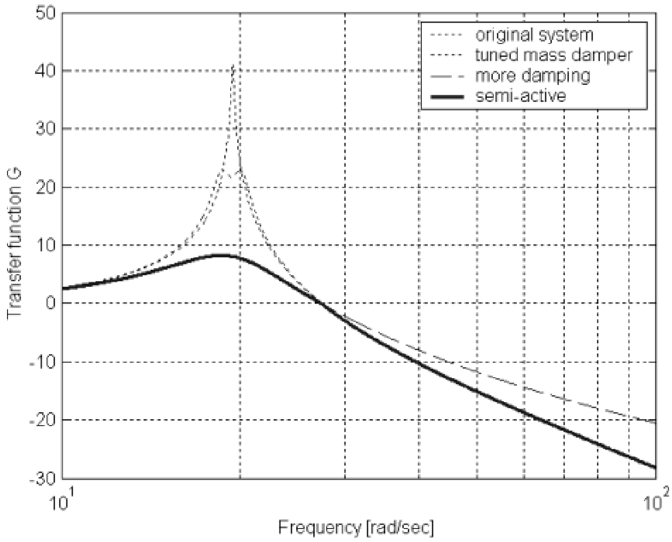


Figure 5.20: Transfer function of a SDOF system equipped with a TMD, a high viscous damper and a semi-active device command by an on-off control law

5.4.3 Theoretical comparisons

A theoretical comparison among different strategies for reducing the resonant peak of the transfer function can be easily performed. In figure (5.20) it is shown the comparison among the transfer functions of the original SDOF system and then equipped with a TMD, a high viscous damper and a semi-active device commanded by an on-off control law. The switching frequency was chosen to be around 9 rad/s. It clearly comes that a proper semi-active device can achieve both the advantage of law and high damping systems. This control strategy is also much more robust against uncertainty than TMD, where optimality in efficiency is strictly related to a very precise tuning of the mass and stiffness values (Bechmann and Weber 1996). On the other hand there is no risk of control instability, because the semi-active device, for a given current, behaves as a passive damper. The real practical problem is connected with the construction of a semi-active device with a damping range between zero and very high values.

5.4.4 Comparison with competitive techniques

5.4.4.1 The HYDE project

The HYDE system consists of two main parts: a Primary Horizontal System (PHS) and a Secondary Horizontal System (SHS).

The PHS must be very stiff in order to concentrate horizontal displacements in its Seismic Link (SL). There are stiff-ductile HYsteretic DEvices (HYDEs) placed. To dissipate a large amount of input energy, they should show almost ideal stiff-plastic behaviour. Then, the SL's can only transmit the maximum HYDE force as storey shear to the adjacent structural members. The conventional structure is protected from overloading.

The SHS must stabilise this mechanism with respect to the P-Δ effect. It should be stiff enough to perform this task, but soft enough not to diminish the efficiency of the HYDE system (Dorka and Schmidt 2000; Schmidt and Dorka 2002). Any additional stiffness of the SHS will draw energy away from the PHS.

A well-designed HYDE system combines the advantages of stiff non-ductile system with a soft ductile system: small displacements and small forces. Linear behaviour of the conventional part of the structure even under serve earthquake is achieved.

Application to the Baby-frame The upper floors are blocked with diagonals in the steel frame, to provide a stiff rigid block. The SL is placed in the ground floor. The HYDEs are connected between stiff diagonals and the beam of 1st floor (figure (5.21)). The initial stiffness of the SL hystereses includes the behaviour of the construction for the HYDEs in the ground level. To not loose necessary stiffness, the connections should avoid any slag. Therefore all connections will be welded.

As HYDEs, inexpensive Shear Panel Device (SPD) are used. They are made of mild steel to use the long yielding level. The storey drift in the ground floor is the governing variable. Assuming brittle connection failure, the elastic deformation limit is reduced to 5 mm maximum inter-storey drift at the ground floor.

Figure 5.21: Passive HYDE device installed on the Baby-Frame.

The elastic FE-model was verified with the test results at the reaction wall. The values to define the hystereses loop of the SL in the numerical model were gained by cyclic sinusoidal wave characterisation tests. One set of SPDs was tested several times to show their behaviour under few numbers of cycles. The SPD showed buckling behaviour because it was test with cycles up to $\pm$ 10 mm. But because the SPD hystereses was still stable, this set was also used for first pseudodynamic test.

Results Some differences between the pseudodynamic test and the numerical results caused from the already damaged SPD. The SPD was used for several times and showed buckling behaviour and cracks at the weld. A positive thing, which figures out very clearly, is the reliability of the SDP. Also after a large number of cycles it protects the building with a very good performance against earthquakes. The HYDE system showed a drastically displacement reduction. Comparison of test and numerical results show that the HYDE system can be calculated with the proposed method without much time effort. The significant reduction of the displacement is the main positive effects of these devices. The upper floors are working as a rigid block.

5.4.4.2 Jarret devices

Passive dissipators manufactured by JARRET were characterised with a testing campaign similar to the LORD semi-active devices and then installed on the Baby-Frame (figure (5.22)).

Figure 5.22: Passive JARRET device installed on the Baby-Frame

The devices were installed on a K bracing system that moves a steel plate acting on the dissipators device.

Characterisation tests In order to characterise the cyclic behaviour of the JARRET BC1BN dissipator (Molina et al. 2002), sinusoidal cycles of different decreasing amplitudes (10, 8, 6, 4, 2, 1 and 0.5 mm) were imposed to a single device prestressed at 11 mm, which is the centre of its run. The first amplitude (10 mm) was repeated four times in order to obtain a stabilised behaviour before starting the decreasing amplitude series. Those cycles were applied at a frequency of 2.0 Hz (for the original reference speed) and followed by a random history with significant frequency content up to 6 Hz. This history of displacement was executed at the nominal reference speed (λ=1) and then at speeds λ=3, 10, 30, 100 and 300 times slower in order to analyse the strain-rate effect (SRE) on the measured force.

Due to the strain-rate effect, the force measured at the slow test has smaller amplitudes, even though the shape is similar to the one at the fast test. This fact suggests a way to correct the force measured at the slow test by multiplying it by a constant factor γ, which should depend on the testing speed. Thus, the corrected force was obtained as the measured force at the slow test multiplied by constant plus a constant offset and it should approximate the force at the real speed test, that is to say,

$$f^{corr}_{\lambda=300} = B f^{meas}_{\lambda=300} + A_0 \approx f^{meas}_{\lambda=1} \tag{5.4.4}$$

Using the measured forces for the fast and the slow tests during the decreasing sinus, the value of those constants was estimated by least squares as

$$B = 1.184 \qquad\qquad A_0 = -0.48\text{kN} \tag{5.4.5}$$

Analyses have shown that this corrected force significantly reduces the error due to the SRE. Applying the same constants to a piece of the random part of the tests, the effect of the correction can also be assessed for an arbitrary displacement history.

For the case of using two BC1BN devices acting together but in opposite direction, as in the current specimen, the offset term would not be needed because the offset of one device would cancel with the one of the other and the corrected force would be obtained just by multiplying by the factor γ. Figure (5.23) shows the value of the correction factor obtained for the different tested speeds by using the same method as for the shown case (λ=300). Thus, depending on the speed of the test, the measured forces should be corrected by applying the factor γ read at this graph.

Results A clear difference in the behaviour is observed for the original structure with respect to the protected cases. In general, the introduction of the devices increases the frequency and the damping ratio.

Even if with relatively small devices (and consequently with small forces), the damping related to the first mode increase up to 12%, i.e. more than 20 times the original one. A numerical analysis on a similar case can be found in (Taucer et al. 1999). The inverse proportionality between damping ratio and displacement amplitude is observed for this structure when retrofitted with the spring-damper devices.

5.5 The ACE-CASCO cable-stayed bridge

5.5.1 Bridge description

The bridge mock-up is a cable stayed cantilever beam that basically represents a cable-stayed bridge under construction. Due to the availability of the ELSA laboratory, a very large-scale set-up was chosen for the mock-up. This means that

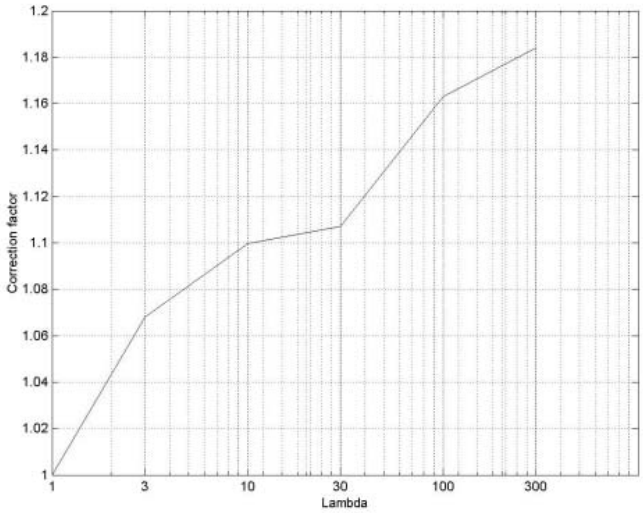


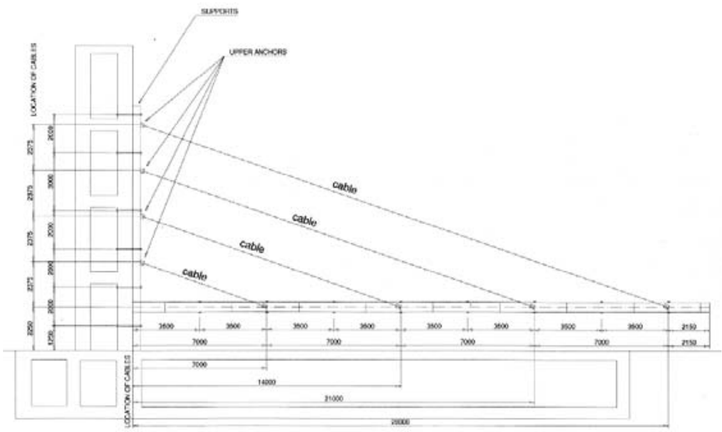
Figure 5.23: Value of the force-correction coefficient γ as a function of the testing time scale

all constitutive parts of the bridge are not special ones, but common industrial components.

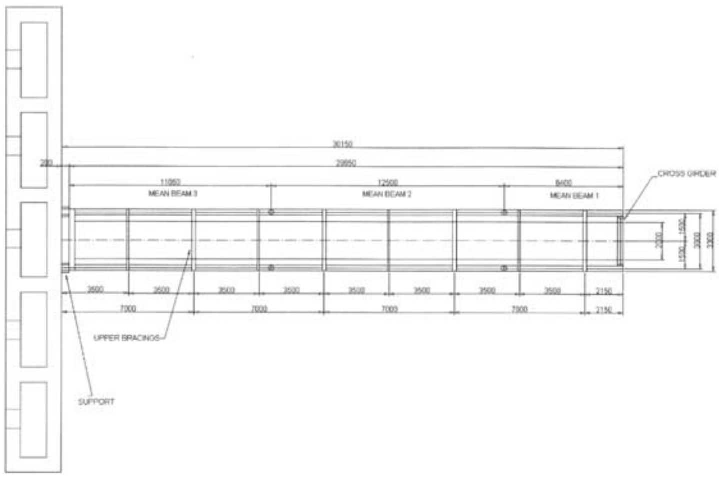
Half span of the real bridge was constructed positioning the deck 2 m about the laboratory floor and fixing the antennas at the Reaction Wall. The deck, about 30 m long, is mainly composed of two H-beams whose axes are spaced 3.0 m apart (figure (5.24)).

They are appropriately linked each 3.5 m with transversal H-beams to provide to the whole structure sufficient transverse and torsion stiffness. Each H-beam is fixed to the Reaction Wall. The vibration excitation source is anchored at the free end of the deck. Four pairs of parallel stay cables support the deck. Each stay-cable is composed of one T13 strand (7 non-circular wires, total section of 112 mm^2) with a slope of 1/3. This slope is very closed to that of the longest stay cable of modern bridges. The static tension being about 65 kN, the first natural frequency of the longest stay cables (29.5 m long) is higher than 5 Hz, which can be rarely met on modern cable stayed bridges. To give the stay cables enough sag and consequently reduce their natural vibration frequencies, they are heavily overloaded with split lead cylinders (figure (5.26)).

This increases their average mass to an amount of about 15.4 k/m. In this way, the sag of the longest stay cables is about 0.8 % of their length and their first



(a) side view



(b) plant

Figure 5.24: Sketch of the cable-stayed bridge mock-up

natural frequency in the vertical plane is closed to 1.2 Hz. With this arrangement the first vertical free frequency of the complete structure (flexion) is very close to that of the longest stay cable.

Figure (5.25) shows the completed bridge. Cables anchor on a cell force connected to the moving rod of the actuator. Some security chairs are also placed under the deck, jointly with reference frames for velocity and displacements measurements.

By positioning an intermediate support under the deck, the vibration frequencies of the whole structure can be varied. It is possible to merge or to separate modal frequencies and to create critical situations for the structural behaviour. As designed, the mock-up allows the complete analysis of numerous particular situations. The mock-up has to be seen as a demonstrator allowing the verification of the efficiency of the active control system in the worst conditions that can be faced by real structures. Without the intermediate support, the first modal frequencies of the structure were 1.01 Hz for the first flexional vertical mode, 1.10 Hz for the first torsion mode and 1.5 Hz for the first flexional horizontal mode.

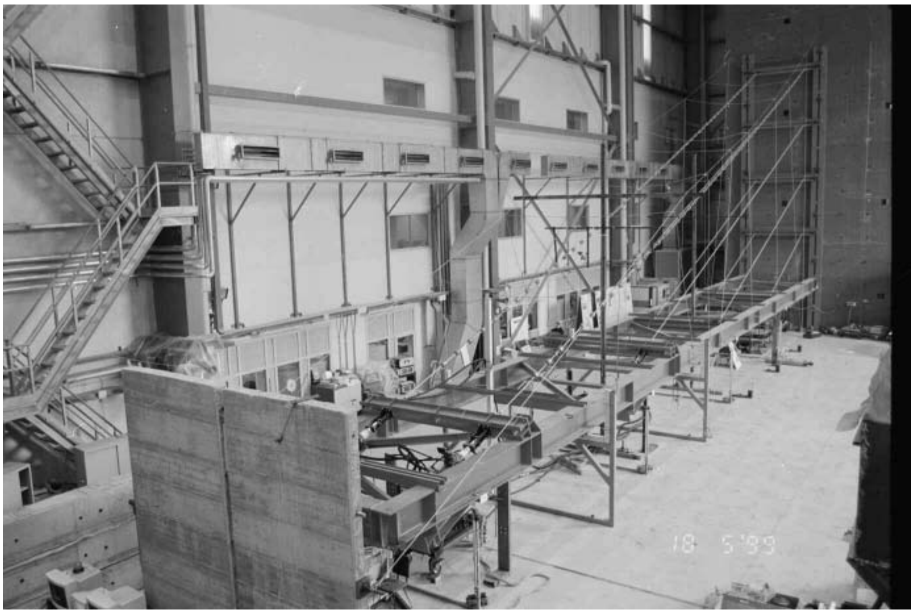


Figure 5.25: Large scale cable-stayed bridge mock-up

5.5.2 The ACE project

5.5.2.1 Objectives

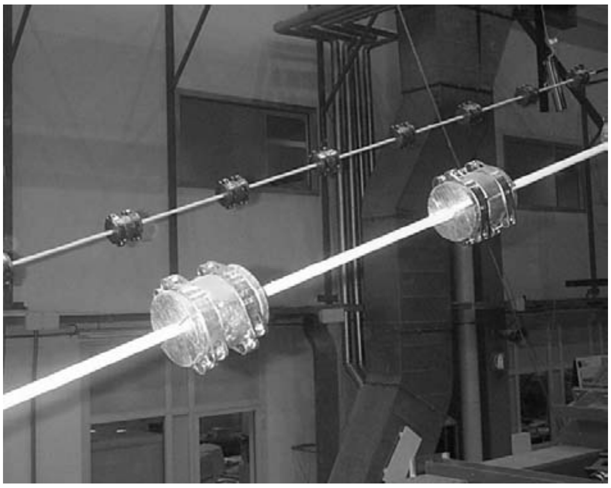


Figure 5.26: Additional masses on cables

The aim of ACE (Active Control in Civil Engineering) research project was to bring the laboratory development in the field of active control to an industrial level and to implement it into complex civil structures. The main thrust of the research programme was to produce active tendon devices. Consequently two types of actuators were developed and tested:

- a large magnetostrictive actuator, as an extension of the devices used with the small-scale mock-up, but of course a very innovative but previously unexplored solution due to the high forces required in civil engineering applications;
- an hydraulic actuator, more "classical" in the field of civil engineering cable-supported structures, but more sensitive to phase lag and response speed requirements.

To demonstrate and validate the application of this technology to civil engineering structures, the large-scale cable-stayed mock-up described in the previous paragraph was built using industrial components and the developed actuators. The aim was to improve knowledge of controlled structure dynamics. In fact, while substantial progress has been made in the study of the components of active damping systems, little attention has been paid to the overall performance of the system applied to a realistic structure.

The structural control system consists of a number of important components such as sensors, controllers, actuators and power generators that must be part of an integrated system. Moreover a number of implementation aspects must be addressed such as intermittent and fail-safe operations, integrated safety, reliability and maintenance. Although these issues require experimental verification under realistic conditions, the validation of the active control system prototype on a large-scale mock-up has given a better knowledge of the non-linear behaviour of the cables and of the real loads in the cables and the anchorages.

To apply these techniques to real structures, the devices would need to be scaled up, but the technology can address any vibration problems. Moreover, by taking into account the theoretical and experimental results, it was possible to include a theoretical analysis on how this technology could be applied to real structures from a technical and economic point of view.

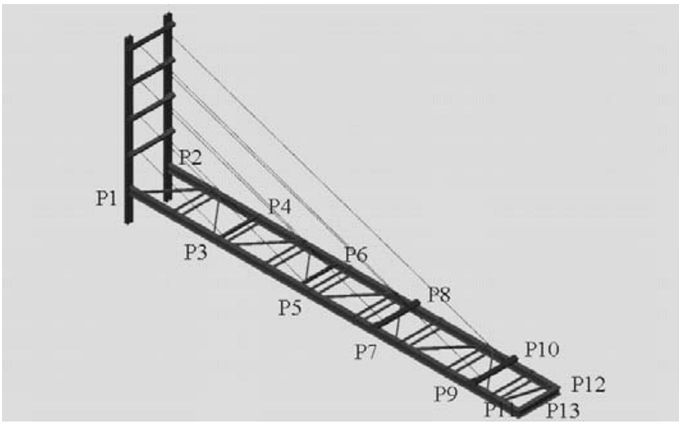
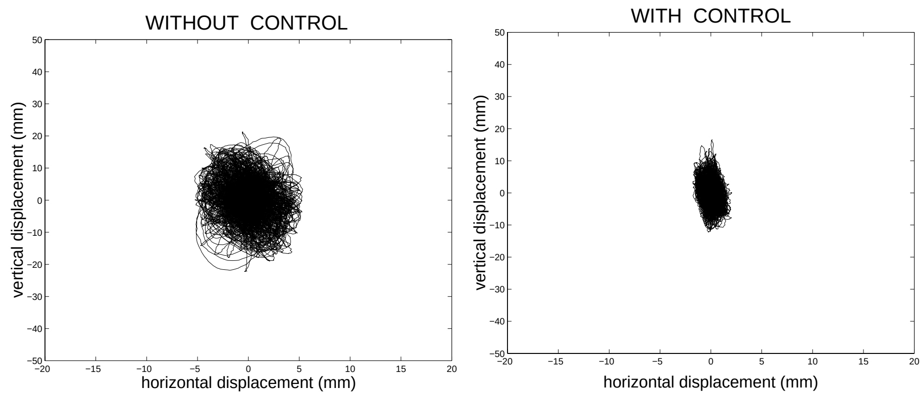


Figure 5.27: Sketch of ACE and CaSCO1 sensors positions

5.5.2.2 Results

The final results obtained with the large-scale cable-stayed mock-up were better than what initially expected. Figure (5.28) shows the comparison between the two trajectories of the longest cables (see figure (5.27) for measurement points locations) with and without control. These trajectories were obtained in the cable's orthogonal planes with an ELSA developed laser scanning system. The bridge was subject to bandwidth noise excitation between 0.6 Hz and 1.3 Hz given by a special developed hydraulic shaker. It can be clearly seen that the displacement reduction is very evident.



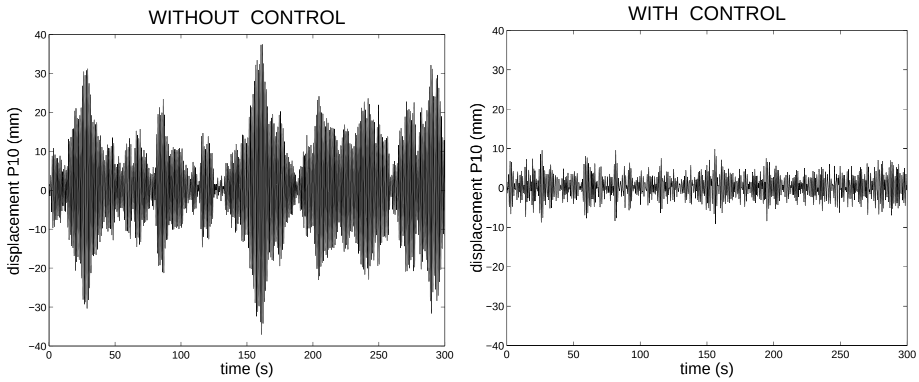
(a) without control (b) with control

Figure 5.28: Cable behaviour during tests with bandwidth noise excitation (0.6-1.3 Hz)

As regarding the deck, an important reduction in vertical displacement is also obtained. Figure (5.29) shows the comparison between the displacement of point P10 during the previous described tests. Snap-back test with imposed displacement gave also very good results, because the damping of the overall structure is increased more than ten times by the active control system.

Concluding, an active damping technique based on a tendon actuator collocated with a force sensor is now ready to be used. These actuators have been manufactured and widely tested on a large-scale cable-stayed mock-up. These technologies are directly applicable to real structures by scaling up the devices. Moreover, direct theoretical application to real structures by the civil engineering experts including technical and economic aspects has exhibited the efficiency of the active control system and the large field of application to cable-supported structures.

The details of the results can be found in (Magonette et al. 1999; Marazzi et al. 2000; Preumont et al. 2000; Bossens et al. 2001; Magonette et al. 2001; Aupérin et al. 2001; Bouygues 2000).



(a) without control

(b) with control

Figure 5.29: Bridge deck behaviour (point P10) during tests with bandwidth noise excitation (0.6-1.3 Hz)

5.5.2.3 Open problems

Even if the application of the active damping technology to real structures is direct by scaling up the devices, the first application to real structures will need additional industrial development in order to address all of the requirements of the structure owner. This technology does not conform to nowadays construction practices and regulations and this will widely hamper dissemination and direct application in the short term.

Even if this technology appears very promising when applied in the construction stage, there are still some concerns about the energy consumption under ordinary conditions.

5.5.3 The CASCO project

5.5.3.1 Background

Concern for the environmental and the quality of life of citizen forces policy makers to transfer the transportation of goods from the traditional heavy trucks to railways. However railway companies are subjected to tremendous resistance and objections to build new railway lines until the problem of vibration exposure to citizen is solved. This problem statement can be put into a more general context where high intensity traffic, wind, earthquakes and other dynamic impacts (e.g.

rotating machinery) generate high noise levels (above 55 dB) and vibrations on systems and structures.

The effects of noise and vibration on European citizen were proven to have a negative impact on the quality and comfort of life, performance levels, health and working environment. Among the most familiar ones are motion thickness, deprivation of sleep, physiological effects such as fatigue and psychological effects such as irritability. Reducing or eliminating noise and vibration to the human body will thus have a positive impact on health, safety and well being of citizen.

An innovative method is required to notably reduce noise and vibration, at the same time minimising energy and material use.

5.5.3.2 Objectives

The goal in CaSCo project was to use advanced materials (rheological fluids) to increase the effectiveness of damping elements, while at the same time minimise their geometric dimensions. Rheological actuators are installed at critical locations throughout structures and underneath railway tracks to eliminate vibration. Thus, kinetic energy is dissipated locally before it is transferred to other components or to the ground. As a consequence, the size of individual viscous dampers is reduced, allowing a more effective use of materials and minimisation of resource consumption. This technology introduces state of the art knowledge into the manufacturing sector and thus increases the competitiveness of European industries and creates employment.

The main project outputs and milestones were:

- viscous and rheological fluid technology;
- controllability assessment and procedure for actuator miniaturisation and placement;
- optimisation tools to reduce product size and energy requirements;
- phenomenological models to effectively portray the behaviour of rheological fluids;
- technical expertise needed to manufacture miniaturised viscous and rheological dampers;
- scientific know-how on operational mechanism and characteristics of semi-active dampers;
- prototype dampers for testing and dissemination of project results;
- knowledge on semi-active control strategies for viscous and rheological energy dissipation;

- coupled analysis and optimisation codes to minimise product weight and geometry;
- dissemination of results in technical journals and conference papers.

In CaSCo research project, two parallel ways have been pursued towards the development of size reduced and optimised method for noise and vibration reduction. The first one has focused on the development of control laws for viscous hydraulic actuators. The innovation in this part of the project consists of the control algorithms for variable orifice dampers, implementable using available computing hardware. The second and much more demanding goal was the development of size reduced energy dissipating devices based on magnetorheological fluids as describe in the previous section. This research effort has been one of the very first attempts to install small energy dissipators underneath high-speed railway tracks to control motion, vibrations and noise.

5.5.3.3 Tested structures

Within the CaSCo project, three structures were tested at ELSA:

1. the cable-stayed bridge used during ACE project, but now equipped with semi-active devices instead of active actuators;
2. the reinforced concrete slab shown in figure (5.30) representing a real-scale portion of a typical the Mass-Spring System commonly adopted in railways tunnels. In this case vibrations are mitigate by additional dampers placed under the slab and working in parallel of more traditional rubber bearings;
3. the Rohrbach bridge shown in figure (5.31), has been tested on field in Austria[4], then dismounted and transported to ELSA to be extensively tested.

5.5.4 Main results

The same cable-stayed bridge used for ACE project (figure (5.25))was evaluated with a very similar testing campaign. The used excitation signals used for the shaker were exactly the same of the ACE project in order to allow a faster and easier comparison among the results. The controller was implemented as reported in the previous chapter and an optimal gain was chosen accordingly with (Bonefeld 2001).

The main results are summarised in the following figures: figure (5.32) shows the comparison between the displacements at point P10 when control is inserted

[4]It was situated 19 kilometres outside Vienna on the north western line and was removed in April 2001.

Figure 5.30: The Mass-Spring System (CaSCo2 mock-up)

or not. Values from P9 are analogous. There is a mean reduction of more than 70% in the deck displacement when control algorithm is working.

Figure (5.33) shows the comparison among the strokes of actuator I (i.e. placed at point P10) in the following conditions:

1. the actuators are off (and control too, obviously): there is no stroke on the actuators, because the cable are lift by a nut on the anchorage. In this case the bridge behaves as a common (passive) cable-stayed bridge.

2. the actuators are on and the control law is inserted: now a big stroke is present, because they are damping out the shaker induced vibrations. In this case the bridge is an actively controlled structure.

3. the actuators are on, but the control law is not active: there is a fixed stroke on the actuators, even if it is much less in comparison with the previous described case. In this configuration the actuators carry the cables tension. Their internal control law keeps constant the pressure inside, but no control displacements are imposed. In this case the actuators behave as passive devices with visco-elastic properties.

Figure 5.31: The Rohrbach bridge (CaSCo3 mock-up) installed inside the ELSA laboratory

Another very important point is the forces behaviour at the anchorage between cables and actuators with and without control. Figure (5.34) shows this comparison. The dynamical component of the force has been plotted, the control algorithm being effective on that. It can be clearly seen that the force is considerably reduced when control is inserted. This leads to a reduction in fatigue effects and in cables ageing. Energy dissipation cycles are shown in figure (5.35).

The previous figures were obtained with an optimal gain value ($g = 0.000009$ m/s/N) obtained with analytical calculations. A testing campaign was then conducted in order to check if this gain value is really the best one. The results are summarised in table (5.3), where the minimum and maximum values for the most important signals coming from bridge position P10: the anchorages forces of the longest cable, the actuators strokes and the deck displacements.

The tests are then ordered by crescent gain in table (5.4). This table confirms the vary good performance of control algorithm. The cable forces and the deck displacements are considerably greater in the uncontrolled case than with the control inserted. An inverse proportional relation between the gain and the cable force value is present here: the higher is the first, the lower is the second. This

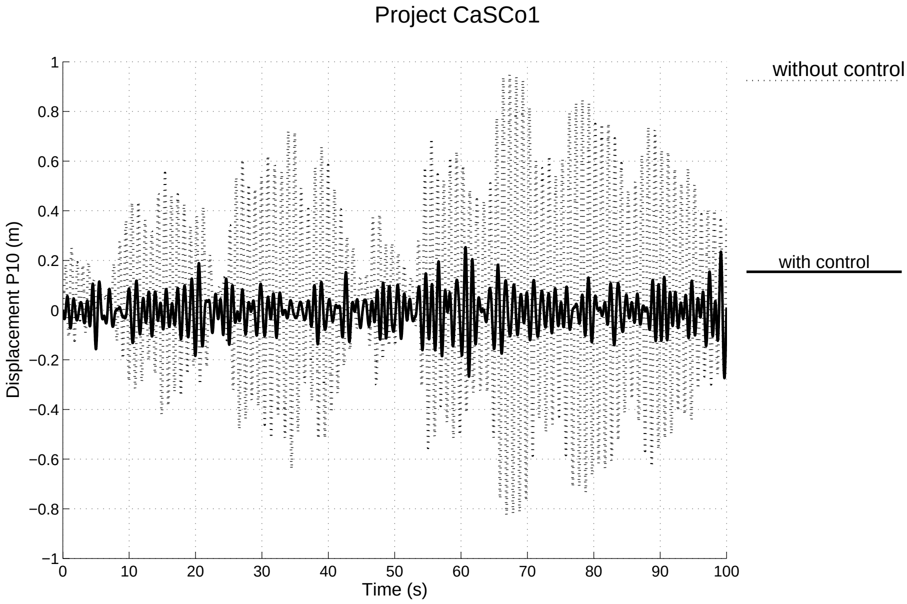


Figure 5.32: Bandwidth noise test (0.6-1.3 Hz): displacement of point P10 with and without control inserted

behaviour cannot be found in deck displacements: it seems that the maximum displacement is not strictly related to the gain value. This point should be further investigated in future. As regarding the maximum stroke that has been registered during these tests, it is less than a half of the maximum allowable stroke of the actuators (60 mm).

Sinusoidal tests were also conducted, but these tests have only a limited value, because when the control is inserted the natural frequencies of the bridge tend to decrease. For the first bending frequency, for example, it will pass from 1 Hz to 0.75 Hz (approximately) with infinite gain. So, if the test is conducted with an excitation frequency equal to the first bending natural frequency of the bridge, to switch on the control will give a very large reduction in displacement amplitude. Two effects will cause this reduction:

1. the damping effect itself given by the control law;

2. the fact that the excitation frequency is no more equal to the natural frequency of the damped system, so it will not excite the bridge so much as before switching on the control.

test number	gain [$ms^{-1}N^{-1}$]	force [N] max	force [N] min	displacement [m] max	displacement [m] min	stroke [m] max	stroke [m] min
c1021	0.0000090	-1947	2028	-0.0278	0.0249	-0.0094	0.0110
c1022	0.0000100	-2073	1665	-0.0282	0.0278	-0.0100	0.0110
c1023	0.0000050	-2042	2207	-0.0228	0.0228	-0.0060	0.0053
c1024	0.0000075	-1987	2383	-0.0281	0.0294	-0.0114	0.0088
c1025	0.0000150	-1216	1499	-0.0268	0.0196	-0.0099	0.0108
c1026	0.0000200	-1407	1308	-0.0287	0.0302	-0.0149	0.0137
c1027	0	-10930	10901	-0.0871	0.0902	-0.0001	0.0004

Table 5.3: Summary of results (referred to actuator I) with different gains

test number	gain [$ms^{-1}N^{-1}$]	Δ^{max}_{force} [N]	$\Delta^{max}_{displacement}$ [m]	Δ^{max}_{stroke} [m]
c1027	0	21831	0.1773	0.0005
c1023	0.0000050	4249	0.0456	0.0113
c1024	0.0000075	4370	0.0575	0.0202
c1021	0.0000090	3975	0.0527	0.0204
c1022	0.0000100	3738	0.0560	0.0210
c1025	0.0000150	2715	0.0464	0.0207
c1026	0.0000200	2715	0.0589	0.0286

Table 5.4: Performance of the controlled system with different gains

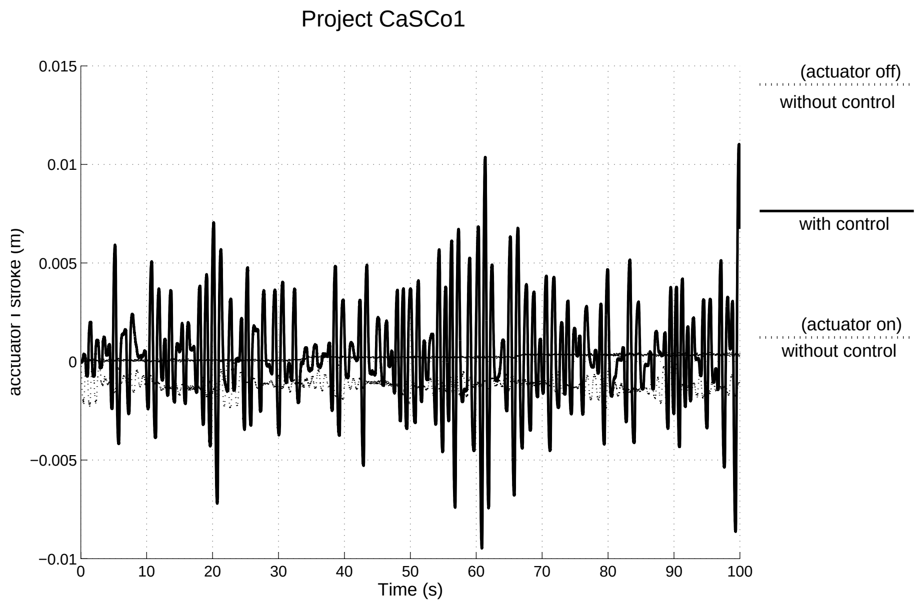


Figure 5.33: Bandwidth noise test (0.6-1.3 Hz): stroke of actuator I with control inserted, without control inserted and with actuator turned off

5.5.4.1 Comparison with ACE project

From and effectiveness point of view, it can be affirm that the CaSCo actuators give the same excellent results than the ACE ones. From an energy consumption point of view, the CaSCo actuators are better that the ACE one, because, once the accumulator is pressurised, the ELSA pumping system is no longer needed. The exact evaluation of their electric consumption is still under investigation because all the operating conditions must be considered.

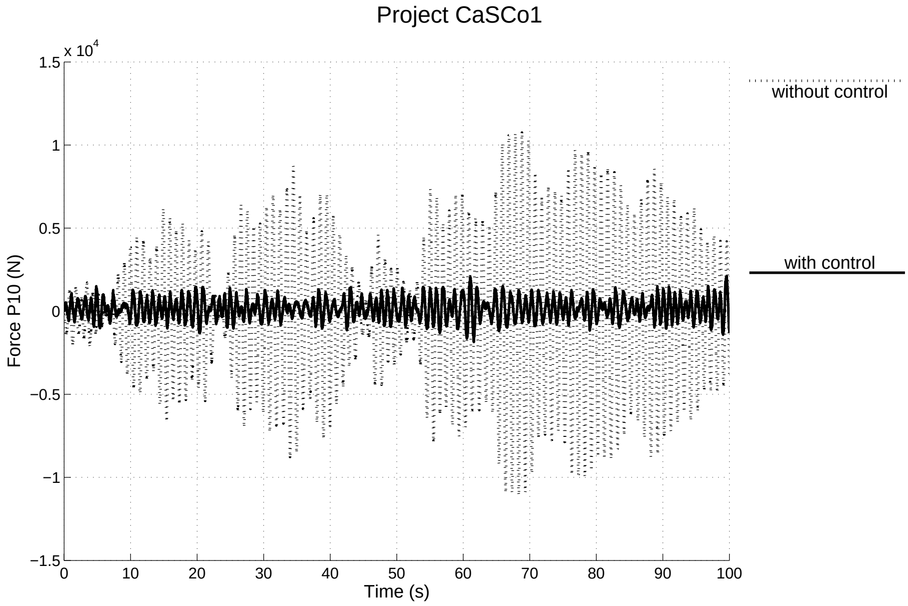


Figure 5.34: Bandwidth noise test (0.6-1.3 Hz): force at point P10 with control inserted and with actuator turned off

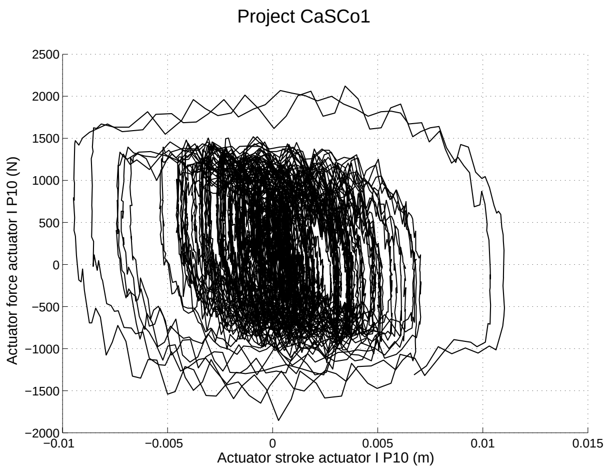


Figure 5.35: Bandwidth noise test (0.6-1.3 Hz): dissipation cycles at point P10

Chapter 6

Conclusions and further developments

6.1 Conclusions

The main goal of this thesis is the investigation of the semi-active control concept applied to civil engineering structures with particular reference to implementation aspects.

The necessary definition is given in the first chapter. This was a very hard job, because structural control is not yet an established field of application and, sometimes, there are misunderstandings among researchers about the terminology. This is particularly true for semi-active control because it is the last discipline born in this field. Commonly used notations in structural control have been recalled, even if there is substantial agreement among scientists about it. The motivations for a semi-active thesis are presented and a very short comparison among different strategies is provided.

Before entering the core of this work, a parenthesis about collocated and non-collocated control systems is reported. The positive and negative aspects of both these two strategies are enumerated. The first part of this chapter is essentially bibliographic, but an original numerical study developed by the author has been reported in the second part. It is shown that, even in a structure with a limited number of storeys, there are thousands of possible combinations among sensors and actuators at each floor. In practice, only some degree of complexity in non-collocated systems is useful, because no substantial improvements can be reached with a "full" non-collocated system with respect to a "slight' non-collocated one.

The third chapter is divided into two parts because the first one, ancillary to the second, is a bibliographical review of the most studied semi-active techniques, while the second is an original tentative of translating these control laws into a flowchart and into the sequence of instructions for implementation. At this level of complexity there are no saturation effects, no time delays, no disturbance. The only target is the choice of the most appropriate quantity to be measured and how to elaborate these informations to get a meaningful response on the semi-active device.

In the forth chapter the implementation of a semi-active control strategy into a controller is documented. Controller hardware and software developed at ELSA are described in details to properly understand the semi-active implementation. The three types of filters reported in this chapter are the most suitable for a real scale control law. These filters are always running when the controller is running, never minding if the control law is active or not. This means that, when the control law is inserted, the filter is already active even if the mean is performed on thousands points. Another positive aspect of these filters is connected with the very short computational time, so being well suited for real-time control problems as in semi-active structural control, where calculation time is a critical aspect. The main part of the control algorithm and of the routines developed by the author is reported with appropriates comments. These programs were used during the experiments reported in the fifth chapter.

Fifth chapter is devoted to the experimental activities conduced in these three years regarding the implementation aspects of the semi-active control laws. A big part refers to the characterisation campaign that was conducted on a magnetorheological device, a variable valve device and a hydraulic semi-active device. This study is needed because the control law must take into account the particular characteristic of the device that is performing it. For example, in a variable valve device the resulting damping is not proportional to the command voltage given to the servo-valve by an input signal, so a calibration is mandatory in order to properly control that parameter.

Another important section is related to a preliminary study on a SDOF system damped with a MR device. Although focusing on a very simple model, this experimental campaign shows several important aspects related to implementation. For example, the results demonstrate that the usable range of the MR inside the structure should not be overestimated for good performances and that the noise related to the load cell measurement can heavily influence the behaviour of the clipped IFF strategy.

The Baby-frame test structure is presented in details because is a very good benchmark structure to test several vibration mitigation solutions. Taking advantage from the other projects on dissipation devices carried on using this structure, some comparison among different techniques is presented.

The presentation of the ACE and CaSCo projects ends the chapter. This last part is very important because it allows a comparison of the results referring to two different types of devices tested on the same large-scale mock-up with the same measurement equipment and the same excitation level. In particular, the use of the IFF clipping control strategy programmed by the author on the ELSA controllers is analysed. Some more calibrations should still be performed on the devices parameter to fully optimised their performance, nevertheless the obtained results are very promising.

6.1.1 Original aspect of this thesis

Very accurate definition of active, passive, semi-active and hybrid control is given at the beginning of this thesis. Since structural control can still be considered as an “adolescent” discipline, no coherent definitions have been found in the literature. So special effort was devoted to introduce more clear definitions in order to avoid misunderstandings.

An original elaboration of the material found in the literature is presented treating of collocated and non-collocated techniques. Numerical examples are added to further clarify this concept. A dedicated model of building storey is developed in Simulink language defining also a simple connection path among input and output.

The semi-active control implementation scheme for each strategy is elaborated by the author both in term of flowcharts and in an hypothetic programming language. This is a fundamental preliminary step for implementation of semi-active control laws in a real controller.

Forth chapter reports the current status of development of the controller implemented at ELSA both in terms of hardware and software. After recalling some properties of digital filters, the list of the subroutine written by the author to implement these filters is reported and explained. The control algorithm was coded in C++ and loaded into the ELSA controller.

The last chapter deals with several testing campaign directly performed by the author during the last 3 years at ELSA. The laboratory characterisation tests of various devices were conducted using a special actuating system. The preliminary study on the SDOF system equipped with MR devices was conducted jointly with the Université Libre de Brussels. The Baby-frame study has contributed to some master thesis on modal analysis (with the direct involvement of the author). The same test set-up was used in relation to some European projects on additional dissipation in civil engineering structures. Regarding the ACE and CaSCo projects, the author has conducted all the experimental runs since the beginning of the testing phase (more than 220 runs for ACE project, 60 runs for CaSCo project). The author was also the main in charge of the database management and of the analysis of experimental data. In particular, in CaSCo project, one of the used control law was implemented by the author in a ELSA controller.

6.1.2 Principal implications

This thesis is a further step in the way towards structural control applied to civil engineering. Some of the main problems arising from large-scale implementation are solved and the effectiveness of semi-active control techniques is shown. Only networking and multidisciplinary approach can be the successful way to face such a complex issue.

6.2 Further developments

6.2.1 Application of a Semi-Active Tuned Mass Damper on the Baby-Frame

The most direct application of the present work would be the construction and test of a Semi-Active Tuned Mass Damper (STMD) applied at the top floor of the Baby-Frame benchmark structure. A preliminary testing campaign should confirm the numerical study about the TMD conducted in the previous chapter. The TMD would be equipped with a variable orifice damper or a magnetorheological damper to obtain variable damping characteristics. The studied control laws could then be applied and the performances of the unprotected, passively protected and semi-actively protected structure should be evaluated in term of inter-storey drift and forces.

6.2.2 Long term monitoring

During a testing campaign conducted inside a laboratory, no fatigue tests are usually performed on the structural control devices. On the contrary, in practical cases, the device will be continuously work and, therefore, could be damage by fatigue effects. Fatigue can be present not only in mechanical components, but also in electronic parts because of switching on/off the controller or in hydraulic auxiliary components as for example seals and piping systems.

A long term monitoring should be conducted on the devices to assess how long each component will last before damaging. A timetable with the maximum working hours for each part should be prepared in order to avoid unwanted failures, for instance following the experience collected in the aeronautical industry.

6.2.3 Real scale in field applications

Semi-active devices and control laws has been tested in laboratory conditions, i.e. with constant temperature and weather conditions. A challenge for future will be the validation of those techniques in adverse working conditions. It must be guarantee that adverse factors as for example heavy rains, ice and strong sunshine will not compromise the working behaviour of the controlling system. The controller devices, installed on real structures such for example bridges, must be protected against vandalism acts, terrorist or accidental damage due, for example, to car accidents.

Even if all these aspects can in principal be solved without many problems, they must still be addressed and a deep interdisciplinary knowledge on human behaviour, weather evolution, electronic hardware and software must be achieved.

A further point refers to the device manufacturing and installation price that must come to a minimum. Design of the protected structure should result in a simpler process. New standards and regulation governing new applications should be developed in order to reduce design times and costs (maintaining the required safety margin).

6.2.4 Miniaturisation of controllers

One of the main targets for future works should be the miniaturisation and integration of the controllers. This doesn't mean that the civil engineering community needs a controller of peanut size, but that the controllers should become more compacts than at present.

During the present research projects, in fact, attention was focused on general controllers to be set time-to-time to the requirement. For debagging purposes it was much more important to have access to all the internal variables than an easily moveable controller. During tests it was necessary to see on a screen the status of the evolving variables and the setting parameters in order to change them in real-time. Moreover, because of the size of some projects, the structural control algorithm and the actuator placement control algorithm were developed by two different partners.

When considering a commercial device, the biggest part of the functions needed during development are no longer needed, so a thinner controller can be developed regarding both hardware and software. This will also pave the way to an easy integration of the controller into the device itself, with great benefits in terms of reliability of the complete system.

6.2.5 Dissemination of results

Dissemination of results shall not be considered exactly a development, nevertheless it can be deemed as a necessary further step in the direction of real world applications. The technical articles should not be addressed only to the most relevant journals in structural control field. It will be very useful to publish the main results on worldwide accessible database (see SAMCO[1] or CONVIB[2] networks, for example) and on the Internet, too. The concepts and techniques should be presented to the vast civil engineering community in a form easy to understand, comprehensible and comprehensive.

Is compulsory to try to convince municipalities and governments about the effectiveness of these techniques in vibration mitigation: they should promote the applications of new technology for the protection and the well being of the citizens. They should adapt construction regulations in order to allow the use of modern techniques.

Finally, these techniques should be taught at university in order to enable engineers and architect to understand them and to make at least preliminary calculation and costs-benefits analysis. Doing nothing in this direction, the "common" engineer will simply ignore these new possibilities. New textbooks about semi-active control are welcome: this topic is usually not deeply treated in the literature about passive devices for vibrations protection.

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[2]Innovative Control Technologies for Vibration Sensitive Civil Engineering Structure, network funded by ESF - European Science Foundation.

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Appendix A

Review on semi-active devices

In the following section the main types of semi-active devices for civil engineering applications will be taken into consideration. This means that passive and active devices will not be mentioned here, even if some of the described devices can be seen as an adaptation of passive or active ones.

It must be notice that, from a semi-active control point of view, it doesn't make sense to speak about actuators (like in active control), because the semi-active devices can only generate forces in a passive way, but they cannot freely give any force. The force that the semi-active devices can generate is always related to the relative velocity and displacement of their ends.

A.1 Variable viscous devices

This first class of devices is the oldest and still most studied one. A viscous device can be obtained by an hydraulic piston in which a flux is allowed to pass from one chamber to the other. If the orifice that allows the flux between the two chamber has a constant opening, the device is a passive viscous damper, but if the flux intensity can be adjusted on-line, the device become semi-active. Note that, for a fixed position of the servo-valve, the behaviour of the device is exactly like that of a passive one, but the advantage, in this case, is that the viscous coefficient characterising the device can be adjusted on-line accordingly with a prescribed control law.

If the fluid inside the piston is oil, the flux intensity can be adjusted on line by mean of a servo-valve orifice. This type of actuator is shown if figure (A.1).

The phenomenological model of these devices is usually constituted by a linear viscous element with controllable damping characteristics given by

$$F(t) = C_{adapt}(u)\dot{\delta}(t) \qquad \text{with} \qquad C_{min} \le C_{adapt} \le C_{max} \tag{A.1.1}$$

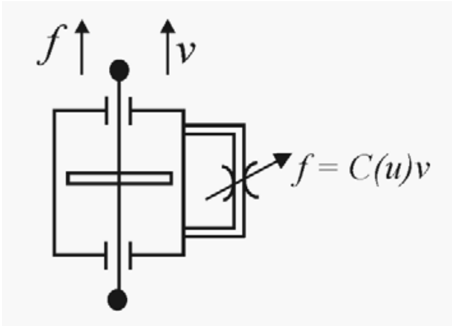

Figure A.1: Variable orifice damper

where C_{adapt} is the actual damping value of the device (that is function of the control variable u), $\dot{\delta}(t)$ is the velocity of device deformation (given by the difference between the velocity at the two ends of the device) and C_{min} and C_{max} are the minimum and the maximum damping value that the device can achieve with the orifice completely opened or closed.

It must be notice that a linear opening of the valve doesn't necessarily reflects in a linear changing behaviour through C_{min} and C_{max}: a prior identification of the device is necessary to properly command the control signal u of the servo-valve.

A special case of this continuous variable damping is the "on-off" device, in which the valve can only assume two values: completely open or completely closed. These types of devices are usually simpler than the continuous ones, but they have lower performance.

If the device is equipped with a magnetorheological or electrorheological fluid, the flux between one chamber and the other can be adjust by changing the magnetic or electrical field around the by-pass from one to the other chamber. This kind of devices will be discussed later in the section devoted to magnetorheological devices.

A.2 Variable stiffness devices

This is another well studied class of devices. They were proposed to avoid, in real time, resonance phenomena: by varying the stiffness of the structure, it is possible to vary the natural frequency of the structure in order to have always a good response to external excitations.

The phenomenological model for these type of devices is given by

$$F(t) = K_{adapt}(u)[\delta(t) - \delta_0(t)] \qquad \text{with} \qquad K_{min} \leq K_{adapt} \leq K_{max} \tag{A.2.2}$$

where K_{adapt} is the actual stiffness of the device (that is function of the control variable u), $\delta(t) - \delta_0(t)$ is the deformation of the device (given by the difference between the positions of the two ends of the device). K_{min} and K_{max} are the minimum and the maximum stiffness that the device can induce into a portion of the structure.

Also in this case the "on-off" device is a particular case in which the stiffness can vary between two values, for example a lower value in which a bracing system is ineffective and another value in which the bracing is contributing to the structural stiffness.

One of the most common scheme for these devices consists in bracing that can vary their stiffness accordingly to a control law. This is usually achieved by means of hydraulic devices that can clamp the bracing to the structure. This lead to the fact that often a semi-active stiffness device is coupled with a semi-active damping device obtaining a varying viscoelastic device.

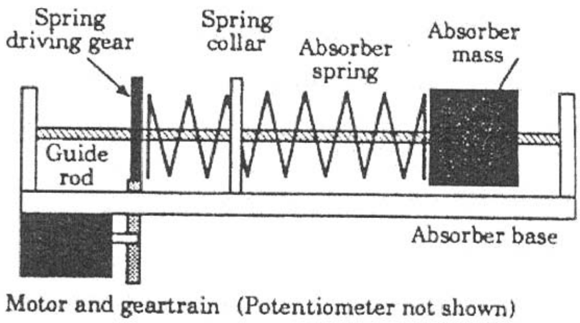

Figure A.2: Variable tuned mass damper

Another interesting way for stiffness variation is the following adapting tuned mass damper (figure (A.2)) in which the stiffness of an helicoidally spring can be modified by varying, by means of an electric motor, the number of coils of the spring.

A.3 Magnetorheological devices

Before describing these devices, some characteristic of magnetorheological fluids (compared with electrorheological ones) will be recalled.

In 1947, W. Winslow observed a large rheological effect (apparent change of viscosity) induced by the application of an electric field to colloidal fluids (insulating oil) containing micron-sized particles; such fluids are called electrorheological

(ER) fluids. The discovery of MR fluid was made in 1951 by J.Rabinow, who observed similar rheological effects by application of a magnetic field to a fluid containing magnetisable particles. In both cases, the particles create columnar structures parallel to the applied field (figure (A.3)) and these chain-like structures restrict the flow of the fluid, requiring a minimum shear stress for the flow to be initiated. This phenomenon is reversible, very fast (response time of the order of millisecond) and consumes very little energy. When no field is applied, the rheological fluids exhibit a Newtonian behaviour.

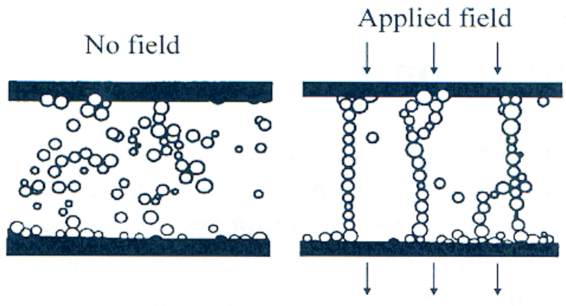

Figure A.3: Chain-like structures formation under the applied external field

Typical values of the maximum achievable yield strength τ are given in table (A.1). ER fluids performances are generally limited by the electric field breakdown strength of the fluid while MR fluids performances are limited by the magnetic saturation of the particles. Iron particles have the highest saturation magnetisation. In table (A.1), we note that the yield stress of MR fluids is 20 to 50 times larger than that of ER fluids. This justifies why most practical applications use MR fluids. Typical particle sizes are 0.1 to 10 μm and typical particle fractions are between 0.1 and 0.5; the carrier fluids are selected on the basis of their tribology properties and thermal stability; they also include additives that inhibit sedimentation and aggregation.

The behaviour of MR fluids is often represented as a Bingham plastic model with a variable yield strength τ_y depending on the applied magnetic field H. The flow is governed by the equation:

$$\tau = \tau_y(H) + \tau\dot{\gamma} \qquad \text{with} \qquad \tau > \tau_y(H) \tag{A.3.3}$$

where τ is the shear stress, γ is the shear strain and η is the viscosity of the fluid. Below the yield stress (at stains of order 10^{-3}), the material behaves viscoelastically:

$$\tau = G\gamma \qquad \text{with} \qquad \tau < \tau_y(H) \tag{A.3.4}$$

where G is the complex material modulus. This model is also a good approximation for MR devices (with appropriate definitions for τ, γ and η). However, the actual behaviour is more complicated and includes striction and hysteresis.

Figure (A.4) shows the three operating modes of controllable fluids: valve mode, direct shear mode and squeeze mode.

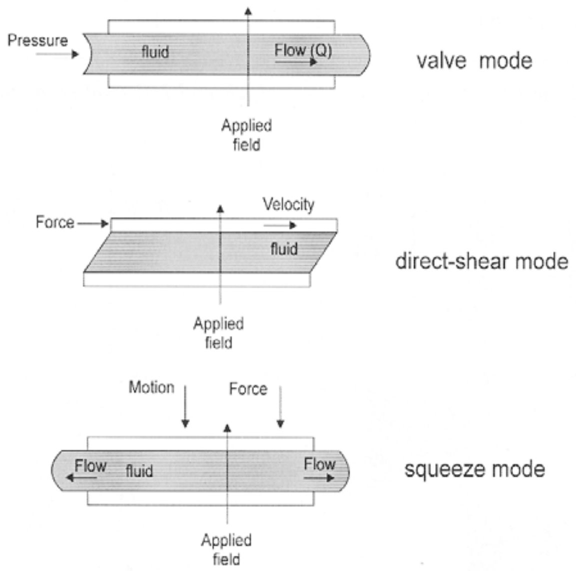

Figure A.4: Operating modes of controllable fluids

The valve mode is the normal operating mode of MR dampers and shock absorbers, the direct shear mode is that of clutches and brakes.

Figure (A.5) shows an example of MR device. It can be seen that it consists in a common viscous damper, but instead of oil it is filled with a MR fluid. The current passing into the coil placed on the rod head generates a magnetic field that is able to polarise the metallic particles of the fluid in order to increase its viscosity.

The cost of the MR fluid contributes significantly to the total cost of the MR device. In order to bring this cost down by reducing the amount of fluid encapsulated, foam devices have been introduced where the MR fluid is constrained in an absorbent matrix by capillarity, without seals.

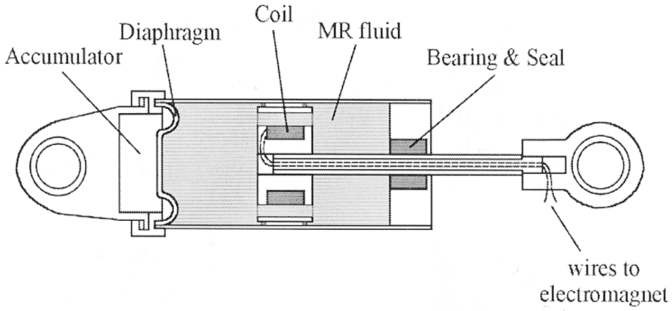

Figure A.5: An example of MR damper

Property	**ER fluid**	**MR fluid**
yield strengh τ	2 - 5 kPa	50 - 100 kPa
max. field	3 - 5 kV/mm	150 - 250 kA/meter
viscosity (no field, 25°C) η	0.2 - 0.3 Pa s	0.2 - 0.3 Pa s
density	1 - 2 g/cm^3	3 - 4 g/cm^3
response time	ms	ms
η/τ^2	10^{-7} - 10^{-8} s/Pa	10^{-10} - 10^{-11} s/Pa

Table A.1: Comparison of typical ER and MR fluid properties

A.4 Friction devices

A.4.1 Semi-active joint connections

In his PhD thesis (Nitsche 2001) noticed that, in many present-day structures, the previous described variable stiffness method is no longer practicable to avoid the excitation of resonance. This happen because the excitation covers a broad frequency band, and there is limited scope for adjusting the mass or stiffness properties of a structure in order to shift resonance frequencies[1]. Because he referred his work mainly to aircraft vibration protection, he than affirms that the above mentioned situation is exacerbated in his case studies by the low mass and by the all welded construction methods often used, which results in low inherent structural damping.

The problem is so to insert damping into the structure. This can be done by

[1] He has in mind the classical passive means of changing natural frequency commonly used in mechanical engineering.

adding special high damping materials or by using high damping alloys, but these methods were, in his opinion, usually expensive and the damping is often frequency and temperature sensitive. He then observeed that, because about 90 percent of inherent damping in most structures arises in the structural joints, it would seem sensible to endeavour to influence the damping in a structure by means of the joint. This can be achieved by controlling the joint clamping forces and hence the relative interfacial slip.

His suggestion was that frictional damping should be deliberately increased and controlled in some structural joints so that the inherent structural damping was increased, thereby reducing the dynamic response, stress, and noise. The energy dissipation mechanism arising from relative interfacial slip in a joint is a complex process, which is largely influenced by the interface pressure. With low joint clamping pressures, sliding on a macro scale takes place. If the joint clamping pressure is increased, mutual embedding of the surfaces starts to occur. Sliding on a macro scale is reduced and micro slip is initiated, which involves very small displacements of an asperity relative to its opposite surface. A further increase in the joint clamping pressure will cause greater penetration of the asperities. The pressure on the contact areas will be the yield stress of the softer material. Relative motion causes further plastic deformation of the asperities.

In most joints he studied, the previously described mechanisms were working. The relative significance of these mechanisms depends on the joint conditions and the magnitudes of the forces. In joints with high normal interface pressures and relatively rough surfaces, the plastic deformation mechanism is significant. Many joints have to carry great pressures to satisfy structural criteria, such as high static stiffness. A low normal interface pressure would tend to increase the significance of the slip mechanisms. An improvement in the quality of the surfaces in contact will also facilitate the slipping. With the macro slip mechanism, the dissipated energy is proportional to the product of an interface shear force function and the relative slip. Under high pressure, the slip is small and under low pressure, the shear force is small: between these two extremes the product becomes a maximum.

Then he passes to describe the "semi-active intelligent bolt" he studied. In order to set and maintain the normal force, a clamping arrangement more elaborate than simple bolts or rivets may be necessary, although this may only mean the addition of an active washer. In any event, the force and moment transfer mechanisms, as well as the damping in structural joints, must be understood if an efficient structure has to be designed. A piezoelectric stack disc is used as a washer to control in real-time the normal force in the friction interface based on feedback from sensors outputs. If a voltage is applied to the piezoelectric washer, the stack disc tries to extend, which results in increasing the normal force. This idea of semi-active friction damping in joint connections has been patented by Prof. Gaul (Gaul and Nitsche 2001; Gaul and Lenz 1998).

A.4.2 Friction controllable sliding bearing

The idea of this type of devices is this: the friction between the bearing and the ground can be controlled by adjusting the pressure between the two sliding surfaces. For example, as shown in figure (A.6), the fluid pressure in the chamber can be increased or decreased in order to diminish or augment the forces coming from the superstructure and acting on the sliding surfaces (Feng et al. 1993).

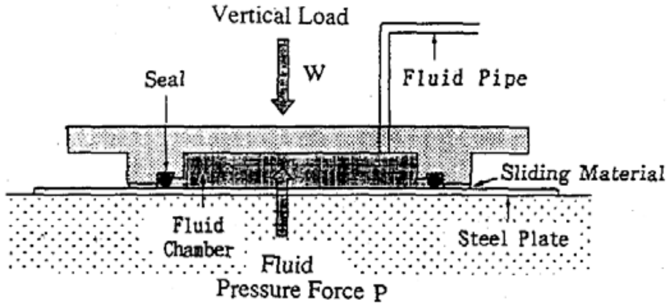

Figure A.6: Idealised view of friction controllable sliding bearing (from Feng)

In this case the computer calculates an appropriate signal to control the fluid pressure based on the observed structural response, such as response acceleration and sliding displacement, and transmits it to the pressure-control device.

The system can be a passive sliding isolation system as long as the pressure of the bearing chamber, and thus the friction, is kept at a constant value. But the pressure is not constant because it can be changed with a little energy expense, so the device can be classified as semi-active. This has the great advantage that the device can became operative also at a low excitation level while maintaining high performance also for stronger excitations.

One important point regards the time response of the system: for practical application a very fast control algorithm should be used together with a good pumping systems in order to avoid excessive delays.

A.4.3 Semi-active slip bracing system

This device, shown in figure (A.7), dissipates energy by allowing slippage to take place along a Coulomb friction interface. The load F_f at which this interface slips is controlled by the clamping force N. It is envision that the typical use is to mount it on a lateral bracing system. The device will allow a brace's axial elongation or contraction through slippage when the brace loads reach the clamping force times the friction coefficient at the interface.

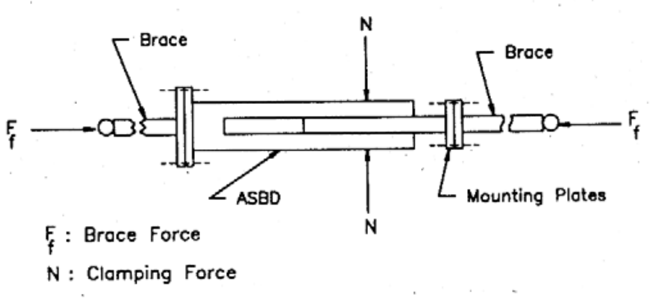

Figure A.7: Operational principle of the semi-active slip bracing system (from Kannan)

By installing this kind of device, a brace's strength can be altered independently of its stiffness. The other advantage of this device with respect to a passive one is that it can become operative also for small forces (i.e. small earthquakes or winds) but maintaining good performances also for stronger solicitations.

This device was called by the author "Active Slip Bracing Device" (Kannan et al. 1995) because the clamping force can be varied by means of an actuator. Then, in the paper describing its characteristic, the author himself said that the device should be classified as hybrid because it monitors and actively alters the energy dissipation of the building structure during its response to vibration excitation. Following the definitions given in § (1.2), however, the most proper category where the device should fall is the semi-active class, because it is a passive device in which the characteristics can be modified on-line accordingly with some measurements and a control law.

Appendix B

Bases for signal acquisition, treatment and analysis

B.1 Introduction

A complete review on signal acquisition, treatment and processing of signals is out of the goals of this thesis. The aim of the following sections is only to give some ideas that can be useful for a proper understanding chapters four and five. Much more information can be found in (Lynn 1982) and (Biondo and Sacchi 1996). A good summary can be found in (Battaini 1998).

B.2 Signal acquisition

In this chapter, some ideas on signal acquisition will be given. The hardware will not be analysed, assuming that enough channels are always available, the sampling time can be chosen arbitrarily, the hard disk space for data recording is large enough and so on. These hypotheses can be not verified in practice, so same arrangements must be thought in order to match the performance of the acquisition system.

B.2.1 Rules for signal acquisition

To pass from the *time domain* to the *frequency domain* is easy with the numerical method known as *Fast Fourier Transform* or *FFT*. It consists in an algorithm that has a signal as input and gives the power spectra of that signal as output. This algorithm is very fast, in the sense that can lead to a maximum number of multiplications that is in the order of magnitude of $N \log_2 N$. At present, it is the most widely used method for signal processing and all the principal softwares for signal analysis embedded it in their subroutine.

The FFT is governed by three basic rules that must be recalled to obtained the searched result: the first one is related to the length of the acquisition in time domain, the second one to the acquisition in frequency domain and the third one is related to the maximum frequency that can be determined with a fixed sampling time.

B.2.1.1 Acquisition in time

The FFT algorithm assumes that the signal is sampled N times at a uniform sampling time Δt. The total sampling period starts at $t = 0$ and ends at $t = T_{TOT}$, given by:

$$T_{TOT} = N\Delta t \tag{B.2.1}$$

where T_{TOT} and Δt are expressed in seconds.

B.2.1.2 Acquisition in frequency

The FFT algorithm assumes that the frequency spectrum of the signal has $N/2$ points uniformly spaced with frequency interval equal to Δf. The frequency range is so defined into the interval $[0, F_{max}]$ with

$$F_{max} = \frac{\Delta f N}{2} \tag{B.2.2}$$

where F_{max} e Δf are expressed in Hertz.

B.2.1.3 Nyquist frequency

The maximum frequency that can be obtained passing from time domain (§ **??**) to frequency domain is related to the sampling time used during acquisition by the equation:

$$F_{max} = \frac{1}{2\Delta t} \tag{B.2.3}$$

This frequency is called *Nyquist frequency*. In the same way, the sampling time must be obey to the following formula:

$$\Delta t \leq \frac{1}{2F_{max}} \tag{B.2.4}$$

where F_{max} is the maximum frequency that can be obtained.

The sampling time is only related to the maximum obtainable frequency and not to the frequency spectrum resolution. In the practical case the factor $1/2$ of the (B.2.4) should be diminish at least to 1/3, better to 1/4. By substituting the (B.2.1) and the (B.2.3) into the (B.2.2), it can be obtained:

$$\Delta f = \frac{1}{T_{TOT}} \tag{B.2.5}$$

that shows how the frequency resolution of the FFT is inversely proportional to the acquisition time.

In most practical cases, taking into account the given formulas, the first choice is done on Δf so obtaining T_{TOT} from the (B.2.5). Then F_{max} is fixed in the (B.2.2) and N is calculated. At this point it is possible to obtain the sampling time Δt from the (B.2.1).

B.2.2 Errors due to sensors

The measurement errors are related to the type of used sensors. As regards accelerometers and displacement sensors (the most commonly used sensors in civil engineering applications), their characteristics can be summarised as follows:

- **the adopted measurement scale**: it should be great enough to enable enough resolution in measurement and, in the meantime, to be little enough to avoid saturation in normally working conditions;
- **the frequency characteristics**: each sensor has its own transfer function that defines the frequency range in which it is fully operative. For accelerometers, for example, a distinction can be made on piezoelectric and capacitative devices. The first ones are used for high frequency measurements, the second ones for low frequency analysis;
- **the electric noise**: this factor corresponds to the perturbation of the physical measurement. This noise is related to the particular sensor and it is generally a function of the measurement scale. Its value is normalized with respect of the measured signal value, so high measured signal values give better results for a given noise level.

B.2.3 Errors due to sampling

Since the mainly part of the sensors and actuators are analogical (while the control is usually executed via a digital computer), it is very common to have to convert data from the analogical format to the corresponding digital one and vicevera. The Analogical to Digital converter (A/D converter) is the necessary instrument to convert an analogical signal into a digital one. It consists usually in a sampling and hold circuit in which the original signal passes thought.

Since the digital signal can only have a limited number of levels between the maximum and the minimum one, the input signal must be divided into "steps". The number of available bits for recording the digitalised signal determines the height of these steps. With a 8 bit circuit, for example, the available levels are $\pm 2^7 = \pm 128$ (7 bit used for the levels, one for the sign) while whit a 16 bit circuit (more often used) there are $\pm 2^{15} = \pm 32768$ possible levels.

The minimum precision ΔV of the A/D converter is so given by:

$$\delta V = \frac{\Delta V}{2^n} \tag{B.2.6}$$

where δV is the range of measurement of the A/D converter and n the number of bit of the sampler, typically 8 or 16, as in the previous examples.

This lead to a truncation in the measured number: the entity of this approximation must be small in comparison with the intensity of the acquired signal. The maximum range of the sensor must be chosen in a proper manner in order to avoid saturation but also to stay close to the maximum foreseen value. Keeping ΔV as small as possible, δV will be at a minimum.

B.2.4 Sensors localisation

From a theoretical point of view, a structure can be instrumented with an arbitrary number of sensors placed in every interesting point. From a practical point of view, the sensors number that can be placed on a structure is strictly related to the available economical resources both from the measurements side and from the complexity of the control low.

It is very important to have the minimum number of sensors in order to have the all necessary information (Soong 1989). To solve this problem, it is very useful to have an approximate idea of the modal shape of the structure. In fact, it is very important to avoid the placement of the sensors near the nodal points of the natural modes shapes, because in these positions the observability is very low.

This can be seen in figure B.1, where some positions are very good for some sensors but bad for some other ones. In this particular case, if a sensor is placed in middle span of the hinged-hinged beam, it will be in an optimal position for odd modes shapes but will be blind for even modes.

In general, it is always better do not place sensors on symmetrical axis to avoid that the asymmetrical modes will be invisible. In the illustrated case a sensors placed at 0.3 distance from the hinge will be the best choice, because also the second mode can be very well detected.

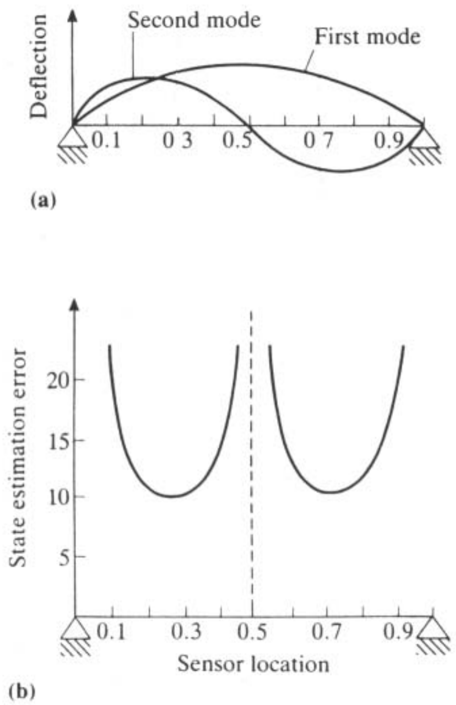

Figure B.1: Sensors placements: (a) first and second modal shapes; (b) error in the measurement of the second mode (from Soong).

B.3 Signal analysis

B.3.1 Shannon theorem

Claude Shannon of Bell Laboratories first proved the following theorem in the late 1940s:

Theorem 1 (Shannon theorem) *Suppose x is a continuous-time signal with no frequency larger than some f_0 Hertz. Then x can be recovered from its samples if $f_0 < f_s/2$, where f_s is the Nyquist frequency.*

B.3.2 Aliasing

The aliasing phenomenon is strictly related with the Shannon Theorem. Its effects are well illustrated in figure (B.2).

Let's consider a sinusoidal signal at 1 Hz. Following theorem (1), this signal must be acquired at least at 2 Hz, otherwise the reconstruction will be impossible. If we use a sampling frequency of 1.2 Hz, for example, the Nyquist frequency will be 0.6 Hz, which is lower than the frequency of the signal we want to catch. In this case the 1 Hz signal will be "mirrored" around 0.6 Hz and will appear at 0.2 Hz (see figure (B.2)).

From another point of view, aliasing dirties low frequencies with high frequencies. A frequency higher than the Nyquist one is always reflected to a low frequency.

The aliasing problem originates from sampling, i.e. in the digitalisation process. This implies that no digital filter can be used to suppress it. Only analogical filter can be used to filter the unwanted frequencies.

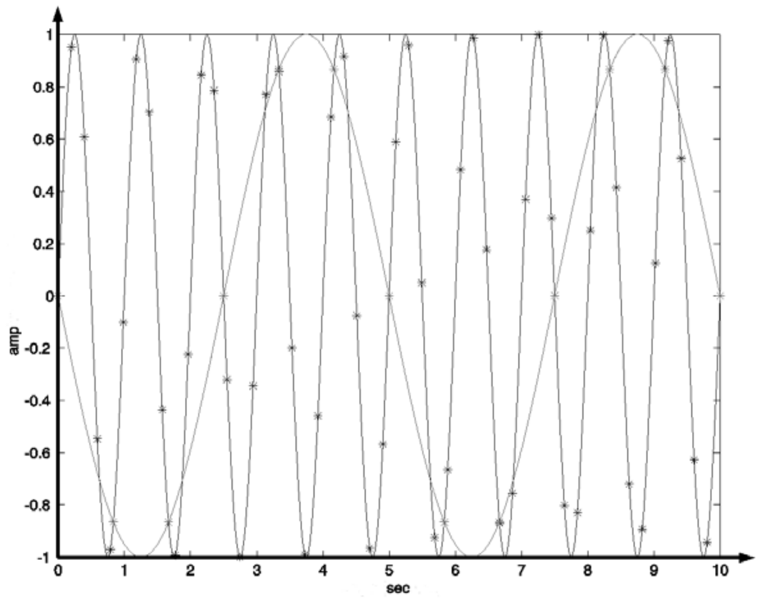

Figure B.2: Example of aliasing

Appendix C

Modal analysis

C.1 Damping

The phenomenon by which mechanical energy is dissipated in dynamic systems is called damping. For this reason this thesis will uses the term *damping* as synonymous of *energy dissipation*. Damping always generate hysteretic loop in the Force vs. Displacement plane. The loops dissipate energy, either as internal thermal energy or as structural damage.

It is very important to characterise the damping of a dynamical system because it allows understanding the major mechanisms associated with mechanical-energy dissipation in the system. A suitable damping model should be chosen to represent the associated energy dissipation. Damping values (model parameters) are determined, for example, by testing the system or a representative physical model, by monitoring system response under transient conditions during normal operation, or by employing already available data.

C.1.1 Types of damping

The damping types can be classified as follows:

- **Internal damping**: refers to a damping that is internal to the structures, i.e. it is not related to the particular ambient where the structure is placed nor on the boundary conditions. Its origin can be found in the material properties of the structural members: it originates from the energy dissipation associated with microstructure defects, such as grain boundaries and impurities, thermoelastic effects caused by local temperature gradients resulting from non-uniform stresses, dislocation motion in metals and chain motion in polymers.

Several models have been employed to represent energy dissipation caused by internal damping. This variability is primarily a result of the vast range of engineering materials; no single model can satisfactorily represents the internal damping characteristics of all materials. Nevertheless, two general types of internal damping can be identified: *viscoelastic damping* and *hysteretic damping*.

The first type of internal damping is usually not related to damage, at least in a first stage. On a Displacement vs. Force (or Stress vs. Strain) graph, this type of damping generates ellipses that are turned clockwise around the origin (figure C.1).

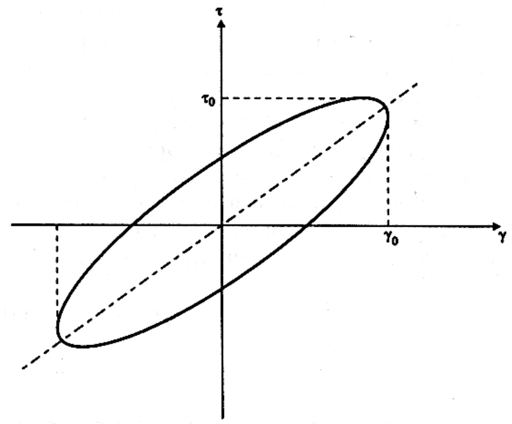

Figure C.1: Stress Vs. Strain plot for a viscoelastic material (form Soong & Dargush)

For a linear viscoelastic material, the stress-strain relationship is given by a linear differential equation with respect to time, having constant coefficients. A commonly employed relationship is:

$$\sigma = E\varepsilon + E^{*}\frac{d\varepsilon}{dt} \tag{C.1.1}$$

which is known as the Kelvin-Voigt model. In equation (C.1.1), E is Young's modulus and E^{*} is a viscoelastic parameter that is assumed to be time independent. The elastic term $E\varepsilon$ does not contribute to damping, and, as noted before, its cyclic integral vanishes. Accordingly with (De Silva 2000), the damping capacity per unit volume under un harmonic load of given frequency ω is:

$$\Phi = \frac{\pi \omega E^* \sigma_{max}^2}{E^2} \tag{C.1.2}$$

where σ_{max} is the maximum stress caused by the harmonic load. For modelling purpose, this damping is usually taken into account in the equations of motion by adding the term $C\dot{x}$, where C is a constant that is derived from the energy dissipated by the material for each cycle.

The *hysteretic damping* is usually related to yielding[1] and relative displacements among elements[2]. In this case it is no longer possible to simply add a term in the force balance of the equation of motions: this type of damping has clearly non-linear characteristics. In order to bring computation difficulties to an acceptable level, an equivalent C_{eq} value of the damping coefficient is often obtained for a given level of excitation and added to the C value of the viscous damping in order to allow linear analyses.

- **Structural damping**: it is a result of mechanical-energy dissipation. This dissipation is caused by the rubbing friction resulting from relative motion between components and by the impacting or the intermittent contact at joints in a mechanical system or structure. Energy-dissipation behaviour depends on the particular mechanical system details. It is extremely difficult to develop a generalised analytical model describing the structural damping. Energy dissipation caused by rubbing is usually represented by a Coulomb-friction model. Energy dissipation caused by impacting, however, should be determined from the coefficient of restitution of the two members that are in contact.

  The common method for estimating the structural damping is by measurement. The measured values, however, represent the overall damping in the mechanical system. The structural damping component is obtained by subtracting the values corresponding to other types of damping (such as material damping present in the system estimated by environment-controlled experiments, previous data, etc.), from the overall damping value.

  This damping is originated at the interface position between the structure and the world around. It can be found at the bridge bearings or in the expansion joints, for example.

- **Fluid damping**: refers to a damping that is external to the structures, i.e. it is related to the ambient where the structure is. Some examples

[1] Especially for metallic elements, but also for composite elements as for example reinforced concrete ones. This damping is usually strictly connected with irreversible damage.

[2] This damping can be found in structures constituted by several assembled elements, as for example in composites structures.

are dissipative effects due to flow-structure interaction (both with air and water), soil-structure interaction at the foundation level and dissipating link between the structure and the surroundings.

C.1.2 Why having a damping matrix?

In modal analysis, there is generally no need to express the damping of a typical viscously damped system by means of the damping Matrix. The damping is more conveniently represented in terms of the modal damping ratios ξ_n (where n is the degree of freedom of the considered structure. In case that the response of the structure cannot be obtained by superposition of the uncoupled modal response, an explicit damping matrix is needed, however. This is true if the structure itself is non-linear (for example in seismic analysis, where the structure can be damaged, so the stiffness and the damping are not constant in time) or because there is the need of simulations in the time domain (as for example in this work, where a non-linear model for the actuators leads to the request of a model for the structure, even if linear).

C.1.3 Rayleigh damping

The simplest way to formulate a proportional damping matrix is to make it proportional both to the stiffness and the mass of the structure. More details about this technique can be found in (Clough and Penzien 1993).

C.1.4 Interstorey damping

Once the damping matrix has been obtained, it is easy use modal techniques to obtain an approximate value of the single damping values for the various structural elements. This can be done only in particular situation, as for example in the case of a building modelled in shear. Under some hypothesis (Paz 1985) the mass is concentrated at each storey and has only one translational degree of freedom. Stiffness and damping are related to the inter-storey displacement and velocity. In this case is interesting to have the values of the inter-storey damping in order to perform simulations.

Appendix D

Some considerations about Time Domain Analysis

The responses of active, passive and semi-active controlled systems can always be compared in the time domain. If the same excitation is used, time-histories can be elaborated in order to find some others comparable indexes as for example the maximum inter-storey drift, the maximum shear force at the ground floor, the maximum acceleration on building's upper storey and so on. Time-histories analysis can be also very useful for the assessment of the control devices behaviour. In the case of a viscous-elastic passive or semi-active device, the energy dissipation due to force vs. displacement cycles can be obtained, for example.

These analyses can be performed taking into account some drawbacks of a time-domain comparison:

- if structure is non-linear, a step by step integration method must be adopted. These methods are usually much more expensive (in terms of computational time) than the corresponding linear ones;
- the structural response is strictly related to the particular forcing signal that has been chosen[1]. Any inference about structural safety or performance must be conducted very carefully;
- the prediction of the expected response under some frequencies content input is still not yet established even if many analysis has been conducted.

[1] Some important characteristics of a seismic input are:

– the duration (for how many seconds the input signal is relevant);

– the amplitude;

– the frequency content;

In principle, it wouldn't possible to speak of 'transfer function' for a semi-active system, because its behaviour is inherently non-linear. However, (Pinkaew and Fujino 2001) proved that the frequency response of a semi-active controlled structure can be obtained under defined conditions.