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Hybrid Semi-Active Mass Dampers in Structures; Assessing and Optimising Their Damping Capacity

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Abstract

The paper discusses the "software"-optimisation of a novel, energy-, cost- efficient hybrid semi-active tuned mass damper configuration applicable to earthquake and wind structural vibration mitigation. Namely, an arrangement of both active and semi-active vibration control components coupled with a range of practical-to-use control algorithms are assessed towards an optimal and fail-safe holistic solution. For brevity, the testbed is the simplest sway single-degree-of-freedom structure under harmonic loading. The analysis for the hybrid vibration mitigation device builds on top of previous findings on the effects of control constraints, such as the stroke and force saturation limits, on the effective structural damping performance. The outcome produced is a hyperstable control solution that while waiving the cumbersome requirement for full-state feedback enables superior performance both in terms of response and energy demand. Essentially such an option satisfies both strict serviceability and sustainability requirements that are often found to govern modern structural applications, yielding a practical, reliable option with broad applicability and efficiency.

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Keywords: Active Control; Semi-Active Hybrid Devices; Vibration Mitigation; PID Controller; Hyperstability;

1. Introduction

Protecting structures from both foreseeable and unforeseeable dynamic loads so as to prevent serviceability or collapse failure, preserve human life as well as improve occupant's comfort, are all factors contributing to the development of technologies and tools, that push the boundaries of structural control innovation. To this end, over

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the years, the civil engineering domain experienced a move from traditional passive control mitigation systems to more sophisticated ones that have the ability to adjust system's dynamic properties in real-time, depending on the structural response or the type of perturbation. A great example of this natural progression, is evident through the evolution of the conventional tuned-mass damper (TMD) to the repeatedly proven superior active mass damper (AMD). Such superior structural control approaches, however, rely on external power for operation which inevitably increase the risks of failure, but also the costs associated with their operation. Additionally, when considering large scale structures, the performance of such configurations is typically limited by the capacity of the installed actuators and/or the auxiliary mass strokes [1-4]. Despite the attempts made to overcome these limitations, either by using different, more efficient and novel-at-the-time AMD configurations such as the swing-style AMD presented in [5], or the electromagnetic device with semi-active control properties presented in [6], amongst many other configurations [3, 7], the crucial absence of a fail-safe mechanism limits the options to structural engineers to an approach that is based on the hybridisation of the AMD device with a component able to prevent undesirable operation upon active component failure. For this reason, most practical structural control configurations comprising a form of active dynamic vibration absorber (DVA) are found in an active-passive hybrid state [8]. Extending the concept of hybridization, most recently Demetriou and Nikitas [9] proposed an alternative form of hybridization of DVA devices, termed semi-active hybrid mass damper (SHMD) which is based on the combination of semi-active and active components for the provision of the required control action, demonstrating that the novel control configuration requires significantly lower energy and actuation demands for achieving a substantial performance increase.

Even though the gains from appropriate configuration selection are evident, a successful control system does not rely exclusively on the individual performance of the hardware, but it is the combined action of both hardware and software that make the control system superior to another. For this reason, this paper, discusses the modification of an existing and popular controller, namely the Proportional-Integral-Derivative(PID) controller in an attempt to optimise the 'Software' part of the SHMD solution. The success of the proposed solution, termed aPID will be evaluated based on the simplicity of implementation, performance/efficiency and robustness compared to the control methods used in previous studies.

1.1. Conventional control approach for ATMD and SHMD

The conventional and simplest approach for the control of both the ATMD and SHMD configurations is based on direct velocity feedback (DVF). In a DVF scheme, the inputs to the actuators are the measured structural velocities (at the location of the actuator i.e. collocated setup) multiplied directly by a gain matrix. The drawback of this simple approach is that even at low gains, the lower frequency poles of the system (which relate to the dynamics of the actuator) rapidly destabilize, placing an upper limit on the range of stable feedback gains that can be used. As a consequence, lowering the feedback gains of the controller to ensure stability will result in a reduction of the effective damping in the actuator, suggesting that the force output (stroke of the mass damper) will be increased. The limitation of rapidly destabilising poles at increasing gains, as well as the associate increase in damper strokes at low gains is believed to be the primary reason why many elaborated control strategies have been introduced for the case of ATMD control. To this end, over the years classical [10], fuzzy [11-13], pole placement [14-16], Lyapunov [17], optimal [18], H-infinity [19], slide mode [20] controllers have been presented in literature.

1.2. Proposed control approach/modification of PID to aPID controller

It is understood that the absence of a zero between the lower frequency pole and the first pole of the structure is the reason why the poles of the system move in the right-hand plane of a pole/zero map [21] . Compensating for the absence of a zero between the pole of the damper and the pole of the structure, a modification of the conventional PID controller via the addition of as second order filter and appropriate tuning of the controller's gains is proposed. This modification, ensures that the resulting closed-loop poles always remain in the left-hand plane of the pole/zero map, satisfying the Routh-Hurwitz criterion. The control architecture in consideration is shown in Fig. 1:

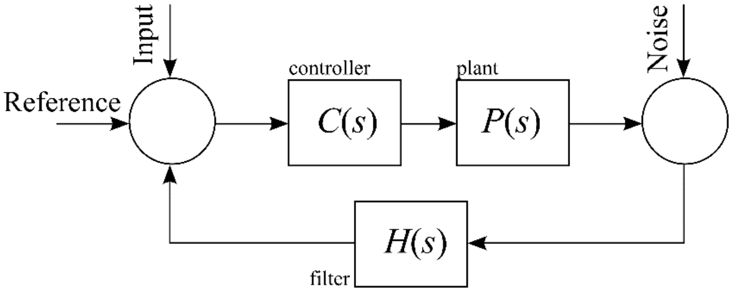

Fig. 1. Robust aPID control system architecture

For the particular control system architecture, $H(s)$ is defined as a second order filter described by the transfer function:

$$H(s)=\frac{s}{as\left(s+\omega_c\right)/2} \tag{1}$$

In which ω_c is the cut-off frequency of the high pass filter and a is the hyperstability parameter (parameter that ensures that the poles of the system remain in the left-hand plane of the pole/zero map). For the tuning of the parameter, a, the frequency of the poles of the system post-application of the TMD must be determined. In this regard, the low frequency pole of the system has a frequency ω_a and a structural frequency ω_b. The robust controller entails that as long $a\in[\omega_a;\omega_b]$ the Routh- Hurwitz criterion is satisfied. After designing the second order filter, one needs to consider the PID controller's transfer function:

$$C(s)=K_p+\frac{K_i}{s}+K_d s=K_p\left(1+\frac{1}{T_i s}+T_d s\right)=K_p\frac{T_iT_d s^2+T_i s+1}{T_i s} \tag{2}$$

In which, K_p K_i and K_d are respectively the proportional, integral and derivative gains of the controller, and $T_i=K_p/K_i$ and $T_d=K_d/K_p$ are the integral and derivative time constants respectively. In order to achieve hyperstability, the following parameter tuning is proposed.

$$K_p=a^2g, \qquad K_i=\frac{a^3g}{2} \qquad K_d=\frac{ag}{2} \tag{3}$$

Where, $g\in[0;\infty]$ is the constant gain of the controller and the hyperstability parameter $a\in[\omega_a;\omega_b]$.

2. Numerical investigation

In this study, the structural systems in consideration are the 2-DOF ATMD and SHMD equipped systems described in [9]. The system's mechanical properties have the following values: $m=1000kg$, $m_d=10kg$, $c=50Ns/m$, $c_d=1.22Ns/m$, $k=1000N/m$ and $k_d=10N/m$. From this, the state space matrices are constructed and converted to the equivalent plant transfer function, $P(s)$ using:

$$P(s)=C(sI-A)^{-1}B+D \tag{4}$$

For the case of SHMD control, the equivalent linear transfer function of the plant must be obtained using identification techniques similar to the ones described in [9].

2.1. Proof of stability

In this study, the Routh-Hurwitz stability test, Bode plots and Root-locus diagrams are used to determine stability. Firstly, the Routh-Hurwitz stability test is used to determine whether the roots of the characteristic polynomial have negative real parts. For this, tables are constructed from the coefficients of the characteristic polynomial, with the number of sign changes in the first column of each table indicating the number of non-negative poles. Fig. 2 illustrates how for values of $g \in [0;\infty] \leftrightarrow a \in [\omega_a;\omega_b]$, the values in the first column of the Routh matrix do not change sign, suggesting the stability of the system.

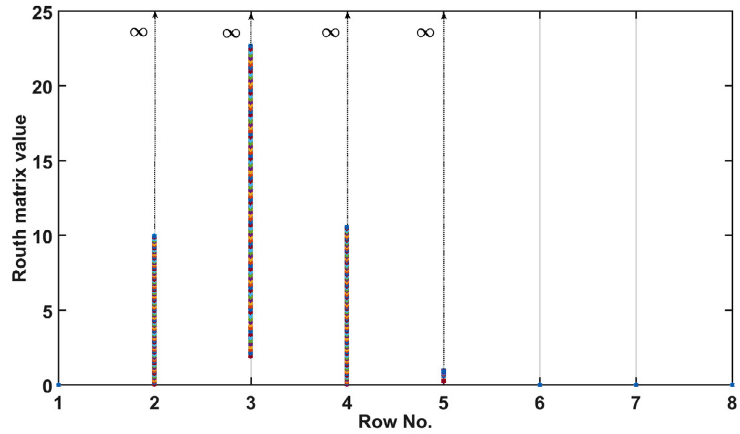

Fig. 2. Values in the first column of the Routh matrix for $g \in [0;\infty] \leftrightarrow a \in [\omega_a;\omega_b]$

Similar observations can be made with reference to the root-locus diagram of the open-loop transfer function of the controlled system (Fig. 3). Evidently, post-application of the robust controller, the system has loci that remain in the left-hand plane demonstrating that regardless the gain, the system's poles attain positive damping values. Complementing this observation, the Bode plot and Nyquist diagram of the system exhibit the infinite gain and phase margins of the controlled system (Fig. 4).

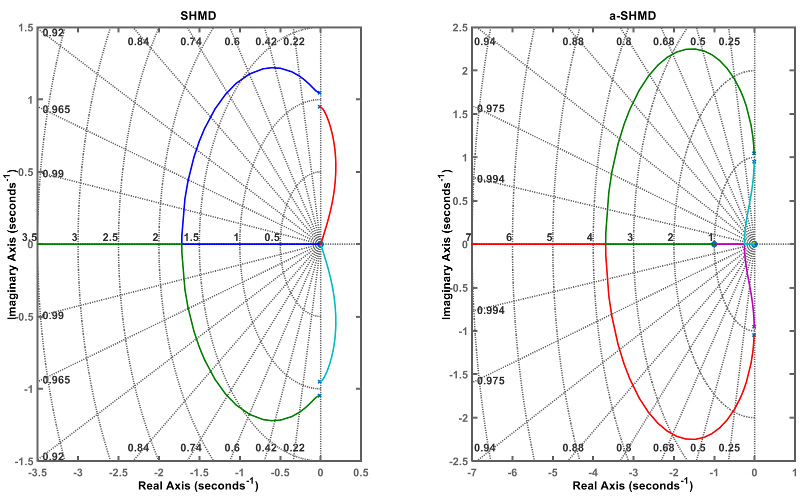

Fig. 3 Root-locus diagram of (a) SHMD and (b) a-SHMD configuration

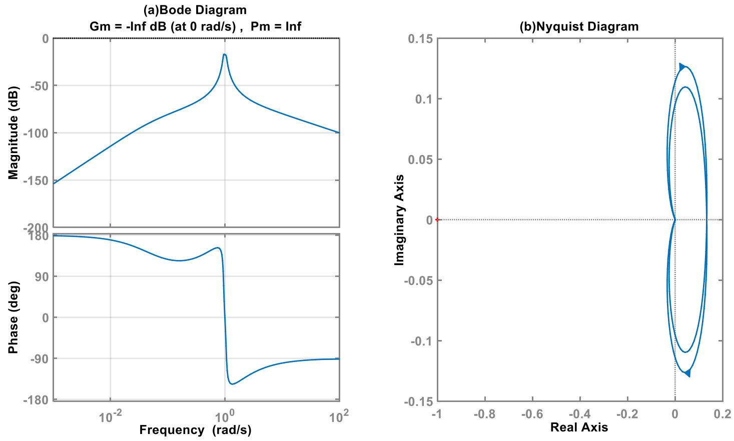

Fig. 4 (a) Bode plot and (b) Nyquist plot for the aPID-SHMD system

3. aPID-SHMD Performance

A linear quadratic regulator (LQR) has been designed for comparison with the proposed robust controller. The design of the LQR is based on the linearized semi-active system following the description found in [9]. In their approach, the Q and R weighting quantities have been iteratively selected such that maximum performance (i.e. vibration attenuation) has been extracted from the system with no limitation to the control effort. This performance-wise optimised LQR allowed the establishment of a performance based comparison reference with the aPID-SHMD counterpart. The response of the LQR system in terms of acceleration, control energy and force demands is shown in Fig. 5.

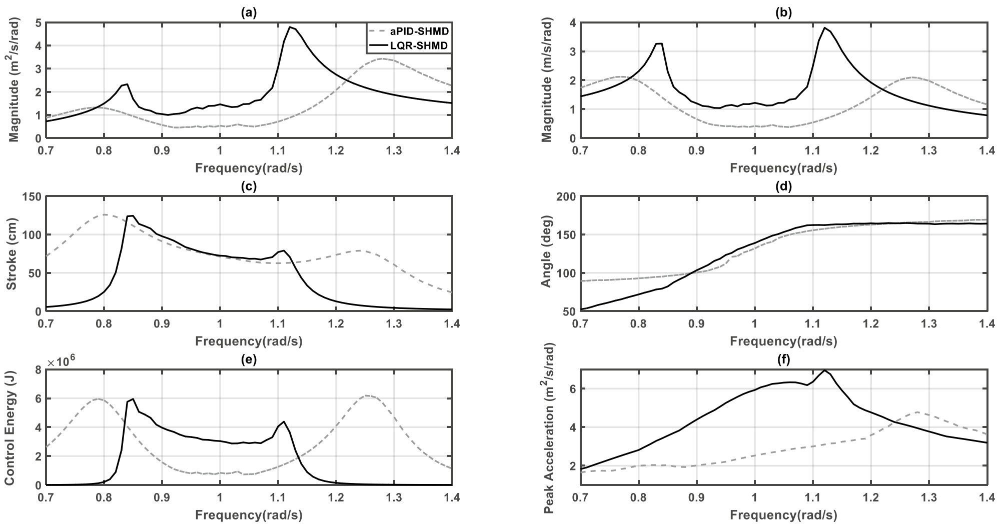

Fig.5(a)Acceleration response, (b) displacement response, (c) damper stroke, (d) relative phase
(e) Control energy and (f) peak acceleration response for the aPID-SHMD and LQR-SHMD systems

Direct comparison of the two algorithms based on similar actuation and control energy demands, demonstrates a clear superiority of the aPID-SHMD over the LQR-SHMD configuration in almost every control design aspect, with only exception the higher strokes required by the aPID-SHMD. But it is not only the enhanced performance of the aPID-SHMD in terms of vibration response reduction that make it superior to the LQR-SHMD. The requirement of the latter configuration for system identification for deriving the state matrices to be used in the solution of the Riccati matrix, make it a complex and non-trivial task. Additionally, using a LQR, full state feedback is required either from direct measurement of all the states (one sensor at each DOF) or using state observers (Kalman filters etc.) further increasing the complexity and practicality. On the other hand, an aPID is based on simple direct velocity feedback control principles, capable of providing control actions using the measurement of only one state.

4. Conclusions

One of the most widely used controllers in the control industry, the PID, has been modified based on the principle of hyperstability in an attempt to provide a simple and effective solution for the control of hybrid ATMD and SHMD devices. To this end, hyperstability conditions have been identified and tuning of the resulting robust aPID controller has been proposed. Proof of stability of the resulting aPID-SHMD system has been demonstrated using Routh-Hurwitz tests, bode plots, Root-locus and Nyquist diagrams. Finally, a comparison with the performance-wise optimised LQR-SHMD system, showed that beyond the simplicity in design and need for a single state measurement of the aPID-SHMD system, the latter system is shown to be superior to the LQR-SHMD configuration in almost every control design aspect, with only exception the marginally higher strokes required by the aPID-SHMD.

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