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Simultaneous active and passive control for eigenstructure assignment in lightly damped systems

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ABSTRACT

The assignment of the eigenstructure (i.e. eigenvalues and eigenvectors) in vibrating systems is an effective way to improve their dynamic performances. System controllability ensures that the poles of the controlled system are exactly assigned but it does not allow to assign arbitrary desired eigenvectors. To this purpose, this paper proposes a novel method for vibration control in lightly damped systems through the concurrent synthesis of passive structural modifications and active state (or state derivative) feedback control gains. Indeed, the suitable modification of the inertial and elastic parameters allows to enlarge the range of assignable eigenvectors. The problem is formulated as an optimization problem, where constraints are introduced to assure the feasibility of the physical system modifications while avoiding spillover phenomena. The experimental application to the eigenstructure assignment on a manipulator proves the method effectiveness.

1. Introduction

1.1. Background and motivations

Dynamic performances of vibrating systems can be defined through their eigenstructure, i.e. the eigenvalues and the eigenvectors. In control theory more attention is usually paid to the closed-loop system eigenvalues, i.e. the poles, since they set damping, stability and the speed of response. Nonetheless eigenvectors also play a fundamental role since they fix the eigenvalue sensitivities, and therefore the controller robustness, and define the spatial shape of the vibrations. The simultaneous assignment of both the eigenvectors and the eigenvalues is usually referred to as eigenstructure assignment (EA). A typical application of EA is, for instance, vibration or noise confinement [1,2] and isolation [3], which are aimed at modifying eigenvectors so that vibrational modes have smaller amplitude in some parts of the system rather than in the others. As a result, vibration energy is "confined" to parts of the system where it is admissible [2]. On the other hand, the possibility to shape the spatial distribution of the vibration is also an effective way to optimize the dynamical behaviour of vibration generators, which are required to vibrate at a specified frequency with a desired mode shape and without introducing damping. For example, in [4] such an idea is applied to the design of linear vibratory feeders. Another meaningful example where EA has shown advantages over other control techniques is flight control in aircrafts and helicopters [5]. In such a field, EA decouples control actions to provide simpler and more responsive manoeuvring. EA is therefore a general technique, which can be applied to a wider range of applications and systems, even when damping is unwanted and

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therefore vibration control through active damping cannot be applied (see e.g. the abovementioned vibration generators).

Passive methods, i.e. Dynamic Structural Modification (DSM) of physical parameters, have been often proposed as effective means to provide approximate solution of EA [4,6–8]. While DSM preserves stability and does not require additional sensors and actuators, the achievable performances are limited by the symmetric nature of the modifications and often by technical or economical feasibility constraints.

To overcome the limitations of passive methods, active feedback control (AC) is often exploited. The effectiveness of active EA was first recognized by Moore in [9] and several methods to synthesize the controller have been proposed in literature (see e.g. [10,11]). However some concerns also affect active eigenvector assignment. Indeed, such an approach is effective only with multiple control inputs. For instance, the method proposed in [12] for active vibration confinement requires the number of actuators to match the number of system degrees of freedom (dofs). If less actuators are adopted, the feasible eigenvectors are constrained to lie in an allowable subspace and the desired eigenvectors are usually approximated by projection to the feasible set [5]. This condition often leads to a rough approximation of the desired eigenvectors in the case of highly underactuated systems.

1.2. Research objectives

Since passive and active control approaches when used alone have both shown limitations, a novel hybrid active-passive approach to EA is proposed in this paper, in order to solve the simultaneous synthesis of a state feedback controller and of the modification of inertial and elastic parameters. The method is suitable for lightly damped vibrating systems. The capability to assign the desired eigenstructure is enlarged by properly combining AC and DSM through an integrated approach, thus ensuring a much closer fulfilment of the desired eigenstructure specifications.

The idea of hybrid control (HC), as a combination of active and passive actions, has been already shown to be successful in some different applications of vibration control. However only a few examples have been proposed. Yet it has never been applied to the general problem of EA. In [13] a hybrid active-passive scheme is suggested to perform vibration confinement, where passive mechanical absorbers are used to reduce active power requirements. An hybrid vibration confinement technique making use of a network of piezoelectric actuators has been presented in [2]. In [14] the simultaneous optimization of structure and control is proposed using mixed H_2/H_∞ norms of the transfer function from the disturbance to the output. In [15] DSM and AC are sequentially combined to effectively perform pole placement in asymmetric systems.

The main issues to be tackled in developing an effective HC scheme for EA is synthesizing active (controller gains) and passive (physical modification) terms in a concurrent and integrated way, by accounting for all the mutual effects. Following this idea, this paper proposes an hybrid approach to tackle the challenging problem of EA. Indeed, passive control can modify the set of allowable eigenvectors so that the desired eigenpairs can be assigned through some suitable controller gains. An optimization problem, which includes simultaneously active and passive terms, is cast by taking advantage of the formulation of the DSM problem adopted in [4] and by posing constraints ensuring the technical feasibility of the passive modifications and avoiding spillover. Some suitable mathematical tools are adopted in the formulation to improve the problem solvability and to increase its robustness with respect to model uncertainty.

The method proposed is general and is suitable to all those applications where EA is usually applied, or whenever performances of state (or state derivative) feedback control are to be improved by setting the eigenvectors. The method can be also interpreted, with a different perspective, as an approach to control–oriented mechatronic design, where the system inertial and elastic parameters are designed concurrently with the controller in accordance to the achievable performances of the controlled system [18]. Indeed, with a mechatronic approach, mechanics and control aspects should be analysed simultaneously [19].

The experimental assessment of the method is also performed in order to show the method effectiveness in enlarging the range of assignable eigenvectors and to corroborate its robustness in the presence of the unavoidable uncertainties affecting experimental devices. The experimental validation is presented for an application to a sample two-link planar manipulator with a single control force. The test case is significant and has been carefully chosen: EA in the presence of rank-one control is not trivial since the single control force does not allow to assign exactly arbitrary desired eigenpairs.

2. Eigenstructure assignment: traditional problem formulation

The theory of EA through either purely passive or active approaches is discussed in this Section, with reference to an arbitrary N-dimensional multi-dof linear system modelled as follows:

$$\mathbf{M}\ddot{\boldsymbol{q}}(t) + \mathbf{C}\dot{\boldsymbol{q}}(t) + \mathbf{K}\boldsymbol{q}(t) = \mathbf{B}\boldsymbol{v}_A(t) \tag{1}$$

where $\mathbf{M}$, $\mathbf{C}$, $\mathbf{K} \in R^{N\times N}$ are the mass, damping and stiffness matrices, (R is the set of the real numbers), $\boldsymbol{q}(\mathrm{t}) \in R^{N\times 1}$ is the displacement vector, $\boldsymbol{v}_A(t)$ is the control force vector, $\mathbf{B} \in R^{N\times b}$ is the control force distribution matrix (b is the number of actuation forces).