

Active Control of Engine-Induced Vibrations in Automotive Vehicles through LPV Gain Scheduling

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ABSTRACT

In this paper, a control approach for the active reduction of engine-induced vibrations in automotive vehicles is presented. As a controller, a discrete-time multiple input multiple output (MIMO) disturbance-observer-based state-feedback controller is designed using linear parameter-varying (LPV) gain-scheduling techniques. The use of LPV control design techniques has the advantage that the stability of the overall system is guaranteed even when the gain-scheduling parameters are changing. The control approach is validated experimentally with an active vibration control system installed in a Golf VI Variant. Two inertia-mass actuators (shakers) and two accelerometers are attached to the engine mounts. Nine frequency components are targeted in the reduction and excellent results are achieved in vehicle driving tests for constant and time-varying engine speeds.

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1. INTRODUCTION

Noise, vibration and harshness (NVH) characteristics of cars and trucks are becoming increasingly important over the last few years (see [1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11] and the references therein). This work focuses on the active reduction of the engine-induced vibrations in automotive vehicles using linear parameter-varying (LPV) gain-scheduling techniques.

Passive methods for vibration isolation like rubber mounts require a trade-off between vibration isolation and engine movement. A low stiffness gives good vibration isolation but leads to a large static engine displacement and to a low resonance frequency which might coincide with the resonance frequencies of the suspension. A little internal damping leads to a high resonance frequency and might cause an excessive engine movement if the resonance frequency is excited. Vibration isolation and engine movement are weighted to achieve a desired performance and comfort.

An alternative that solves this problem is the use of active noise control (ANC) or active vibration control (AVC) techniques [1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12]. Adaptive filtering updating the filter weights through the Filtered-x LMS (FxLMS) algorithm is the common approach in ANC/AVC. The main disadvantage of this approach is that the analysis of the closed loop is difficult. The stability and transient behavior (convergence speed) depend on the system input. Also, to the best of the authors' knowledge, only "approximate stability results" are available for the FxLMS algorithm [13].

Another approach is the use of gain-scheduling controllers for the rejection of periodic disturbances with time-varying frequencies. LPV techniques can be used and the stability is then guaranteed for changes in the gain-scheduling parameters [14, 15, 16, 17, 18, 19, 20, 21, 22, 23, 24, 25, 26, 27, 28, 29, 30, 31]. All controllers contain a model of the harmonic disturbance to achieve disturbance rejection according to the internal model principle [32]. Different control approaches as general output-feedback controllers [14, 15, 16, 17, 18, 19, 20, 21, 22, 23, 24, 25, 26] or observer-based state-feedback controllers [27, 28, 29, 30, 31] can be used to obtain a controller including the model of the disturbance.

In this paper, a discrete-time multiple input multiple output (MIMO) observer-based gain-scheduling controller is designed and LPV techniques are used. The obtained controller is validated experimentally in a Golf VI Variant to reduce the engine-induced vibrations in the chassis of the vehicle. Two inertia-mass actuators (shakers) and two accelerometers are attached to the engine mounts. The controller uses the measurements of the accelerometers to generate the cancelling signal of the shakers. Anti-aliasing filters are applied to the output signals and reconstruction filters to the control inputs. The controller is designed for the rejection of nine frequency components and is implemented on a rapid prototyping unit (MicroAutoBox from dSPACE) with a sampling frequency of 2 kHz. The two gain-scheduling parameters depend on the engine speed measured by the CAN bus and the engine speed is used for the interpolation of the controller from three controller vertices. Excellent results are achieved for

test drives with constant engine speeds and for time-varying engine speeds. The stability of the controller is guaranteed even for fast changes of the engine speed since LPV control design techniques were used.

This paper is an extension of previous work of the authors. The present approach can be seen as an extension of [6, 7]. In [6, 7], an observer-based state-feedback approach was used for the reduction of engine-induced vibrations in a passenger car. However, heuristic gain-scheduling was used and stability of the closed-loop system could not be guaranteed (the design was based on “frozen stability”). Also, a single input single output (SISO) system was used.

Here, a discrete-time LPV gain-scheduling MIMO observer-based controller for the rejection of harmonic disturbances is designed, a polynomial approximation to reduce the number of scheduling parameters is used and the controller is tested on a vehicle to reduce the engine-induced vibrations. The controller has only two gain-scheduling parameters and it is interpolated from only three controller vertices with a very simple interpolation. Excellent results for the reduction of nine frequency components are obtained for a specified engine speed range.

The remainder of this paper is organized as follows. The control design techniques used to obtain an LPV gainscheduling controller for the suppression of harmonic disturbances is explained in [Sec. 2](#). The experimental setup for the validation of the controller in a Golf VI variant is described in [Sec. 3](#). In [Sec. 4](#), experimental results for the reduction of the engine-induced vibrations in vehicle driving tests are shown. A summary and some conclusions are given in [Sec. 5](#).

2. CONTROL DESIGN

In this section, the control design to obtain an MIMO LPV observer-based controller for the reduction of the engine-induced vibrations is described. According to the internal model principle [32], for asymptotic disturbance rejection, the controller has to include a model of the disturbance. The LPV control design techniques used in this section to obtain a controller for the rejection of harmonic disturbances are based on the internal model principle. A time-varying disturbance model is needed for the control design since the spectral content of the engine-induced vibrations are time varying (due to the varying engine speed). An LPV representation of the disturbance model with a polynomial approximation to reduce the number of gain-scheduling parameters is obtained in [subsection 2.1](#). The polytopic LPV (pLPV) control structure that contains a model of the disturbance is explained in [subsection 2.2](#). Finally, in [subsection 2.3](#) the vertex gains are obtained.

2.1. LPV Disturbance Model

The vibration caused by the engine can be modeled as disturbances with harmonically related time-varying frequencies.

A state-space representation of a discrete-time $n \times n$ MIMO disturbance model with n_d time-varying frequencies is given by

$$\mathbf{x}_{d,k+1} = \mathbf{A}_{d,k} \mathbf{x}_{d,k}, \quad (1)$$

$$\mathbf{y}_{d,k} = \mathbf{C}_d \mathbf{x}_{d,k} \quad (2)$$

with

$$\mathbf{A}_{d,k} = \begin{bmatrix} \mathbf{A}_{d_1,k} & \cdots & \mathbf{0} \\ \vdots & \ddots & \vdots \\ \mathbf{0} & \cdots & \mathbf{A}_{d_{n_d},k} \end{bmatrix}, \quad (3)$$

$$\mathbf{A}_{d_1,k} = \cdots = \mathbf{A}_{d_{n_d},k} = \begin{bmatrix} \tilde{\mathbf{A}}_{d_1,k} & \cdots & \mathbf{0} \\ \vdots & \ddots & \vdots \\ \mathbf{0} & \cdots & \tilde{\mathbf{A}}_{d_{n_d},k} \end{bmatrix}, \quad (4)$$

$$\tilde{\mathbf{A}}_{d_i,k} = \begin{bmatrix} r \cos(\Omega_{i,k}) & r \sin(\Omega_{i,k}) \\ -r \sin(\Omega_{i,k}) & r \cos(\Omega_{i,k}) \end{bmatrix}, \quad \Omega_{i,k} = 2\pi f_{i,k} T,$$

$$f_{i,k} \in [f_{i,\min}, f_{i,\max}] \text{ for } i = 1, \dots, n_d \text{ and}$$

$$\mathbf{C}_d = \begin{bmatrix} \mathbf{C}_{d_1} & \cdots & \mathbf{0} \\ \vdots & \ddots & \vdots \\ \mathbf{0} & \cdots & \mathbf{C}_{d_{n_d}} \end{bmatrix}, \quad \mathbf{C}_{d_1} = \cdots = \mathbf{C}_{d_{n_d}} = [1 \ 0 \ \cdots \ 1 \ 0]. \quad (5)$$

An LPV representation of this model is obtained choosing $r \cos(\Omega_{i,k})$ and $r \sin(\Omega_{i,k})$ as time-varying parameters. The resulting LPV disturbance model for n_d time-varying frequencies has $2n_d$ gain-scheduling parameters.

To reduce the number of gain-scheduling parameters a model for constant frequencies with the matrix

$$\tilde{\mathbf{A}}_{d_i,k} = \begin{bmatrix} 0 & 1 \\ -r^2 & 2r \cos(\Omega_i(k)) \end{bmatrix} \quad (6)$$

can be used. In this case n_d gain-scheduling parameters are needed even if the frequencies are harmonically related. This model is correct only for fixed frequencies.

A polynomial approximation is presented in [22, 23] to reduce the number of gain-scheduling parameters if the frequencies are harmonically related. In [22, 23], this approximation is used with the model for constant frequencies (6) to solve an SISO AVC problem in simulation. Here, this polynomial approximation is used and modified for a time-varying disturbance model for an MIMO AVC problem. The controller is implemented in real time and tested experimentally on a vehicle.

Using the polynomial approximation of [22, 23], the cosine function and the sine function can be written as

$$r \cos(\Omega_{i,k}) \approx a_{i,0} + a_{i,1}\Omega_{0,k} + \dots + a_{i,N_p}\Omega_{0,k}^{N_p} \quad (7)$$

$$r \sin(\Omega_{i,k}) \approx b_{i,0} + b_{i,1}\Omega_{0,k} + \dots + b_{i,N_p}\Omega_{0,k}^{N_p} \quad (8)$$

for n_d harmonically related time-varying frequencies

$$\mathbf{f}_k = [f_{1,k}, f_{2,k}, \dots, f_{n_d,k}] = [h_1, h_2, \dots, h_{n_d}] \mathbf{f}_{0,k} \quad (9)$$

The coefficients $a_{p0}, \dots, a_{pN_p}$, and $b_{p0}, \dots, b_{pN_p}$ can be determined through a least squares fit. A very good approximation is achieved in experiments and simulations with three coefficients. Therefore here the cosine and sine function are approximated as

$$r \cos(\Omega_{i,k}) \approx a_{i,0} + a_{i,2}\Omega_{0,k}^2 + a_{i,4}\Omega_{0,k}^4 \quad (10)$$

$$r \sin(\Omega_{i,k}) \approx b_{i,0} + b_{i,2}\Omega_{0,k}^2 + b_{i,4}\Omega_{0,k}^4 \quad (11)$$

The time-varying parameters

$$\theta_{1,k} = \Omega_{0,k}^2 = (2\pi f_{0,k}T)^2, \quad (12)$$

$$\theta_{2,k} = \Omega_{0,k}^4 = (2\pi f_{0,k}T)^4 \quad (13)$$

are introduced and then, the system matrix $\mathbf{A}_{d,k}$ in (1) can be written as

$$\tilde{\mathbf{A}}_{d,k} \approx \begin{bmatrix} a_{i,0} & b_{i,0} \\ -b_{i,0} & a_{i,0} \end{bmatrix} + \begin{bmatrix} a_{i,2} & b_{i,2} \\ -b_{i,2} & a_{i,2} \end{bmatrix} \theta_{1,k} + \begin{bmatrix} a_{i,4} & b_{i,4} \\ -b_{i,4} & a_{i,4} \end{bmatrix} \theta_{2,k} \quad (14)$$

for $i = 1, \dots, n_d$ using only two gain-scheduling parameters independently of the number of frequencies. An LPV representation of the disturbance model for n_d frequencies is obtained with the polynomial approximation with only two gain-scheduling parameters.

Powerful control design methods can be used if the parameters $\theta_{1,k}$ and $\theta_{2,k}$ vary inside a polytope. Since the relationship between the parameters is known

$$\theta_{2,k} = \theta_{1,k}^2 \quad (15)$$

a triangle as polytope is in the next subsection obtained and polytopic control design methods are used.

2.2. pLPV Control Structure

In this subsection, the disturbance-observer state-feedback control structure is introduced. The control structure is a state-feedback gain combined with an observer for the augmented system.

The plant is described by the state-space representation

$$\mathbf{x}_{p,k+1} = \mathbf{A}_p \mathbf{x}_{p,k} + \mathbf{B}_p (\mathbf{u}_{p,k} + \mathbf{y}_{d,k}), \quad (16)$$

$$\mathbf{y}_{p,k} = \mathbf{C}_p \mathbf{x}_{p,k}, \quad (17)$$

where the disturbance $\mathbf{y}_{d,k}$ is modeled as the output of an unforced linear time-varying exo-system given by (1) and (2) acting at the input of the plant.

The augmented system containing the states of the plant and the disturbance model can be written as

$$\mathbf{x}_{k+1} = \mathbf{A}_k \mathbf{x}_k + \mathbf{B} \mathbf{u}_{p,k}, \quad (18)$$

$$\mathbf{y}_{p,k} = \mathbf{C} \mathbf{x}_k \quad (19)$$

with

$$\mathbf{x}_k = \begin{bmatrix} \mathbf{x}_{d,k} \\ \mathbf{x}_{p,k} \end{bmatrix}, \quad (20)$$

$$\mathbf{A}_k = \begin{bmatrix} \mathbf{A}_{d,k} & \mathbf{0} \\ \mathbf{B}_p \mathbf{C}_d & \mathbf{A}_p \end{bmatrix}, \quad (21)$$

$$\mathbf{B} = \begin{bmatrix} \mathbf{0} \\ \mathbf{B}_p \end{bmatrix} \quad (22)$$

and

$$\mathbf{C} = [\mathbf{0} \quad \mathbf{C}_p]. \quad (23)$$

Once the augmented system is obtained, an observer-based state-feedback controller can be designed. This control structure is shown in Fig. 1. The control law is given as

$$\mathbf{u}_{p,k} = -\mathbf{K} \hat{\mathbf{x}}_k \quad (24)$$

where