

Integrated Solutions for Noise and Vibration Control in Vehicles

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ABSTRACT

The automotive industry is aiming at both reducing the weight of the vehicles while improving a high level of comfort. This causes contradicting requirements for the systems used for noise and vibration control. Thus, active systems are investigated which may enhance the performance of passive noise and vibration control in vehicles without adding excessive weight. In this paper, basic principles for the implementations of those systems with a focus on the control systems are reviewed. Examples from implementations in automotive applications are presented, including control of engine vibrations, structure borne noise transmitted from the road into the passenger compartment and low-frequency chassis vibrations. Based on adaptive filter systems already widely used in active noise control adaptation of the control algorithms to the specific application scenarios are discussed. This includes different configurations for feedforward and feedback control, single- and multichannel control systems and the utilization of different actuator concepts like active mounts, inertial mass actuators and active tuned absorbers, as well as different control sensors like accelerometers and microphones. Finally, current research topics and respective concepts for the improvement of active control systems are introduced.

CITATION: Mayer, D., Militzer, J., and Bein, T., "Integrated Solutions for Noise and Vibration Control in Vehicles," *SAE Int. J. Passeng. Cars - Mech. Syst.* 7(3):2014, doi:10.4271/2014-01-2048.

INTRODUCTION

The automotive industry is more and more facing the problem of reducing the weight of the vehicle but guaranteeing an equivalent level of comfort in terms of noise, vibration, and harshness (NVH). Improvement of vehicle noise and vibration without affecting other performances is proving to be extremely difficult if not impossible with state-of-the-art technology. Besides advanced passive material systems, active or smart concepts are being increasingly considered for the NVH optimization of vehicles. However, to overcome contradicting requirements traditional design and material choices must be revisited. Frequently, new technologies in the fields of smart materials and active control provide potential solutions although they have mainly been proved in the laboratory.

The potential for active noise and vibration control in automotive applications has been demonstrated already some time ago. Extensive research was committed to active noise control (ANC) with loudspeakers [1], and those systems have been industrialized in the last years also in the automotive industry. Also the principles of active vibration control (AVC) and active structure acoustic control (ASAC), which utilize dynamic active forces for disturbance compensation, have been investigated. Mainly, the transmission of vibrations from the engine has been tackled by active systems, in some cases also in combination to ANC systems [2]. There are some main principles for the implementation of the actuation, like active

absorbers or inertial mass actuators for inducing point forces [3], active mounts for enhancing isolation between two subsystems [4], or foil actuators directly attached to a vibrating panel [4, 5].

Also for the control systems, some basic methods can be distinguished, regarding the type of disturbance to be tackled and also the layout of the control system. This paper addresses these different active measures for NVH control with a focus on the control approaches and experiences with system integration from a number of conducted projects. The rest of the paper is organized as follows: The different vibration problems and types of control systems are briefly recapitulated. The considered control algorithm is summarized. Afterwards, the application of this approach to several different noise and vibration problems from the automotive industry is described. In each section, the necessary adaptation of the algorithm, the system integration and experimental results are introduced. In the final section, open issues and related current research are summarized, which can enhance the level of system integration and enable broader applicability of these systems.

PROBLEM DEFINITION

The tasks for active noise and vibration control in vehicles can be distinguished into some basic classes. Regarding mechanical design, an active noise control system, an active engine mount and an active inertial mass actuator are

completely different systems; the design of the control system mainly depends on the characteristics of the disturbances to be cancelled, the possibility to get proper sensor signals and the characteristics of the control path.

Types of Disturbances

With respect to the control system, active noise and vibration control can be reduced to the cancellation of disturbance signals. In the following, mainly two kinds of signals will be distinguished:

Periodic signals are mainly excited by the engine and the drivetrain. Usually, several orders are present. However, since the frequency of the signal only varies slowly with time, the characteristics are predictable, which is a prerequisite for gaining a high level of attenuation with an active system.

Random signals are originating e.g. from the contact from the car to the road. Irregularities in the surface are transmitting structure borne sound into the car, or excite global vibrations of the chassis. These signals are mainly unpredictable, but at least for a certain time stationary and mostly always broadband. However, the respective sensor signals measured at the structure are filtered with the respective characteristic, i.e. the transfer function from the disturbance source to the sensor. Thus, even if the disturbance would be white noise, the autocorrelation of the sensor signal reflects the characteristics of the structure.

Control Systems

Two main control system layouts have been established for active noise and vibration control: Feedback and feedforward systems. Basic configurations are shown in Figure 1. In both cases, the goal is to minimize the signal e , which is a superposition of the disturbance signal d and the output y of the control path S . The latter describes the characteristics of the path from the actuator to the sensor, where the disturbance is measured. The main difference is the input signal for the controller W . While for a feedback system, the output signal is used, a feedforward system utilizes a reference signal r . As well known, the feedback configuration causes stability issues. For the feedforward system however, a reference signal correlated to the disturbance is needed, which might not be easy to generate in all cases.

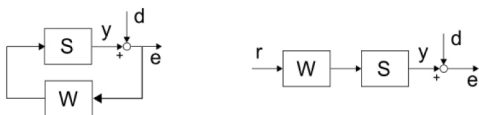


Figure 1. Feedback and feedforward configuration for disturbance cancellation.

Feedforward systems for disturbance cancellation are quite often implemented on basis of adaptive digital filter systems, which will be introduced in the next section. Due to the high stability, this configuration is often favored, if a reference signal

can be found in the system. If no such signal can be used, feedback systems have to be applied. In some cases, even simple control schemes for feedback control can be applied successfully. Feedback of velocity signals can be used to enhance damping and in turn reduce resonant oscillations [5]. However, the feedback problem can be turned into a feedforward one, enabling the application of the similar algorithms for disturbance cancellation.

ADAPTIVE ACTIVE CONTROL SYSTEMS

Disturbance Cancellation with the LMS Algorithm

The basic idea of the active control systems introduced in the following sections is the cancellation of disturbances with a suitable signal, which is gained from filtering a reference signal (Figure 2 a)). Characteristics of disturbance signals most probably will change during operation. This motivates the application of adaptive digital filters for this task. In the following paragraphs, the derivation of the stochastic Least Mean Square (LMS) algorithm invented by Bernard Widrow [6] is briefly summarized.

In the discrete time domain, the superposition of the disturbance and the reference convolved with the impulse response of the filter can be written as:

$$\sum x(n-l)w(l) - d(n) = 0 \quad (1)$$

The goal of the adaptation is to minimize the average power of the output, the *error signal* $e(n)$:

$$\sum e^2(n) = \min \quad (2)$$

This can be achieved by the stochastic LMS algorithm, which reads:

$$\mathbf{w}(n+1) = \mathbf{w}(n) + \mu \mathbf{x}(n)e(n), \quad (3)$$

with μ being the adaptation step size (Figure 2 b)). This algorithm can be readily used for cancellation of electrical disturbance signals. In active control applications however, the disturbance signal is not cancelled by the output of the adaptive filter W directly, but at the output of the path from the control actuator to the sensor, the secondary path $S(z)$. To stabilize the adaptation process, the related phase shift and amplitude variation of the filter output is compensated by the integration of a model of the secondary path $\hat{S}(z)$, leading to the Filtered-X LMS (FXLMS) algorithm (Figure 2 c)):

$$\begin{aligned}
 x'(n) &= \sum \hat{s}(l)x(n-l) \\
 \mathbf{w}(n+1) &= \mathbf{w}(n) + \mu \mathbf{x}'(n)e(n)
 \end{aligned}
 \quad (4)$$

This algorithm is the basis for many active noise and vibration control systems implemented [7], and also for the examples studied in the remainder of this paper.

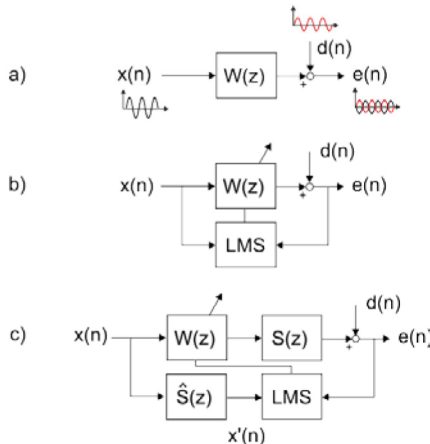


Figure 2. a) Principle of disturbance signal cancellation, b) implementation with an adaptive filter and c) the FXLMS algorithm.

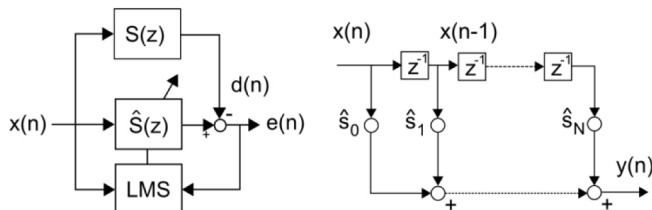


Figure 3. a) Adaptive LMS for system identification.

A prerequisite for the implementation of the FXLMS algorithm is the identification of the secondary path. A practical approach is to utilize the adaptive LMS scheme for this task as well (Figure 3a)). Due to their inherent stability and nice convergence properties, adaptive finite impulse response (FIR) filters are widely used [6]. In active vibration control systems, the secondary path is characterized by the resonant behavior of the dynamic structure under control. Thus, the impulse response is usually quite long which is directly reflected by a high number of coefficients of the FIR filter (Figure 3b)). This might lead to issues when an implementation on a real-time system is considered.

EXPERIMENTAL IMPLEMENTATIONS AND RESULTS

Cancellation of Engine Vibrations

In vehicles, engine excited vibrations contribute a large amount to the interior noise. Due to unbalanced mass excitation and the periodical combustion process, the excitation is correlated to the rotational speed of the engine. Thus, one can easily find a signal that is strongly correlated to the disturbance by

measuring the rotational speed and generating a reference signal by means of a controlled harmonic oscillator [8]. Because of the availability of a suitable reference signal a feedforward control system can be implemented.

Beside the positive effect that the reference signal is well correlated to the disturbance, there is no feedback of the controlled actuator to the synthetically generated reference signal. So no additional feedback compensation has to be implemented.

Implementation of Algorithm

In the presence of a multi-order disturbance signal, several harmonic reference signals (one for each order considered) ought to be generated by means of harmonic oscillators.

Figure 4 shows the applied narrowband (NB) FXLMS algorithm. The harmonic cosine signal is extended to a quadrature reference signal by means of a 90° phase shift which can be realized using a first-order all-pass filter [7].

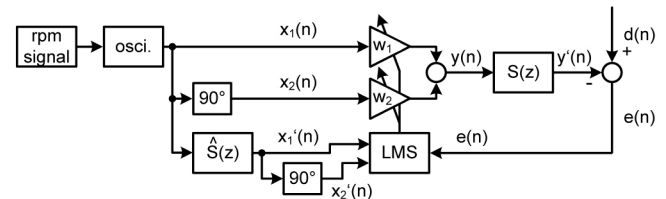


Figure 4. Narrowband FxLMS control algorithm.

The actuator control signal $y(n)$ is a linear combination of the sinusoidal reference signals $x_1(n)$ and $x_2(n)$ whose adaptive weights w_1 and w_2 are readjusted in every single sampling step by means of the gradient based NB FXLMS adaptation algorithm.

$$w_{1,2}(n+1) = (1 - \gamma\mu) \cdot w_{1,2}(n) + \mu x_{1,2}'(n)e(n)
 \quad (5)$$

The adaptive weights w_1 and w_2 are updated iteratively using the filtered reference signals $x_1'(n)$ and $x_2'(n)$ and an adaptation step size μ . In order to avoid divergence and improve stability, an additional leakage factor γ with $0 < \gamma \leq 1$ is added to the update equation [7, 9].

Realisation of an Active Engine Mount Control System

The described control system was applied to a resilient active engine mount. The active mount is located at the gear box site of the engine and designed to have similar stiffness compared to the serial part in the loading force axis direction. Additional dynamic forces are induced by a stroke amplified piezo actuator which is decoupled from static loads by means of an additional high viscous damping element. A brief presentation of this novel active engine mount and its model based development can be found in [10, 11]. However, two different control setups were considered to be evaluated during experimental testing on a test bench. Within the first setup, the

acceleration at the mounting point of the active mount at the chassis side in vertical direction was a target for vibration suppression (AVC) whereas a lowered acceleration level should result in an also lowered sound pressure level. In the second setup the sound pressure at the driver's ear serves as a target for the reduction of harmonic disturbances (ASAC).

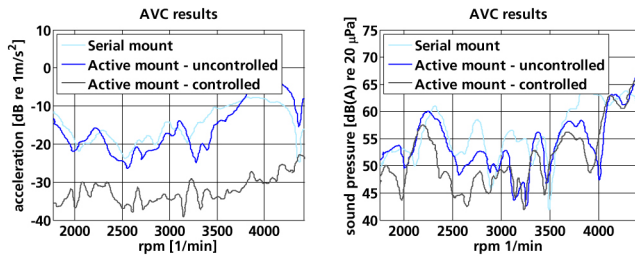


Figure 5. Comparison of serial and novel active engine mount in case of AVC. Acceleration (left); Sound pressure (right).

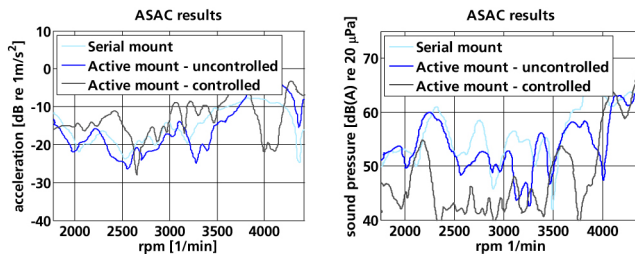


Figure 6. Comparison of serial and novel active engine mount in case of ASAC. Acceleration (left); Sound pressure (right)

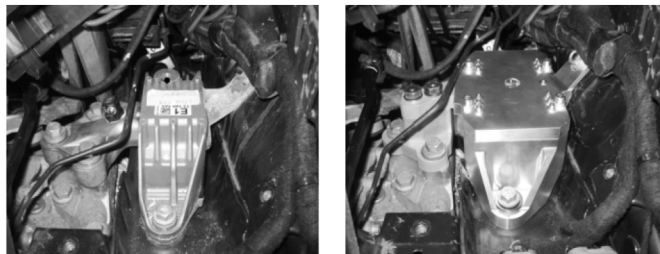


Figure 7. Serial mount (left) and novel active engine mount (right) at the gear box side of the engine.

Figure 5 and Figure 6 show the results for the dominating second engine order and the two test setups. For both scenarios the active engine mount shows comparable results to the serial engine mount in the case that no control signal is applied to the actuator. In the first controlled setup (AVC), the acceleration at the mounting point is significantly lowered compared to the serial mount. The sound pressure at the driver's ear is also lowered due to the reduced excitation of the car's chassis and the improved dynamic isolation of the active engine mount. In the second controlled setup (ASAC) the sound pressure could be lowered once more. However, considering the acceleration level, it is slightly higher compared to the serial mount, as the control algorithm tries to reduce the sound pressure that is induced over the non-active mounting points of the engine as well. This effect was also observed in other experimental realizations of active mounts [12].

Realisation of a Multichannel System

In many applications, multiple actuators are required for the active vibration control system. For larger Diesel engines, all four engine mounts are contributors to the transmitted vibrations. Thus, the control system includes four actuators and four accelerometers as error sensors. As an example, the Diesel engine of a small rescue ship is presented [5]. The experimental set up comprises the ship engine, the active mounts, a segment of the hull and an electrical generator connected to the engine to simulate the drivetrain (Figure 8 a)). Four active mounts were used as actuators; each collocated with an accelerometer mounted hull-side (Figure 8 b))

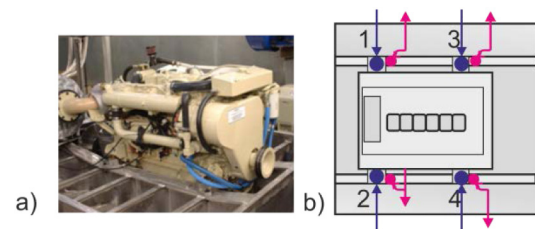


Figure 8. a) Diesel engine of a ship on a hull segment. b) configuration of the active mount system

The extension of the single input-single output (SISO) FXLMS algorithm to a multiple input-multiple output (MIMO) system can be realized straightforwardly [7]. The computation of each actuator signal requires two filter coefficients. Equation 6 shows the adaptation equation for the first adaptive weight of actuator k for a total amount of M error sensors.

$$w_{1,k}(n+1) = (1 - \gamma\mu) \cdot w_{1,k}(n) + \mu \sum_{m=1}^M x'_{1,km}(n) e_m(n) \quad (6)$$

Issues mainly arise when the implementation on a real-time platform is considered. The aforementioned FIR filters still perform well, when enough coefficients are used. However, since each channel has to be modelled, in the case of the active engine mount system, 16 secondary path models with 512 taps each have been identified, which imposes a significant computational load on a real-time control system (Figure 9). Several ways to decrease the complexity of the algorithm have been studied. Obviously, the secondary paths possess different amplitudes. Since the cross-coupling between the positions is smaller than the coupling of the actuator to the collocated sensor for most of the paths, a straightforward solution is to neglect the non-diagonal terms. The controller then consists of four independently working single-channel FXLMS algorithms. This decentralized approach (D-SISO) still allows some vibration reduction, but the performance is decreased compared to the MIMO case (Figure 10). However, in an industrial application, where standard embedded microcontrollers have to be used, a compromise between performance and system complexity will have to be found and deterioration might be tolerated. In the last section of this paper, more approaches to decrease the computational load of adaptive filter systems are discussed.