

Experimental Studies on Different Actuator-Sensor Configurations of Active Control Systems for the Reduction of Noise and Vibration in Vehicles

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ABSTRACT

Current developments in the automotive industry such as downsizing, the use of cylinder deactivation and consistent lightweight construction increasingly enable the application of active control systems for the further reduction of noise and vibration in vehicles. In the past few years, different configurations of actuators and sensors for the realization of an active control system have been investigated and evaluated experimentally. Active engine mounts, inertial mass actuators and structural integrated actuators can be used to reduce either structural vibrations or the interior noise level. As a result, a variety of different topology concepts for the realization of an active control system arises. These can be divided into an active vibration control scenario, the direct influence of the sound field with loudspeakers or the application of structural actuators for the reduction of the interior sound pressure. In the latter case, microphones are used as error sensors for the active control system. With regard to the selection of a suitable actuator-sensor configuration special attention is required as several transfer paths for airborne and structure borne noise are existing. Furthermore, the attainable bandwidth of the applied control systems topology depends on the selection of a suitable actuator-sensor configuration and an adequate number of actuators and sensors in order to enable global control and to avoid local effects. This paper summarizes and compares different actuator-sensor configurations for the control of noise and vibration in vehicles. Beside the control performance, it highlights the electrical power requirements for the control of engine induced interior noise as a function of the selected actuator-sensor configuration.

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INTRODUCTION

Noise, vibration and harshness (NVH) engineering is an essential component of today's vehicle development process. Its goal is to reduce operational vibrations and noise of vehicles and furthermore to design a pleasant vibro-acoustic feedback for the passenger while driving. Beside road and environment excited NVH, engine induced noise and vibrations have a large contribution to the vehicle interior noise [1] and vibrations of comfort points like the passenger seats and the steering wheel [2, 3]. In addition, current developments in the automotive industry such as three-cylinder engines, the use of cylinder deactivation and consistent lightweight construction are leading to an increased vibration level that passive NVH components might no longer compensate for. Therefore, active NVH components are one focal point in current research activities. There may be many reasons for the application of active systems: An increased and/or varying excitation due to downsizing or cylinder deactivation, lightweight construction, the possible neglect of passive tuned mass dampers (linear and rotatory) and balance shafts and an adaptive behavior in the case of a time-variant (e.g. aging, temperature, mechanical preload, etc.) mechanical system behavior, just to mention a few.

Active noise (ANC) and active vibration control (AVC) systems have been realized during the scope of various applications within the last decades. Concerning ANC and AVC in vehicles, there are primarily two different approaches, namely the control of engine induced noise and vibration and the control of broadband stochastic noise and vibration caused for example by rough road excitation.

In the first case, additional active systems can be used to suppress the vibration caused by the engine by the use of active engine mounts. These have been realized with various actuation concepts including electromagnetic [4], electrodynamic [5] and piezoelectric stack or patch actuators [6,7]. Beside the further reduction of vibration, active engine mounts are commonly used in the case of cylinder deactivation and are often combined with an additional ANC system for a further improvement of the sound profile [8] in the lower frequency range from 50 to 150 Hz. Early stage ANC systems included additional loudspeakers whereas present ANC systems utilize the existing sound system [9]. In addition to the reduction of undesired engine noise also active sound design is a common application [10].

Beside loudspeakers also structural integrated actuators or inertial mass actuators attached to vibrational sensitive points of the chassis have been used to enhance either the acoustic [11] or the vibrational behavior [12], in some cases this also applies for multi-variable systems incorporating several actuators and sensors [13]. Concerning the exterior sound quality, loudspeakers in the air intake system [14] and active exhaust systems for the retouching of the sound profile in case of cylinder deactivation or active sound design are taken into consideration or are already in use [10]. Also the reduction of rotational vibrations within the drive train by either active measures like rotational inertial mass actuators [15] or an existing electrical motor [16] in a hybrid drive train is becoming a subject of current research.

In the case of broadband and stochastic noise and vibration, actuators can be integrated at the axles [17], the chassis or the strut bearings [18] which allows for an active compensation of the transmitted dynamic loads. Global vibrational modes (e.g. torsional vibration of convertibles) can be reduced by means of active vibration absorbers [19]. Additionally the loudspeakers can be used to decrease the interior noise caused by stochastic sources [20].

To summarize, a variety of active NVH systems have been developed and demonstrated in laboratory and different applications. However, a systematic comparison of different actuator-sensor configurations for a similar application in the same vehicle is rarely documented. In the scope of the presented work, a common middle-class passenger car of series production is equipped with different active systems for the purpose of engine induced interior noise reduction. Four different actuator-sensor configurations have been tested so far. These include an active vibration control system incorporating a novel active engine mount and three inertial mass actuators, an active noise control system utilizing the existing loudspeakers, a second active noise control system with structural integrated actuators and a promising hybrid active noise control system combining an inertial mass actuator and two loudspeakers.

Beside the analysis and comparison of the just mentioned different actuators-sensor configurations, the paper is organized as follows: In the beginning the main different active control concepts are briefly recapitulated and the considered control algorithm is summarized. Afterwards the test vehicle, the main results of a transfer path analysis, the experimental set-up and the utilized actuators are described. In the final section, the four different actuator-sensor configurations are finally compared and a brief overview of future activities is given.

ACTIVE CONTROL CONCEPTS

The systems for active control of noise and vibration can be distinguished into a few basic system concepts. In most cases, the target of automotive applications is the reduction of noise in the audible frequency range. The most established approach for this is active noise control (ANC). This concept utilizes acoustic sensors (microphones or sound particle velocity), and loudspeakers which directly influence the sound field in the passenger compartment.

However, since noise is correlated to vibration of radiating surfaces, dynamic actuators influencing the vibrations in conjunction with sensors which acquire dynamic displacement, velocity, acceleration, force or strain can be used in an active vibration control (AVC) configuration.

However, it is known, that not every vibration mode of a structure is a good sound radiator. Thus, several system layouts have been developed aiming at influencing of the relevant radiating modes of a structure. This includes either the direct measurement of the sound pressure level [21,22] or the incorporation of a model in the control system which represents the radiation characteristics of the structure, which estimates the sound radiation characteristics from structural measurements (active structural acoustic control, ASAC). This causes either a reduction of the amplitudes of the radiating modes [22], or the structural modes are actively reconfigured in order to minimize the radiated sound [22]. To complement, also slightly varying definitions are used whenever structural actuators and acoustic sensors are applied. This includes active noise-vibration control (ANVC) [22]. For the scope of simplification this paper refers to AVC whenever structural sensors are applied and to ANC whenever acoustic sensors (regardless of the actuator type being used) are applied. Figure 1 illustrates the basic configurations for the active reduction of noise by means of ANC with acoustic actuators (left), AVC (middle) and ANC with structural actuators as well as ASAC (right).

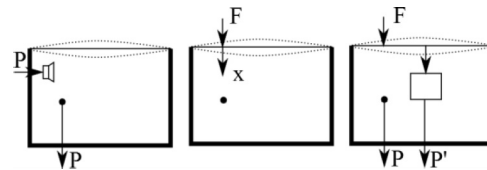


Figure 1. Basic configurations for the active reduction of noise inside a compartment

Regardless of the actual actuator-sensor configuration being used, two main control system layouts have been established for active noise and vibration control: Feedback and feedforward systems whereas the used control algorithm mainly depends on the present application [23]. For the control of harmonic engine induced disturbances adaptive feedforward control systems are commonly used [24]. The feasibility of active systems for the control of narrowband disturbances was successfully demonstrated in different applications [13,25,26]. Due to the feedforward topology, the control system offers a high robustness. Furthermore, a suitable reference signal can be generated synthetically by measuring the current rotational speed of the engine and by making use of a discrete time oscillator. In previous work, the commonly applied single frequency FxLMS algorithm [24] was extended to a multi-channel topology [26]. Here, the control algorithm utilizes a multiple-input multiple-output (MIMO) topology allowing the application of several actuators and sensors. The control algorithm provides the control of several engine orders by means of a parallel controller design for each engine order considered. Figure 2 shows the applied control algorithm in a narrowband feedforward control scenario where K actuators, M error sensors and N engine orders are considered. The primary disturbance $\mathbf{d}(n)$ is a multi-order harmonic signal. An additional secondary source $\mathbf{S}(z)$ serves the cancellation of the primary waveform $\mathbf{d}(n)$ by means of the secondary waveform $\mathbf{y}'(n)$.

The residual waveform $e(n)$ is being used for the adaptation of the adaptive control filter weights $w_0(n)$ and $w_1(n)$. In case of an active noise control system the residual waveform, or the so-called error signal $e(n)$, equals the remaining in-vehicle sound pressure measured by M microphones. For an active vibration control system, the error signal $e(n)$ equals the measured acceleration at some associated measurement points on the mechanical structure.

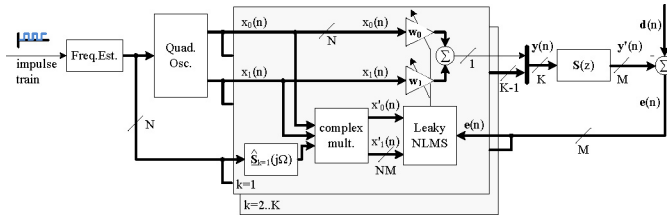


Figure 2. Multi-order multiple-input multiple-output (MIMO) Filtered Reference Least-Mean-Squares algorithm. The control algorithm utilizes a synthetic harmonic reference signal from a rotational speed measurement.

The control algorithm focuses on high computational efficiency. This is mainly achieved by an alternative secondary path modeling technique. Here, the commonly utilized Finite Impulse Response (FIR) secondary path model is replaced by a complex frequency response model $\hat{S}_{km}(j\Omega)$. A detailed introduction to the implemented control algorithm is given in [26]. Within the scope of this work, the control algorithm is extended by a normalized adaptation step size α [24] and an additional leakage factor $0 < v < 1$ [24,27] in order to avoid effects of longtime divergence and to improve the stability of the applied FxLMS algorithm. Equation (1) summarizes the update equation for the adaptive filter weights for a single order MIMO narrowband FxLMS algorithm ($i = 1, 2$). (See also Table 3: Formula symbols.)

$$w_{ik}(n+1) = vw_{ik}(n) + \alpha \cdot q_{km} \sum_{m=1}^M \frac{x'_{ikm}(n) \cdot e(n)}{\max(P_{x'_{ikm}}(n), P_{\min})} \quad (1)$$

Here, k denotes the actuator index and m stands for the sensor index. Beside the extension to a normalized adaptation step size for a MIMO control system, the update equation is also extended by an additional weighting factor q_{km} [24]. By means of this additional factor, some secondary path transfers functions $\hat{S}_{km}(j\Omega)$ with small actuator-sensor coupling can be neglected by setting q_{km} to zero [24,26]. For the subsequent AVC scenario, only collocated actuator sensor positions are taken into account by the control algorithm ($q_{km} = 0$ if $k \neq m$) which leads to a multi-SISO control topology for the AVC scenario. In contrast to this, the subsequently presented ANC scenarios require the consideration of all secondary path transfers functions ($q_{km} = 1$ for all k, m) due to the relatively large cross-coupling of the secondary path transfers matrix $S(z)$.

TEST VEHICLE OPEL ASTRA

During preceding research activities the high potential of active measures for vibration reduction and noise reduction could be shown. These investigations were taken at a test bench for active mounts and a full vehicle test bench. To provide representative results, the test bench was equipped with an Opel Astra Sports Tourer representing a middle-class passenger car of series production [6]. Consistently for

the current investigations in driving tests the same vehicle model is used. The vehicle is road-approved and allows for a close examination of different active NVH measures during real driving scenarios. It incorporates a four cylinder in line Otto engine installed in transvers direction. The 1.4 l Otto engine with a power of 103 kW is supported at four positions. The engine and the gear are attached by hydraulic mounts, located on the sides of the engine. The mounts of the front and rear torque support are rubber bearings. All passive NVH components of the car are retained at the car during test runs. However, to analyze the engine dominated acoustic characteristics of the test vehicle, engine run-ups with different gear stages and different driving conditions were performed in a previous step. Some of the tests were carried out on an airport runway and some tests were done on public roads. The measurements of the interior sound pressure level show a dominance of the second engine order. The considered 4-cylinder in-line gasoline engine shows two combustion cycles per revolution, thus the second engine order is the mainly exciting engine order. Assuming an idle speed of 600 rpm and a maximum speed of 6000 rpm, the relevant frequency range for the second engine order ranges from 20 to 200 Hz. The interior sound pressure level averaged over all four microphones is depicted in Figure 3. The dominance of the second order is evident and the results show that the second order is striking, especially at engine speeds between 1000 and 2000 rpm. Within the scope of this paper, different actuator-sensor configurations for the reduction of this low-frequency engine induced interior sound pressure are investigated. In this connection, a transfer path analysis is carried out in order to detect relevant transfer paths at the engine's mounting positions. Different structural actuators are being used for both, the active control of vibration as well as the active control of noise. Figure 4 shows the test vehicle and the associated coordinate system.

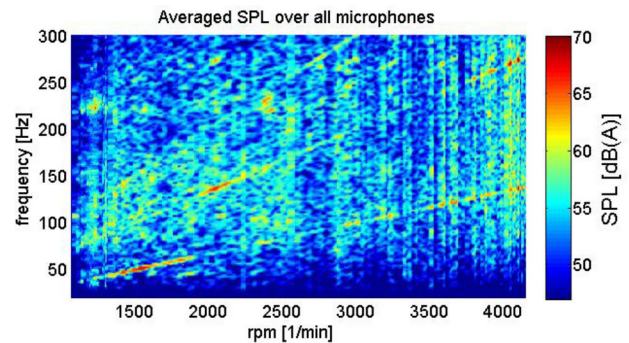


Figure 3. Averaged sound pressure level during a 4th gear run-up.



Figure 4. Test vehicle Opel Astra Sports Tourer and associated coordinate system.

Transfer Path Analysis

In order to identify relevant transfer paths from the source (engine) to the target (vehicle interior) and in addition to the preliminary analysis of the test vehicle, an experimental-based transfer path analysis (TPA) based on the second engine order was realized. The noise in the entire vehicle structure is transmitted through structure-borne and airborne noise paths. The result of a TPA enables a qualitative and quantitative assessment of the contribution of each noise transfer path to the overall noise characteristic and therefore allows NVH refinement. Beside the noise transfer paths, also the acting external loads at the mounting points of the engine are determined. The knowledge of both, the transfer paths and loads, are a requirement for the dimensioning and effective integration of active isolation components into the vehicle.

The TPA-methods differ in the “classical” and “operational” approach with distinctions concerning implementation, complexity and computation. Depending on the application case each method has advantages and disadvantages. Within the classical TPA, the load estimation can be achieved among other by the mount stiffness and/or matrix inversion procedure. The mount stiffness procedure requires the dynamic stiffness of each mount in X, Y and Z direction and can be used when the transfer paths are decoupled from each other by almost adequate elasticity. The matrix inversion procedure is used in cases with rigid connections and high mount stiffness values. For further information see [17].

Within the characterization of the test vehicle, the classical TPA with matrix inversion method was used. Therefore, the measurement of the vibro-acoustical transfer functions between the force input points and the comfort points in the vehicle interior, structural transfer functions between all points of force transmission as well as operational measurements at each engine connection point and in the vehicle interior is required for the analysis of the transfer paths. For this purpose, the accelerometers were placed at the two engine mounts (EM) and the front and rear torque support (TS) as well as in the vehicle interior at the vibrational comfort points.

The main results of the TPA are shown in Table 1 and Figure 5. In Figure 5 a detailed overview of all partial noise transfer paths results is given for all four connection points of the engine to the car-body in all three transversal directions. The comparison between the measured interior noise characteristic and the complex sum of the synthesized transfer path contributions shows especially around 1500 rpm a good accordance. The accordance between the measured and the synthesized characteristic represents conformity regarding all relevant transfer paths. However, a variance between the measured and synthesized characteristic is caused by missing transfer paths. As shown in Figure 5, the vertical (Z) noise transfer contribution is dominant compared to the longitudinal (X) and the lateral (Y) direction. Table 1 shows the partial contribution for each transfer path in percentage of total sum in the range of 1500 up to 1650 rpm. This rpm range includes the maximum of the second engine order sound pressure level observed. Especially the rear torque support in Z-direction has a high contribution (74.08 %) in this rpm range. Except for the engine mount on the left engine side, all other mounting points show a considerable contribution to the interior

sound pressure especially in the vertical Z-direction. Hence, the actuators for the subsequently introduced active vibration control set-up are chosen to work along the vertical axis. The contribution of the vertical engine’s mounting positions to the interior sound pressure is highlighted in Table 1.

Table 1. Partial contribution for each transfer path in percentage of the total sum in the range of 1500 to 1650 rpm.

Transfer path	Contribution to the overall sound pressure level in [%]		
	X	Y	Z
Engine mount left (EM left)	5.99	0.21	3.60
Engine mount right (EM right)	2.69	0.05	6.33
Torque support front (TS front)	0.36	0.21	3.23
Torque support rear (TS rear)	1.83	1.45	74.08

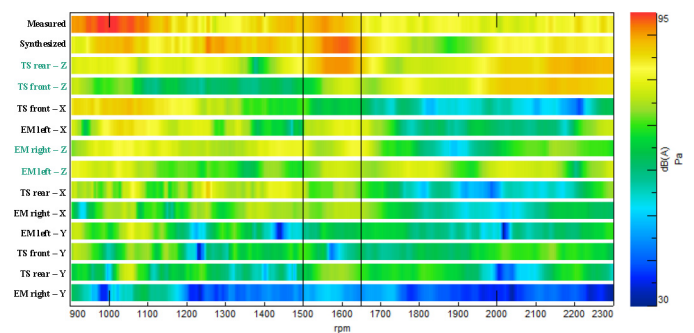


Figure 5. Partial contribution characteristic of structure-borne noise transfer paths in the test vehicle for the second engine order. The engine mounts (EM) at the left and right engine side as well as the torque supports (TS) have been considered within the transfer path analysis.

Experimental Setup

Applied Structural Actuators

As aforementioned, the vehicle is equipped with different actuators at several positions of the car body namely loudspeakers, inertial mass actuators and an active engine mount. These actuators use different physical principles to produce the necessary force for counteracting the disturbing vibrations or sound waves, respectively. The loudspeakers and the inertial mass actuators are based on the electrodynamic principle while the active engine mount uses a piezoelectric stack actuator to generate a mechanical force. The utilized actuators will be described in the following. The focus will be on the self-developed electrodynamic inertial mass actuator and the piezoelectric active engine mount.

Electrodynamic Inertial Mass Actuator

The inertial mass actuators (sometimes also referred to as proof-mass actuators) were developed at Fraunhofer LBF for the scope of an all-purpose actuator. Figure 6 shows the inertial mass actuator and a schematic representation of its structure and the attachment to a mechanical structure. As one can see, the actuator can be placed anywhere on a mechanical structure since the actuator itself contains an inertial mass. Making use of electrodynamic forces and the mechanical realization, this concept is fairly simple and based on the known principle of voice coil actuators. In this particular case an