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Experimental study on active noise and active vibration control for a passenger car using novel piezoelectric engine mounts and electrodynamic inertial mass actuators

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Abstract

Active noise control (ANC) is an efficient technique to reduce noise of passenger cars using a cancelling anti-noise waveform, which is introduced by additional secondary sources. In the same way, engine caused vibrations which are transferred from the mounts and the adjacent structures. This results in reduced driving comfort and can be enhanced either by active vibration control (AVC) or active noise control (ANC) strategies. To show the potential of AVC, a medium-class passenger car is equipped with an active engine mount featuring a piezoelectric actuator and three additional inertial mass actuators (IMAs) located at the mounting positions of the combustion engine. The active engine mount as well as the inertial mass actuators are used to compensate engine induced vibrations close to the source. Besides the AVC system, a loudspeaker based ANC concept for reduction of interior vehicle noise is investigated. Here, the conventional loudspeaker system is extended by a subwoofer mounted in the trunk.

Due to the fact that the distortion is mainly correlated to the rotational speed of the engine, both concepts use a feedforward control algorithm in an adapted topology for the compensation of harmonic disturbances.

The integration of both measures in a single test vehicle enables a comparison of different active concepts for the improvement of noise, vibration and harshness (NVH) under real driving conditions and an investigation of different strategies for the development and system integration of active NVH components. Experimental results for vehicle road tests of the AVC concept and the ANC concept are discussed in a comparative analysis. Key aspects of this experimental study are vibration reduction at the engines' mounting positions and vehicle interior noise reduction. Based on these results an evaluation of the achievable benefit is carried out for both concepts.

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1. 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 vibroacoustic 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]. Therefore current research activities are focusing on engine noise and vibrations. In the last few years different actuator concepts as well as control algorithms were developed and demonstrated in laboratory or different applications [4, 5]. However, a comparison of different concepts for a similar application in the same vehicle is rarely documented.

Therefore scientists at Fraunhofer LBF equipped a common middle-class passenger car of series production with different active concepts for NVH reduction, analyzed each of it separately and benchmarked them. In this paper first results of this benchmark study are presented dealing with the question of advantages and disadvantages for active noise control (ANC) and active vibration control (AVC) in the field of vehicle NVH.

Thus the test vehicle is equipped with an AVC system as well as an ANC system for compensation of engine induced disturbances. For the AVC system the passenger side hydro engine mount of the car is substituted by a novel active piezoelectric engine mount [6]. Furthermore, three inertial mass actuators (IMAs) are implemented at the mounting points of the engine. The topology of the active engine mount allows to separate the piezoelectric actuator from static loads and to induce cancelling forces from only a few Hertz up to 200 Hz. A more detailed description of the actuators used for AVC is given in Chapter 2. The ANC which uses two car loudspeakers integrated at the front doors and an additional subwoofer mounted in the trunk concept is presented in Chapter 3 and. Both active concepts use a Multi-Order Multi-Input-Multi-Output (MIMO) Narrowband Filtered-Reference Least Mean Squares (FxLMS) algorithm for actuator control. Using adaptive feedforward control is advantageous because these algorithms are easy to adapt to different hardware configurations, robust and enable the actuators to compensate several harmonic signals such as different engine orders simultaneously. In Chapter 4 the control algorithm itself is presented whereas in Chapter 5 the test setup, the signal conditioning hardware and the test vehicle are introduced. Experimental results for vehicle road tests of the AVC concept and the ANC concept are comparatively presented in Chapter 6. Key aspects of this experimental study are vibration reduction at the engines' mounting positions and engine induced vehicle interior noise reduction.

2. ACTIVE VIBRATION CONTROL – SYSTEM COMPONENTS

Active vibration control is a concept which is used to actively reduce vibrations induced by the combustion engine. Therefore one or more additional actuators are placed at vibration sensitive positions or are integrated directly into the engine mounts. These actuators generate a force output which counters vibrations or vary the dynamic characteristics of the engine mount (e.g. damping). Several actuator concepts are published in the last years, as for example electrodynamic [7] and piezoelectric engine mounts [8], magneto rheological engine mounts [9] or additional electromechanical shakers [2, 3, 10]. However, force generating systems should act at the car body rather than on the engine side in order to make use of the passive isolation of the engine mounts and thus requiring less force and energy. Beside actuators, sensors for measuring input signals, amplifiers and an electronic control unit including the control software are components of an AVC system. In this paper an AVC system combining a piezoelectric engine mount and three inertial mass actuators based on electrodynamics principle is presented. These actuators are used to reduce the remaining engine induced vibrations at the mounting positions in vertical vehicle direction of a four cylinder inline Otto engine. The actuator and sensor placement is shown in Figure 1. The active engine mount is installed at the passenger engine side and replaces the conventional hydro mount. The IMAs at located at the left engine mount and the two swivel supports. Four accelerometers measure the vibration of the mounting positions at the car body in vertical direction. The control algorithm features a multi-SISO topology whereas each actuator pursues the aim to reduce the engine induced vibrations at its related mounting position (collocated positioning).

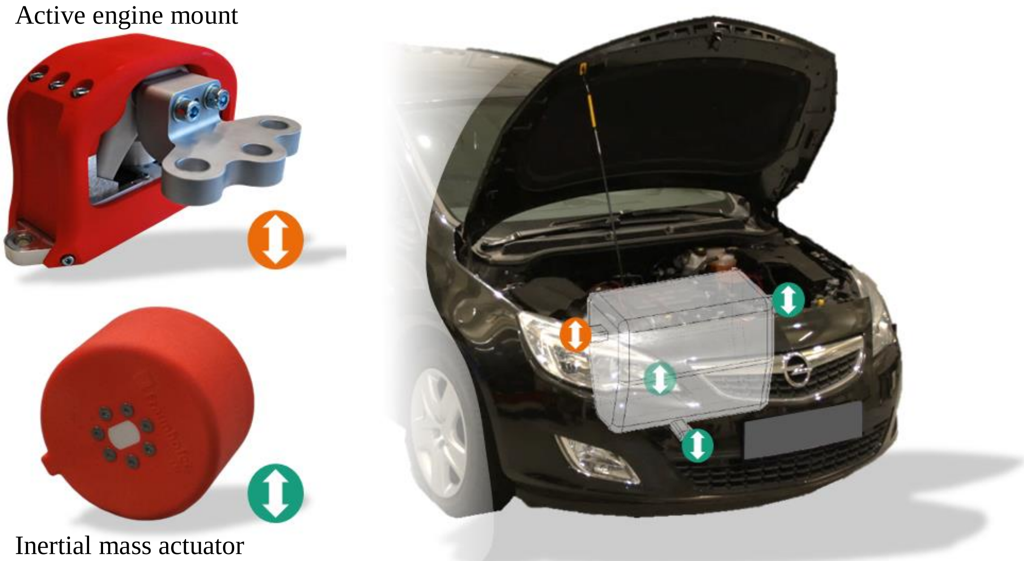


Figure 1. Actuator and sensor placement for active vibration control.

2.1 ACTIVE PIEZO-ELECTRIC ENGINE MOUNT

Mechanical loads acting on the engine mounts can be divided into dynamic and static or quasi-static components. Quasi-static load components result from the engine's mass as well as the driving torque. These loads may exceed the dynamic load components, which primarily result from the combustion process and inertia forces of accelerated engine components. These dynamic load components effect the vehicle's NVH and therefore have to be kept small and/or further reduced by means of an active measure. For the presented piezoelectric engine mount [11] a system topology with two separate, in parallel arranged load paths was chosen. This enables the decoupling of the actuator from static loads and therefor allows both: minimizing the actuator loads and transferring the dynamic counterforces to the car body efficaciously. The decoupling is realized by means of a serial arrangement of the actuator and a viscous damper $\mathbf{b}$ at the active path and a parallel suspension spring $\mathbf{k_l}$, which bears the majority of the static and quasi-static loads at the passive path. The counteracting actuator force is introduced to the structure through the viscous damper whose dynamic stiffness increases as the frequency rises. Figure 2 a) shows the topology of the active engine mount whereas Figure 2 b) shows a laboratory setup of the active load path. In this previous laboratory setup, the active path includes an additional coupling element $\mathbf{k_c}$ to adjust the resulting stiffness of the active path as the actuator's stiffness $\mathbf{k_a}$ was relatively high and was not supposed to be varied.

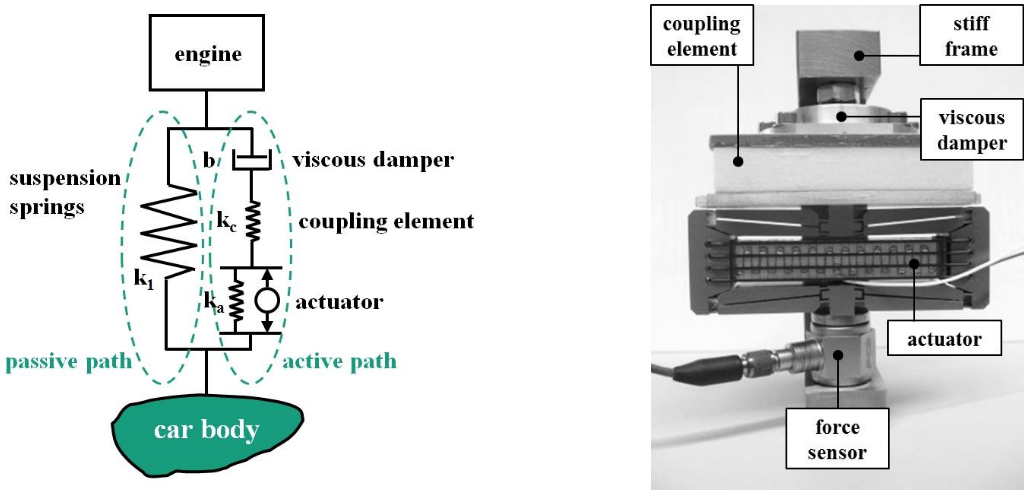


Figure 2. a) topology of the active engine mount and b) laboratory setup of the active load path.

Since piezoelectric actuators feature the capability to introduce high forces but only little strokes, a stroke amplification mechanism is used. Another advantage of this topology is that in case of actuator failure the passive isolation effect remains. After laboratory tests, the proposed system topology is converted to a near series design which enables better vehicle integration (see Figure 3). Here, the actuator's stiffness $\mathbf{k_a}$ is adapted to the application and the additional coupling element $\mathbf{k_c}$ can be omitted.

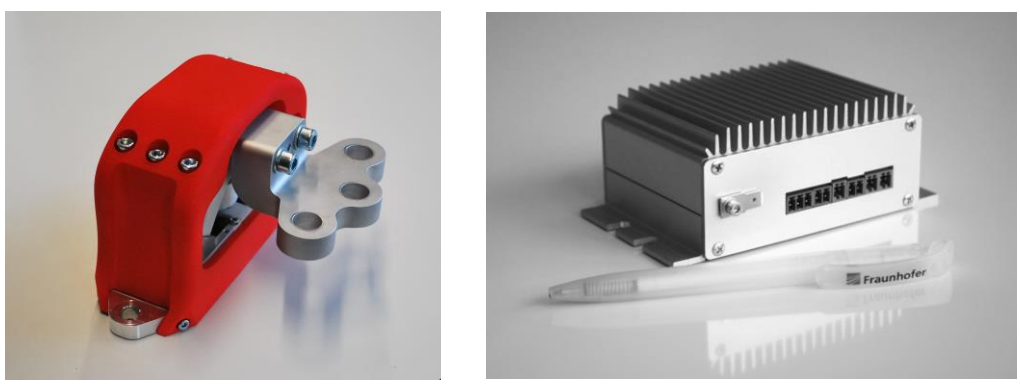


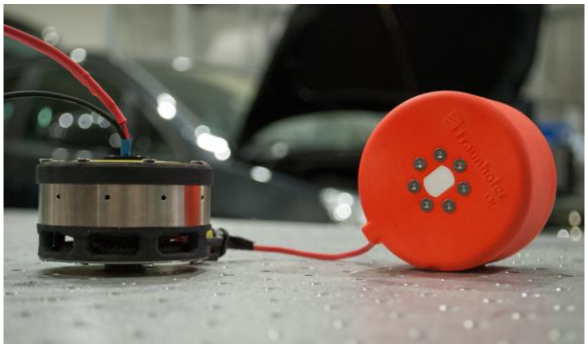
Figure 3. Final design of the piezoelectric engine mount (left) and piezo amplifier (right).

There are important requirements concerning power electronics and power amplifiers when utilizing piezoelectric actuators. For example a high linearity over a wide frequency range and a low signal-to-noise ratio are desirable. Furthermore, driving high capacitive loads especially at high frequencies requires large dynamic currents. For the driving tests a self-developed power amplifier is used to drive the piezoelectric engine mount. The amplifier is supplied via the 12 V on-board power supply. The maximum driving voltage of 150 V is provided by means of an internal DCDC-converter. The piezoelectric engine mount is then controlled by means of an integrated class-AB amplifier. The amplifier drives the piezoelectric actuator with a maximum peak-to peak voltage of about 150 V up to a frequency of 200 Hz and a total harmonic distortion of less than 0.25 % for the driving voltage and 0.5 % for the driving current.

2.2 ELECTRODYNAMIC INERTIAL MASS ACTUATOR

The three inertial mass actuators (IMAs) used for AVC are based on the electrodynamic principle and were developed at Fraunhofer LBF as multi-purpose actuators for vibration reduction. Therefor the actuators are not tailored for the presented application. Tailoring the IMA to the presented application could further reduce their size and weight. Main goals of the development were high linearity, wide operation range, good force to mass ratio and compact and robust actuator design for easy system integration. Each actuator (Figure 4 a)) has a total mass of 560 g with an inertial mass of about 500 g, a diameter of about 79 mm and a height of 53 mm. The electrodynamic coupling coefficient is shown in Figure 4 b). Each actuator can produce a continuous force of about 50 N and a peak force of up to 120 N. The nominal electrical resistance of the voice coil actuator is about 4 Ohm so that the actuator can be driven by common HIFI amplifiers. Figure 4 b) shows the dynamic characteristics of an IMA. The resonance frequency of the inertial mass is about 20 Hz. Higher resonance frequencies in force direction are above 1 kHz. The total operating range is electrical restricted by the actuators inductivity. The 3 dB cutoff frequency is about 250 Hz.

a)



b)

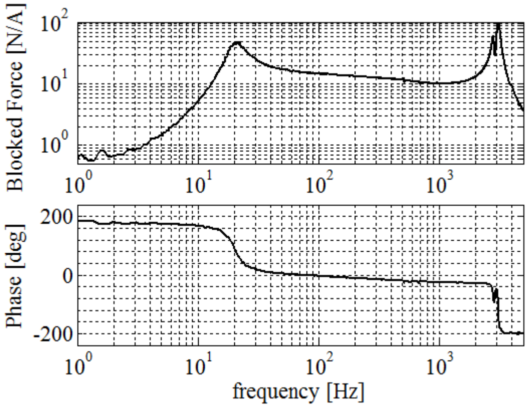


Figure 4. a) Design of the Inertial Mass Actuators and b) dynamic characteristics of an IMA.

3. ACTIVE NOISE CONTROL - SYSTEM COMPONENTS

Several active noise control (ANC) concepts for vehicle and other technical applications have been published in the last years. Different actuator concepts as for example loudspeakers or electromechanical shakers are discussed and various control approaches are presented. In contrast to the AVC concept the ANC approach can be understood as a secondary measure for the reduction of engine induced vehicle interior noise. ANC is often utilized for active receptor isolation. Here, the sound pressure is reduced by means of a cancelling secondary waveform introduced by loudspeakers. In this study an ANC approach based on two common car HIFI speakers and a subwoofer is presented to actively tackle harmonic engine induced cabin noise. The speakers are integrated in the front doors of the car whereas the subwoofer is mounted in the trunk. Four microphones are mounted at the front and rear headrests and serve the measurement of the disturbing interior noise. According to the AVC approach the ANC system uses the same dSPACE MicroAutoBox as control unit. Due to the relatively lager cross-coupling between each loudspeaker and the microphones, a Multiple-Input-Multiple-Output (MIMO) controller topology is chosen.

4. ADAPTIVE NARROWBAND FEEDFORWARD CONTROL

The feasibility of active systems for the control of narrowband disturbances was successfully demonstrated in different applications [11, 12, 13, 14, 15]. For the control of harmonic engine induced disturbances adaptive feedforward control systems are commonly used [16]. 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 [16] was extended to a multi-channel topology [13]. Here, the control algorithm utilizes a 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.

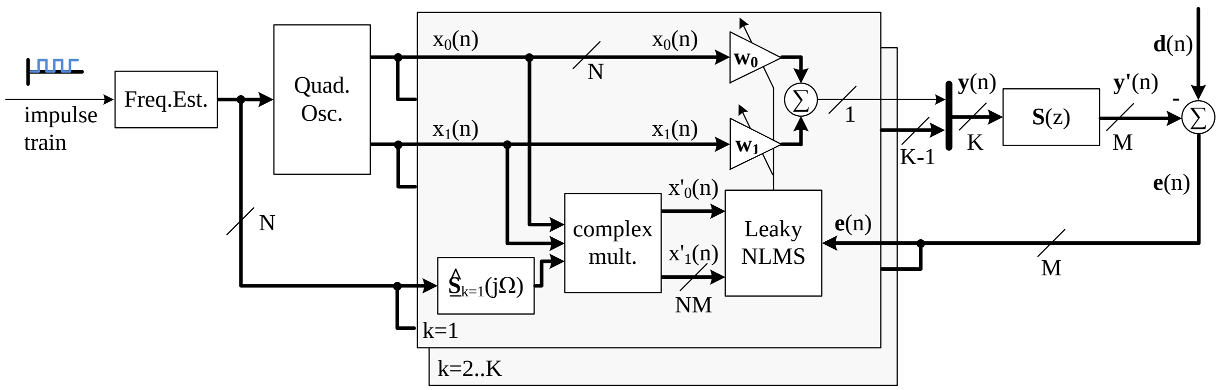


Figure 5: Multi-Order Multiple-Input-Multiple-Output Narrowband FxLMS control algorithm.

Figure 5 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 **d**(n) is a multi-order harmonic signal. An additional secondary path **S**(z) serves the cancellation of the primary waveform **d**(n) by means of the secondary waveform **y**'(n). The residual waveform **e**(n) serves the adaptation of the adaptive control filter weights $\mathbf{w}_0(n)$ and $\mathbf{w}_1(n)$. 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.

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 $\hat{\mathbf{S}}(j\Omega)$. A detailed introduction to the implemented control algorithm is given in [13].

Within the scope of this work, the control algorithm is extended by a normalized adaptation step size α [16] and an additional leakage factor $0 < \nu < 1$ [16, 17] 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.

$$\mathrm{w_{ikn}}(\mathrm{n}+1) = \nu \mathrm{w_{ikn}}(\mathrm{n}) + \alpha \cdot q_{km} \sum_{\mathrm{m}=1}^{\mathrm{M}} \frac{\mathrm{x'_{ikmn}}(\mathrm{n}) \cdot \mathrm{e_m}(\mathrm{n})}{\max(\mathrm{P_{x'_{kmn}}}, \mathrm{P_{min}})} \qquad i = 1,2 \tag{1}$$

Here, k denotes the actuator index, m denotes the sensor index and n stands for the index of the considered engine order. 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} [16]. 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 [13, 16]. For the active vibration control (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 ANC scenario requires the consideration of all secondary path transfers functions ($q_{km} = 1$) due to the relatively large cross-coupling of the secondary path transfers matrix $\mathbf{S}(z)$.

5. TEST SETUP

The test vehicle (Opel Astra Sports Tourer) is a middle class series production passenger car with 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 mount and the gear mount, located on the sides of the engine, are hydro mounts. 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 the test runs.

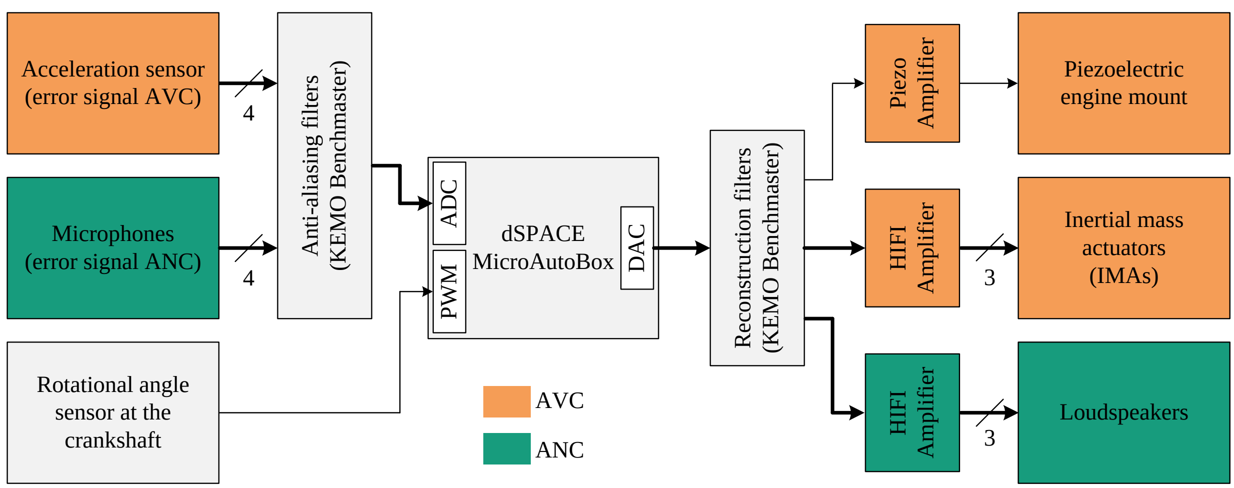


Figure 6: Structure of the ANC and AVC system.

The control algorithm is implemented on a dSPACE MicroAutoBox which allows real time signal processing. The control algorithm is fed with error and reference signals. For the AVC scenario the error signal is given by acceleration sensors at the mounting points of the inertial mass actuators. In the ANC scenario the error signal is given by the four microphones near the headrests. In both scenarios the engine speed is used as a reference signal. Kemo benchmaster lowpass filters are used to avoid aliasing and to amplify the measured error signals.

The control signals are fed into a common car HIFI amplifier which drives either the actuators mounted at the car body or the loudspeakers. The piezoelectric engine mount is driven by a piezo amplifier which has been developed at Fraunhofer LBF.

6. EXPERIMENTAL RESULTS

As mentioned in the beginning, two different investigations are carried out. The first setup focuses on the reduction of the second engine order vehicle interior noise caused by the engine. In a second investigation the second engine order acceleration at the mounting points, briefly mentioned in Section 2, should be reduced by means of an active engine mount and some additional inertial mass actuators.

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 parts of the tests were carried out on an airport runway and some tests were done on public roads. Figure 7 shows the interior sound pressure level averaged over all four error microphone positions during an exemplary engine run-up in the fourth gear.

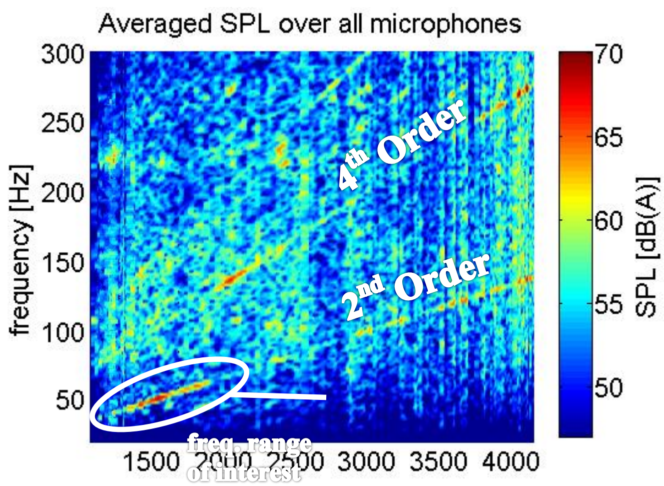


Figure 7: Average sound pressure level for an exemplary run-up during a driving test.

The results show that the second engine order is dominant particularly at engine speeds between 1000 rpm and 2000 rpm. To ensure comparability between all measurements, the ANC systems as well as the AVC system are tested for an engine run-up between 1000 rpm and 2000 rpm at the sixth gear stage under full engine load. The control algorithm pursues the aim to minimize the second order of the respective error signals. These are the sound pressure close to head microphones for the ANC scenario and the acceleration in vertical direction at the four mounting points of the engine. The following sections show the performance of the ANC and the AVC system by means of the second engine order cut of the associated error sensors.

6.1 RESULTS OF THE ACTIVE NOISE CONTROL SCENARIO

Classic ANC systems for the reduction of vehicle interior noise are somehow limited to a lower frequency range of approx. 100 Hz, depending on the positioning of the error microphones [14]. At higher frequencies active noise reduction can only be achieved close to the error microphone positions with the risk of increasing the sound pressure level at other locations in the sound field. The limited frequency range does not only depend on the complex sound field, it is also influenced by the different phase delays of the transfer paths from the engine to the passenger compartment. For this reason the investigations in this ANC scenario are focusing on the engine speed range from 1000 rpm to 2000 rpm. That implies disturbance frequencies from 30 Hz to 70 Hz.

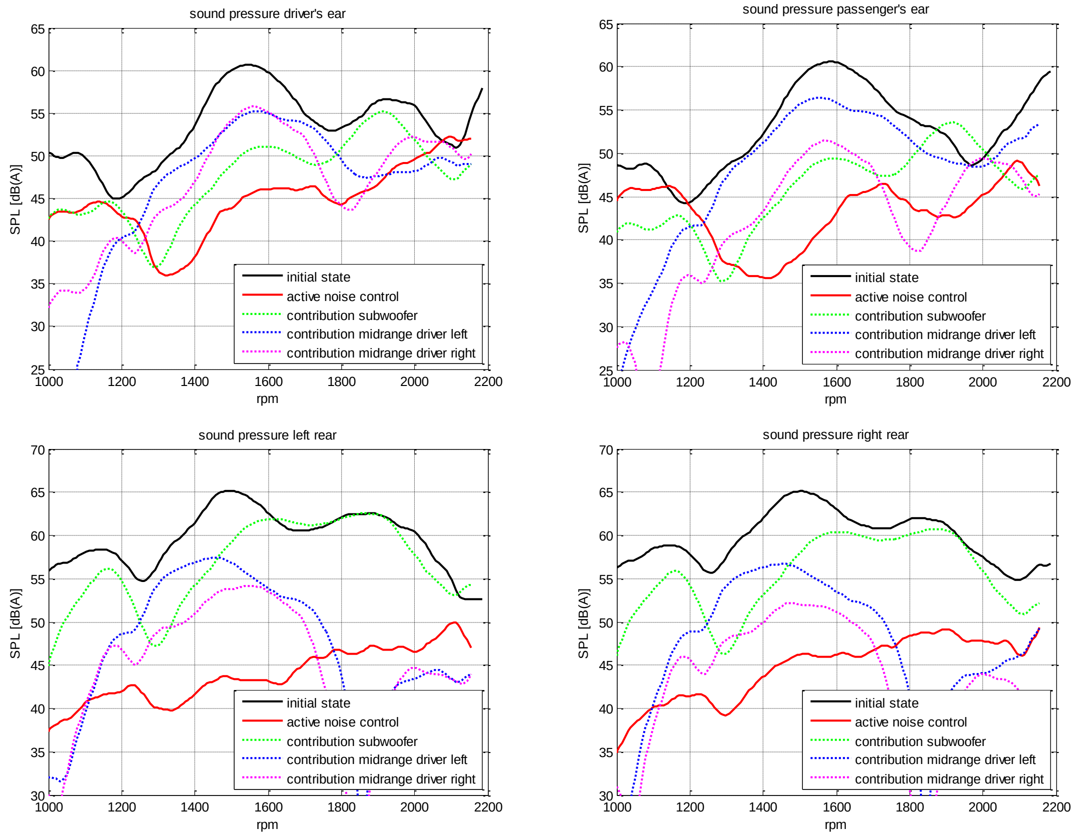


Figure 8: Reduction of the second engine order sound pressure at the four microphone positions caused by the ANC system. In addition, the contribution of each actuator is also illustrated.

Figure 8 illustrates the performance of the ANC system whereas two midrange drivers located in the front doors of the car and a subwoofer located in the trunk serve as additional secondary sources for the ANC system. A significant reduction of the second engine order can be observed for all microphone positions. For a close investigation, Figure 8 also illustrates the contribution for each loudspeaker position which is calculated by means of the second engine order order-cut of the actuator control signal $\mathbf{y}(n)$ and the secondary path estimate $\hat{\mathbf{S}}(j\Omega)$. The results show, that the subwoofer shows the largest contribution for the reduction of the second engine order in the rear microphone positions. The reduction of the sound pressure at the driver's and passenger's ear results from a collaboration of all loudspeakers. A worsen control performance in the low frequency range up to approx. 45 Hz could be observed for the microphones located in the front. This might be due to the first Eigen frequency of the midrange drivers which is close to 45 Hz. Below its first Eigen frequency, the loudspeakers can hardly generate a proper sound pressure level. In order to increase the performance in a consequent next step, the midrange drivers should be substituted by some subwoofers to be located in the front of the car.

To summarize, the proposed ANC systems can achieve a global reduction on the interior sound pressure in the considered frequency range. A similar behavior of the vehicle was observed for different driving conditions and gears.

6.2 RESULTS OF THE ACTIVE VIBRATION CONTROL SCENARIO

As stated before, the second setup makes use of an active engine mount and three additional inertial mass actuators in an AVC scenario. The same control algorithm is applied for both, the ANC as well as the AVC scenario. However, due to the relatively small cross coupling of the secondary path transfer matrix only collocated secondary path transfer functions are regarded in the update equation for the adaptive filter weights which leads to a multi-SISO controller topology (see Section 4).

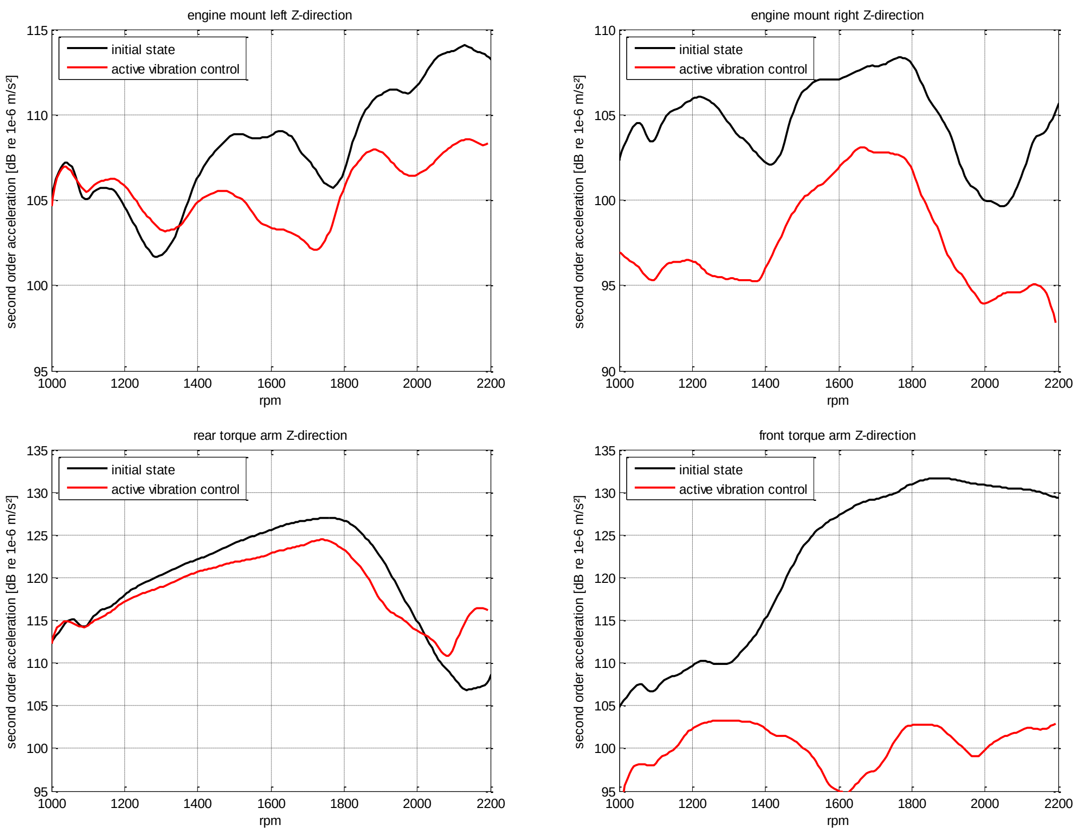


Figure 9: Second engine order acceleration at the mounting positons of the actuators for the AVC scenario

Figure 9 illustrates the control performance by means of the second engine order order-cut of the acceleration at the associated actuator positions. A reduction of the acceleration can be observed for all mounting position whereas the acceleration at the mounting point of the active engine mount and the acceleration at the front torque support show the largest reduction. In order to compare the performance of the AVC system, Figure 10 shows the resulting second engine order sound pressure at the four microphone positions mentioned in the previous section. A reduction of the sound pressure is mainly achieved for the rear microphone positions. Obviously, the ANC system achieves better results compared to the AVC system which is mainly due to two reasons: On the one hand, the ANC system copes with all sound sources that are related to the second engine order (e.g. the exhaust system). On the other hand, the AVC system tackles only four transfer paths out of a multitude of potential transfer paths regarding the various mounting positions and the degrees of freedom of the four engine mounts. In the latter case, the position of the actuators was selected considering an easy implementation of the actuators and an initial test of the control algorithm.

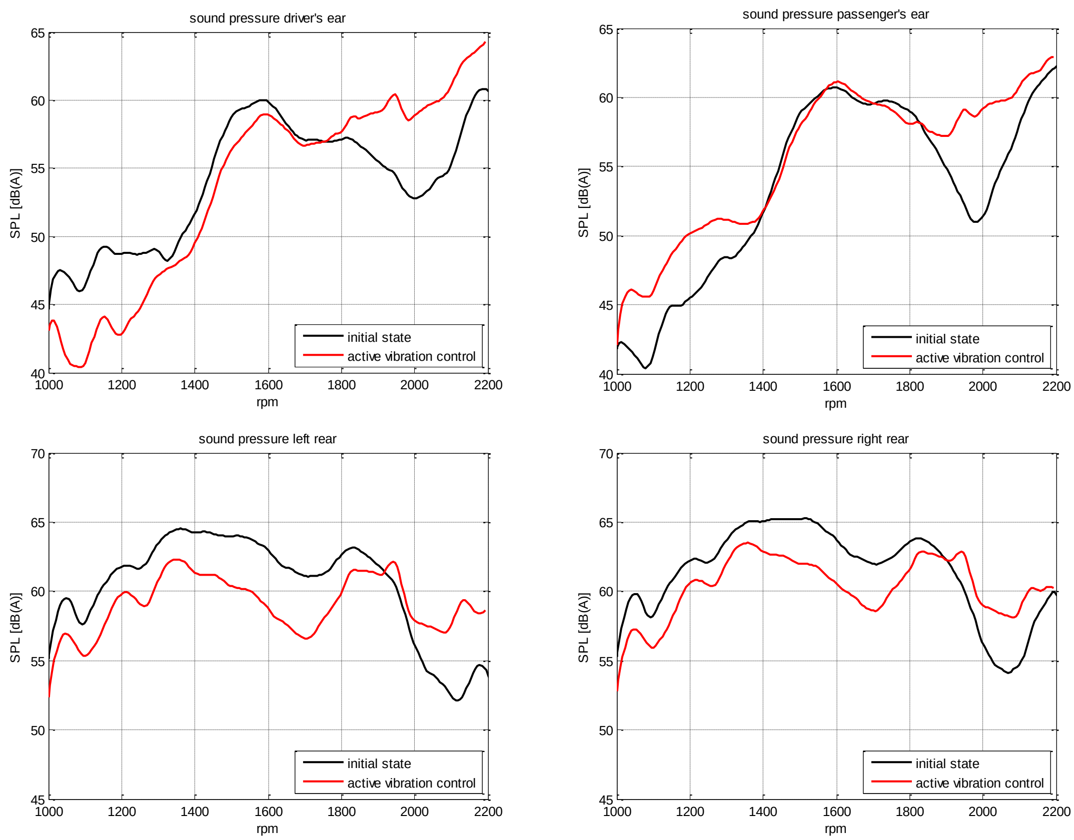


Figure 10: Resulting second engine order interior sound pressure for the AVC scenario.

7. CONCLUSION AND FUTURE TOPICS

This paper summarizes the first results of a comparative study on active systems which are applied in an NVH scenario. Both, the active control of engine induced interior vehicle noise as well as the control of engine induced vibrations was successfully tested in a single test vehicle in order to compare the performance and effects of ANC and AVC systems. The AVC setup makes use of a novel active engine mount based on a stroke amplified piezo actuator and a self-developed power amplifier. Both systems were successfully implemented and tested during driving tests on public roads. However, these studies imply a first step in a holistic comparison of active measures for the control of sound and vibration for the application in vehicles.

Current studies are focusing on an extensive transfer path analysis for the evaluation of major acoustical and vibrational transfer paths of engine induced disturbances. In a logical next step, the placement of the actuators and their direction of working will be adapted with respect to the results of the transfer path analysis. The evaluation of the performance of the AVC system will be expended with regard to common comfort points like the driver's seat and the steering wheel in this context. Also different actuator concepts and their power requirements might be evaluated in an AVC application.

Concerning the control algorithm, the current research focuses on the reduction of the computational burden [13], the improvement of the convergence behavior [15], online secondary path estimation [16, 18] for a time variant system behavior and extensions of the control algorithm by self-tuning adaptation step sizes and self-tuning leakage factors [19]. The current work also deals with the adapted implementation of the control algorithm on embedded systems and the determination of requirements for low-cost sensors, amplifiers and digital signal processors.

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