Automotive Applications of Active Vibration Control[1]

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1. Introduction

In recent years, commercial demand for comfortable and quiet vehicles has encouraged the industrial development of methods to accommodate a balance of performance, efficiency, and comfort levels in new automotive year models. Particularly, the noise, vibration and harshness characteristics of cars and trucks are becoming increasingly important (see, e.g., (Buchholz, 2000), (Capitani et al., 2000), (Debeaux et al., 2000), (Haverkamp, 2000), (Käsler, 2000), (Wolf & Portal, 2000), (Sano et al., 2002), (Mackay & Kenchington, 2004), (Elliott, 2008)).

Research and development activities at ContiTech and the UniBwM have focused on the transmission of engine-induced vibrations through engine and powertrain mounts into the chassis (Shoureshi et al., 1997), (Karkosch et al., 1999), (Bohn et al., 2000), (Svaricek et al., 2001), (Bohn et al., 2003), (Kowalczyk et al., 2004), (Bohn et al., 2004), (Kowalczyk & Svaricek, 2005) (Kowalczyk et al., 2006), (Karkosch & Marienfeld, 2010). Engine and powertrain mounts are usually designed according to criteria that incorporate trade-offs between vibration isolation and engine movement since the mounting system in an automotive vehicle has to fulfil the following demands:

- holding the static engine load,
- limiting engine movement due to powertrain forces and road excitations, and
- isolating the engine/transmission unit from the chassis.

Rubber and hydro mounts are the standard tool to isolate the engine and the transmission from the chassis. Rubber isolators work well (in terms of isolation) when the rubber exhibits low stiffness and little internal damping. Little damping, however, leads to a large resonance peak which can manifest itself in excessive engine movements when this resonance is excited (front end shake). These movements must be avoided in the tight engine compartments of today's cars. A low stiffness, while also giving good isolation, leads

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to a large static engine displacement and to a low resonance frequency (which would adversely affect the vehicle comfort and might coincide with resonance frequencies of the suspension system).

Classical mount (or suspension) design therefore tries to achieve a compromise between the conflicting requirements of acceptable damping and good isolation. It is clear that this, as well as other passive vibration control measures, are trade-off design methods in which the properties of the structure must be weighted between performance and comfort.

An attractive alternative that overcomes the limitations of the purely passive approach is the use of active noise and vibration control techniques (ANC/AVC). The basic idea of ANC and AVC is to superimpose the unwanted noise or vibration signals with a cancelling signal of exactly the same magnitude and a phase difference of 180° (i.e. the ''anti-noise" principle of Lueg (Lueg, 1933)). In the case of ANC, this cancelling signal is generated through loudspeakers, whereas for AVC, force actuators such as inertia-mass shakers are used. Various authors have addressed the application of ANC and AVC systems to reduce noise and vibrations in automotive applications (Adachi & Sano, 1996), (Adachi & Sano, 1998), (Ahmadian & Jeric, 1999), (Bao et al., 1991), (Doppenberg et al., 2000), (Dehandschutter & Sas, 1998), (Fursdon et al., 2000), (Lecce et al., 1995), (Necati et al., 2000), (Pricken, 2000), (Riley & Bodie, 1996), (Sas & Dehandschutter, 1999), (Shoureshi et al., 1995), (Shoureshi et al, 1997), (Shoureshi & Knurek, 1996), (Sano et al., 2002), (Swanson, 1993). ContiTech has implemented prototypes of AVC systems in various test vehicles and demonstrated that significant reductions in noise and vibration levels are achievable (Shoureshi et al., 1997), (Karkosch et al., 1999), (Bohn et al., 2000), (Svaricek et al., 2001), (Bohn et al., 2003), (Kowalczyk et al., 2004), (Bohn et al., 2004), (Kowalczyk & Svaricek, 2005) (Kowalczyk et al., 2006), (Karkosch & Marienfeld, 2010). Honda has developed a series-production ANC/AVC system to reduce noise and vibration due to cylinder cutoff in combination with the engine RPM as reference signal (Inoue et al., 2004), (Matsuoka et al., 2004). A recent overview of such series-production AVC systems can be found in (Marienfeld, 2008).

Most of these approaches rely on feedforward control strategies (either pure feedforward or combined with feedback). The feedforward signal is either taken from an additional sensor (usually an accelerometer in active vibration control) or generated artificially from measurements of the fundamental disturbance frequency (Kuo & Morgan, 1996), (Hansen & Snyder, 1997), (Clark et al., 1998), (Elliot, 2001). Contrary to the major fields of application for active noise and vibration control (military and aircraft), the automotive sector is extremely sensitive to the costs of the overall system. It is therefore desirable to use an approach that requires only one sensor. Also, most approaches rely on adaptive control strategies such as the filtered-x LMS algorithm (Kuo & Morgan, 1996), (Hansen & Snyder, 1997), (Clark et al., 1998), (Elliot, 2001). This seems necessary as the characteristics of the disturbance acting upon the system are time varying. In automotive applications, for example, the fundamental frequency (engine firing frequency, which is half the engine speed in four-stroke engines) varies from 7 Hz at idle to 50 Hz at 6000 rpm. The adaptive approach will adjust the disturbance attenuation of the control system to the frequency content of the disturbance. Whereas this works well in many applications (see the references given above), some critical issues such as convergence speed, tuning of the step size in the adaptive algorithm and stability remain. Discussions between the authors and potential customers (automobile manufactures) have indicated that particularly the issues of convergence speed, tracking performance (this is related to the attenuation capability of the

algorithm during changes in engine speed such as fast acceleration) and stability are crucial. A non-adaptive algorithm might have the benefit of a higher customer acceptance.

Another advantage of a non-adaptive algorithm is that the behavior of the closed-loop system can be analysed independent of the input signals. In an adaptive algorithm, the optimal controller depends on the external signals that act upon the system; thus, it is very difficult to analyse the performance off-line.

Both kind of algorithms have been implemented in an active control system for cancellation of engine-induced vibrations in several test vehicles. The remainder will present an overview of ANC/AVC system components, control algorithms, as well as obtained experimental results.

2. System description

A schematic representation of an AVC system in a vehicle is shown in Figure 1. The disturbance force originating from the engine and transmitted into the chassis through the engine mounts is actively cancelled by an actuator force of the same magnitude but of opposite sign.
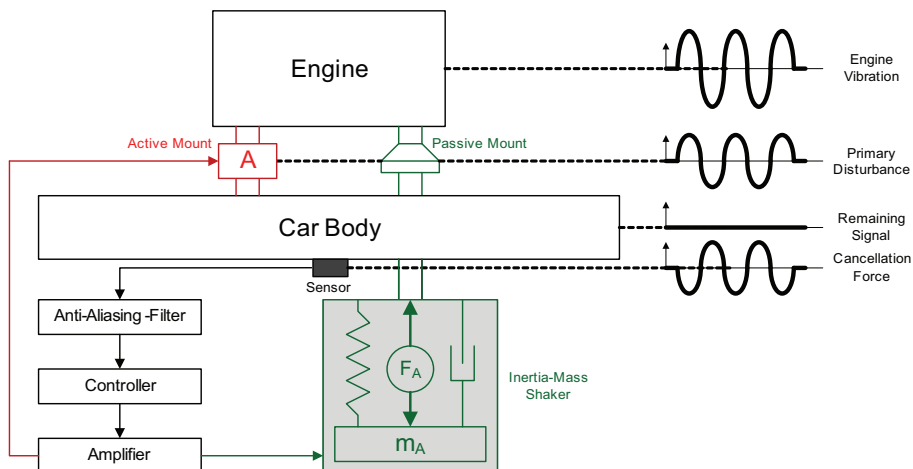

Fig. 1. Schematic representation of an AVC system with an active mount (red) or an inertia-mass shaker (green).

The basic components of the system include actuators, sensors and an electronic control unit (ECU). The electronic hardware consists of an amplifier and filter unit that contains the power amplifier and the anti-aliasing filter for the sensor signal. Figure 1 presents two alternative principles, the inertial mass actuator (green) and active mount with integrated actuator (red), whereby the ECU, sensors and the actuator's basic components can be identically used. The two principles are similar in how they function, forces are fed into the system in targeted fashion so that the resulting dynamic forces at the base of the mount (attachment point) are reduced. In this example, attachment point acceleration is measured and supplied to the controller. The countersignal calculated in the control unit powers the actuator via power amplifiers. Ideally, the superimposed forces cancel out one another so that no annoying engine vibration is disseminated via the chassis.

Generally, there are two possible ways of active vibration cancellation, the Inertial mass shakers, attached at suitable points, cancel out the disturbing vibration by a force signal of opposite phase. On the other hand, active engine mounts compensate the displacement between engine mount and the car body. Hereby the car body is kept free from the vibration forces emitted from the engine. With regard to the specifications of the AVC system the suitable system configuration has to be chosen. In (Hartwig et al., 2000), (Karkosch & Marienfeld, 2010) the electrodynamic and the electromagnetic actuator principle and the two system configurations are compared. Figure 2 shows the electrodynamic and electromagnetic actuator principles.

Electrodynamic, linear (Lorentz force)   **Electromagnetic, nonlinear (reluctance principle)**   **Electromagnetic, linear (reluctance+Lorentz force)**
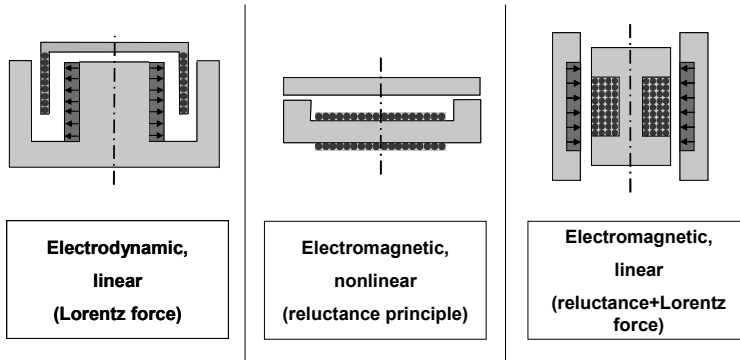
Fig. 2. Electrodynamic and electromagnetic actuator principles.

An electromagnetic actuator has the benefit to an electrodynamic actuator in higher actuator force at decreased magnet and design volume such as more cost efficient. On the other side, the electrodynamic actuator principle has the advantage in a simple design of the iron core and the absence of magnetic forces lateral to the deflection direction. A comparison between an active absorber and an active hydromount configuration is shown in Figure 3.
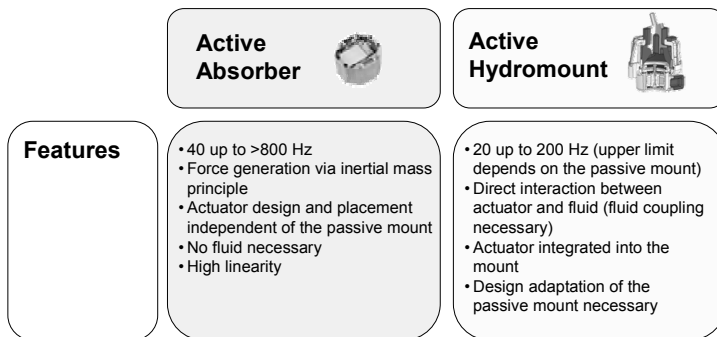

Active Absorber

Active Hydromount

Features

- 40 up to >800 Hz
- Force generation via inertial mass principle
- Actuator design and placement independent of the passive mount
- No fluid necessary
- High linearity

- 20 up to 200 Hz (upper limit depends on the passive mount)
- Direct interaction between actuator and fluid (fluid coupling necessary)
- Actuator integrated into the mount
- Design adaptation of the passive mount necessary

Fig. 3. Active engine mount system configurations – principle comparison.

3. Control system design

The problem of active control of noise and vibrations has been a subject of much research in recent years. For an overview see e.g. (Kuo & Morgan, 1996), (Hansen & Snyder, 1997),

(Clark et al., 1998), (Elliot, 2001) and the references therein. The main part of the published literature makes use of adaptive feedforward structures. Adaptive feedback compensation (Aström & Wittenmark, 1995), in which the feedback law depends explicitly upon the error sensor output has found little application in the active noise and vibration control field.

Feedforward control provides the ability to handle a great variety of disturbance signals, from pure tone to a fully random excitation. However, the performance of feedforward control algorithms can be degraded if disturbances are not measurable in advance (e.g. road or wind noise) or the transmission path characteristics change rapidly. Contrarily, a feedback controller can be designed to be less sensitive to system perturbations. Robustness and performance, however, are conflicting design requirements. To achieve a good attenuation of the vibrations the cancellation wave has to be very accurate, typically within ±5 degrees in phase and ±0.5 dB in amplitude.

3.1 FxLMS approach

The FxLMS algorithm has been originally proposed in (Morgan, 1980) and is described in detail in (Kuo & Morgan, 1996). The basic idea is to use the feedforward structure shown in Figure 4. The transfer path between the disturbance source and the error sensor is called *primary path*. The *secondary path* is the transfer path between the output of the controller and the error sensor. The aim in the control loop is to minimise the output signal (error signal).
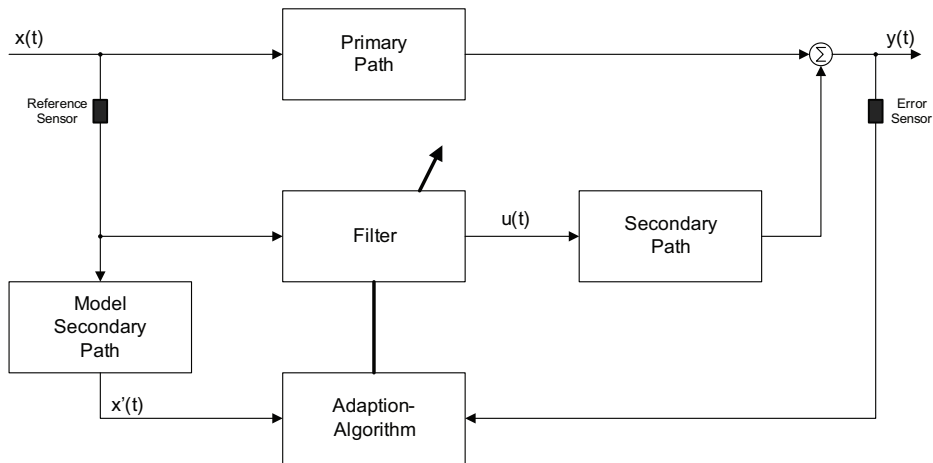

Fig. 4. Block diagram of FxLMS algorithm.

The adaptive filter has to approximate the dynamics of the primary path and the inverse dynamics of the secondary path. For the on-line adaptation of a FIR-filter (finite-impulse-response filter), two signals are used: error signal and reference signal filtered with the model of the secondary path (filtered-x).

The discrete-time transfer function of a FIR-filter has the form

$$F(z) = \frac{U(z)}{X(z)} = \frac{w_0 z^m + w_1 z^{m-1} + ... + w_m}{z^m} \ , \tag{1}$$

whereas the filter coefficients w_i, $i=1,...,m$ can be represented as a vector:

$$w(k) = \begin{bmatrix} w_0(k) & w_1(k) & \dots & w_m(k) \end{bmatrix}^T . \tag{2}$$

The adaptation of the filter weights w_i is performed through the well-known LMS (least mean square) algorithm originally proposed in (Widrow & Hof, 1960). A performance index J is built from the sum of squares of the sampled error signal:

$$J = \frac{1}{N} \sum_{i=1}^{N} y^2(i) . \tag{3}$$

This performance function depends on the filter coefficients and can be described through a hyperparaboloid as shown in Figure 5. The optimal values for the adaptive filter coefficients are located in the deepest point of the performance surface. The LMS-algorithm is searching on-line for the coordinates of the deepest point. The control signal is generated as the output of the adaptive filter.
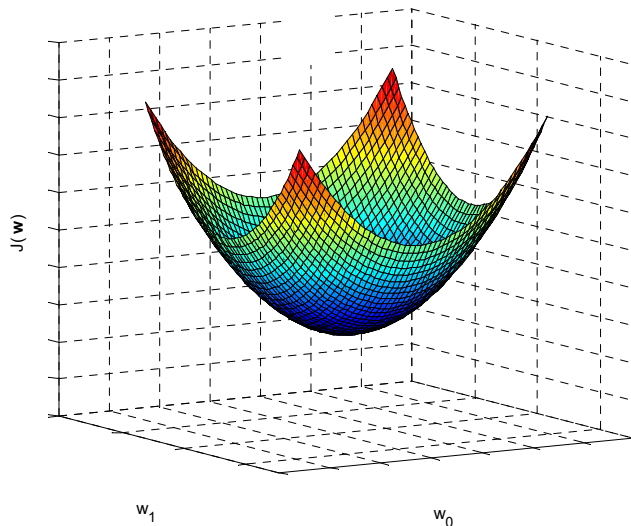

Fig. 5. Example of a performance surface for a two-weight system.

3.2 Disturbance observer approach

This method is based on state observer and state feedback and has been proposed in (Bohn et al., 2003), (Kowalczyk et al., 2004), (Bohn et al., 2004), (Kowalczyk & Svaricek, 2005). It is assumed that the disturbance enters at the input of the plant S, see Figure 6.

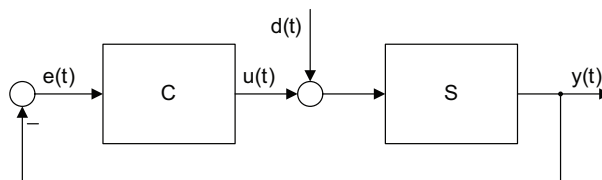

Fig. 6. Control loop with a plant S and a controller C.

The disturbance is modelled as a sum of a finite number of sine signals, which are harmonically related:

$$d(t) = \sum_{i=1}^{N} A_i \sin(2\pi f_i t + \varphi_i). \qquad (4)$$

This disturbance is time-varying and needs frequency measurements to be fed into the model. The disturbance attenuation is achieved through producing an estimate of the disturbance d and using this estimate, with a sign reversal, as a control signal u. To generate the estimate, a disturbance observer is used. The observer is designed off-line assuming time-invariance and investigating the property of robustness over a certain frequency region for a single observer. Later on, a gain-scheduling is implemented to cover the whole frequency region of interest by a stable observer. This provides a non-adaptive approach, where the frequency is used as a scheduling variable.

The transfer function of the controller C has infinite gain at the frequencies included in the disturbance model. The controller poles show up as zeros in the closed-loop transfer function. Figure 7 shows the frequency response magnitude of the sensitivity function $1/(1+CS)$.
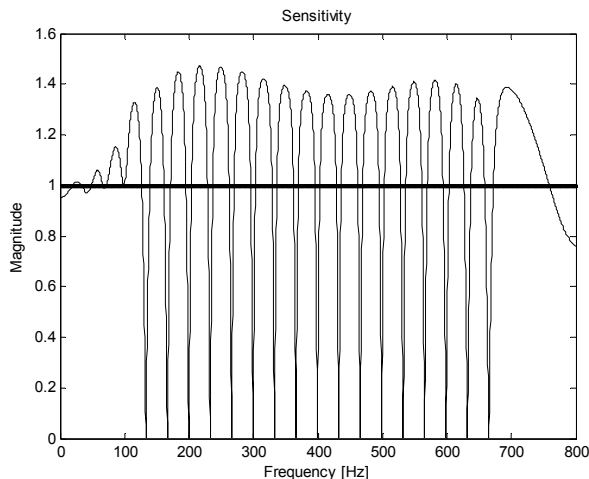

Fig. 7. Frequency response magnitude of the sensitivity function.

It can be seen that the magnitude of the sensitivity function is zero for the frequencies specified in the disturbance model, which corresponds to complete disturbance cancellation. The improvement of the disturbance attenuation for these frequencies leads to some disturbance amplification between these frequencies. This effect is in accordance with Bode's well-known sensitivity integral theorem and is called *waterbed effect* (Hong & Bernstein, 1998). For more details on this algorithm, see (Bohn et al., 2004).

Finally, both approaches can be combined to give a two-degree-of-freedom control structure, which is referred to as a hybrid approach in the ANC/AVC literature (Shoureshi & Knurek, 1996), (Hansen & Snyder, 1997). The implementation of all control algorithms is usually done on digital signal processing hardware.

Due to a large number of influence parameters, no definite statements can be made with regard to which control scheme will give a better performance. Rather, control strategies have to be chosen with regard to the characteristics of the vibration problem to be addressed, such as available sensor signals (e.g., costs associated with additional feedforward sensors, possible use of existing sensors), Type of excitation (periodic, e.g. engine vibrations, or stochastic, e.g. road excitations), Frequency range of interest (e.g. 25 – 30 Hz for idling speed or 25 – 300 Hz for the whole engine speed range), Spectral characteristics of excitation (narrowband, e.g. distinct frequencies, or broadband; e.g. fixed/varying frequencies).

The decision for one particular control strategy and the determination of suitable controller settings is a very important step in the development of ANC/AVC schemes. Therefore, simulation studies and real-time experiments on vehicles are carried out to identify a suitable strategy for a given noise and vibration problem. For the real-time experiments, the control strategies, together with auxiliary function such as signal conditioning and monitoring routines, are implemented on a rapid prototyping system.

4. Experimental results

In the last years, several vehicles — with different problems — have been equipped with active absorber systems to attenuate the transmission of the engine vibrations into the vehicle cabin. As mentioned earlier, the control algorithms have to be chosen with regard to the particular problem of the considered vehicle. ContiTech has equipped a test vehicle with an AVC system with inertia-mass shaker attached on the transmission cross-member. Figure 8 shows the location of the system components on the transmission cross-member in the test vehicle.

Fig. 8. Location of the AVC components in the test vehicle.

The control algorithm is implemented on a rapid prototyping unit, the dSPACE MicroAutoBox. The electronic hardware consists of an amplifier and filter unit that contains the power amplifier and the anti-aliasing filter for the sensor signal, and the electronic control unit. A remote control on/off switch is used to turn the control algorithm on and off during vehicle tests (Kowalczyk et al., 2006).

In control engineering terms, the transfer function from the amplifier input to the (filtered) sensor output is the transfer function of the plant to be controlled (assuming linearity and

time invariance). In accordance with the active noise and vibration control literature (Kuo & Morgan, 1996), (Hansen & Snyder, 1997), (Clark et al., 1998) this is called the secondary path S. To design a control algorithm, a model for the secondary path is required. Quite often models for vibration control systems are derived from physical principles (Preumont, 1997), from finite-element models or through experimental techniques such as modal analysis (Heylen et al., 1997). Physical principles are mostly applied to fairly simple mechanical structures such as beams or plates for which analytical solutions can be found. Finite-element models or models derived from modal analysis will give a model of the structure only, that is, without the dynamics of the electrical and electromechanical components (amplifier, actuator, sensor). The approach taken here is to excite the system with a test signal and record the response. Any of the discrete-time black-box system identification techniques (such as the least squares approach for equation-error models) can then be used to identify a model (Ljung & Söderström, 1983). Figure 9 shows the amplitude and phase responses of an identified system transfer function. The amplitude response would be dimensionless, since it corresponds to the output voltage, i.e. the filtered sensor signal, over the input voltage of the amplifier. However, for interpretability, the output signal has been scaled to acceleration (m/s², using the sensor sensitivity) and the input signal to current (A, using the amplifier gain). Such models are used for the subsequent controller design and for simulation studies.
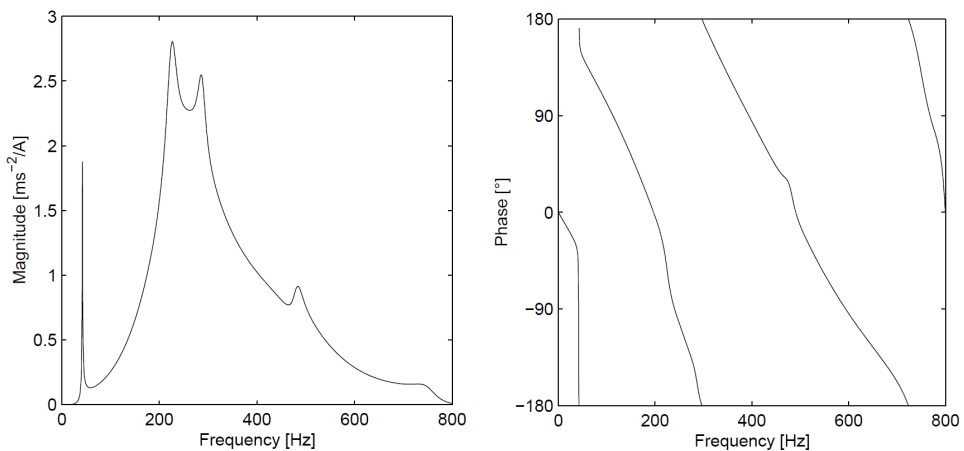

Fig. 9. Amplitude and phase plots of an identified system transfer function (actuator current to filtered sensor output; the first peak corresponds to the resonance of the inertia-mass actuator).

For instance, the stationary behavior of the controlled system is of interest when the comfort under idling speed conditions should be improved.

A typical real-time result for such a problem is given in Figure 10. Here, a comparison of the error signals (measured accelerations at the frame) is shown for control off and on. It can be seen that the engine orders 2, 4 and 6 are predominant at idling speed without active control. However, a significant reduction (up to 37 dB) of these engine orders can be achieved by using an AVC system.
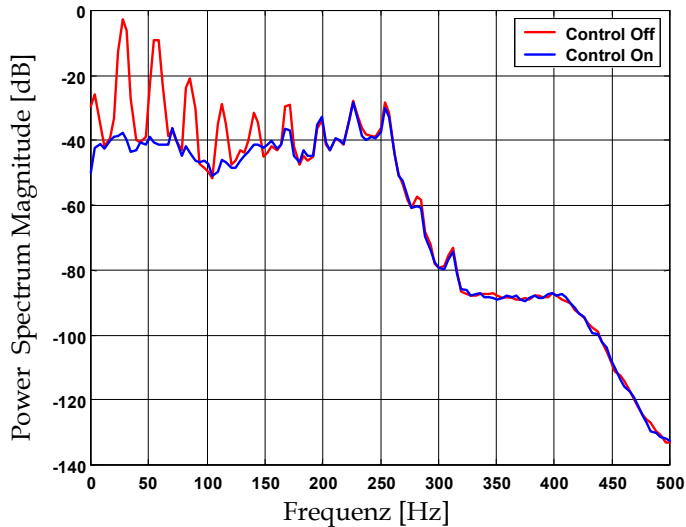
Fig. 10. Power spectrum of the measured frame vibrations at idle.

In other applications, the active system should work over a wide engine speed range. For such applications the tracking behavior of the active system must be considered. Figure 11 gives an impression of the dynamic behavior of the adaptive FxLMS algorithm. To illustrate the adaptation of the controller, the decrease in the measured frame vibrations after switching on the control algorithm at t=1[s] is shown.
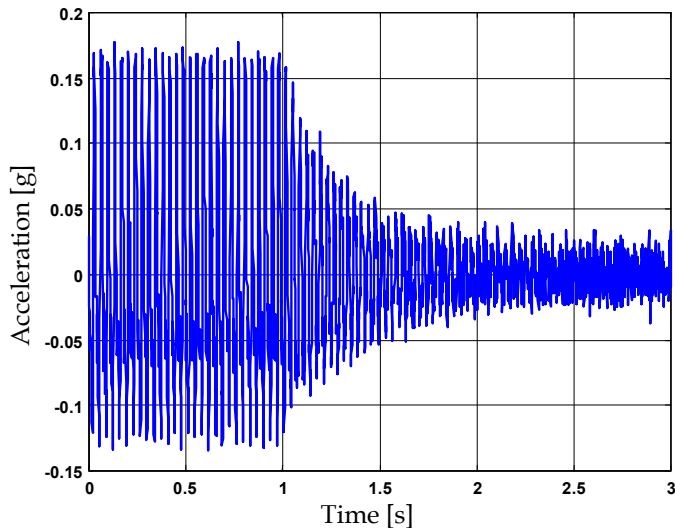

Fig. 11. Adaptation behavior of the FxLMS algorithm.

It is well–known that parts of the transmitted vibration energy through the mounts pass through the chassis and emanate in the vehicle passenger compartment in the form of structure–borne noise. Figure 12 shows an order analysis of a sound pressure level measurement at the passenger's left ear of a test vehicle that has acoustic problems in the frequency range between 200 and 300 Hz.
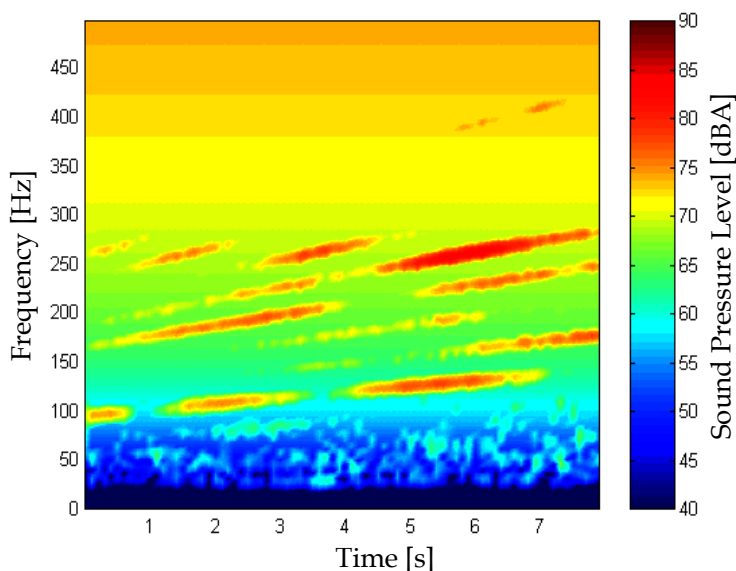

Fig. 12. Order analysis of sound pressure level (passenger's left ear) of a road test (acceleration from 1800 to 4500 rpm, full throttle, 3rd gear, control off).

Here a lot of engine orders (2.5, 3, 3.5, ...) are visible since the transmission mount is the major path for this engine–induced noise. The improvement with control on is shown in Figure 13. The sound pressure level measurement at the passenger's left ear points out a significant reduction in sound for frequencies higher than 120 [Hz].

Due to the fact that the measured vibrations at the transmission are well correlated to the cross member vibrations a classical FxLMS algorithm has been chosen for this application. An impressive reduction of the sound pressure level, achieved by the small (weight about 0.6 kg) active absorber at the transmission mount, can be registered in Figure 13. The remaining 2nd order line is a result of the vibrations that are still transmitted through the two front engine mounts.

The active absorber system has not only a great impact on the interior noise of the vehicle but also on vibrations at comfort relevant points. Such an interior comfort improvement for the passengers can be observed from a control on/off comparison of the power spectrum of the measured acceleration signal at the steering wheel, see Figure 14.
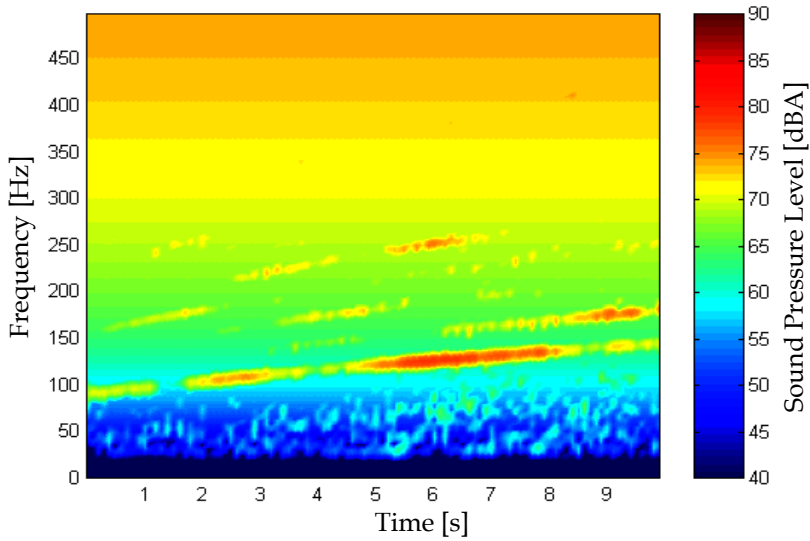
Fig. 13. Order analysis of sound pressure level (passenger's left ear) of a road test (acceleration from 1800 to 4500 rpm, full throttle, 3rd gear, control on).
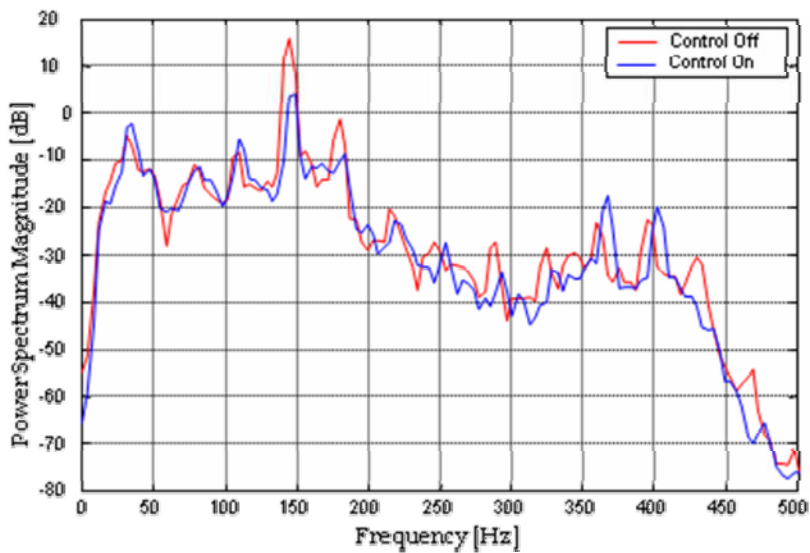

Fig. 14. Power spectrum comparison of the measured steering wheel acceleration for constant drives with 4400 RPM.

5. Conclusion

This chapter has given an overview of recent research and development activities in the field of active noise and vibration control in automotive applications. The design of an ANC/AVC system with its components is described in general such as two control approaches, a feedforward and a feedback approach, are presented in detail. Experimental results from a test vehicle, equipped with an AVC system with inertial-mass shaker and a dSpace MicroBox, were discussed.

Recent advances in NVH (Noise Vibration Harshness) design and analysis tools, development of low cost digital signal processors, and adaptive control theory, have made active vibro–acoustic systems a viable and economically feasible solution for low frequency problems in automotive vehicles.

Further experimental results and a comparison of the presented control approaches can be found in (Kowalczyk et al., 2004) and (Kowalczyk & Svaricek, 2005).

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Vibration Control

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Vibrations are a part of our environment and daily life. Many of them are useful and are needed for many purposes, one of the best example being the hearing system. Nevertheless, vibrations are often undesirable and have to be suppressed or reduced, as they may be harmful to structures by generating damages or compromise the comfort of users through noise generation of mechanical wave transmission to the body. the purpose of this book is to present basic and advanced methods for efficiently controlling the vibrations and limiting their effects. Open-access publishing is an extraordinary opportunity for a wide dissemination of high quality research. This book is not an exception to this, and I am proud to introduce the works performed by experts from all over the world.

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