An Overview of Recent Automotive Applications of Active Vibration Control

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ABSTRACT

Modern control applications are becoming increasingly important in the area of vehicle riding comfort. An attractive application in this area is the use of active vibration control in engine mounting concepts, particularly since conventional mounts are approaching their inherent limitations. The standard approach is to isolate the engine and the transmission vibrations from the chassis with rubber or hydro mounts. This mount design is always a compromise between the conflicting requirements of acceptable damping and good isolation.

Continental has developed and implemented prototypes of active mounting systems on various test vehicles and demonstrated that significant reductions in noise and vibration levels are achievable. This paper will present an overview of the system components, control algorithms and experimental results.

1.0 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., [1-6]).

Research and development activities at Continental have focused on the transmission of engine-induced vibrations through engine and powertrain mounts into the chassis. [7-13] Engine and powertrain mounts

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are usually designed according to criteria that incorporate trade-offs between vibration isolation and engine movement since the mounting system in a 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 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 [14]). 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 [15-31]. Continental has implemented prototypes of AVC systems in various test vehicles and demonstrated that significant reductions in noise and vibration levels are achievable [7-13].

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 [32-35]. 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 [32-35]. 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 behaviour 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 of this paper will present an overview of the system components, control algorithms, as well as obtained experimental results and is organised as follows. In Section 2, the AVC system installed in the test vehicle is described. This system is used to validate the algorithms described in 3. Experimental results are presented and discussed in Section 4. A summary and some conclusions follow in Section 5.

2.0 SYSTEM DESCRIPTION

A schematic representation of an AVC system is shown in Fig.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.
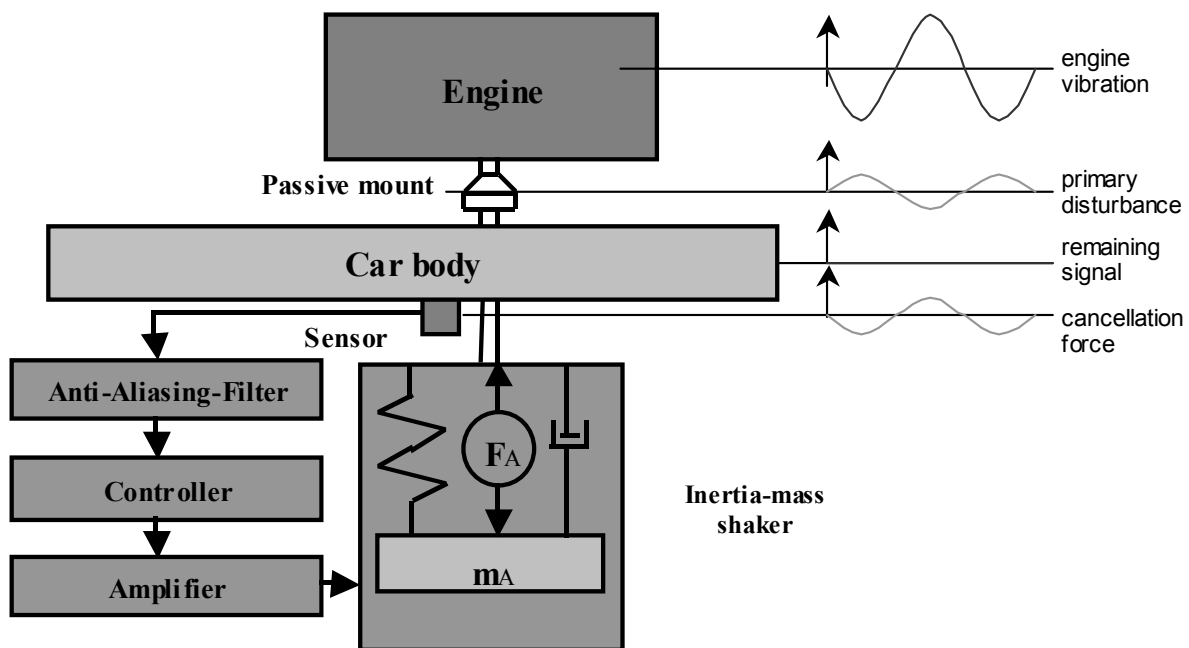

**Figure 1: Schematic representation of an AVC system.**

In active noise and vibration control, the reference signal is commonly zero; therefore, the controlled signal is also the error (with a sign reversal), which is why the output sensor is usually called the error sensor. The cancelling force is generated through an electromagnetic inertia-mass actuator of the type described in [36]. This actuator is driven by a power amplifier that converts the voltage signal from an electronic control unit into an actuator current. The control signal u is the input of the power amplifier (a voltage), and the output signal y is the acceleration sensor output (also a voltage). Since the controlled signal is the chassis acceleration, the output signal is scaled to m/s^2 (which is more meaningful than the sensor voltage). Also, the current flowing through the actuator is more relevant than the amplifier input voltage, therefore, the input signal u is scaled to amperes. Figure 2 shows the location of the system components on the transmission cross-member in a test vehicle.

Figure 2: Location of the active absorber and the sensors in a 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 (see Fig. 3). A remote control on/off switch is used to turn the control algorithm on and off during vehicle tests.
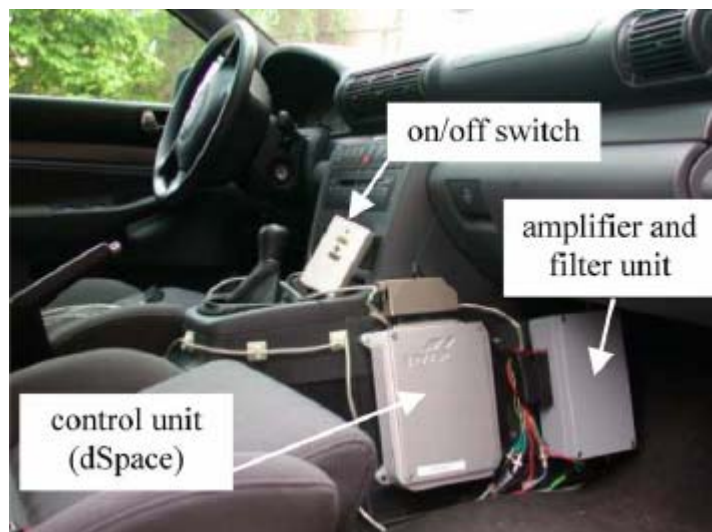
Figure 3: Location of the electronic hardware in the test vehicle.

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 [32-34] 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 [37], from finite-element models or through experimental techniques such as modal analysis [38]. 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 therefore 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 [39]. Figure 4 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.
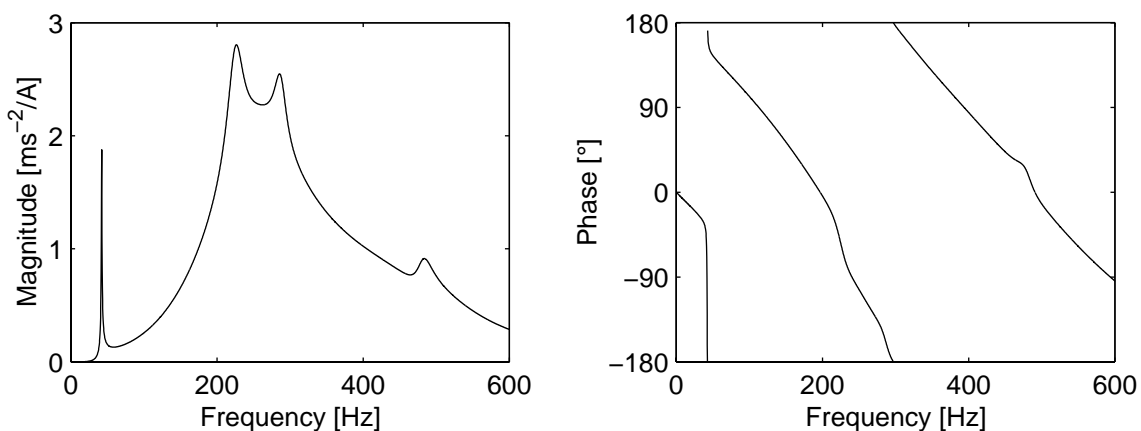
Figure 4: Amplitude and phase plots of an identified system transfer function (actuator current to filtered sensor output; the first peak corresponds to the resonance of the inertia-mass actuator).

3.0   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. [32-35] and the references therein. The main part of the published literature makes use of adaptive feedforward structures. Adaptive feedback compensation [40], 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 in Figure 1 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 [41] and is described in detail in [32]. The basic idea is to use the feedforward structure shown in Figure 5. 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).
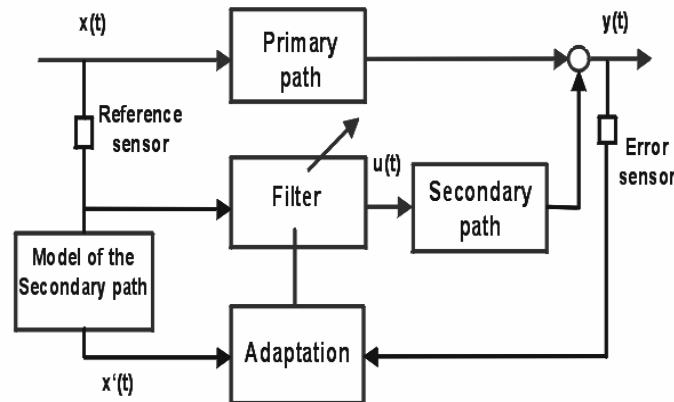
Figure 5: Block diagram of FXLMS algorithm.

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

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

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

$$\boldsymbol{w}(k) = [w_0(k) \quad w_1(k) \quad ... \quad w_m(k)]^T \qquad (2)$$

The adaptation of the filter weights w_i is performed through the well-known LMS (least mean square) algorithm originally proposed in [42]. A performance index J is built from the sum of squares of the sampled error signal:

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

This performance function depends on the filter coefficients and can be described through a hyperparaboloid as shown in Fig. 6.
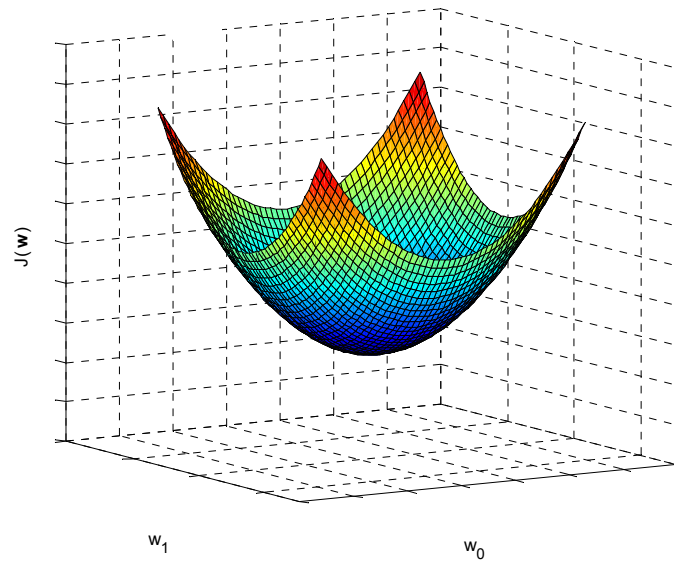
Figure 6: Example of a performance surface for a two-weight system.

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.

3.2   Disturbance Observer Approach

This method is based on state observer and state feedback and has been proposed in [11-13]. It is assumed that the disturbance enters at the input of the plant *S* (see Fig. 7).
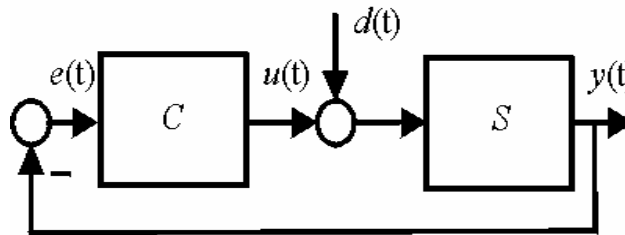

Figure 7: Control loop with a plant *S* and a controller *C*.

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

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

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 8 shows the frequency response magnitude of the sensitivity function $1/(1+CS)$.
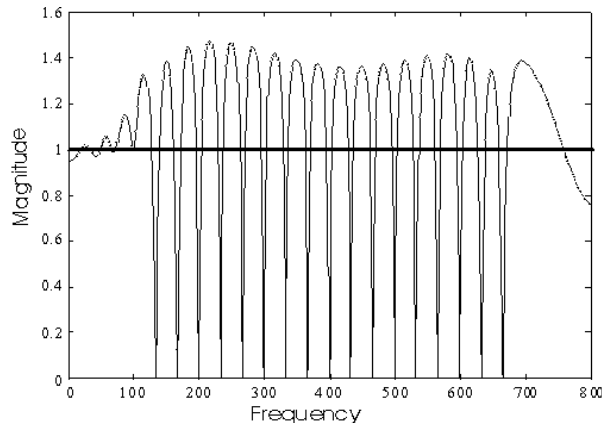
Figure 8: Frequency response magnitude of the sensitivity function.

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* [43]. For more details on this algorithm, see [13].

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 literature [29,33]. 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 regards 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; 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 AVC/ANC schemes. In our development, 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.

4.0 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.

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 9.
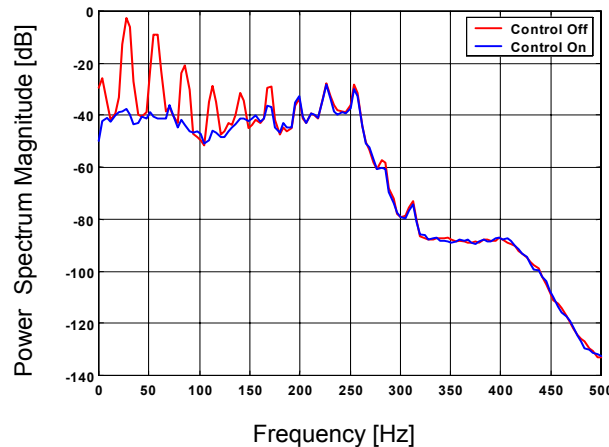

Figure 9: Power spectrum of the measured frame vibrations at idle.

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 AVC.
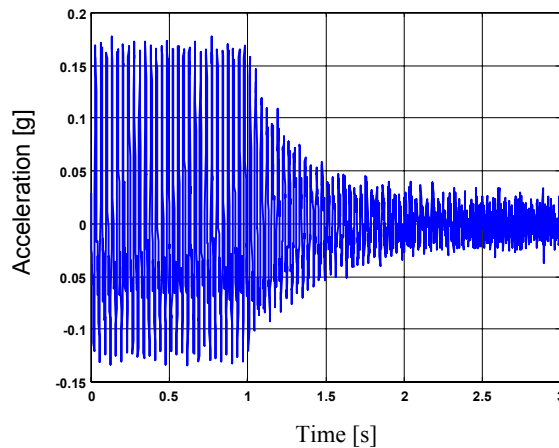

Figure 10: Adaptation behavior of the FXLMS algorithm.

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 10 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.

It is well–known that part of the transmitted vibration energy through the mounts passes through the chassis and emanates in the vehicle passenger compartment in the form of structure–borne noise. Figure 11 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.
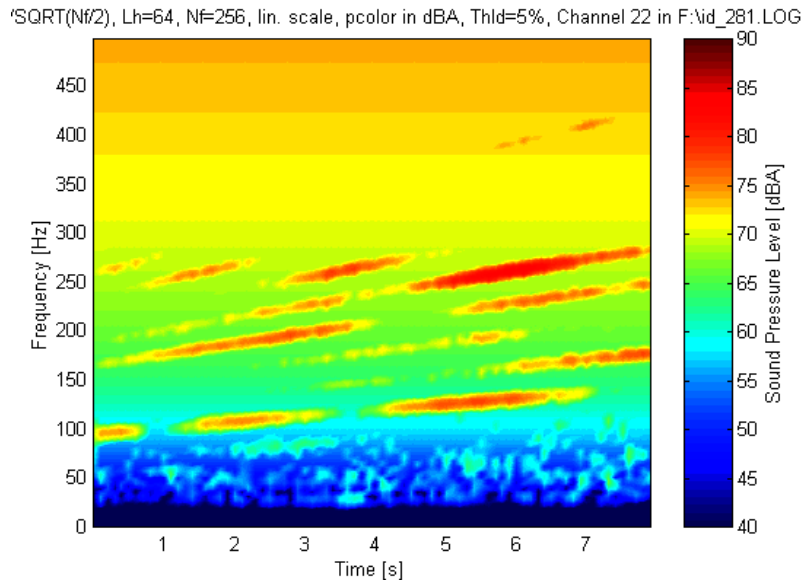
**Figure 11: Order analysis of sound pressure level (passenger's left ear) of a road test
(acceleration from 1800 to 4500 rpm, full throttle, 3rd gear, control off).**

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. To improve the acoustic behavior, the test vehicle has been equipped with the active absorber system shown in Figure 2. 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.
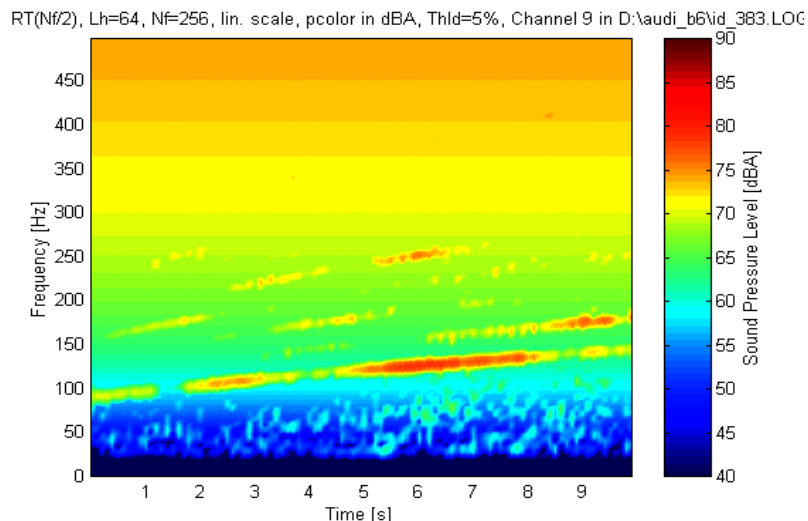
**Figure 12: Order analysis of sound pressure level (passenger's left ear) of a road test
(acceleration from 1800 to 4500 rpm, full throttle, 3rd gear, control on).**

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 12. 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, Fig. 13.
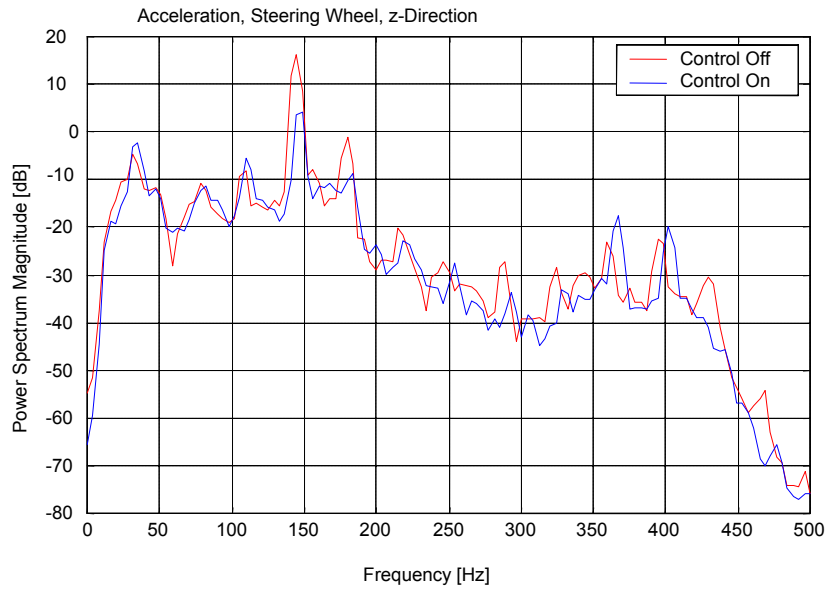

Figure 13: Power spectrum comparison of the measured steering wheel acceleration for constant drives with 4400 RPM.

5.0   CONCLUSION

This paper has given an overview of recent research and developments activities in the field of active noise and vibration control of the automotive supplier Continental AG and the University of the German Armed Forces, Munich. Recent advances in NVH 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. Future work at Continental will focus on the suitability of the components and the algorithms for the implementation of active vibration control systems in series production vehicles.

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Detailed Analysis or Short Description of the AVT-110 contributions and Question/Reply

The Questions/Answers listed in the next paragraphs (table) are limited to the written discussion forms received by the Technical Evaluator. The answers were normally given by the first mentioned author-speaker.

P24   K. Kowalczyk, F. Svaricek, C. Bohn 'An overview of Recent Automotive Applications of Active Vibration Control', (Univ of the German Armed Forces, DE)

This paper focus on AVC (Active Vibration Control) and more specifically on the isolation of the engine in a Ground Vehicle. The developed Control basically lies on the fact that 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. The control algorithm was implemented on a rapid prototyping unit, the dSPACE MicroAutoBox. The author gave the analytical background of the control adopted at Continental and demonstrated that a significant reduction of vibrations and noise was obtained: the coupling between noise and vibration was not discussed.

Discussor's name: M-C Tse
Q. Is it a close-loop control? If so, how do you take into account of the time delay constant and how to determine it? Has the sensor location an impact on the control?
R. Yes, the disturbance-observer approach is the closed-loop control technique. The
Model of the secondary path contains information about the time-delay constant. The controller design is performed for a ready set-up. The control is designed for the specific sensor location.

Discussor's name: C. Petiau
Q. How do your algorithms work with unsteady excitation (like 'asynchronous' vibrations due to nonlinearities of engine bearings)?
R. The control approaches are robust for non-harmonical disturbances. We have done a number of experiments trying to destabilize the control loop. The controller stayed stable.