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# AN ACTIVE ABSORBER TO IMPROVE THE SOUND QUALITY IN THE PASSENGER COMPARTMENT OF VEHICLES

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**ABSTRACT**
In comfortable vehicles further improvement of sound quality requires additional optimisation of the low frequency acoustic comfort. To reach this goal a highly compact active system, consisting of microphones as sensors, loudspeakers as actuators, and of a feedback control system circuit, was developed. The system can be regarded as an active absorber formed by an active device featuring the characteristics of a broad-band Helmholtz resonator. In addition to the description of the optimum construction and tuning parameters, detailed results as to its acoustic overall performance and with regard to its many possibilities offered to influence the sound quality in the passenger compartment of vehicles are presented.

## 1 - INTRODUCTION
It is a well-known fact that luxurious (chauffeur driven) sedan vehicles offer a high acoustic comfort. Due to the implementation of an effective passive sound attenuation treatment package, the major part of the acoustic annoyance can be reduced to an acceptable level. This especially applies to the higher frequency range, where the acoustic treatment, e.g. with regard to its absorption capabilities, can significantly contribute to improve the acoustic performance of the vehicle. On the other hand, passive treatments do not allow to noticeably influence the low frequency comfort in the passenger compartment. This must be considered as a great disadvantage since it is a well-known fact that especially a further improvement of the low frequency noise characteristics could significantly improve the noise quality perceived by the passengers.
In the low frequency range (up to 200 Hz), cavity-eigenmodes significantly influence the acoustic characteristics in the passenger compartment of a vehicle. The use of Helmholtz resonators is a well-known and very effective method for damping acoustic resonances in buildings and also in vehicles [1]. However, a conventional Helmholtz resonator shows two disadvantages: Sound attenuation is limited to a small frequency band near its resonance frequency and good effectiveness requires a resonator cavity of a large size [2], which could not be realised in a vehicle.
To overcome these disadvantages, a highly compact active system, consisting out of microphones as sensors, loudspeakers as actuators, and of a feedback control system circuit, was realised and will be presented. Special focus will be placed on the system's operation characterised by its broad-band effective frequency behaviour to improve the sound quality in the passenger compartment of vehicles.

## 2 - DYNAMIC RESPONSE OF A COUPLED CAVITY-RESONATOR SYSTEM
A general modal description of the sound field in a cavity allows evaluating the possibilities for noise attenuation at low frequencies [3, 4]. Let us assume, that a vibrating area $A_{exc}$ at position $\mathbf{x}_{exc}$ with displacement $\xi_{exc}$ excites a cavity with nearly rigid walls (Fig. 1). In addition, a resonator system is coupled to the cavity at position $\mathbf{x}_{res}$ through the coupling area $a$. The dynamic behaviour of this resonator system is generally described by its dynamic compliance $\xi^{res}/F(\omega)$ and it should be excited

only by the sound pressure $p(\mathbf{x}_{res})$ in the cavity at the position of the coupling area. The dynamic response of this cavity-resonator system at the position $\mathbf{x}$ can be derived from:

$$\frac{p(\mathbf{x})}{\ddot{\xi}_{exc}} = -\sum_{r=0}^{N} \frac{\Theta_r(\mathbf{x})}{K_r} \cdot \frac{A_{exc}}{\omega_r^2 - \omega^2 + 2j\omega\omega_r\vartheta_r} \cdot \left[ \Theta_r(\mathbf{x}_{exc}) + \Theta_r(\mathbf{x}_{res}) \cdot \frac{a^2 \cdot \frac{\xi^{res}}{F} \cdot \omega^2 \cdot \sum_{i=1}^{N} \frac{1}{K_i} \frac{\Theta_i(\mathbf{x}_{res}) \cdot \Theta_i(\mathbf{x}_{exc})}{\omega_i^2 - \omega^2 + 2j\omega\omega_i\vartheta_i}}{F} \right] \qquad (1)$$

with $\omega$, $\omega_r$, $\vartheta_r$, $K_r$, $\Theta_r$: frequency; resonance frequency, damping ratio, modal stiffness of and sound pressure distribution in the $r$-th eigenmode.

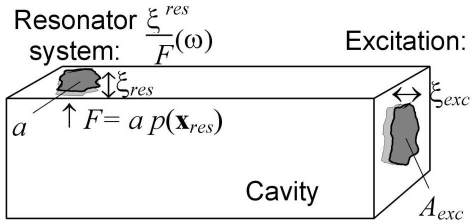


**Figure 1:** Cavity with coupled resonator system.

With regard to the effectiveness for sound attenuation Eq. (1) reveals, without going into details, the importance of the size of the coupling area $a$ and the dynamic compliance of the resonator system [4]. The effect on the sound pressure in the cavity increases with both magnitudes.
On the basis of this theory, the effectiveness and the optimal compliance of broad-band effective systems can be evaluated. In a first step, the excellent narrow-band sound attenuation capabilities of a Helmholtz resonator [2] are expanded to a wider frequency range. At the resonance frequency, the damping factor $d$ solely determines the magnitude of the resonator's dynamic compliance. If a resonator would be in resonance for all frequencies simultaneously, the dynamic compliance of such an "ideal" broad-band resonator would be:

$$\frac{\xi^{res}}{F}(\omega) = \frac{1}{j\omega d} \qquad (2)$$

Indeed, by coupling this system to a cavity, sound pressure attenuation at all frequencies can be observed (Fig. 2), and the resonance peaks are completely eliminated. The effectiveness is especially high at the high sound pressure levels in the resonances, due to the fact that the resonator system reacts on the sound pressure field in its coupling area.
However, this dynamic compliance cannot be achieved, if it has to be realised simultaneously at different frequencies. The dynamic compliance of a set of (non-interacting) Helmholtz resonators with different resonance frequencies realised in a single resonator compliance is a good approach for the approximation. The compliance derived can be expressed as

$$\frac{\xi^{res}}{F}(\omega) = \sum_{i=1}^{N_{res}} \frac{1}{-\omega^2 m_i + j\omega d_i + k_i} = \sum_{i=1}^{N_{res}} \frac{1}{m_i} \frac{1}{\omega_i^2 - \omega^2 + 2j\omega\omega_i\vartheta_i} \qquad (3)$$

with $m_i$, $k_i$, $\omega_i$, $\vartheta_i$, $d_i$: eff. air mass, stiffness, resonance frequency, damping ratio, damping coefficient of the $i$-th resonator.
The frequency response is shown in Fig. 3. In the band of the resonance frequencies, large amplitude values and phase angles around 90° can be found. The effectiveness of this compliance coupled to the cavity gives the possibility for sound attenuation in a wide frequency range quite similar to that of the ideal broad-band resonator (Fig. 2).
The optimum construction parameters are similar to those of a (mono-frequent) Helmholtz resonator: The best effectiveness can be achieved by using a small vibrating mass $m_{res}$, for a high compliance, and a large coupling area $a$. Here all masses $m_i$ are set to the same value $m_{res}$. The delta between the different resonance frequencies has to be small, and the damping has to be selected large enough in relation to this delta, so that the resonance peaks overlap.

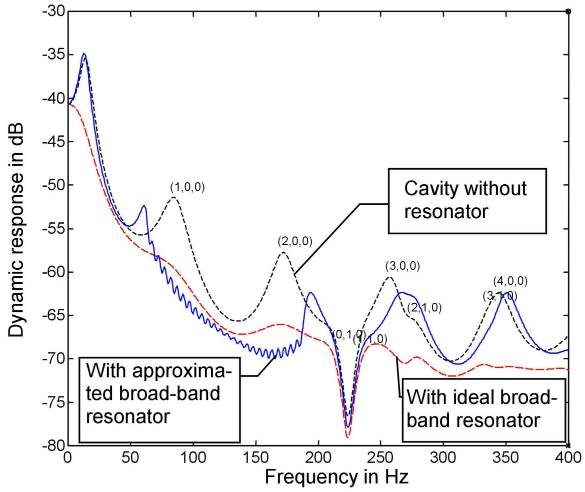


**Figure 2:** Dynamic response $\dfrac{p\left(\mathbf{x}_{res}\right)}{\ddot{\xi}_{anreg}}$ of a cavity (dimensions: 2 m, 0.8 m, 0.4 m) with and without broad-band resonators.

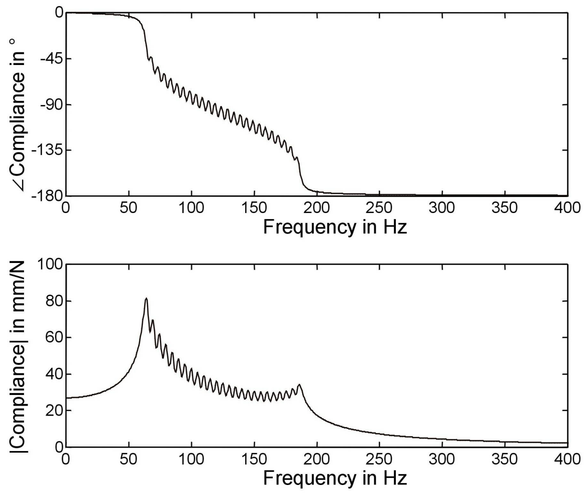


**Figure 3:** Compliance of the approximated broad-band resonator ($N_{res}$=25, $\omega_i = 2\pi\,(65, 70, \ldots, 185)$ Hz).

Besides the use of the optimum tuning parameters for the dynamic compliance of the resonator system, the dynamic response of the coupled cavity-resonator system in Eq. (1) shows - due to the sound pressure distribution $\Theta_r\left(\mathbf{x}\ldots\right)$ in the eigenmodes - a strong dependence on the positioning of the resonator relative to the position of the excitation and the position of the receiver, i.e. the customer. A detailed analysis [4] reveals that for a best use in a vehicle, the positioning of a resonator system for broad-band sound attenuation has to be near to the receiver position, especially if a reduction of the noise level is required not only in the cavity eigenmodes. These positions are the head positions of the passengers. Therefore, this positioning was also used as a basis for the foregoing analysis.

## 3 - THE ACTIVE RESONATOR

The system used to realise the broad-band effective dynamic compliance consists out of a microphone as a sensor and a small electrodynamic loudspeaker as actuator (Fig. 4). A feedback control system circuit drives the loudspeaker on the basis of the sound pressure detected by the microphone installed directly in front of the system. This control circuit is targeted to react on an incoming sound pressure wave in a similar way as a large number of slightly different passive Helmholtz resonators (see Eq. (3)).
The frequency response of the controller has to be designed according to the electrical and mechanical dynamic behaviour of the loudspeaker-box system. The controller eliminates the resonance formed by the mass of the loudspeaker diaphragm and the stiffness of the membrane-suspension together with the

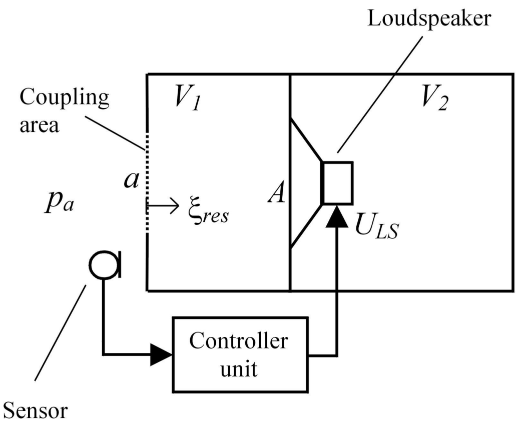


**Figure 4:** Schematic illustration of the active resonator.

stiffness formed by the air in the cavity of the loudspeaker housing. Additionally, the target compliance, as indicated in Eq. (3), has to be implemented. The controller design also takes into account a small cavity $V_1$ in front of the loudspeaker and a reduction of the cross-sectional area by a protection grid as shown in Fig. 4. The resulting frequency response of the controller is given by

$$\frac{U_{LS}}{p_a} \approx \frac{1}{m_{res}} \cdot \frac{R_s}{BL} \cdot \frac{a^2}{A} \cdot \sum_{i=1}^{N_{res}} \frac{-m_{LS}\omega^2 + j\omega d_{LS} + k_{LS}}{-\omega^2 + 2j\omega\omega_i\vartheta_i + \omega_i^2} \tag{4}$$

with $m_{LS}$, $d_{LS}$, $k_{LS}$, $R_S$, $BL$ - Parameters of the electrodynamic loudspeaker-system.
The frequency response is plotted in Fig. 5 together with the approximation by an IIR-filter of 4-th order, which is used in the applications with a digital controller unit.

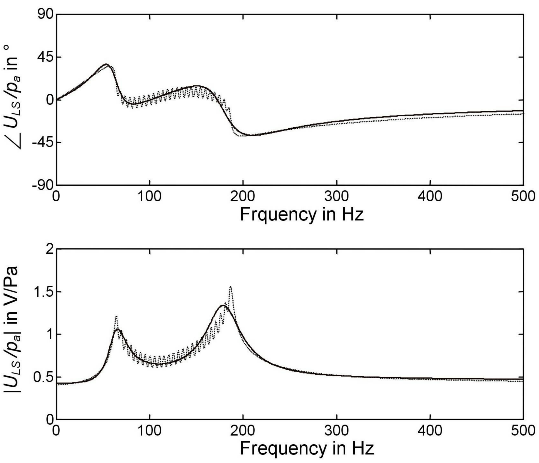


**Figure 5:** Frequency response of the controller; dashed line: according to Eq. (4), continuous line: approximation with IIR-filters of ord. 4.

With the mathematical description of the active resonator and of the interaction with the sound pressure in a defined cavity, a detailed analysis of the effectiveness of the resulting system for sound attenuation and the behaviour of the feedback control circuit is possible, especially with regard to its stability properties [4]. The sound pressure generated by the loudspeaker is fed back via the microphone. A closed loop system results therefrom, which, because of stability considerations, is not allowed to show a positive feedback characteristic with values greater than one. This analysis and practical experience reveal, that the stability criterion forms the limitation of effectiveness for the active resonator. This reasoning can be explained on the basis of Eq. (4): Thus reduced controller amplification is equivalent to an increased vibrating mass $m_{res}$ and accordingly a decreased simulated dynamic compliance, which entails a reduced sound attenuation.

## 4 - APPLICATIONS FOR SOUND OPTIMISATION IN A VEHICLE

The interior noise in the passenger compartment of a vehicle can be divided into three main components: 1) engine noise, 2) rolling noise and 3) wind noise. In the following, for two applications, the benefits of the active resonator to optimise the acoustic level and the sound quality in the passenger compartment of a vehicle will be presented. One application deals with the attenuation of engine noise, another with the attenuation of wind and rolling noise.

### 4.1 - Engine noise attenuation

Engine noise spectra are characterised by a variety of harmonics in context with the engine rotation. Especially in 4-cylinder engines, the second harmonic is dominant. A significant acoustic improvement can be achieved by the attenuation of the noise components related to the second engine harmonic. In this application it is not necessary for the active absorber to simulate the dynamic compliance of a broad-band resonator at all frequencies at the same time. It is sufficient to just simulate the compliance of a huge, slightly damped and thus very effective Helmholtz resonator at its resonance frequency. This resonance frequency is continuously tuned to the actual second order excitation frequency. In this system, an easy to realise controller can be used, and besides the optimisation of the interior noise, the effect of the theoretical ideal broad-band resonator on the sound pressure in a real car can be analysed. The system consists out of four separate resonators, placed near the headrests of the passenger seats.
The sound attenuation during a slow run-up on the rear seat passenger's head is shown in Fig. 6. In accordance with the theoretical results, derived by making use of the ideal broad-band absorber, the sound pressure in the second engine harmonic is reduced by more than 10 dB in the complete frequency range from 50 Hz (1500 rpm) to 200 Hz (6000 rpm). Because of the dominance of the second harmonic, the overall level is also reduced by 5 - 10 dB(B). Besides the attenuation of the sound pressure level and thus the loudness - in a typical boom-rpm range around 4200 rpm from 30.7 sone to 21.7 sone - the subjective annoyance is reduced by far more due to the elimination of the tonal boom component. The composition of the interior noise is more harmonic, and subjectively the sound quality improves a lot.

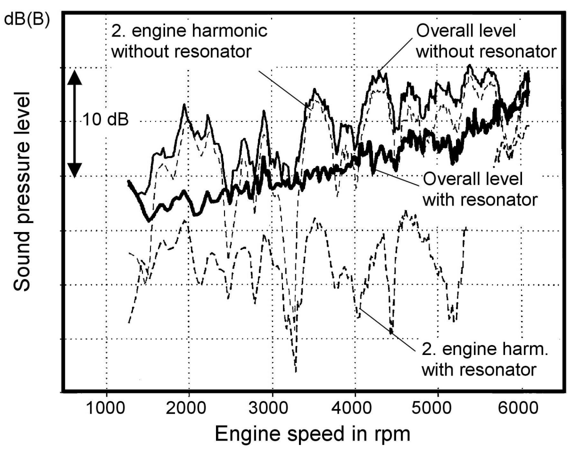


**Figure 6:** Sound pressure attenuation with a frequency-variable active resonator.

A great advantage of the active absorber - besides its high compactness - is the possibility to easily tune the effectiveness to the actual driving conditions. In the case the customer is driving very sporty, the sound attenuation can be adaptively reduced during full load acceleration in order to offer more feedback from the engine. Of course it is increased again to maximum effectiveness under steady state conditions to offer the highest comfort. A further step in this active sound design is possible by introducing additional load-dependent engine sound components by the speaker system [5].

### 4.2 - Attenuation of wind and rolling noise

Rolling noise is characterised by broad-band, stochastic character with most of its energy in the low frequency range. It is excited in the tire-road contact patch area and it is transferred through the noise-transfer-paths form the suspension through the car body to the passengers' ears. In addition to this structure-borne rolling noise, aero-acoustic effects (wind noise) excite the low frequency band in the passenger compartment especially at higher driving speeds. The sound pressure distribution at frequencies in the vicinity of the eigenfrequencies of the lower cavity eigenmodes (with pressure maxima at the ends of the cavity) has the effect that the low frequency (50-150 Hz) sound pressure level is higher

at the back seat area than at the front seats. This is especially a problem in highly comfortable, chauffeur driven vehicles.
In this example of an application, the active absorber is designed to reduce the increased sound pressure level locally on the rear seat positions in the frequency band from 50 to 150 Hz. Because of the feedback controller structure, the system is best suited for this application due to the fact that every noise contribution recorded in the front of the system is reduced, regardless of its excitation mechanism. In particular, also the low frequency component of wind noise is attenuated. The system is integrated behind the headrests in order to achieve a high local effectiveness with two microphone-loudspeaker units on each seat (11 cm-loudspeakers in a 3.3 l housing).
This compact resonator configuration allows to reduce the sound pressure level by 4-5 dB from 50 to 150 Hz (Fig. 7), which is equivalent to an attenuation of the specific loudness in the dominant second critical band by 17% from 5.9 to 4.9 sone/Bark. Thus it is possible to achieve low frequency interior noise levels at the rear seats equivalent to those at the front seats. The overall loudness reduction from 31.9 sone to 29.1 sone alone does not represent the subjective improvement of sound quality, because besides the loudness reduction also the sound characteristic is changed. On the one hand, the spectrum is better equalised, and the typically annoying low frequency sound pressure is reduced. On the other hand, a reduced fluctuation strength can be observed, which contributes a lot to the improvement of the overall acoustic annoyance [6].

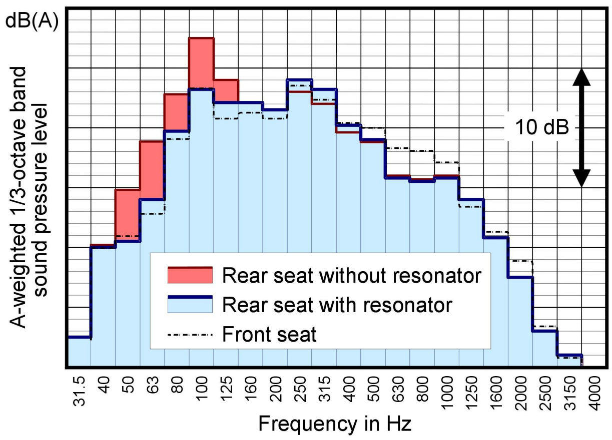


**Figure 7:** Effectiveness of a broad-band active resonator for sound pressure attenuation (constant speed on motorway).

## 5 - CONCLUSION

Highly compact active absorbers allow to effectively reduce the acoustic levels in the passenger compartment of a vehicle in a wide frequency range. The effectiveness for noise attenuation of huge Helmholtz resonators can be achieved in a wide frequency band. Two practical applications of active absorbers validate the theoretical fundamentals for the acoustic optimisation. The advantage to improve the sound quality in the passenger compartment of a vehicle was demonstrated.

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