

ANC in Automobiles

Eduard Reithmeler
Institut für Meß- und Regelungstechnik
Universität Hannover
Hannover, GERMANY

Ferit Küçükay
Institut für Fahrzeugtechnik
Universität Braunschweig
Braunschweig, GERMANY

George Leltmann
College of Engineering
University of California at
Berkeley, USA

Abstract:

Car driving usually produces a variety of noise phenomena mainly caused by rolling friction of wheels, wind flow around the car body, combustion and rotating parts within engine and gears. In the past strong efforts have been made to optimize the design of tires, body and frame as well as engine and suspension elements in order to attenuate noise generation. Aside from these „parameter optimizations“ recent developments demonstrate that closed loop control may play an important role in that field. In this work it will be shown how to actively suppress noise inside the car. The control scheme acts via sound generation through the car loudspeakers.

1 Problem Statement and General Modelling

Driving a car means also being influenced by a series of noise sources such as a running engine, the exhaust pipe, the power train, rolling wheels or the windflow around the car body (see Fig. 1). Significant efforts have already been made in order to employ techniques for passive noise attenuation by insulating materials, for instance. And recent discussions raise the question if there is a possibility to influence the sound pattern near the passengers ears by generating "anti noise" through some noise actuators. The noise actuators need to be located in the car and car loudspeakers may be used for that purpose.

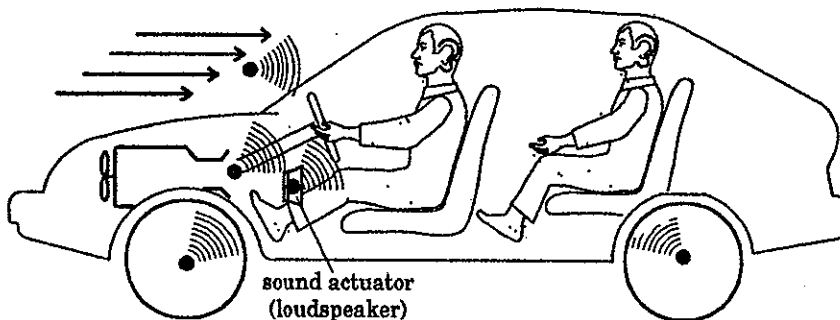


Fig. 1: Noise Sources in Cars

To get an answer to that question, we consider a general (i. e. nonlinear) transfer model according to Fig. 2. The car interior will be our plant. The power train, wheels and windflow cause the disturbances z_p , z_T and z_w . The applied voltage u_L at the car loudspeaker is considered to be the input variable. And the sound pressure p near the ears' location - recorded via a measurement microphone - is the output. Later on it will be reasonable to model the sound pressure p by some appropriate observer who gets his information through the Cars Area Network Data Bus (CAN-Bus). That is, an estimation \tilde{p} of p will be reconstructed by essential information such as car/engine speed ω_E / ω_f , gas pedal position α or gear ratio n .

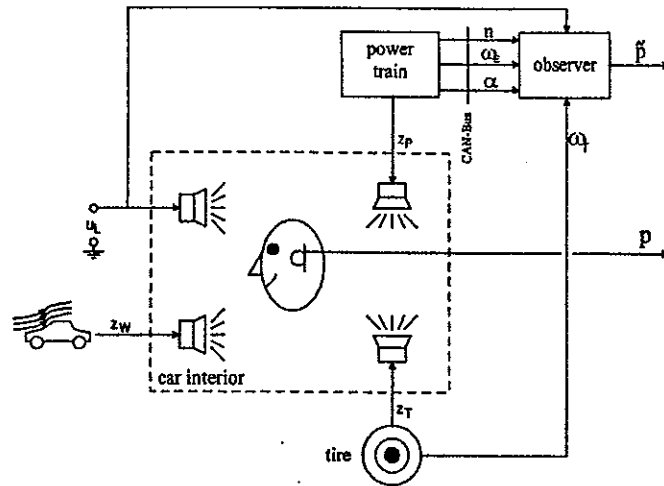


Fig. 2: General Transfer Model

Since the sound field inside the car has quite a complex structure and differs from car to car and with the number and position of objects and passengers in the car, we refrain from any mathematical modelling. Instead, we will seek to identify the plant by means of an experimental approach.

2 Acoustic Path and Plant Transfer Function

Our experimental set-up is a test car which is driven on a roller test stand inside a semi-anechoic chamber (see Fig. 3).



Fig. 3: Test Car

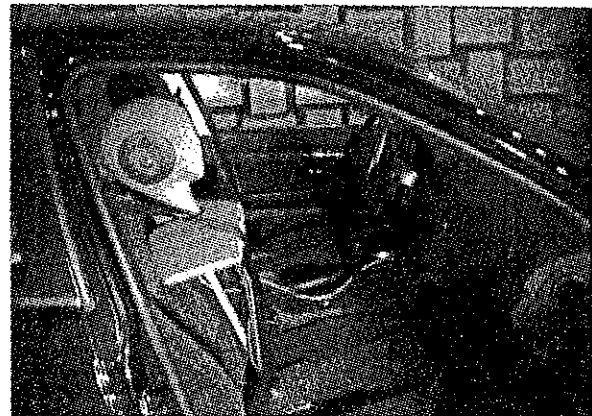


Fig. 4: Head Acoustical Equipment

In order to record the sound patterns near the passengers' ears we use appropriate head acoustical equipment (see Fig. 4) and a standard signal processing toolbox. To determine a BODE plot of the plant shown in Fig. 5,



Fig. 5: Plant for Recording BODE Plots

we feed the loudspeaker(s) with a sweeping sine (amplitude 1 V, frequency between 20 Hz and 20 kHz). The detected signal of the microphone is amplified and recorded with a sampling rate of 5 kHz. While recording the data, the car is operated in a stationary mode (i. e. constant speed), or operated with some characteristic noise pattern such as a change in gears, e. g. from gear 5 to gear 2 etc. Fig. 6 shows ten different measurements between 20 Hz and 1 kHz. The measurements were taken with slightly different microphone positions and different objects between loudspeaker and microphone such as legs or arms. As it turns out, sound pressure amplitudes show increasing sensitivity with increasing frequencies.

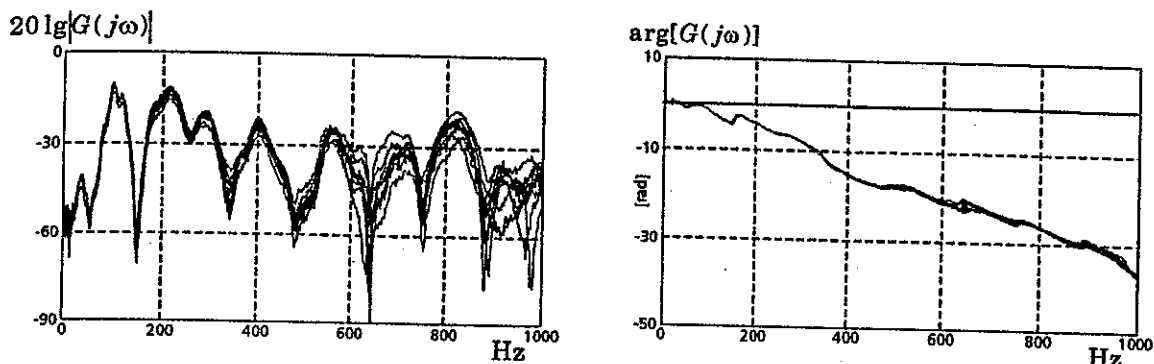


Fig. 6: BODE Plot of the Considered Plant

We refer to the stationary operated car or some gear change sound pattern as a "reference condition". In a neighborhood of that reference condition we model the plant as a linear SISO with transfer function $G(s)$ (see Fig. 7), loudspeaker input $U(s)$ and sound pressure output $P(s)$. All the disturbances z_p , z_r and z_w are collected in z near the ears' location and the corresponding sound pressure is expressed by $P_z(s)$ in the frequency domain. The linearly increasing phase shift marks the time delay T_t of different sound wave frequencies. In our case we obtain $T_t \approx 3$ ms, since the distance between ear and loudspeaker is around 1.1 m. The amplitude-frequency plot shows a series of resonances, mainly caused by the different modes of the loudspeaker diaphragm.

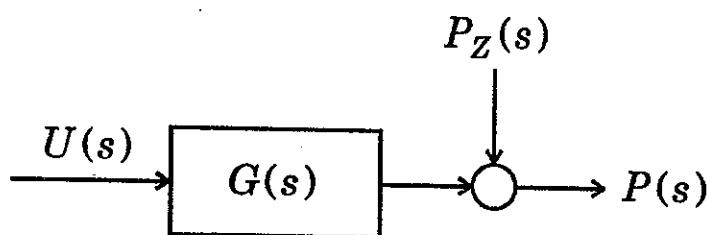


Fig. 7: Linearized Model Around Reference Condition

Usually the sound pattern near the ears' location need not be influenced over the whole frequency range. The main goal is to suppress primarily annoying sound parts. They appear inside certain intervalls $\Omega_i := [a_i, a_{i+1}]$; $i = 1, 2, \dots, N$. Inside those ranges, we replace the transfer function G by some elementary rational function

$$G_i(s) = \frac{Q_i(s)}{P_i(s)} = K_i \frac{(s - q_1^{(i)}) \cdot \dots \cdot (s - q_m^{(i)})}{(s - p_1^{(i)}) \cdot \dots \cdot (s - p_n^{(i)})}$$

Using a band-pass-filter $H_i(s)$ in addition leads to the approximation

$$\tilde{G}(s) = G_i(s)H_i(s)$$

of $G(s)$ within Ω_i (see Fig. 8). The still unknown parameters K_i , $p_k^{(i)}$ and $q_k^{(i)}$ may be determined via some least square procedure between ω_{\min} and ω_{\max} such as

$$\int_{\omega_{\min}}^{\omega_{\max}} |G(j\omega) - \tilde{G}(j\omega)|^2 d\omega \rightarrow \min!$$

where

$$\tilde{G}(s) := \sum_{i=1}^N G_i(s) \cdot H_i(s) e^{-jsT_i}$$

Finally, for control design purposes, we replace $G(s)$ by $\tilde{G}(s)$.

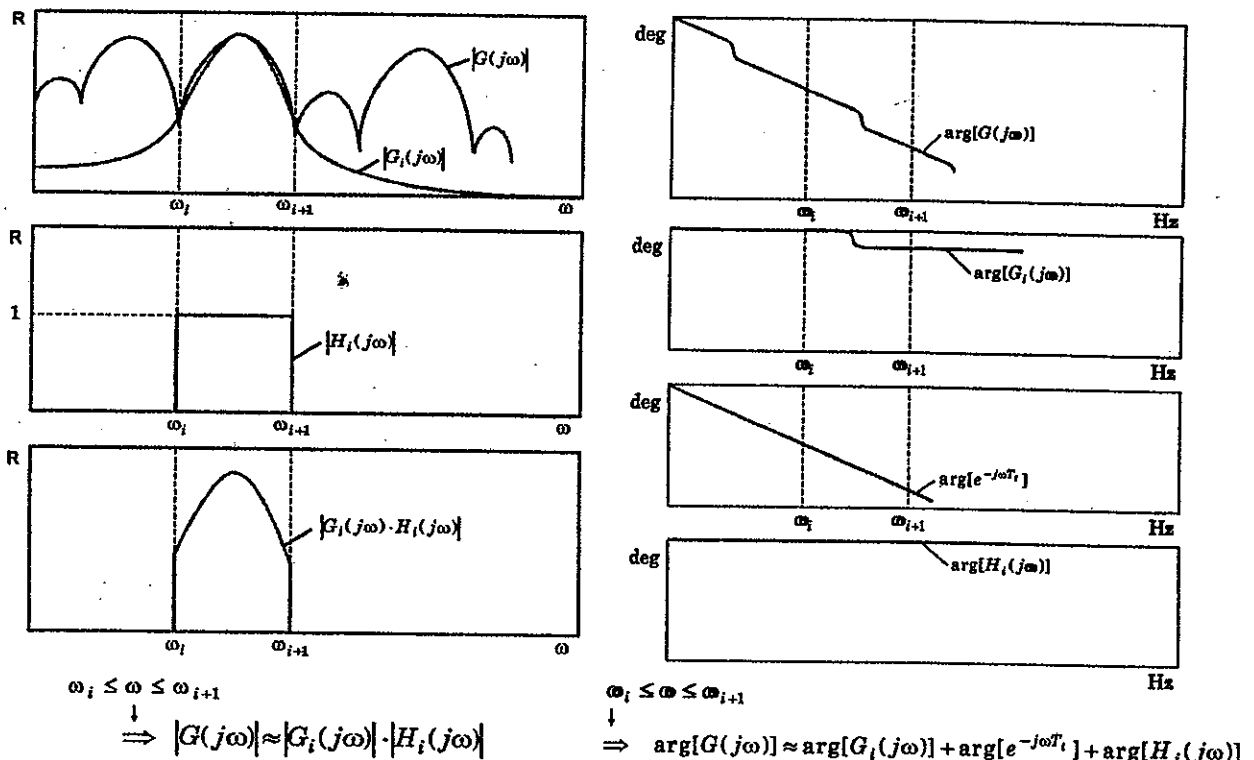


Fig. 8: Plant Approximation

Fig. 9 shows $\tilde{G}(s = j\omega)$ versus $G(s = j\omega)$ between 20 Hz and 900 Hz which is the range of interest.

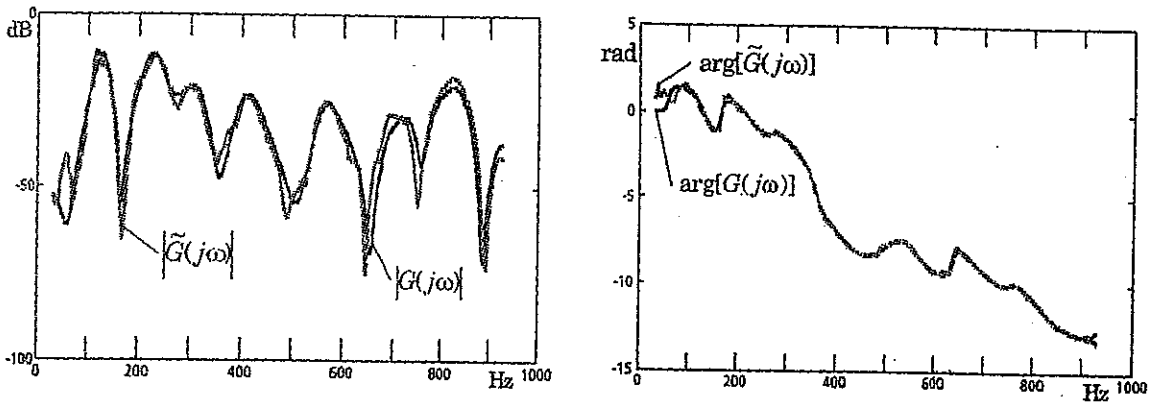


Fig. 9: $\tilde{G}(j\omega)$ versus $G(j\omega)$

3 Controller Design via Error Transfer Function

The plant approximation $\tilde{G}(s)$ of $G(s)$ is required for our controller design. Based on a single closed loop according to Fig. 10, we use an inexpensive micro controller board with CAN-Bus connector. The CAN-Bus supplies mainly the information about the current operating conditions of the car, which determines our reference noise pattern for linearization. $P_D(s)$ is some desired noise pattern. $G_c(s)$ represents the controller which uses the deviation $\Delta P(s)$ between $P(s)$ and $P_D(s)$ as input and supplies the appropriate voltage $\Delta U(s)$ to generate the anti noise-pattern. $\bar{U}(s)$ arises in addition if the radio or CD-Player is on (see Fig. 10).

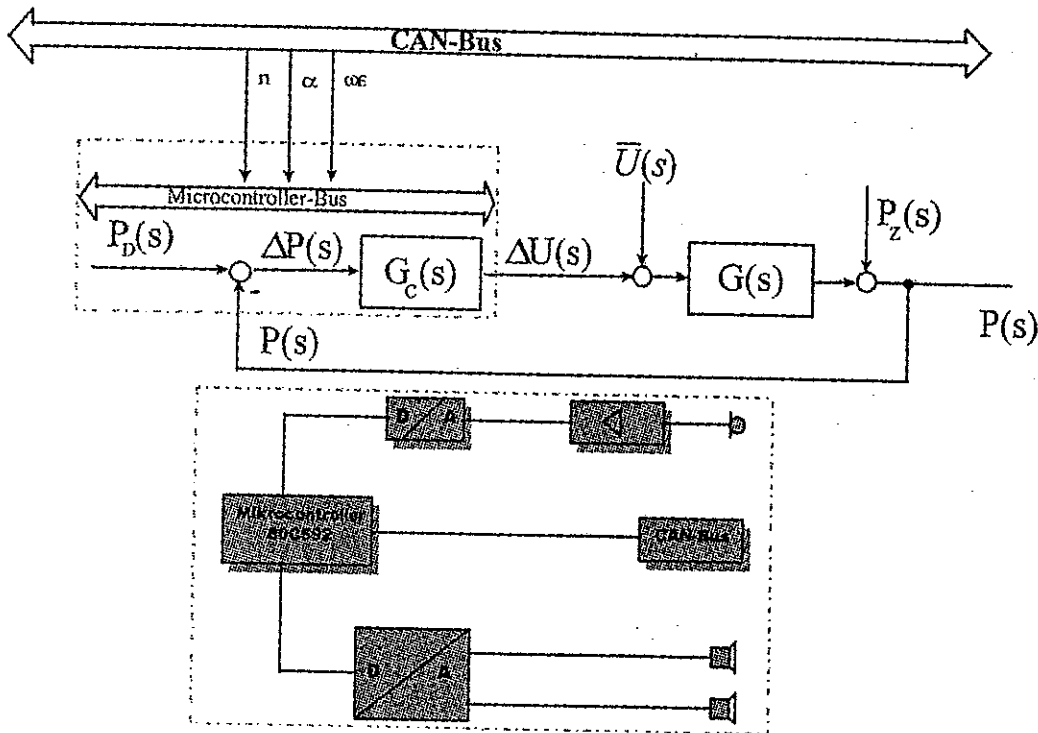


Fig. 10: Closed Loop Control

In case $P_z(s)$ is entirely unknown, one may use the error-transfer-function F according to Fig. 11 in order to design $G_c(s)$.

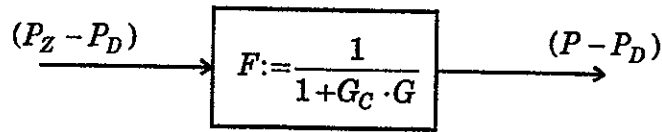


Fig. 11: Error-Transfer-Model

Choosing

$$G_C(s) := \frac{-K}{\sum_{i=1}^N G_i(s) H_i(s)} \quad \text{where} \quad K \geq 2$$

yields approximately

$$F(s) \approx \frac{1}{1 - K e^{-sT_i}}$$

This transfer function will always tend to suppress any deviation ΔP because

$$\frac{|P(j\omega) - P_D(j\omega)|}{|P_z(j\omega) - P_D(j\omega)|} \approx |\tilde{F}(j\omega, K)| \leq \frac{1}{|K - 1|}$$

The amplification is limited by the technical specifications of the loudspeaker. And if $2 \leq K \leq K_{\max}$ holds, we get the best results for $K = K_{\max}$. In our case we obtain a reduction of a deviation ΔP from 10 dB to 13 dB (see Fig. 12).

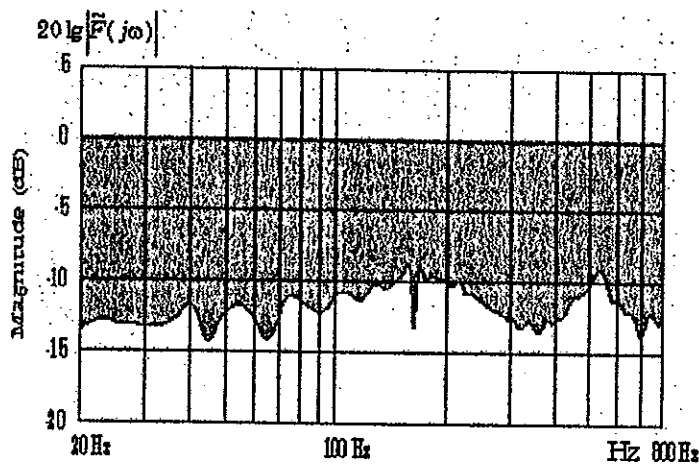


Fig. 12: Suppression of Deviations ΔP

Of course, if $P_z(s)$ is better known, say some characteristic engine noise, G_c may be designed differently in order to take advantage of additional information. This will be done in future work of the authors.

4 Literature

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