10th Workshop on Dynamics & Control

Complex Dynamical Systems With Incomplete Information

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ANC IN AUTOMATIC TRANSMISSION AUTOMOBILES TO IMPROVE DRIVING COMFORT

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The drivability and ride comfort of automatic transmission cars, particularly while shifting, is influenced by the change in the interior noise due to a change in engine speed. Tests show that this is also true even if the longitudinal vibration behaviour of the vehicle while shifting is rated as comfortable. The main objective of our work is to improve the subjective sound impression while shifting. This is realised by influencing the car interio sound spectra via the car loudspeakers.

Therefore several steps will be taken into account: The first part is to determine the transfer function of the acoustic path between the engine compartment and the interior space. In a second part, head acoustical equipment is used to record significant driving situations in order to set up actual reference patterns. The third part is concerned with psychoacoustical methods and sound-engineering techniques to design desired or nominal sound patterns, respectively, compared to actually occuring gear shift sound patterns. Finally, the last part is focused on developing a active noise control (ANC) concept which reduces differences between actual and desired sound-spectra. The authors discuss the current state of this project and first results.

1 Problem Statement

The ride comfort and drivability of automatic transmission cars are significantly influenced by the shift quality of the transmission. In position D of the selector lever (see Fig. 1), the gears of an automatic transmission are shifted depending on vehicle speed v and gas pedal position α. The corresponding shift curves are implemented in the transmission controller.

The controller has basically the following tasks:

- Determination of shift point positions (α, v),
- Design of shift processes (gear shift transition),
- Error diagnostics.

The transmission controller communicates via the CAN-Bus with other control units such as engine control (EC), anti-lock brake system (ABS) and automatic stability control (ASC).

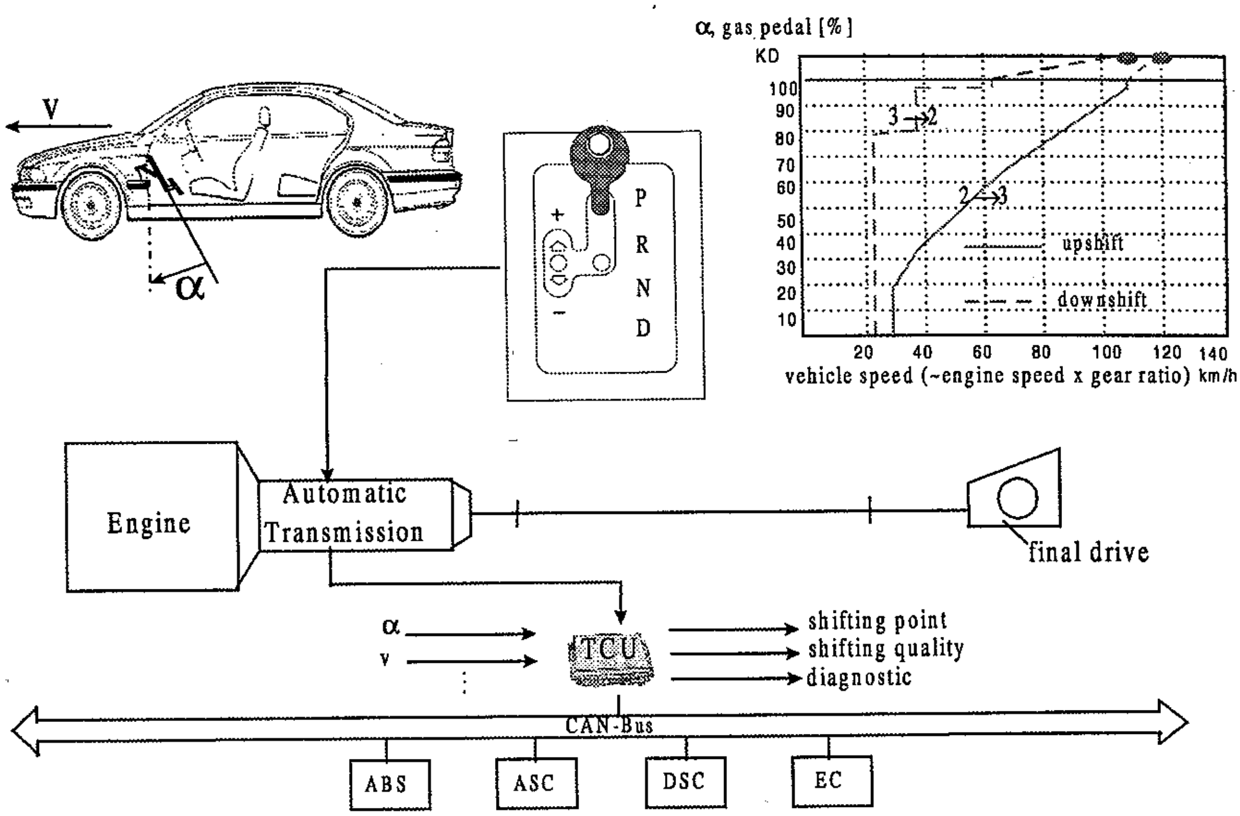

Fig. 1: Automatic Transmission Automobil

Various parameters affect the subjective impression of the shift comfort. The most importan parameter (see Fig. 2) is the jerk depending on the change in the longitudinal acceleration

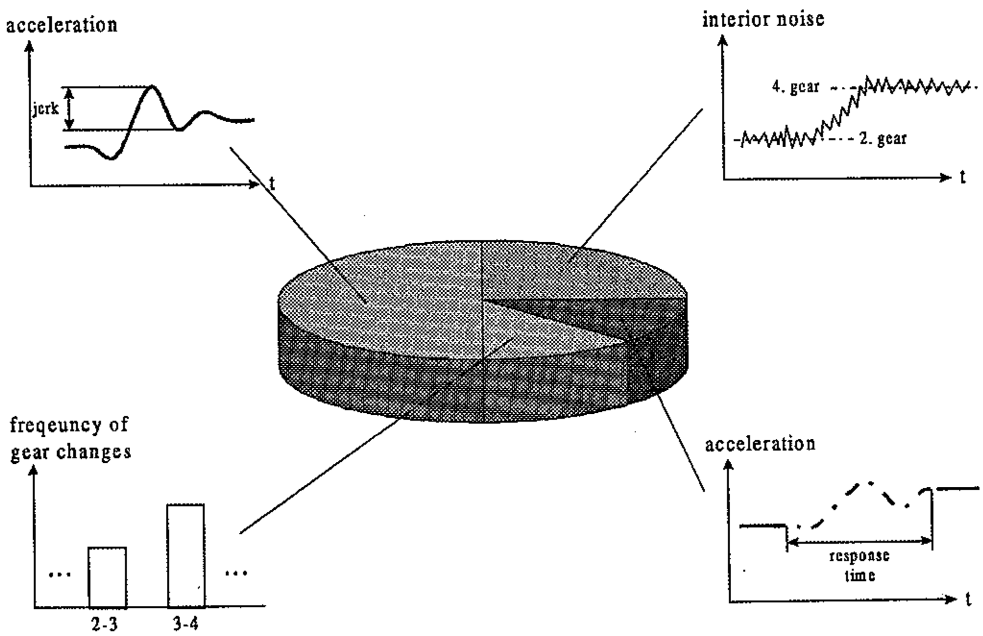

Fig. 2: Parameters with an Influence on Gear Shift Comfort

of the car while shifting. The second most important factor is the change in the interior noise level as a result of a change in engine speed, again, while shifting. Further phenomena which influence the shift comfort are the response time during gear shift transitions and the frequency of gear shifts.

In this work, noise control is considered in order to improve shift comfort. In accordance with several operating conditions, various gear shifts depending on gas pedal positions α and engine speed v will be considered (see Fig. 3).

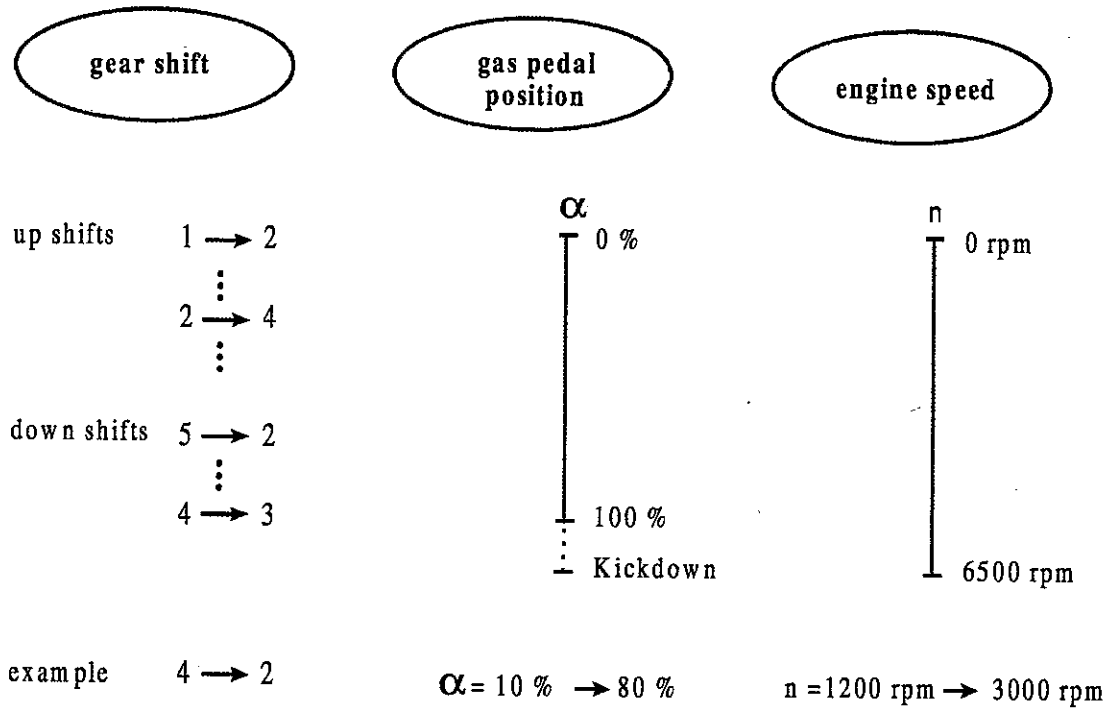

Fig. 3: Gear Shift Parameter Set

For example, if we consider a vehicle speed of 60 km/h in fourth gear, the engine runs at a speed o 1200 rpm and the gas pedal position α is at 10 % of its maximum value. Increasing the gas pedal position to 80 % (i. e. the driver accelerates the car) a down-shift from fourth gear to second gear takes place and the engine speed increases to a value of 3000 rpm. The corresponding sound pressure near the drivers ear while shifting from 4 to 2 is shown in the frequency domain as well as in the time domain on the left hand side of Fig. 4. The right hand side shows the desired sound pattern. The desired sound pattern results from the actual sound pattern (left hand side) by editing tha pattern in an audio-lab until it is rated subjectively as more pleasant than the actual pattern.

The main objective of an active noise control system (ANC) located within the car is to cancel ou the difference between actual and desired sound pressure.

In order to design such a anti-noise control concept, we consider a transfer model according to Fig. 5.

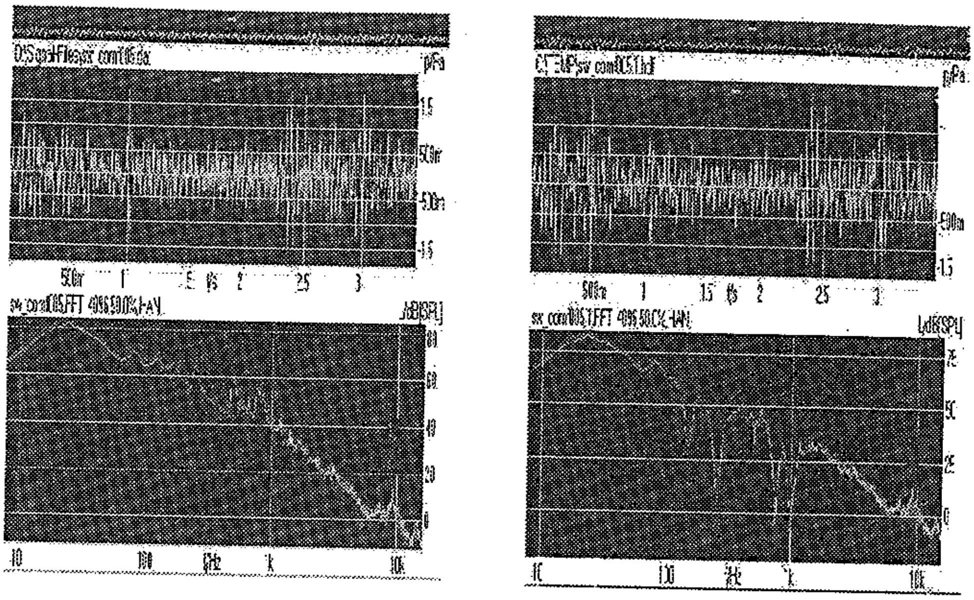

Fig. 4: Actual and Desired Sound Pattern

The car interior is taken as the 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. The sound pressure p near the ears' location - recorded via a measurement microphone - is the output. In general, this transfer model will be nonlinear, that is, the variables u_L, z_W, z_T, z_P and p constitute some nonlinear relationship with each other. However, we may assume that the deviations Δu_L, Δz_W, Δz_T, Δz_P and Δp between the actual situation (given b u_L, z_W, z_T, z_P and p) and some "compatible" reference condition $\bar{u}_L$, $\bar{z}_W$, $\bar{z}_T$, $\bar{z}_P$ and $\bar{p}$ ($\bar{p}$ may be the desired noise pattern around the ears) are small. In that case, the dependencies between u_L, z_W, z_T, z_P and p can be considered as linear. And, in case of slow changes in our reference condition with respect to time (say a downshift from fourth to second gear), the linearity may be even considered as timeinvariant.

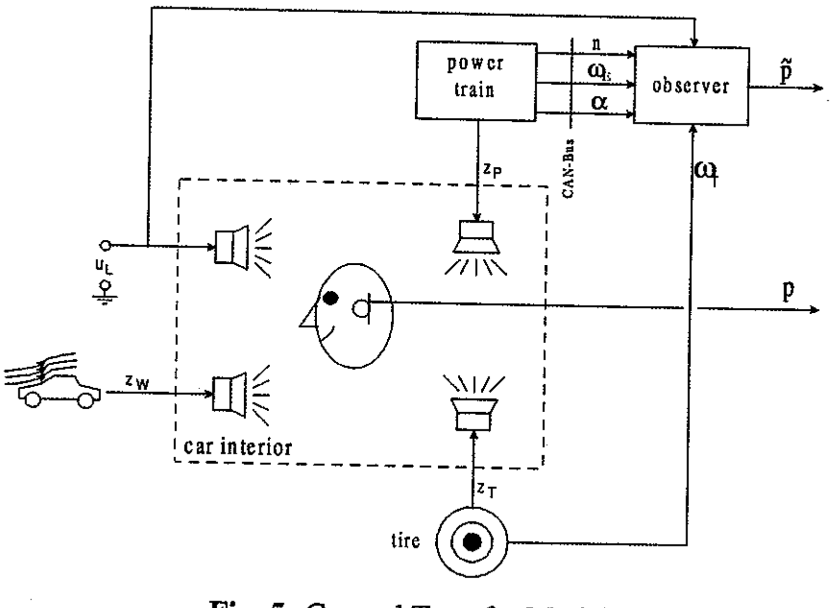

Fig. 5: General Transfer Model

Later on it will be reasonable to model the sound pressure p by some appropriate observer who gets his information through the Car's 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_T gas pedal position α and gear ratio n. 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. 6).

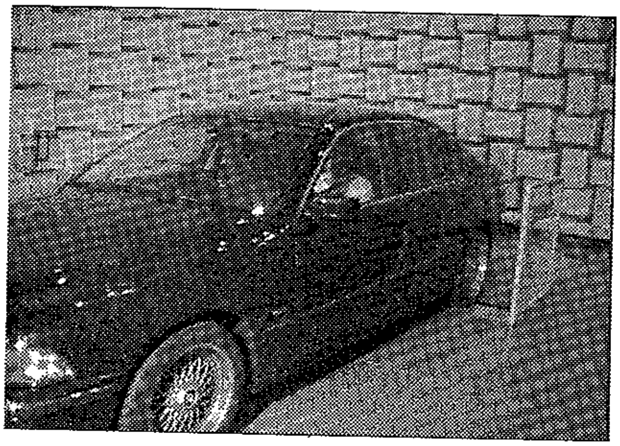

Fig. 6: Test Car

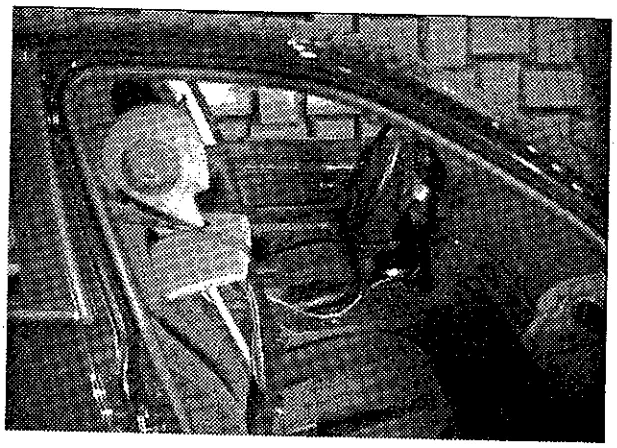

Fig. 7: Head Acoustical Equipment

In order to record the sound patterns near the passengers' ears we use appropriate head acoustical equipment (see Fig. 7) and a standard signal processing toolbox. We refer to some selected 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. 8), loudspeaker input $U(s) = \mathrm{L}\,[\Delta u_L]$ and sound pressure output $P(s) = \mathrm{L}\,[\Delta p]$. All the disturbances, Δz_P, Δz_T and Δz_W, are collected in a single disturbance Δz near the ears, and the corresponding sound pressure is expressed by $P_Z(s) = \mathrm{L}\,[\Delta z]$ in the frequency domain.

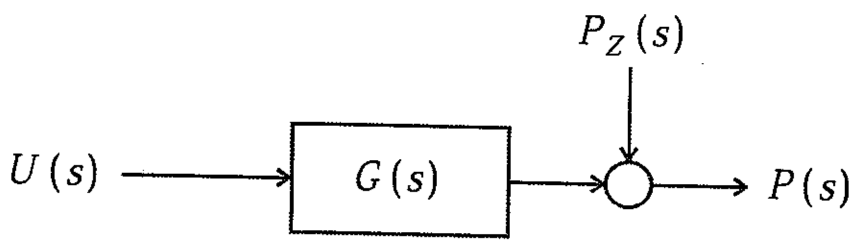

Fig. 8: Linearized Model Around Reference Condition

To determine a BODE plot of the linear plant shown in Fig. 9,

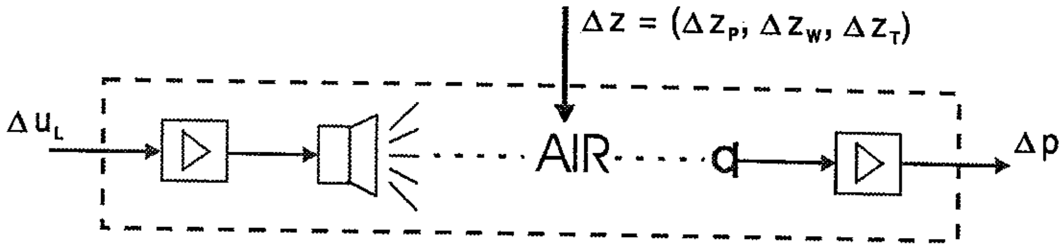

Fig. 9: Plant for Recording BODE Plots

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 with some characteristic noise pattern such as a change in gears, e. g. from gear 4 to gear 2, etc. Fig. 10 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 one can see, sound pressure amplitudes show increasing sensitivity with increasing frequencies. 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 plo shows a series of resonances, mainly caused by the different modes of the loudspeaker diaphragm.

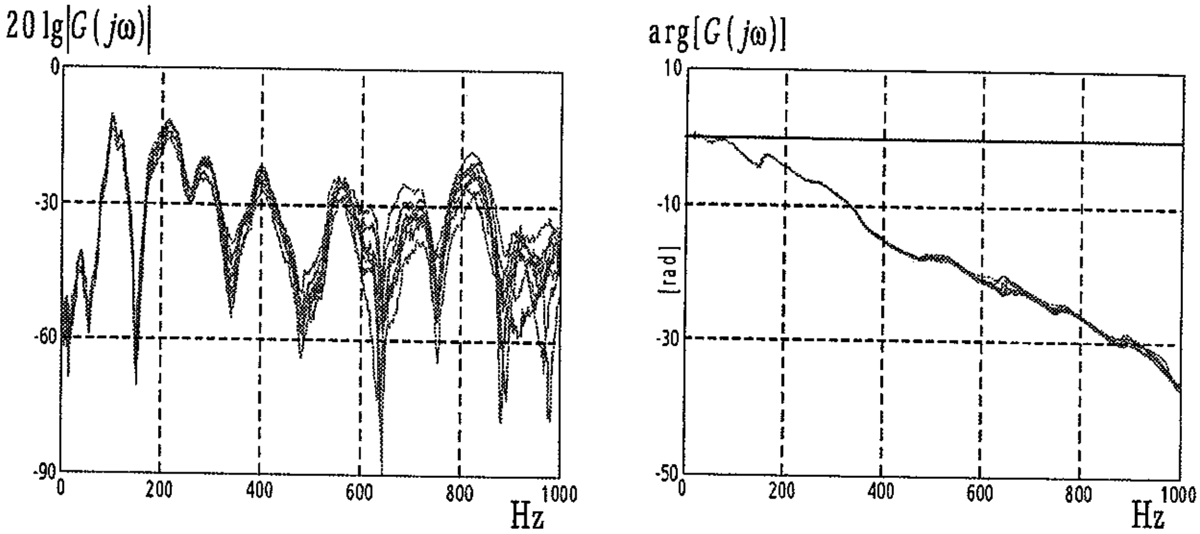

Fig. 10: BODE Plot of the Considered Plant

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

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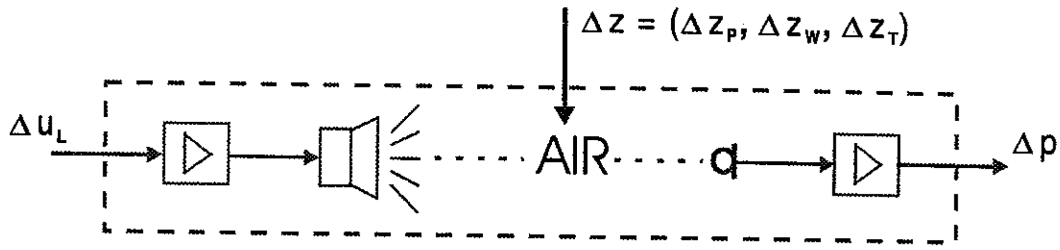

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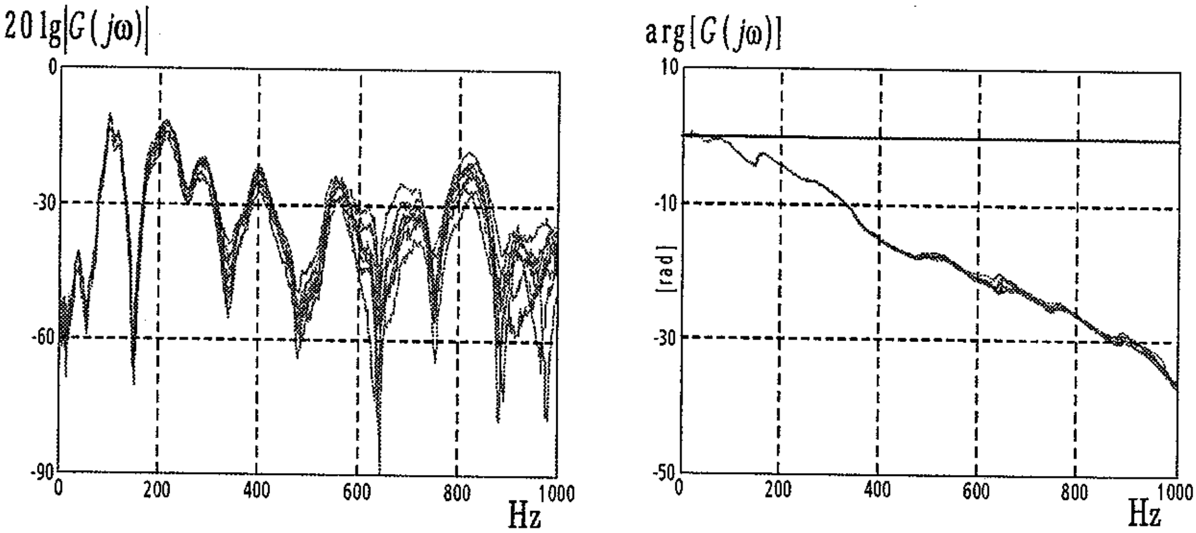

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$$G_i(s) = \frac{Q_i(s)}{P_i(s)} = K_i \frac{(s-q_1^{(i)})\cdot\ldots(s-q_m^{(i)})}{(s-p_1^{(i)})\cdot\ldots(s-p_n^{(i)})}$$

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. 11). The still unknown parameters K_i, $p_k^{(i)}$ and $q_e^{(i)}$ may be determined via a least square procedure between $\omega_{\min}$ and $\omega_{\max}$ such as

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

where

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

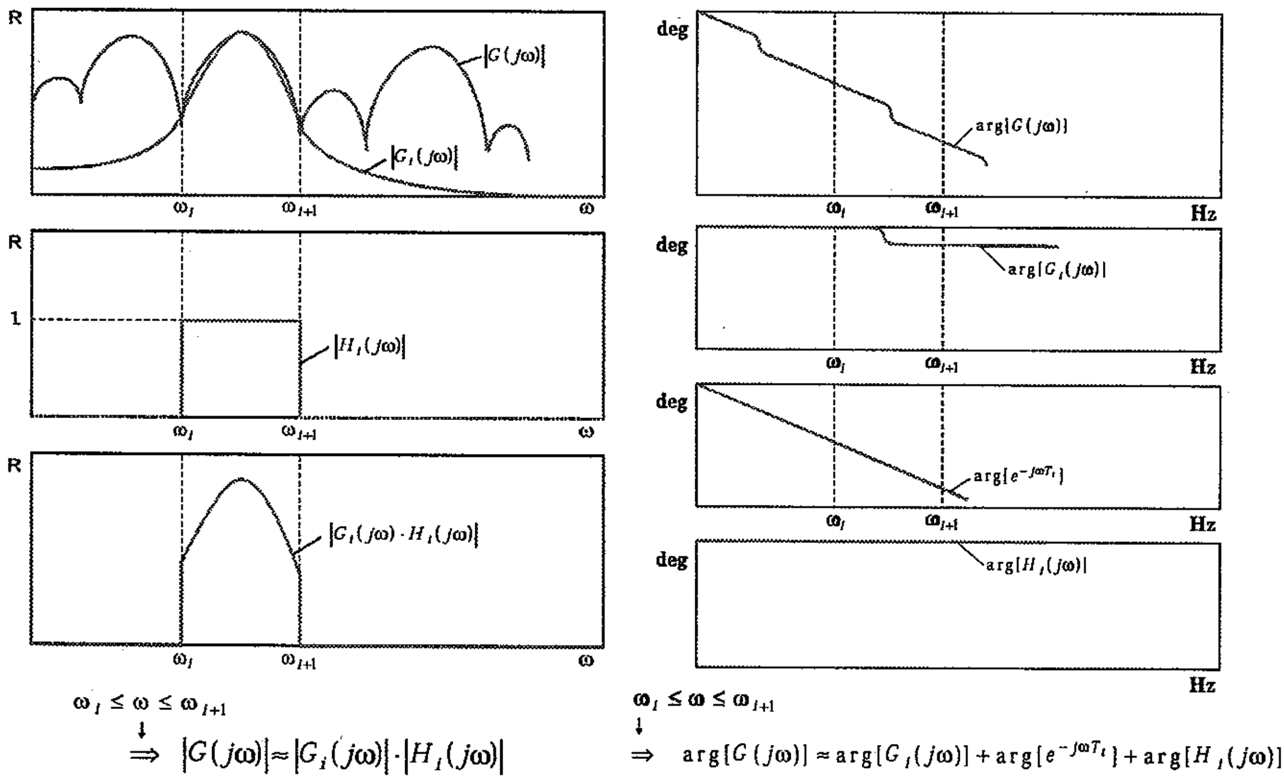

Fig. 11: Plant Approximati

Finally, for control design purposes, we replace $G(s)$ by $\tilde{G}(s)$. Fig. 12 shows $\tilde{G}(s=j\omega)$ and $G(s=j\omega)$ between 20 Hz and 900 Hz which is the range of interest.

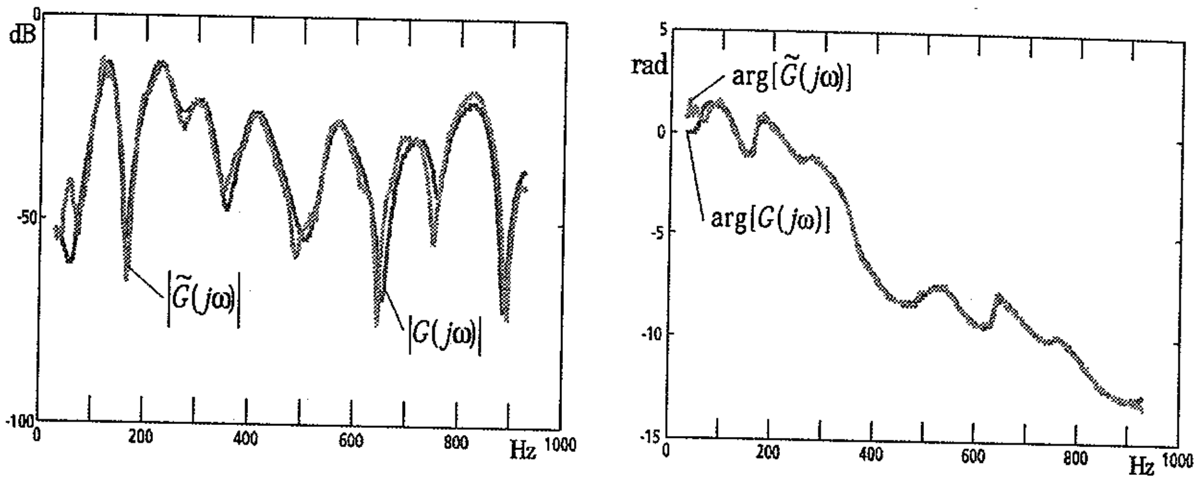

Fig. 12: Comparison of $\tilde{G}(j\omega)$ and $G(j\omega)$

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. 13, 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 or desired noise pattern $P_D(s)$ respectively. That is, if the current operating condition describes a shift from the i-th to the k-th gear, the corresponding desired

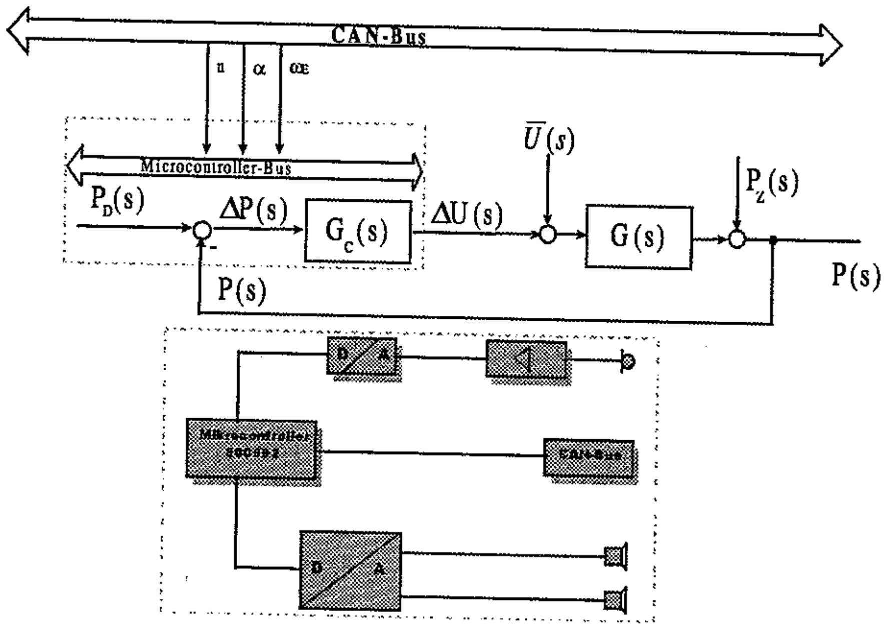

Fig. 13: Closed Loop Control

sound pattern $P_D(s) = P_{ik}(s)$ will be taken say from a look up table (LUT) which contains all desired sound patterns $P_{ik}(s)$. The LUT may be stored on a board memory.

$G_c(s)$ represents the controller which uses the deviation $\Delta P(s)$ betwee $P(s)$ and $P_D(s)$ as input and supplies the appropriate voltage $\Delta U(s)$ to generate the anti-noise pattern. $\overline{U}(s)$ arises in addition if the radio or CD-Player is on (see Fig. 13).

In case $P_Z(s)$ is entirely unknown, one may use the error-transfer-function F according to Fig. 14 in order to design $G_c(s)$.

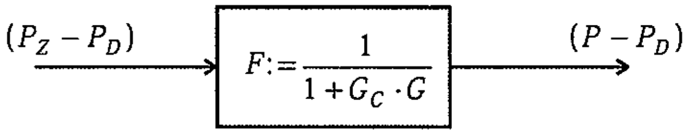

Fig. 14: Error-Transfer-Mode

Choosing

$$G_C(s) := \frac{-K}{\sum_{i=1}^{N} G_i(s)\, H_i(s)}$$

yields approximatel

$$F(s) \approx \frac{1}{1 - K\, e^{-sT_t}}\,.$$

For $K \geq 2$ 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. 15).

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.

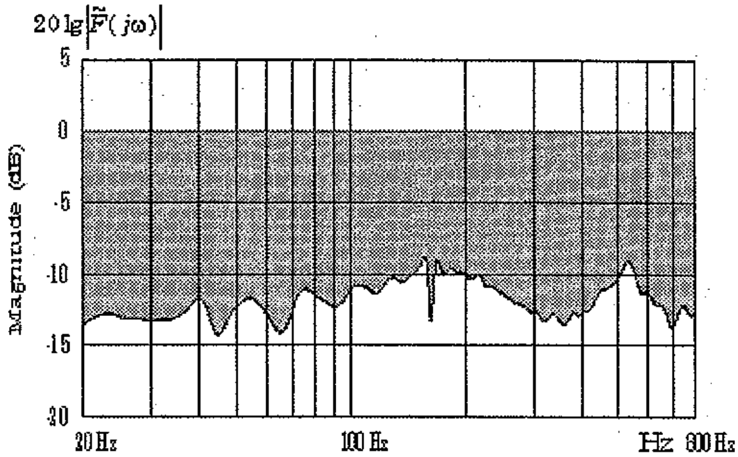

Fig. 15: Suppression of Deviations ΔP

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