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Reducing rear axle gear whine noise inside a car by influencing the structure-borne sound transfer path using structurally integrated piezo-actuators

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Abstract

The gear whine noise of rear axles is a well-known acoustic phenomenon especially for rear-wheel and four-wheel drive vehicles. The acoustical optimization of this problem leads to the major goal conflict: improvement of the vibrational isolation of the rear axle versus excellent driving dynamics of the vehicle. A possible solution for this issue is in focus of a research project at Fraunhofer Institute for Machine Tools and Forming Technology IWU in cooperation with Mercedes AMG GmbH. The aim is to reduce the noise contributions of the rear axle inside the car using an active vibration control system on the transfer path from the axle into the vehicle based on structurally integrated piezo-actuators. This paper describes the development process of the active vibration control system. At first, a FEM-simulation model has been created which is able to represent the operational deflection shapes of the rear axle assembly in critical operating points. In a next step, two simulation approaches have been applied in order to identify promising excitation points for an actuator application: the structural intensity analysis and a sensitivity analysis of transfer functions of the rear axle structure. Furthermore, the geometry and material properties of the piezo-actuator have been implemented in the simulation to calculate the force reduction on the coupling points to the vehicle body. In addition, a rear axle test bench has been set up in order to reproduce the operational deflection shapes and validate the simulation results.

Keywords: Active vibration control, gear noise, vehicle acoustics, NVH

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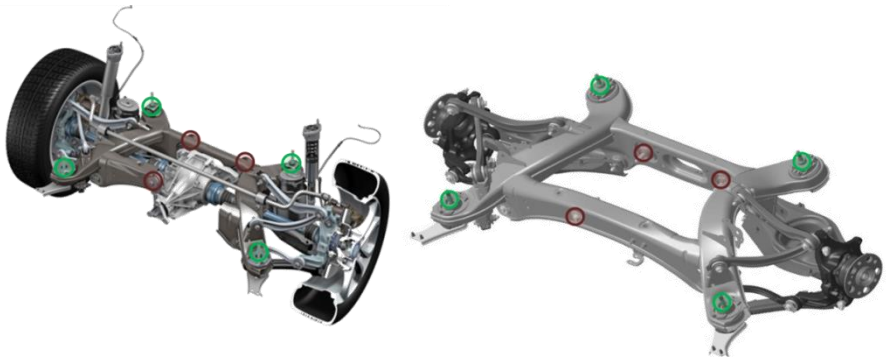
1 Introduction

In general, the gear mesh is the main noise source in transmissions. Due to the changing overall stiffness of the gear pair, a so-called parameter excitation occurs which causes dynamic forces in the gear mesh. Furthermore, there are geometric deviations due to the production process and designed flank modification, which additionally effect a periodical displacement excitation [1].



Source: Daimler AG

Figure 1: Example rear axle assembly (red markers: mounts of differential gearbox, green markers: rear sub frame mounts).

The resulting dynamic forces are transferred over the shaft-bearing system into the housing structure of a gearbox. Beside a direct sound radiation of the housing due to surface velocities, structure-borne energy is transferred over the bearing system into the vehicle body. In terms of rear axles, the excitation is higher compared to typical transmission gearboxes due to the high torque which needs to be transmitted through the complete drive train in combination with the bevel gear stage. Basically, rear axles of passenger cars are decoupled twice from the vehicle body using a rear sub frame (Figure 1). The elastomer elements between axle and frame are typically arranged as a three-point-bearing and the sub frame is coupled to the vehicle body using four elastomer mounts. In terms of a force excitation from the gear mesh, the rear axle is forced to oscillate in its flexible mounting. The resulting rigid-body motions and sensitive frequency ranges depend on the distribution of the dynamic mass and the boundary mounting conditions. Typical operational deflection shapes are pitching, rotating and wobbling. In all cases, a significant amount of structure-borne energy is transmitted through the rear axle mounting in the sub frame. Afterwards, the structure-borne sound is transferred through the frame and its elastomer mounts into the vehicle body. Structural modes of the frame can be excited within certain frequency ranges as well and additionally transferred into the vehicle. In the passenger compartment, gear whine is perceived as a load and drive speed dependent

tonal noise. Because of the narrowband noise characteristics, it is an annoying and unpleasant sound, which is even noticeable at low sound pressure levels. The frequencies of gear whine result from the rotational speed of the driveshaft multiplied with the number of teeth of the bevel wheel which is called the first gear mesh order. The first order is mostly the dominant one from acoustical point of view. Sometimes also its harmonics like 2nd, 3rd and 4th order contribute to the overall sound pressure level [1]. In most cases, the whining noise is not present in all operating modes of the vehicle. Instead, there are often just a few speed ranges where the gear noise is dominating the interior noise.

2 Problem identification

For analyzing the gear whine phenomenon, measurements have been carried out on different test vehicles under different driving conditions. Beside the measurement of sound pressure level at different positions in the passenger compartment, acceleration signals in front of and after each mounting position in three dimensions as well as the rotational speed of the drive shaft have been recorded. From comparison of the overall sound pressure level and the extracted 1st and 2nd gear mesh orders at different rotational speeds, the critical operation modes from an acoustical point of view could be identified.

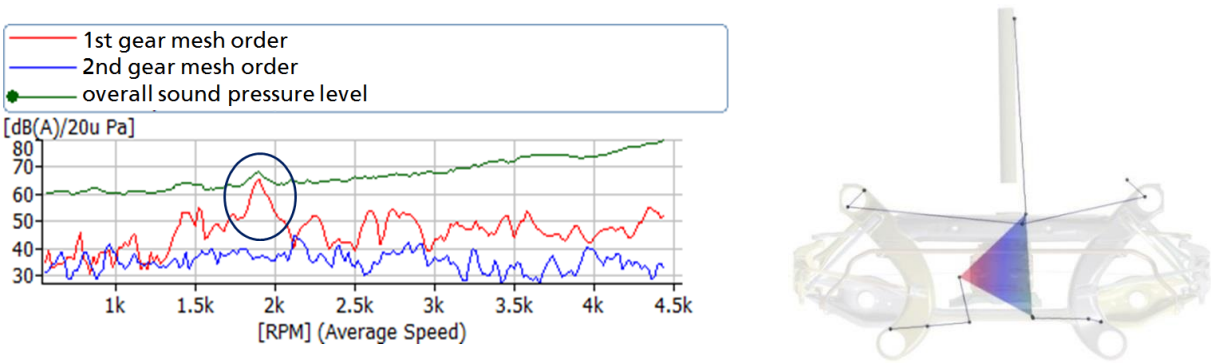


Figure 2: left - Overall sound pressure level at driver's ear and 1st and 2nd gear mesh order over rotational speed of the drive shaft; right - corresponding rigid-body motion of the rear axle

For example, Figure 2 shows a run-up measurement of a vehicle under partly load conditions. Between 1800 and 1900 rpm of the drive shaft, the 1st gear mesh order is dominating the overall level which can be perceived as a distinct gear whine noise inside the car. Two critical driving conditions could be found in this way: approx. 80 $\mathrm{km/h}$ with a corresponding gear mesh frequency of 540 Hz and approx. 120 $\mathrm{km/h}$ with a corresponding gear mesh frequency of 730 Hz. In a further step, the responsible rigid-body motion of the rear axle has been calculated and animated based on measured acceleration signals using a simple structure model. The relevant operational vibrations are wobbling and rotating modes of the rear axle body in its mounting points. On the right side of Figure 2 an example rigid-body motion of the rear axle is shown. Major deformations can be detected at the left rear mount and front mount of the gear housing, which is due to a rotation around the rear right mount. It is obvious, that depending on the rigid-body motion of the differential gear housing the structure-borne energy transfer into the sub frame can significantly vary between the different mounting positions and spatial directions.

The further analysis and modifications have been exemplarily carried out for the operational deflection shape at 120 km/h (730 Hz).

3 Solutions for noise reduction of rear axles

There are several primary and secondary measures to lower gear noise inside a car, which in most cases influence other functionalities and lead to major goal conflicts. The most efficient way is lowering the gear mesh excitation (primary measure). A significant reduction of the gear excitation often causes a worse durability behavior in combination with lower system effectiveness and higher production costs (due to higher production accuracy in terms of micro geometry and surface quality). A second way for gear noise reduction is the optimization of the transfer paths into the passenger compartment (secondary measure). The three mounts of the differential gearbox seem to be most suitable since they are the direct connection to the rear sub frame. Lowering the stiffness would improve the vibration isolation significantly. But the vibrational behavior of the whole drive train (engine, gearbox, drive shaft, rear axle) would be negatively influenced since torsional eigenmodes are shifted to lower frequencies, where the main torsional engine vibrations occur. The ride comfort of the car will be worse. The third possibility for gear noise reduction is the improvement of the isolation of the rear sub frame mounts by lowering the stiffness. This will also affect the driving dynamics significantly because movements of the rear axle relatively to the car body will be increased. In terms of high performance vehicles, the driving dynamics is a major selling point. In addition, the rear sub frame mounts also influence road excited vibrations as well as tire-wheel-excited vibrations and their transfer into the vehicle body [2]. For that reason, the design of all rear axle mounts leads to a major goal conflict in the acoustical development process which results in a need of alternative solutions e.g. active vibration control system (AVC). An active system for vibration reduction is based on the idea to compensate disturbing vibration by inducing an additional vibration with opposite phase (180° phase shift). For generation of the compensating vibration, different actuator types can be used depending on the required force level, acceleration level and the interesting frequency range. Actuators based on piezo ceramics have been used for several practical applications yet, in order to reduce unrequested vibration. The main advantage is the high force level which can be generated with less space and weight compared to other actuator types. The achievable displacements are comparable low. For AVC in the middle and high frequency range, where in most cases force excitations are needed, piezo-actuators in form of patches or stacks are most suitable. In the last years, various active fiber composite actuators based on piezoelectric ceramic materials have been developed and are available at the market as commercial products [3], [4], [5]. A connection to the structure is usually established by gluing the actuator on the surface. Within the collaborative research center Transregio 39 / PT-PIESA an alternative approach has been developed, which is the direct integration of the actuator in a metal structure e.g. between two metal sheets [6]. The developed production process makes it possible to manufacture complex, three-dimensionally shaped piezo-metal-composites with structurally integrated actuator/sensor functionality. The main advantages of this technique are: the actuator can be placed directly in the force flow, the actuator counteracts the material deformation directly in the material and the actuator modules are protected against external influences (moisture, dirt, salt etc.).

The focus of the project described within this paper is to influence the structure-borne energy flow over the rear sub frame using an AVC-system. Therefore, actuators are placed on the frame, where the energy flow can be influenced in a best way. This approach does not influence other functionalities of the sub frame mounts. Furthermore, the required force level of the actuator is quite low which makes the system more efficient. In order to find the right actuator position, several simulation methods have been implemented and validated on a rear axle test bench.

4 Simulation model for rear axle noise

A FEM simulation model has been set up to virtually reproduce the dynamic behavior of the rear axle in order to find the optimal actuator position for an AVC-system. The model consists of the following rear axle components which influence the rigid-body motion of the axle body and the structure-borne energy transfer in the vehicle: rear sub frame, gear housing, drive shaft and all mountings on axle and vehicle side. For simplification and reduction of calculation time, the excitation from the gear mesh is not included in the model. Instead, the accelerations of the gearbox which have been measured in the full vehicle tests have been applied to the model in order to excite the operational deflection shape. The rear sub frame is connected to the vehicle body by four elastomeric mounts which are modelled using a measured damping ratio and dynamic stiffness. They are connected to nodes which are fixed in all six degrees of freedom which represents a defined boundary condition “infinitely rigid”. That´s why, the calculated reaction forces on the four mounting points are so-called blocked forces and will be higher than the real operating forces in the vehicle. Furthermore, the frequency dependent reaction of the vehicle body due to the exciting forces is not included in the simulation model. This model setup is similar to the laboratory test rig which is described later in this paper. The created model (Figure 3, left side) gives a sufficiently accurate simulation of the operational vibrations (Figure 3, right side) as well as the operational force input through the rear sub frame mountings. The model which includes all relevant transfer paths from the gearbox to the contact points of the vehicle body has been used to examine potentials for lowering gear noise contributions in the passenger compartment.

Figure 3: left - FE simulation model of the rear axle; right - simulated operational deflection shape at 730 Hz

5 Simulation methods to investigate actuator positions and noise reduction potential

To find an ideal actuator position on the structure of the rear sub frame with the highest noise reduction potential, two simulation methods have been implemented: a calculation of structural intensity and a sensitivity analysis regarding relevant structure-borne transfer paths.

5.1 Structural intensity

The structural intensity (STI) was used to identify the position on the rear sub frame where the highest energy transfer is located. It is calculated with the complex velocity of each FE element in each direction ($\underline{v}_{x,y,z}$) with regard to the resulting complex tensile and shear stresses in all directions on the element borders ($\underline{\sigma}_{x,y,z}, \underline{\tau}_{x,y,z}$) [7]:

$$\underline{\vec{I}}_S = -\underline{S} \cdot \underline{\vec{v}} = \begin{bmatrix} \underline{\sigma}_{xx} & \underline{\tau}_{xy} & \underline{\tau}_{xz} \\ \underline{\tau}_{yx} & \sigma_{yy} & \underline{\tau}_{yz} \\ \underline{\tau}_{zx} & \underline{\tau}_{zy} & \underline{\sigma}_{zz} \end{bmatrix} \cdot \begin{bmatrix} \underline{v}_x \\ \underline{v}_y \\ \underline{v}_z \end{bmatrix} \tag{1}$$

The real part of the STI gives a qualitative visualization of the active power in a structure when it is excited by a dynamic force. Thereby, high energy concentrations become apparent and an AVC-system can be applied to influence the energy flow through a structure at a single point. The identified position of the highest energy concentration from the STI analysis is not necessarily the best position for influencing the structure-borne sound flow through all mounting points of the rear sub frame into the vehicle body.

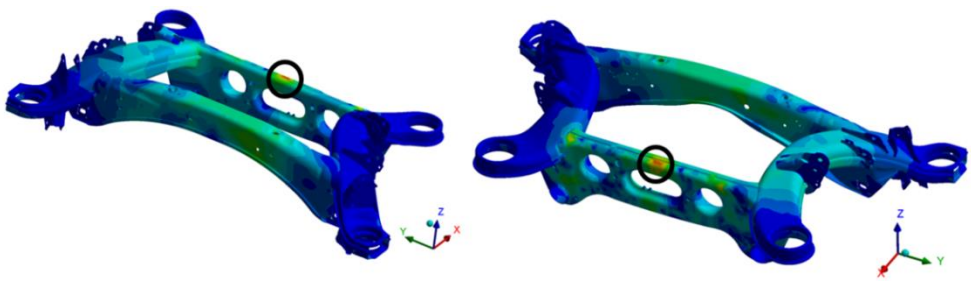


Figure 4: Structural intensity on rear sub frame with the position of highest energy concentration marked

Nevertheless, this method indicates which parts of a structure have the highest or lowest energy concentrations and gives useful information for the actuator positioning. Figure 4 shows the STI at the chosen frequency of 730 Hz with the highest structure-borne sound energy on the rear cross bar between the two rear mounting points of the housing. This is the first position which has been chosen for further tests on the rear axle test bench. Generally, it becomes apparent, that the structure-borne sound flow is mainly concentrated on the front and rear crossbar. Furthermore, it can be stated that the structure-borne power transfer on the right side of the

frame is higher compared to the left side which is due to the unbalance arrangement of the gearbox mounting positions.

5.2 Sensitivity analysis

The second method which has been implemented is based on an analysis of transfer functions (TF) from possible excitation points on the frame structure to the elastic mountings of the gear housing and the fixing points of the frame at the vehicle body. High sensitivity in the TFs corresponds to high potential for influencing the force flow from the gearbox mountings to the rear sub frame mountings. Using a point net on the rear sub frame, the TFs from a normal excitation of $\mathrm{F_n} = 1\ \mathrm{N}$ at a driving point to the three dimensional force reactions at 7 mounting positions have been calculated. Therefor, no operational accelerations have been applied to the model. Since the sub frame mounts are rigidly fixed in the simulations model, there will be no acceleration reaction on the mounting positions. That´s why a force to force transfer function was chosen to give relevant information about influencing the energy flow. As a result, 21 TFs for each excitation point have been calculated. This leads to a three-dimensional $\mathrm{n\ x\ 21\ x\ f}$ matrix $\mathrm{H_{nm}xyz}$ of transfer functions, where n is the number of the driving points, m is the number of rear axle and gearbox mountings $(\mathrm{m} = 1,2,\ldots,7)$, $\mathrm{x,y,z}$ are the spatial direction and f is the frequency which is the third dimension of the matrix (Eq. 2).

$$diag(F_1, F_2, \ldots, F_n) \cdot H_{nm^{xyz}} = \begin{bmatrix} F_{11x} & F_{11y} & F_{11z} & F_{12x} & \cdots & F_{17z} \\ F_{21x} & F_{21y} & F_{21z} & F_{22x} & \cdots & F_{27z} \\ \vdots & \vdots & \vdots & \vdots & \cdots & \vdots \\ F_{n1x} & F_{n1y} & F_{n1z} & F_{n2x} & \cdots & F_{n7z} \end{bmatrix} \tag{2}$$

The root mean square of all reaction forces at the 7 mounting positions in all three spatial directions is shown in Figure 5 for each excitation point at a frequency of 730 Hz.

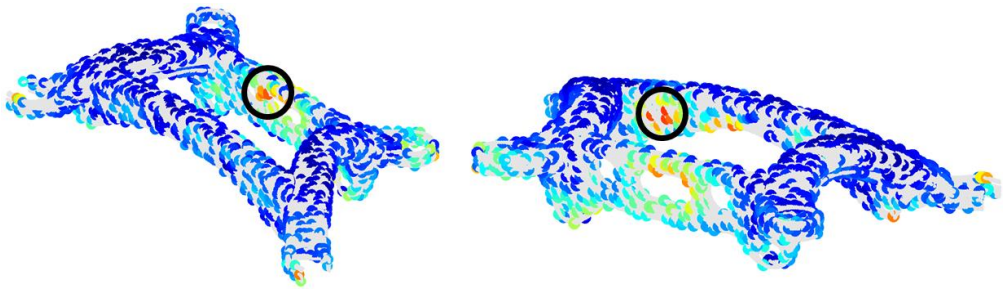


Figure 5: Root mean square of reaction forces at all mounting positions for each driving point at 730 Hz with the most sensitive positions marked

The two marked positions show the highest potential for inducing a counter force. A detailed analysis of the different spatial directions showed that the area on the left picture in Figure 5 is sensitive to all three directions at the mounts while the area marked on the right picture is mostly sensitive to reaction forces in x-direction. For further simulations of the AVC-system, the area marked on the left in Figure 5 has been exemplarily used. The results of the sensitivity

analysis lead to the conclusion that the rear and front crossbar are the most sensitive parts of the rear sub frame which also corresponds to the results of the STI-analysis.

5.3 Estimation of noise reduction potential

To evaluate the noise reduction potential of the possible driving points, a parameter study with the variables amplitude and phase of the applied actuator force has been set up. Therefore, the operational deflection shape of the rear axle has been simulated in the model using measured accelerations at the gearbox mounts. An additional force has been induced on the positions which have been identified by the STI and sensitivity analysis. The resulting parameter space is given by the amplitude and the phase of the induced counter force and the averaged reaction forces on the 4 rear sub frame bearings. The parameter study was exemplarily carried out on two points which has been identified by structural intensity (Figure 4) and by the sensitivity analysis (Figure 5, left). Figure 6 shows the difference of reaction forces at the sub frame mounts caused by an AVC-system with the ideal combination of amplitude and phase. Beside the reaction force reduction in each direction of each mounting, a summation over all mountings and directions (sum all) and a summation of the reduction on the front and rear mountings (sum front, sum rear) are shown.

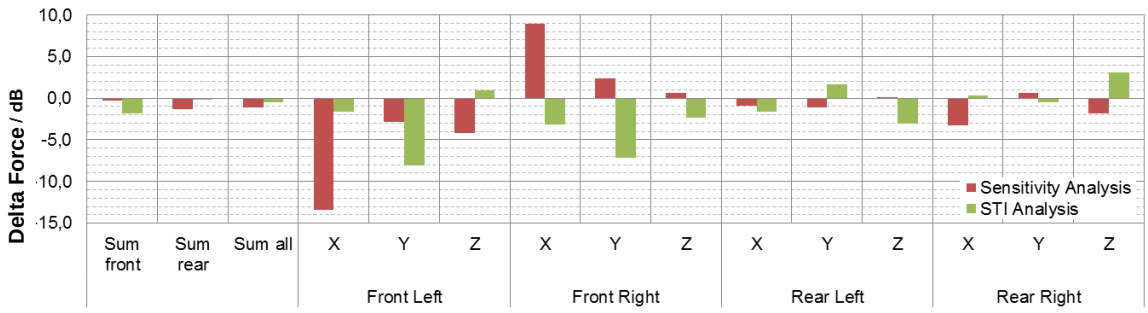


Figure 6: Difference of reaction forces on the rear sub frame mounting points with and without an AVC-system for two different driving points (Sensitivity and STI-analysis)

The results show that it is possible to reach a force reduction of more than 10 dB at single points in single directions using one actuator. Due to a simultaneous increase of the induced forces at other mounting points, the overall reduction is comparatively small. A vibration reduction on several mounting in more than one direction is only possible using more than one piezo actuator which makes the AVC-system more complex. A simulation based investigation of multiple actuators is not reasonable due to the increased parameter space and computing time. For that reason, experimental investigations on a rear axle test bench have been carried out in addition.

6 Rear axle test bench measurement

For verification of the simulation results, full vehicle measurements are not suitable because the excitation effects and transfer paths cannot be separated. For that reasons, a rear axle test bench has been set up. It consists of the complete rear axle assembly except break

components and wheels. For defined boundary conditions at the rear axle mounts, a “blocked force” configuration has been established by a rigid connection to a vibration fundament (Figure 7, left). This mounting is comparable to the simulation model set-up.

Figure 7: left - rear axle test bench with shaker excitation; right - applied piezo-actuators on the rear cross beam

To directly measure operational forces, three-dimensional force transducers are connected between rear sub frame mounts and fundament. The excitation of the gear mesh is not induced by rotating gears but rather emulated by exciting the rigid-body motion of the differential gear housing using a shaker. If the measured accelerations before the differential mounts are similar to the measured ones on the test bench, it can be assumed, that the structure-borne energy transfer in to the sub frame is on a comparable level. The similarity of the deflection shape in the car and on the test bench has been validated using the Modal Assurance Criterion (MAC). The calculated MAC value of the shape at 730 Hz is 0.91 which implies a very good correlation. Several piezo-actuators have been applied on the simulated positions to the rear sub frame (Figure 7, right). The reached force levels for a combination of three piezo actuators (all positions from section 5) in comparison to the initial state are shown in Figure 8. An overall reduction of the forces at the rear mounts of approx. 5 dB and at single positions of max.15 dB shows high potential for noise reduction inside the passenger compartment since the rear mounting positions are mainly responsible for the noise induction into the vehicle.

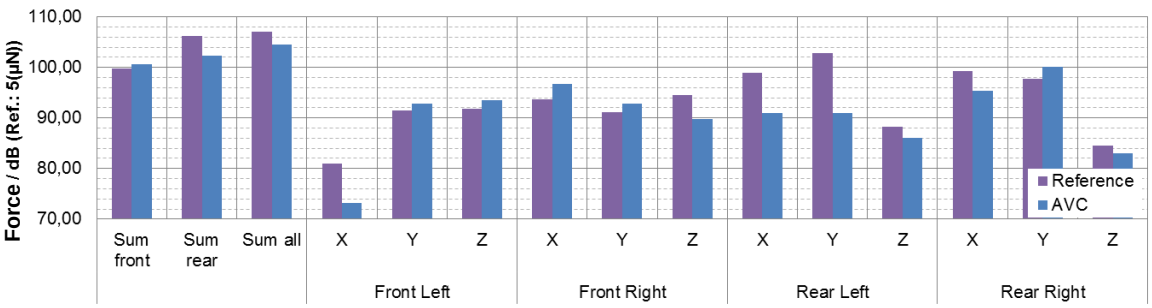


Figure 8: Absolute force levels with and without AVC-system (consisting of three actuators) for different sub frame mounting positions

7 Conclusions

The integration of a piezo based actuator in a sub frame of a rear axle seems to be a promising approach to significantly reduce gear noise of the rear differential in the passenger compartment of a car. In addition, it could be a reasonable alternative to the known primary and secondary measures for gear noise reduction and will also solve major goal conflicts in the development process. In a next step, a control algorithm needs to be applied which is able to control the AVC-system under typical driving conditions. Furthermore, the vibration and noise reduction potential of the AVC-system will be validated in a vehicle under real driving conditions. In terms of costs and complexity, an AVC-system for rear axles has a considerable advantage because new significant possible savings can be generated, e.g. reduction of production costs for acoustically optimized gears (higher excitation of low gear qualities could be compensated by AVC-system) and reduction of mass of rear axle components and vehicle body (worse acoustical behavior could be compensated by the AVC-system). The final concept of an AVC-system for rear axle noise including actuator, production process, control algorithm and high voltage amplifier needs to be analyzed in detail and compared in terms of estimated costs in a series production and potential savings on a full vehicle.

Acknowledgments

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