International Journal of Automotive Technology, Vol. 6, No. 1, pp. 23–28 (2005)

1229–9138/2005/019–04

POSSIBILITIES TO IMPROVE TRANSIENT GEAR SHIFT NOISE (SHIFT CLONK) IN A PASSENGER CAR

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(Received 6 November 2003; Revised 17 May 2004)

ABSTRACT–The presented investigation of shift clonk in a vehicle with front-wheel drive shows how a detailed analysis of the complete acoustic system with respect to excitation, transfer and radiation foremost enables possibilities of noise reduction to be worked out. One of the most important basics for the shift clonk analysis was a synchronous measurement of both, torsional vibrations in the drive train on the excitation side as well as airborne and structure-borne noise signals on the transfer and radiation side. Thus, root causes could be identified and improvement measures of the internal shift system could be worked out. An analysis of the transfer paths by means of airborne and structure borne noise measurements made evident that the side shafts were responsible for the disturbing frequencies in the transfer paths. With the help of the FE-simulation it was possible to develop measures of structure optimisation for the side shaft system. The realisation of these measures clearly reduced the shift-noises in the vehicle interior.

KEY WORDS : Drivetrain, NVH, Shift clonk

1. INTRODUCTION

The customers as well as manufacturer's demands on noise quality and vibration comfort of passenger cars have greatly increased in recent years. While the exterior noises are subject of the homologation, interior noises and vibrations highly influence the quality impression of the vehicle.

In particular, customers complained about transient metallic noises caused by gear shifts (shift clonk) in the transmission of their cars. The shift clonk was assessed to be a defect of the transmission. An investigation showed that tolerance variances during the production process do not cause the issue. Therefore further acoustic investigations were required.

As improvements directly at the excitation source are most effective at eliminating noises, the excitation mechanism has to be known first. Therefore, a detailed analysis is necessary by means of high-resolution data acquisition. In order to separate between overlapping effects like the excitation of torsional and bending vibrations, special NVH-test benches are used (Reitz, 2000).

The possibility to reduce transient shift clonk through design parameters within the transmission are restricted. Consequently, an optimisation of the complete vibration and transfer system has to be elaborated. This means identifying all components which cause interfering resonances. When the critical transfer paths are detected, resonances can be shifted by structure modifications. Thereby, an improved balance of the vibration system results.

If the measures taken at excitation source as well as transfer path are exhausted, the drive train related vibrations have to be isolated so that a transfer onto the body can be prevented. At the body panels, which are large and significantly weaker than the structure, noise can be radiated as through a loudspeakers membrane.

The following section will describe how to elaborate suitable measures to prevent the shift clonk by means of a detailed analysis of the complete acoustic system.

2. INVESTIGATION OF GEAR SHIFT CLONK IN THE VEHICLE

The shift clonk phenomenon is presented in Figure 2 by means of artificial head measurements during a road test. The diagram in the top right shows the vehicles deceleration as a function of time. At the measurements start the vehicle velocity amounts to v=38 kph. The manoeuvre lasts approximately 20 seconds. During this period, the vehicle coasts down with open clutch while the driver shifts from the 2nd into the 3rd gear and back. At the end of the measurement the velocity of the vehicle

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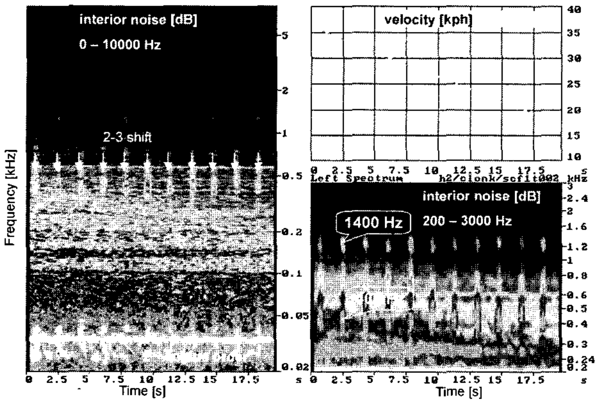

Figure 1. Artificial head measurement on the co-drivers seat during a 2–3 gear shift.

amounts to v=18 kph.

The diagram on the left side shows the airborne-noise spectra measured at the co-drivers seat in dependence of time. The y-axis provides frequencies between 0 and 10 kHz. In the low-frequency range of approximately 25 Hz the 2[nd] engine order is present, this is caused by the test vehicles four-cylinder in-line engine running at idle speed. When the driver shifts from the 2[nd] into the 3[rd] gear broad-banded impulses occur in a range of 30 to 1600 Hz, and therefore are clearly recognised. A more detailed focus on this frequency range is possible as it is given in the spectrogram on the right below.

In order to elaborate those frequencies which are disturbing, a correlation between subjective and objective data was done in the psycho-acoustic laboratory at the Institute of Automotive Engineering in Aachen (ika). Here, ten test persons (experts as well as customers) listened to the measurements recorded by an artificial head. As a result, the frequencies at 400 Hz and 1400 Hz (clearly shown in Figure 1) were assessed as very unpleasant.

The drive train components participated at the excitation are presented in Figure 2.

During the process of a gear shift, the clutch is

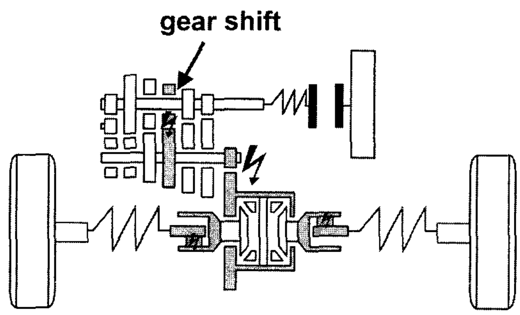

Figure 2. Excitation of shift clonk in the drive train.

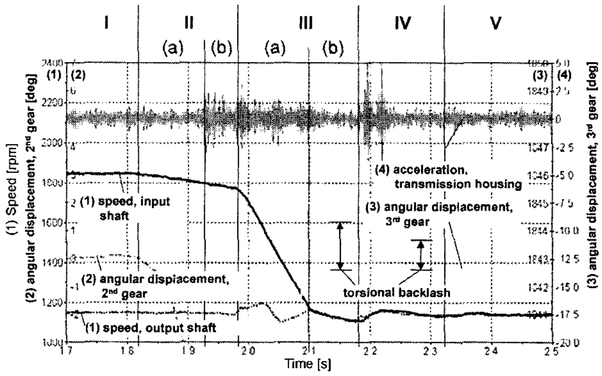

Figure 3. Analysis of the shift clonk during a 2–3 shift.

disengaged, which means that influences caused by the clutch characteristics can be excluded. The excitation follows during the shift process in the synchronisation unit. In consequence, a torque impulse results and is transferred onto the subsequent drive train components. Effected by the torsional backlash and the inertia of the drive train components (transmission gearing, differential and CV joints) structure-borne noise occurs at the transmission housing, which is transferred to the vehicle interior.

For the investigation of excitation mechanisms during the shift process by means of synchronous measurements of torsional vibration, structure-borne and airborne noise, a special analysis system with a very high resolution of torsional vibrations is used. Figure 3 shows the processes, which are responsible for shift clonk during 2–3 shift.

Beside the speed of the input and output shaft (axis (1)), the angular displacement between both shafts as well as the structure-borne noise at the transmission housing (axis (4)) are shown. The speed of the transmission output shaft is calculated referring to the input shaft under consideration of the ratio in the 3[rd] gear. The angular displacement between both shafts was calculated for the 2[nd] gear (axis (2)) as well as the 3[rd] gear (axis (3)).

The processes during gear shift are subdivided into several phases:

Phase I: The clutch is disengaged, i.e. the engine torque amounts to M=0 Nm, the 2nd gear is shifted. Transmission input and output shaft run synchronically, which means the angular displacement between both shafts amounts to φ=0°.

Phase II: The driver shifts from the 2nd gear into idling position.

(a) Initially, a transition of static and slide friction starts in the synchronous unit. Therefore, the input shaft is slowed down by the drag torque in the transmission. The speed of the output shaft keeps constant, since it is linked with the high inertia of the drive train and the wheels.

(b) While the gear shift sleeves are pulled from the locking device, an impulse occurs and leads to a transient structure-borne noise excitation of the transmission housing.

Phase III: The driver shifts from idling position into the 3rd gear.

(a) The synchronous torque, which is supported by the input shaft, reduces the input shafts speed and excites vibrations in the output shaft as well as in the connected drive train. In this phase a slight impulse results from the synchronisation.

(b) If the synchronous torque does not occur because of speed identity the input shaft is slowed down by the drag torque. The relative backlash between both shafts amounts to ö=0° following a short overshooting.

Phase IV: The input shaft freely oscillates during the shifting process from the 2nd to the 3rd gear and reaches a synchronous unity when the 3rd gear is completely formed up. During this phase impulse excitations occur within the limits of a backlash, which is temporarily variable because of the axial relative movement of the synchronous unit. The impulses within this shift phase cause a significant structure-borne noise excitation of the transmission housing.

Phase V: The 3rd gear is shifted, i.e. the gear shift is performed and the vibrations decay.

The parameter of the synchronous unit are widely given by functional demands and therefore offer only minor possibilities to improve the shift clonk. Design parameters, which serve to minimise force impulses during the synchronisation period, are listed as follows (Roeper, 1998):

- Ratio as low as possible between input shaft and the position of the synchronous unit (e.g. by the arrangement of the synchronous unit at the input shaft, for gear ratios larger than 1)
- Pitch circle diameter of the synchronous ring as large as possible and teeth as pointed as possible
- Choice of a small modulus for the locking device
- Inertia of the synchronous ring as low as possible
- Free flight distance of the synchronous unit as short as possible

Both, impact number ε as well as friction value μ during the synchronisation period can hardly be influenced by the engineer. Further optimisation possibilities derive from the transition from the synchronisation to the slide over phase, if the corresponding subsystems transmission input and drive train are taken into consideration (Roeper, 1998):

- Input shaft drag torque as low as possible
- Intertia of all components rotating with the input shaft as low as possible (especially the friction disc of the clutch)

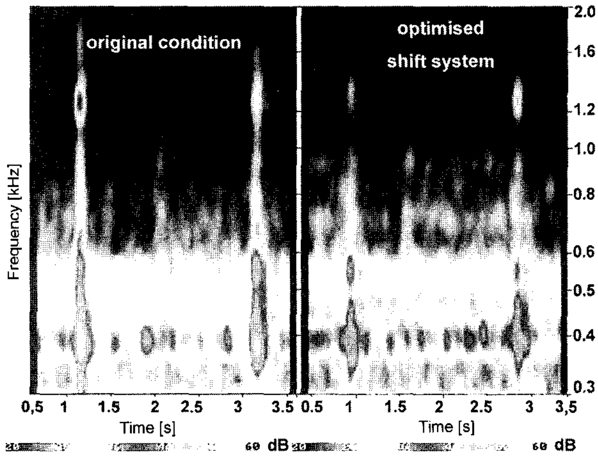

Figure 4. Improvement by measures at the inner shift device and the drive train (2–3 shift).

- Backlash in the drive train as low as possible
- Drive train drag torque as low as possible

Figure 4 shows the influence of measures directly taken at the inner shift device and the drive train onto shift clonk.

The presented data were measured with an artificial head on the co-drivers seat. Two shift periods were analysed for each vehicle setup. With the original setup distinctive resonances at 400 Hz and 1400 Hz are measured. Through the reduction of the drive train backlash and suitable measures at the inner shift device, the amplitudes are reduced compared to the original condition. Despite these measures the disturbing resonances are transferred into the interior.

This comparison shows that the engineer faces limits for drive train related measures as both, the backlash as well as inertia can not be arbitrarily varied. Therefore, the following acoustic system analysis concentrates on the question: What are the sources for unpleasant frequency proportions following a gear shift.

3. TRANSFER PATH ANALYSIS

In order to clarify the phenomena of disturbing resonances transfer paths from the transmission to the vehicle interior are investigated with the help of a transfer path analysis (TPA). Figure 5 shows all transfer paths that are relevant for the transfer of shift clonk.

The transfer of impulses caused by a gear shift can result in both, structure-borne as well as airborne noise. For the noise proportion transferred as airborne noise the soundpackage of the body plays an important role.

Structure-borne noise can be transferred via the engine and transmission mounts as well as over components of the external shift devices. Further transfer paths are

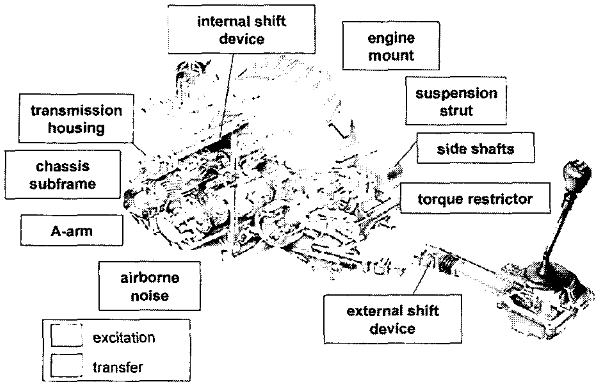

Figure 5. Transfer paths from the transmission to the body.

offered by the side shafts, which transfer the structure-borne noise to the steering knuckles. Here, a transfer is possible via the suspension struts or the A-arms and the subframe onto the body.

During the transfer path analysis the airborne noise in the interior is calculated as a product of excitation forces under driving conditions and the transfer functions are measured individually in laboratory tests. Thereby, the transfer paths have to be subdivided into apportioned transfer paths of the drive train, the engine and chassis mounts as well as the body. The total transfer results from a mathematical linkage of the single components of the transfer paths. As a result, coupling effects occurring by the inertance of the body as well as the dynamic stiffness of the mounts and the dampers between the subdivided transfer paths can be taken into consideration. This mathematical approach to analyse transfer paths is shown in Figure 6.

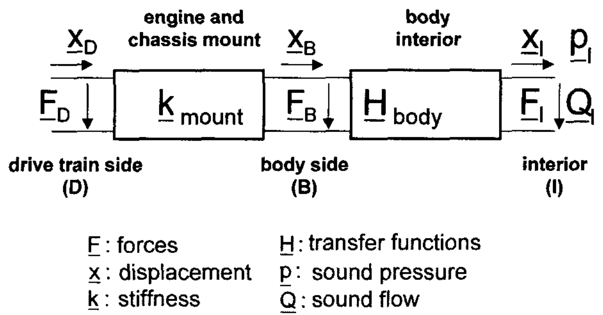

Figure 6. Mathematical approach to analyse transfer paths

The transfer path analysis aims to calculate the audible and distinct system reactions in the interior by analysing the subsystems transfer behaviour and therefore, to identify the significant transfer paths. An energetic and complex superposition of the single transfer paths concentrates on overlapped system reactions at the comfort relevant positions in the interior.

In order to analyse the transfer of shift clonk all transfer paths shown in Figure 5 have to be measured.

The data processing of the numerous measurements is done by the LMS analysis software CADA_X. The TPA module offers two methods to calculate the transfer behaviour:

- Matrix- inversion method
- Complex-stiffness-method

The matrix-inversion method is used for transfer paths, for which the connection between excitation and response position is considered to be stiff. It supports the analysis of drive train and chassis transfer to mounting components on the drive train side.

If mounting elements should be considered the complex-stiffness method is used. This method enables the analysis of engine and chassis mount transfer. Here, the dynamic stiffness of the individual mount components has to be known. For the shift clonk investigation these data were provided by the mount supplier.

For the analysis of each transfer path describing the relationship between the system excitation and the corresponding response at a certain position are measured.

$$H_{ij}(\omega) = \frac{B_i(\omega)}{A_j(\omega)} \tag{1}$$

i : measuring position of the response signal
j : measuring position of the excitation signal

If the drive train is excited by a force A_j at the position j (e.g. transmission housing), it responds with a measurable acceleration B_i at the position i (e.g. at the steering knuckles). The transfer function is a complex function and generally is presented in the frequency range. The

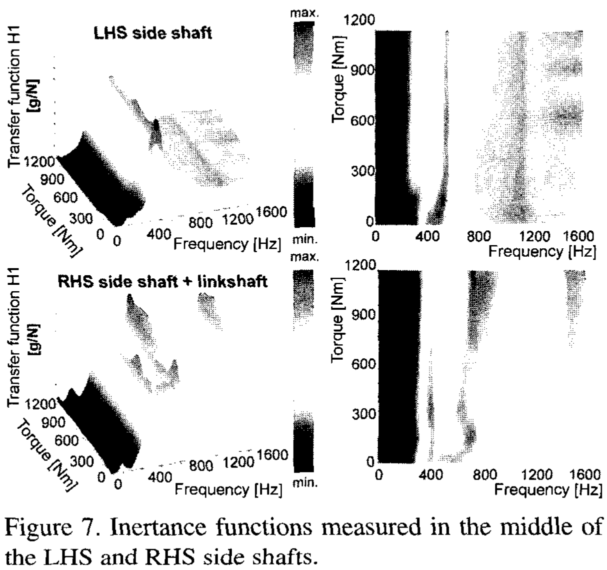

Figure 7. Inertance functions measured in the middle of the LHS and RHS side shafts.

maxima of the transfer function are eigenfrequencies of the transfer path.

Here, the impulse-hammer-method was used for the acquisition of the transfer functions. The resonances depend on the side shafts torque. Therefore, the torque was varied between 0 and 1200 Nm with the help of special devices at the wheels. In order to support the initiated torque the drive train was fixed at the starter ring gear.

Figure 7 shows the inertance functions of both side shafts (LHS: left hand side and RHS: right hand side) measured in the vehicle. In this case the excitation and response signals were measured in the middle of each side shaft.

The upper diagrams of Figure 7 show the transfer function of the left side shaft. The transfer behaviour at the right side consisting of linkshaft and side shaft is presented in the two diagrams underneath.

The first eigenfrequency of the left side shaft varies between 400 Hz and 600 Hz in the vehicle depending on the torque. For the gear shift especially the range of minor torques is of importance as here the backlash of the drive train is passed through. For torque below 100 Nm the left side shafts resonances varies between 400 Hz and 450 Hz, and consequently is also responsible for disturbing noises in the vehicle interior (compare Figure 1).

At the right side resonances at approximately 400 Hz, 700 Hz, and 1400 Hz occur, which also vary significantly as a function of torque. For low torques, the first two resonances seem to have one broad maximum. Foremost, for torques above 100 Nm the two resonances are separated. This effect is caused by the backlash of the CV joints which is not bridged in a range around 0 Nm and therefore a non-linear behaviour of the system occurs. With increasing torque the backlash is bridged and linear conditions are created.

The analysis of all relevant transfer paths shows that the side shaft transfer is responsible for disturbing noises caused by gear shifts in the interior. Structure-borne noise excited at the transmission housing is transferred via the side shafts and intensively forwarded to the steering knuckles. A part of the vibrations is transferred over the suspension strut, another part via the subframe to the body. The transfer via the engine mounts and the outer shift device were uncritical.

The identification of the disturbing transfer paths follows the elaboration of a structure optimisation of the side shaft system in order to reduce the relevant frequency proportions.

4. ELABORATION OF APPROPRIATE TRANSFER PATH TARGETS

A structure analysis of the side shaft system is done by means of simulation using the FE-method. Thereby, Figure 8 shows the FE model of the right side shaft system, which is in contrast to the left-hand side more complex.

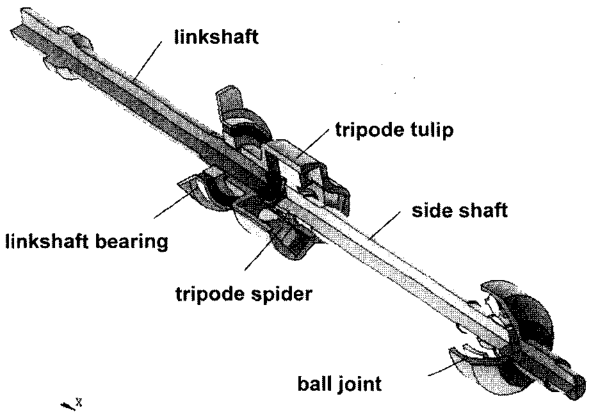

Figure 8. FE model of the RHS side shaft system to elaborate structure modifications.

The model consists of the linkshaft and its corresponding bearing, the tripod joint, the side shaft as well as the outer ball joint. In order to consider the influence on the eigenvibration behaviour special attention was also paid on the meshing of the joints. Similar to the vehicle certain boundary conditions are necessary. Therefore, the FE model is fixed at the differential side of the linkshaft, at the mounting points of

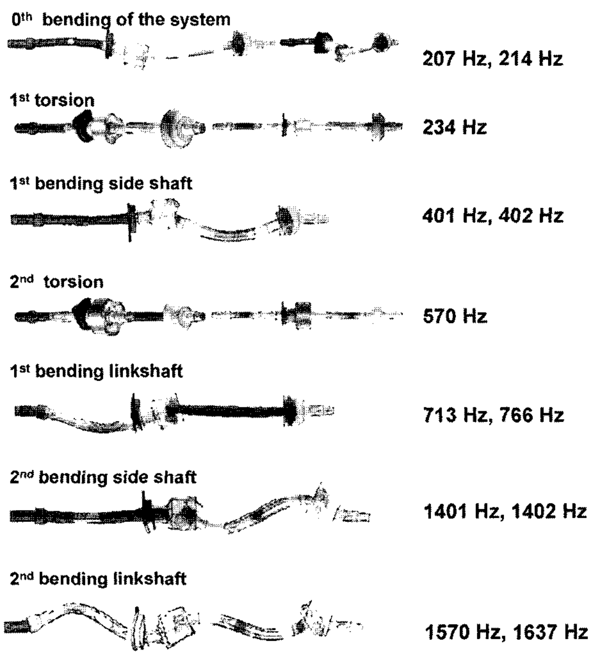

Figure 9. Modal analysis of the side shaft system with linear boundary conditions.

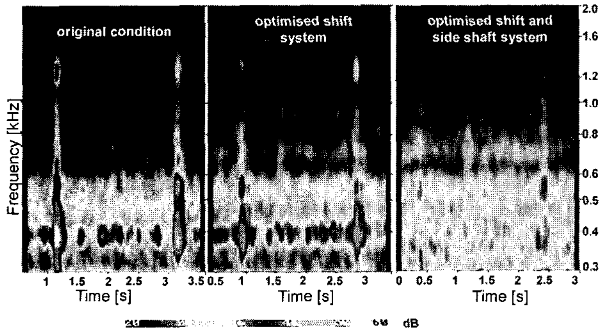

Figure 10. Artificial head measurement of a 2–3 gear shift.

the linkshaft bearing as well as at the wheel hub related end of the ball joint. Only a rotation around the rotation axis is allowed. The validation of this linear calculation model is done by an experimental model analysis in the vehicle with 200 Nm torque application at the wheels, i.e. with a bridged backlash within the joints.

Figure 9 gives an overview of the eigenfrequencies and mode shapes of the side shaft system of the production vehicle.

The fundamental bending mode of the total system with maximum amplitude at the tripod is given as orthogonal mode at 207 Hz and 214 Hz. At 401 Hz or 402 Hz the 1st bending of the side shaft is excited by a gear shift. 713 Hz or 766 Hz form the frequency of the 1st linkshaft bending in dependence on the vibration direction. The 2nd bending of the side shaft amounts to 1401 Hz or 1402 Hz also as an orthogonal mode. The 2nd bending of the linkshaft shows the highest values in the analysed frequency range (1570 Hz and 1637 Hz).

With this simulation model it is possible to execute a sensitivity analysis. Therefore, parameters with the highest effect on the eigenvibration behaviour of the side shafts are determined. As a result, targets can be established for the design of the side shaft system with respect to an improvement of the shift clonk problem.

The largest proportion of the gear shift clonk in the interior is caused by the 1st side shaft bending at approximately 400 Hz and the 2nd bending at approximately 1400 Hz. Consequently, remedial measures have been derived to influence the bending stiffness of the side shaft.

In order to validate the reduction measures in the vehicle a final acoustic system analysis has to be done. Therefore, Figure 10 shows the influence of the elaborated modifications of the shift system and the side shaft system in contrast to the original condition of the vehicle.

With simultaneous use of a vibration system, which is optimised with respect to the vibration transfer, and a gear shift system, which is improved with respect to excitation, a significant improvement of the airborne noise analysis results.

5. CONCLUSION

An optimisation referring the sources of excitation sometimes offer limited possibilities when dealing with customer complaints. Regarding shift clonk in particular, the design of the internal shift system of manual transmissions requires the consideration of shiftability, durability and costs beside acoustic aspects. The methods used in this paper facilitate a systematic analysis of excitation, transfer and radiation of transient gear shift noises (shift clonk).

Improvement measures that have been elaborated within the scope of the investigation presented are already in production. By doing so, the gear shift problem has been improved so that no further customer complaints were registered anymore. The targets, which have been established for the vibration system are available for future vehicle programs.

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