NVH Drive Line Optimisation by MBS

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Introduction

In the last decades great efforts in automotive R&D have lead to new technologies and materials as well as ever increasing reliability of the individual components. One important factor concerning customer satisfaction is a good NVH performance. To achieve the necessary NVH goals all major vehicle components have to be taken into account, with the drive train being one of the most prominent. The drive trains main task is simply to transfer the engine torque to the road, but its design process is complicated by a number of issues like durability, packaging, NVH, driveability and costs. These issues often lead to conflicting requirements and thus the need for reasonable trade-offs. The use of modern simulation tools is essential to find satisfying compromises and fulfill NVH requirements. This process shall be documented with the following study.

Model Set Up

The model is set up using the multi body simulation software ADAMS. It comprises all relevant components, such as power train, shafts, differential and wheels, sub frames as well as suspension in high resolution and the rigid vehicle body. With the help of special purpose models regarding nonlinearities the power train, sub frame and differential mounts are modeled. The model is shown in **figure 1**.

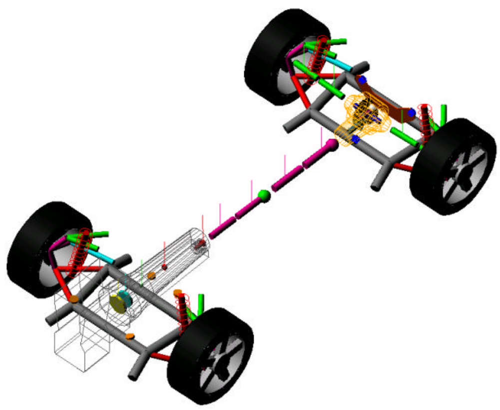

Figure 1: MBS Vehicle Model

In its base configuration the model consists solely of rigid bodies, constraints and forces. The drive train parts are modeled to consider the first torsional modes of the shafts and relevant nonlinear effects, such as clutch friction. This configuration allows simulation work to start early in the design process when detailed design-data of the components are not yet available and input data have been defined by the specifications of a predecessor vehicle. With the ongoing development process the model is updated and, where necessary, rigid parts can be exchanged by flexible structures.

Booming Noise

Torsional vibration in drive trains are the cause of a range of different NVH phenomena such as very low frequency events as shuffle vibration and clonk noises, booming noises in the lower frequency range and high frequency gear rattle noises.

In rear wheel driven vehicles booming noise is frequently brought about by high rotational irregularities in the rear differential, causing the differential to pitch and roll and therefore leading to high dynamic forces at the differential mounts. These excite the vehicle body, either directly or via the sub-frame, depending on the installation of the differential. The frequency range for this phenomena lies between approximately 40 Hz to 100 Hz.

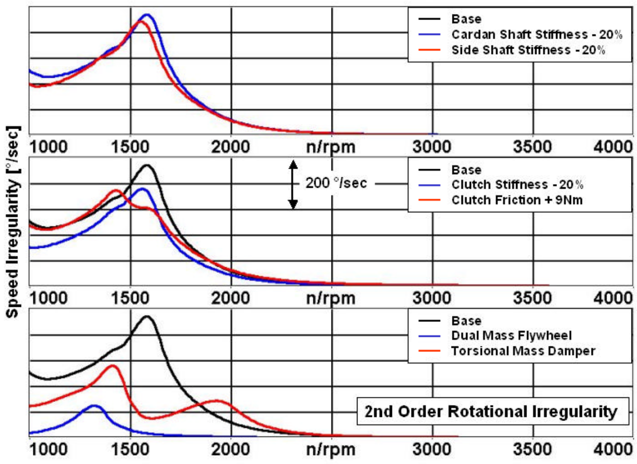

Figure 2: Speed irregularity at cardan shaft

Figure 2 shows the results of a simulated full load run up. The 2nd order speed irregularity at the rear cardan shaft is plotted versus engine speed. The baseline drive train configuration shows a resonance at 1580 RPM respectively 53 Hz. High speed irregularities appear at this engine speed due to the high 2nd order excitation of the diesel engine.

The flywheel and clutch twist against transmission, cardan and side shafts with a vibration node in the transmission input shaft. This mode can be identified as the 3rd drive train mode “cardan shaft”. The 3rd mode, as well as the 4th mode “transmission”, is also known to excite gear rattle due to higher torsional vibration amplitudes in the transmission.

It is difficult to estimate the relevance of rotational irregularities in respect to interior noise compared to the dynamic power train mount reaction due to engine gas and mass forces and moments. One approach is to assess the dynamic forces acting on the vehicle body at the joints. With multiple joints and three orthogonal directions for each it soon becomes difficult to make conclusive statements concerning their interior noise level significance. On top of that the transfer path behaviour has to be taken into account. These problems can be overcome by using the Vehicle Interior Noise Simulation (VINS) developed by FEV.

Vehicle Interior Noise Simulation (VINS)

The Vehicle Interior Noise Simulation has a wide range of applications. In principle it evaluates the excitation by filtering it with the measured vibro acoustic transfer functions of the vehicle body from excitation point to the driver's ear. The resulting individual noise shares are added in correct phase relation yielding the binaural noise at the driver's ear.

The excitation can either be measured, e.g. in an acoustic test cell, simulated or gained by a combination of both. In that way it becomes possible to evaluate a prototype engine in existing vehicles even before the engine can be installed in the vehicles and therefore making an early acoustic troubleshooting possible. Another application is the evaluation of individual noise shares in relation to overall noise to detect acoustic "weak points". This strategy will be used to evaluate the significance of the observed 53 Hz resonance.

Since vibro acoustic transfer functions are not available early in the design process, they can be measured at a predecessor vehicle to enable predictions of the interior noise.

Excitations were considered at the power train mounts, the suspension strut domes and the rear sub-frame mounts. The results from the simulation are shown in **figure 3**. The overall levels of the interior noise as well as those of the two major noise shares, power train mounts and rear subframe mounts, are plotted versus engine speed. While the interior noise is governed by the power train mounts contribution over most of the speed range, it is clearly dominated by the sub-frame mounts in the 1580 RPM range by up to 12 dB. This means that the speed irregularity calculated with the help of the MBS drive train model has to be judged as critical.

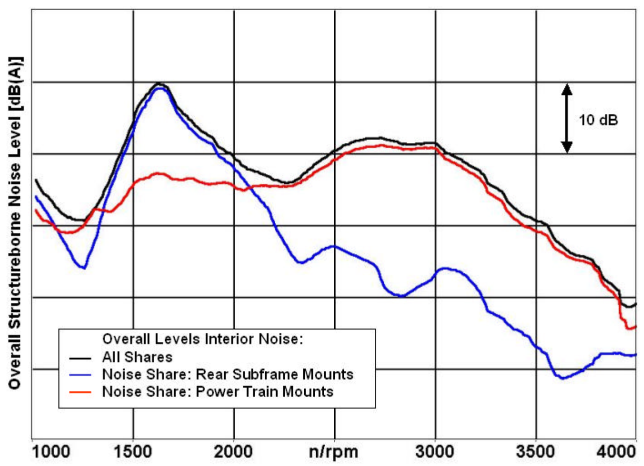

Figure 3: Overall Level: Interior Noise (Shares)

Several modifications to reduce the speed irregularities at 1580 RPM and thus the booming noise are analysed with the MBS model. The results can be taken from **figure 2**. The topmost graph shows the influence of stiffness reduction of cardan and side shafts by 20%. While no benefits can be seen for cardan shaft stiffness reduction, the reduction of side shaft stiffness decreases the speed irregularity by 10%. This would result in a reduction of the interior noise by less than 1 dB.

Modifications of clutch characteristics (**figure 2**, middle) show a more significant influence, a reduction of the clutch stiffness by 20% reduces the speed irregularities by approximately 20%. Alternatively increasing the friction torque in the clutch by 9 Nm leads to a reduction of speed irregularity by 20% as well as a shift of the resonance frequency to 48 Hz (1440 RPM). This measure would yield a reduction of interior noise by approximately 2 dB.

Better results can be achieved by either a dynamic damper on the cardan shaft or a dual mass flywheel (**figure 2**, bottom). The dynamic damper improves speed irregularity by about 40%, the dual mass flywheel be even about 75%. The cut back in interior noise would amount to 4.4 dB and 12 dB, respectively.

While the dual mass flywheel offers the greatest benefits in terms of interior noise, it is also the most expensive alternative. Additionally, limitations in power train length, e.g. due to package restrictions, may prohibit the use of a dual mass flywheel. This would advocate the alternative use of a combination of modifications, for example dynamic damper and a reduced clutch stiffness. This combination could achieve a noise reduction by more than 6 dB, meaning that the maximum level caused by the sub-frame noise share would not rise significantly above the maximum noise level caused by the engine noise share.

Shuffle and Driveabilty

Shuffle is a NVH effect caused by sudden changes in the drive torque resulting in low frequency longitudinal vehicle vibrations due to excitation of the first drive train mode. This phenomenon normally occurs in the lower gears, i.e. 1st and 2nd. It can seriously affect vehicle driveability when higher acceleration amplitudes are reached, for example in stop and go driving situations. Therefore the modifications analysed to reduce booming noise shall be evaluated regarding their effect on the shuffle mode.

The simulation results regarding shuffle can be summarised as follows. It can be observed, that the potential for optimisation by hardware measures is very low. A more suitable measure to improve shuffle behaviour is by engine management system calibration. By a short, controlled reduction of engine torque during the torque increase vibration amplitudes can be reduced substantially by approximately 60%. Stiffness changes have no effect on the acceleration amplitudes, a reduction of side shaft stiffness leads to a slight increase in acceleration amplitudes and a slight decrease in the eigen-frequencies. Since the human body perceives vibration stronger with lower frequencies, a drop in the eigen-frequency should be avoided.