

Gearshift-Comfort Oriented Transmission and Drive Train Simulation at BMW

This report describes the way in which the complete drivetrain was modelled with SIMPACK at BMW on the basis of experimental and theoretical system analysis and how this model enabled analysis and evaluation of all parameters which affect gearshift comfort.

1 INTRODUCTION

Apart from service life, gearshift comfort is one of the most important design criteria for manual transmissions and makes a major contribution towards the driver's general feeling of well-being. Irregularities in the shifting sequence or faults exert a particu-

larly negative influence in this respect. The continuously growing diversity of variants combined with increasing cost pressure and shorter development periods are leading to a situation in which comfort objectives can no longer be achieved by validating transmission designs with testing alone. Therefore a SIMPACK MBS simulation process was enabled at BMW in order to assess effects of drivetrain modifications on shiftability before test components are manufactured.

» CUSTOMER APPLICATION 01

Dr. Bencker, M. Nussbaumer, BMW Group, Prof. B. Schlecht, TU Dresden

Gearshift-Comfort Oriented Transmission and Drive Train Simulation at BMW

» SOFTWARE 06

Dr. Wolfgang Trautenberg, INTEC GmbH

2-D-Plots in the New SIMPACK Plot Module

» SOFTWARE AND APPLICATION .. 08

Dr. Lutz Mauer, INTEC GmbH

Modelling and Simulation of Drive Line Gears

» SIMPACK TIPS AND TRICKS 11

Dr. Christoph Weidemann, Steve Mulski, INTEC GmbH

Sensors and their Definition

2 FUNDAMENTAL PRINCIPLES

The manual gearboxes used at BMW are inline countershaft transmissions, Figure 1. With this gearbox construction, the engine torque is transmitted via the input shaft to the countershaft, where the power flow branches via the various gear stages to the drive shaft according to the selected gear. One exception to this is "direct drive", whereby the gearbox input shaft is directly linked to the output shaft when

Dr. Bencker, M. Nussbaumer, BMW Group, Prof. B. Schlecht, TU Dresden

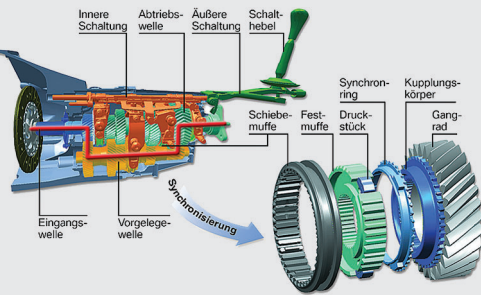


Figure 1: In-line Countershaft Transmissions and Synchroniser

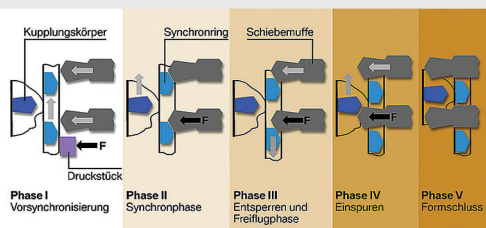


Figure 2: Positive of the Synchroniser showing the five Phases of the Synchronisation Sequence

the gear is selected. When changing gear, the driver uses the gear lever to activate the synchromesh mechanism with the clutch pedal pressed down. The gearshift effort is applied to the sliding sleeve of the synchroniser via the external and internal shift mechanisms and the desired gear is selected.

2.1 SYNCHRONISER

The synchroniser, Figure 1, is used to accelerate or decelerate the gear set (with clutch disk) from the current gear level to the speed of the target gear, enabling gear selection. Positive locking is then established during the subsequent phase and the gear is engaged.

The synchronisation sequence can be divided into 5 phases, which are reflected in the gearshift effort profile at the gear lever, Figure 3.

Joined to the fixed sleeve with a torsion resistant connection, the sliding sleeve is in its neutral position during Phase I and the driver pushes it towards the gear wheel to be selected, Figure 2, Phase I. Pre-synchronisation is achieved by the thrust pieces which move the synchroniser ring into the blocked position. This is perceived as a slight increase in gearshift effort, Figure 3, Phase I.

Phase II is referred to as the synchronising phase. The sliding sleeve presses the synchroniser ring against the friction cone of the synchroniser hub, Figure 2, Phase II, while the rpm of the gear set is matched to the speed of the target gear by means of one or several friction surfaces. The gearshift effort increases again rapidly during this phase, Figure 3, Phase II.

Once synchronism is achieved, the torque between the mating surfaces tends towards zero and the synchroniser ring can rotate freely again, Figure 2, Phase III. This eliminates the blocking effect and the free-flight phase (Phase III) begins. This process is evident as a noticeable dip in the effort profile, Figure 3, Phase III. In Phase III, the sliding sleeve again moves towards the gear wheel until it makes contact with the synchroniser hub, initiating

Phase IV.

The resulting impact, Figure 2, Phase IV, is transmitted via the internal and external shift mechanisms to the driver's hand and is shown as an impulse in the gearshift effort profile, also referred to as the double bump, Figure 3, Phase IV.

Phase V, in which the gear is engaged, begins when the bevels on the teeth of the synchroniser hub have been overcome. The renewed increase in gearshift effort is due to the sliding sleeve making contact with the end stop, Figure 3, Phase V.

3 PROBLEMS ENCOUNTERED IN THE SHIFTING SEQUENCE

Shifting Sequence Reference literature describes four known synchroniser related types of problem occurring in the shifting sequence, which are perceived by the driver acoustically or in the gearshift effort profile at the gear lever. These phenomena are unblocking inhibition, meshing inhibition, double bump and vibration grating. The first three are static problems and can be influenced by geometric synchroniser variables (cone angle, sharpness of selector teeth, etc.), for example. Unlike these, vibration grating is essentially determined by the dynamic performance of the entire drive train, as well as geometric transmission variables and the synchroniser. The reciprocal influences of gearbox, drive train and vehicle on the shifting sequence lead to numerous interactive effects and goal conflicts, which are very difficult to define in terms of testing, which is why simulation is used to examine this vibration phenomenon.

3.1 DESCRIPTION OF THE "VIBRATION GRATING" PHENOMENON

The continuous increase in engine torque is accompanied by an increasing load on the drive train components (clutch, gearbox, etc.). The effects are inevitably felt in their size and, above all, in the inertia of their masses. In an effort to achieve short shifting times with low gearshift effort, efficient multi-cone synchronisers are used, en-

abling acceleration or deceleration of the inert gear set to the level of the target gear. The resulting high synchronising torque – particularly when changing down from 2nd to 1st gear – is supported by the drive train, which is twisted as a result. In the free-flight phase, when the synchronising torque drops to 0 Nm, the drive train settles back within its elasticity and backlash limits. This leads to another difference between the speeds of the sliding sleeve linked to the output shaft in 1st gear and the synchroniser hub, which is linked to the gear set and rotates at virtually constant speed. When the sliding sleeve makes contact with the synchroniser hub (Phase IV) during a grating gear shift, the sleeve is locked out and its teeth skip several times until the speeds are matched. The resulting axial motion of the sleeve is transmitted to the driver's hand directly and is expressed in the form of the manual force profile shown in Figure 4.

3.2 ASSESSMENT CRITERIA AND CHARACTERISTIC VALUES

An assessment of the unpleasant vibration grating subjectively perceived by the driver's hand on the gear lever requires objectification of this process according to measured values. Measured directly at the gear lever, the interface to the driver, the gearshift effort profile, Figure 4, is a particularly suitable evaluation variable. The driver regards a renewed increase in effort after synchronisation as being very unpleasant as he does not instinctively anticipate any further resistance. This gives rise to the relationship between maximum synchronising force ($F_{max,II}$) and the double bump ($F_{max,VI}$), Figure 3, as an evaluation variable. The smaller the ratio $F_{max,VI}/F_{max,II}$, the more inconspicuous the double bump. The time dimension of the fault is expressed by integral I_H for the manual force from the time at which meshing begins through to the time of positive engagement, Figure 4. The more time required for this operation and the higher the force amplitudes, the less easy the gearshift operation as

perceived by the driver. The number of force amplitudes n during the meshing operation must also be known in order to distinguish between grating and faultless gear shifts. $n = 1$ indicates a faultless gear shift, whereas $n > 1$ implies a grating gear shift, Figure 4.

4 SIMPACK SIMULATION

Based on the results of experimental (test rigs, in vehicle tests) and theoretical research the necessary modelling depth and the degree of detail was determined in order to generate an appropriate simulation model. It was found out that it is essential to model the complete drive train system with all of the relevant system characteristics. This is the only way to ensure that the reciprocal effects of the individual subsystems are considered. The studies described below do not therefore simply concentrate on the driver and the influence exerted by variables inside the transmission, but also allow for the vehicle as a whole.

4.1 SIMPACK MBS MODEL

4.1.1 VEHICLE AS A WHOLE

The modular structure of the vehicle as a whole essentially comprises the body and its masses, with the transmission, drive train and rear axle carrier sub models connected to its bearing points, Figure 6 and 8.

4.1.2 GEARBOX MODEL, INTERNAL AND EXTERNAL SHIFT MECHANISMS

The gearbox model comprises a housing, which is connected to the body and the engine, the synchroniser and one gear. Particular importance was attached to the model of the synchroniser (sliding sleeve, synchroniser ring and teeth of the synchroniser hub are mating components), Figure 7. The according SIMPACK library functionality (force element) was developed by INTEC within a project work. The synchronising torque built up during the synchronising phase causes system excitation and it is stored in the model as a measured function of gearshift effort and differential speed. The sliding sleeve and synchroniser hub, which

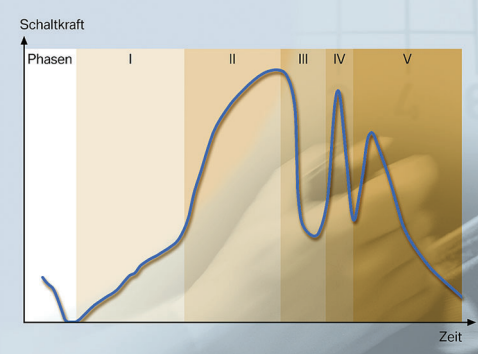


Figure 3: Five Phases of the Synchronisation Sequence

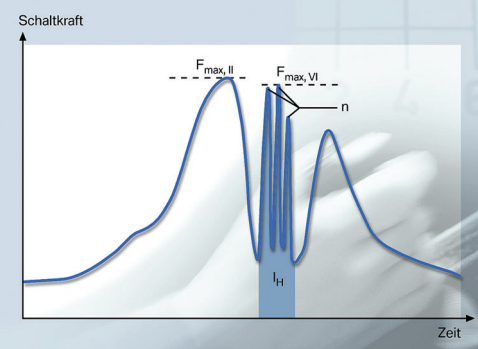


Figure 4: Manual Force Profile showing Synchronisation

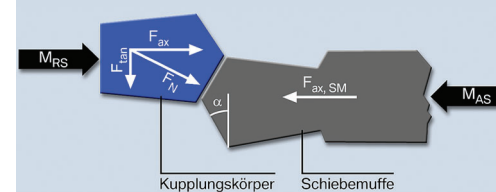


Figure 5: Contact Situation of Synchronisation Phase IV

Dr. Bencker, M. Nussbaumer, BMW Group, Prof. B. Schlecht, TU Dresden

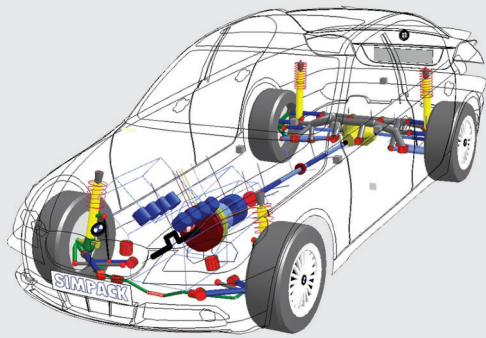


Figure 6: Front View of the MBS Model

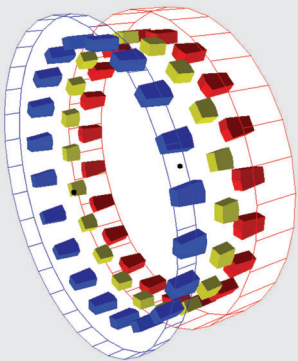


Figure 7: Synchroniser (Sliding Sleeve, Synchroniser Ring and Synchroniser Hub)

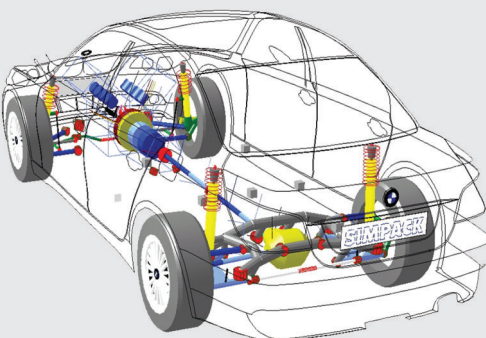


Figure 8: Rear View of the MBS Model

make contact with one another during meshing, are represented by geometric bodies. The forces exerted when these bodies meet are therefore described unambiguously. Furthermore, the flexible internal and external shift mechanisms are modelled with the pertinent bearing points on the body and gear-box. These bearing points constitute a very important variable determining gearshift comfort in transmission development and therefore require in-depth examination.

4.1.3 ROTARY DRIVE TRAIN MODEL

The rotary drive train model comprises flexible coupling, prop shaft, centre bearing, differential, output shafts and tyres. These components are also modelled in detail and are linked to the body at the relevant bearing points.

4.1.4 REAR AXLE CARRIER MODEL

The entire kinematics of the suspension and all bearing points are stored in the rear axle carrier model. This enables modelling of pitch and roll vibration, as well as the translatory longitudinal vibration of the rear axle carrier, which must also be considered when examining vibration grating.

4.1.5 DRIVER MODEL

Great importance is also attached to realistic modelling of the driver as his behaviour essentially determines the overall shifting operation, particularly during the synchronising and meshing phases. A parameterised mass-spring-damper system enables simulation of all relevant driver types (sporty, average and comfort oriented), which are represented by different gearshift effort values and/or shifting times during the synchronising phase. The interaction with the internal and external shift mechanisms and the synchroniser gives rise to the gearshift effort profile. A constant effort value is specified for the driver model during the meshing phase, in which the driver perceives the response of the drive train at the gear lever. The interaction with the vehicle gives rise to a corresponding double bump or grating.

4.2 MODEL VERIFICATION

The model is verified according to variables measured in preliminary tests, i.e. the torque at the prop shaft, the rpm of the sliding sleeve, the rpm of the gear wheel and the manual force profile. Apart from this, the model is calibrated with vibration measurements for the rear axle carrier and the differential. Figure 10 and 11 shows an example of a comparison between simulated and measured synchronising operations and unobstructed drive train vibration.

5 MODEL ANALYSES

With defined boundary conditions, parameter variations or statistical experimental design were performed with the according SIMPACK model. Conclusions regarding the sensitivity of components could be reached quickly and easily in order to enable definition of optimum design in a subsequent stage. Goal conflicts with other phenomena must be considered in this respect, however. These include longitudinal dynamics, stress reversal behaviour (bucking) and acoustics (clacking, knocking noises during load reversal).

5.1 BOUNDARY CONDITIONS

Gearshift effort is varied within certain limits during each simulation run, which comprises numerous individual computations, to ensure that the simulation allows for the gear changing behaviour of both sporty and comfort-oriented drivers. The forces taken as the basis for this are obtained from in-vehicles measurements and they represent the entire driver spectrum to be expected from customers. Apart from this, it is essential to ensure that the meshing conditions between the teeth of the sliding sleeve and the synchroniser hub are defined when they make contact for the first time to enable a comparison of the variants. The starting conditions are therefore varied within the tooth pitch in the model. This method offers a means of examining a situation in which the teeth of the sliding sleeve fit directly

into the gap between the teeth of the synchroniser hub or the bevels on the teeth collide with one another. Vehicle speed is another important variable to ensure comparability. This is also varied within certain limits during a simulation run.

5.2 OPTIMISATION VARIABLES

There are interface variables between the driver, internal and external shift mechanisms, gearbox, synchroniser and rear drive train subsystems. Optimisation of the interface variables between gearbox and drive train (reducing differential speed and torque for meshing, maximum possible axial force) also brings about a reduction in the axial force acting on the sliding sleeve. This must be applied by the driver via the internal and external shift mechanisms or is transferred to the driver's hand and is expressed in the gearshift effort profile. Optimisation of the aforementioned variables therefore also results in an improvement in the characteristic values for gearshift comfort.

5.3 PARAMETER VARIATION

Statistical Experimental Design and Optimisation by systematically varying parameters (statistical experimental design) related to the gearbox, drive train and internal and external shift mechanisms, it is possible to calculate different variants and define one or more optimised designs on the basis of the results. As the influence exerted by the various components depends on the vehicle configuration to a great extent, and the evaluation criteria (gearshift comfort, longitudinal dynamics, stress reversal behaviour) are determined by the design philosophy, no general conclusions can or should be given here.

6 SUMMARY

In an effort to counteract increasing pressure on costs and shorter development periods, the complete SIMPACK vehicle model presented in this article provides a tool that offers a means of comprehensively studying and analys-

ing all of the variables which affect vibration grating. This enables determination of the sensitivity of each influential parameter and identification of the contrary or reinforcing effects of a variation. To this end, a preliminary theoretical and experimental system analysis was carried out to identify the parameters which influence vibration grating and are found in the gearbox, the internal and external shift mechanisms and the rear drive train (including the rear axle bearing arrangement and rear axle kinematics). Based on the knowledge acquired with respect to sequences, processes, sensitivity and system behaviour, the SIMPACK simulation model was produced which is capable of realistically simulating the aforementioned vibration phenomenon. The defined characteristic values can be used to capture the driver's subjective sense of comfort when changing gear in objective measurements, thereby describing vibration grating. These variables offer a means of analysing and evaluating in vehicle measurements and simulation results. Based on the acquired knowledge, various drive train variants can now be tested cost-effectively and quickly according to vehicle model and associated customer requirements (sporty or comfort-oriented), allowing for reciprocal effects and goal conflicts, beginning at the concept phase and going right through to SOP. One result of this is a reduction in iterative loops, particularly with respect to testing and trials, as only the promising variants need to be tested. Using statistical experimental design, the test engineer of the future will be able to perform virtual optimisation and tuning calculations and use these as the basis for definition of test variants.

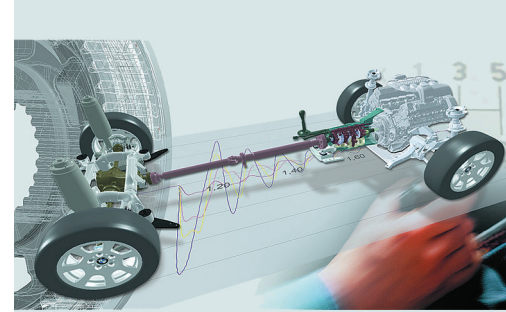


Figure 9: Complete Vehicle Dynamics System

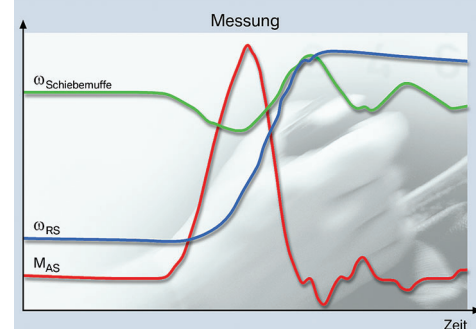


Figure 10: Measured Synchronisation Process

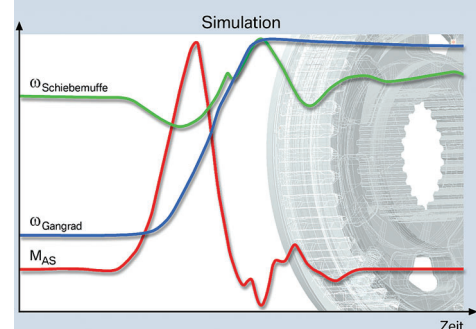


Figure 11: Simulated Synchronisation Process