

CAE-Based Driving Comfort Optimization of Passenger Cars

Timothy Drotar

Ford Motor Co. Research and Advanced Eng

Jacopo Palandri, Friedrich Wolf-Monheim, and Paul Zandbergen

Ford Research Laboratory Aachen GmbH

Bjoern Reff

Ford Werke GmbH

ABSTRACT

One of the key challenges in developing a vehicle for excellent vehicle dynamics is being able to achieve a high level of driving comfort without degrading the steering and handling performance.

The part of driving comfort discussed in this paper are tactile vibrations up to $f = 100$ Hz. This paper describes how Multi-Body Dynamics (MBD) Computer Aided Engineering (CAE) tools are applied to optimize such vibrations in the early phase of the development process. The approach hereby presented combines system level testing with MBD for the study of ride comfort, similar to the way that system level kinematics and compliance testing is combined with MBD to support steering and handling investigations. Laboratory investigations have been executed to fully characterize a reference suspension with respect to frequency and amplitude behavior. The respective MBD models have been subsequently refined and validated versus physical laboratory measurements. Several examples for a front wheel drive passenger car will be given, which show how these models can be used to effectively support chassis development in early design phases. The first example demonstrates the effect of subframe isolation on ride comfort via analyses of the forces transmitted to the vehicle body. The second example demonstrates the sensitivity of bushing stiffness to impact harshness using a full vehicle model. The final example demonstrates how impact harshness can be optimized using different levels of compliance split between the subframe and suspension bushings.

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INTRODUCTION

The term 'driving comfort' generally refers to a broad range of vehicle attributes such as Vehicle Dynamics (ride, steering, handling and braking), NVH (Noise, Vibration and Harshness), Powertrain (Drivability, Performance Feel, and Shift Quality), Climate Control and Ergonomics.

Within the context of this paper the term 'driving comfort' specifically refers to the structural vibrations of passenger cars in a frequency range of up to $f = 100$ Hz induced by surface irregularities of the road. These road induced vibrations are transferred sequentially through the wheel and tire assemblies, the hub and bearing unit, the knuckle, the control arms including various rubber-metal bushing elements, and the subframe structures, into the vehicle body structures. Tactile vibrations are felt by the driver through the seat structures, the driving pedal structures and the steering wheel. Audible vibrations may be heard throughout the vehicle cabin. This paper focuses on the tactile vibration phenomena only. Audible vibration phenomena are not in the scope of this work.

The tactile vibration phenomena predominantly influencing the driving comfort of a passenger car within a frequency range of up to $f = 100$ Hz can be subdivided into three main areas: primary ride, secondary ride and impact harshness. The primary ride behavior of a passenger car is related to various body motion phenomena related to the road profile input in a frequency range of between $f = 0.5$ Hz and $f = 4-5$ Hz. The key area of focus in terms of the primary ride behavior of a vehicle is the body motion control in roll, bounce and pitch degrees of freedom analyzed on road profiles with a pronounced portion of low frequency, long wavelength excitations.

The secondary ride behavior of a passenger car describes various vehicle body vibration responses to higher frequency road inputs in a frequency range between $f = 4-5$ Hz and $f = 30$ Hz covering a wide range of vibration types. For the objective secondary ride testing the vibrations are measured on the driver seat rails as well as on the steering wheel while driving on macroscopic flat road surfaces with a pronounced portion of high frequency, short wavelength excitations.



Finally, the impact harshness performance of a vehicle describes the vehicle body motions and vibration responses to single impacts of different heights, in a frequency range between $f = 30$ Hz and $f = 100$ Hz. Usually the impacts are crossed at a moderate speed of $v = 30$ kph to investigate the impact harshness behavior of the front and rear suspension systems. In terms of the impact harshness performance both the response to the initial impact and the decay (aftershake) are of equal importance [1].

MULTI-BODY DRIVING COMFORT SIMULATION - STATE OF THE ART

Early in the development phase of new vehicle suspensions, in the absence of physical prototype hardware, CAE tools are necessary to evaluate the performance of different design alternatives. In the early stages, CAE analysis will be used to support the decision-making process on the best concept. In the latter stages, CAE tools will be used to optimize the design for multiple attributes in preparation for series production. Typically in the early stage the layout of the vehicle is not yet completely determined and limited design information is available. Although there may be highly detailed data for 'carry-over' body structure components, there may only be estimates for the mass, inertia and geometry of the main chassis components. There will likely be only rough estimates for design and performance characteristics of the suspension bushings, springs, shocks and stabilizer bars. For this reason it is often impractical or even impossible to use traditional Finite Elements (FE) techniques for the assessment of ride comfort since they require data at a level of detail not available at this point in time.

Multi-body techniques are widely applied throughout chassis development to optimize the system level kinematics and compliance (K&C) characteristics of the suspension and to predict low frequency steering and handling attribute performance. They have also proven to be particularly effective to investigate ride attribute performance at a vehicle level ([6],[9]). The approach hereby presented applies system level MBD for the study of ride comfort, similar to the way that system level MBD is used to support steering and handling investigations. Moreover, using common models for ride and handling analysis presents the opportunity for increased efficiency of the multi-attribute optimization process.

The ability of multi-body dynamics models to handle material as well as geometric non-linearity, to integrate component flexibility via model reduction techniques, and describe the non-linear behavior of critical components such as rubber bushings and tires can effectively expand the range of validity of such models to frequencies up to approximately $f = 100$ Hz. This makes them particularly suitable to investigate transient phenomena such as impact harshness or engine startup simulations. Because of their limited number of degrees of freedom, multi-body models generally provide faster simulation times than finite element models, with an adequate level of accuracy for many structural vibration related issues like powertrain and road vibration ([2], [3], [4], [5], [6], [7], [8] and [9]).

A steady state response analysis of the system is still possible whether in the frequency domain by linearizing the system around a known operating condition, or in the time domain using the original non-linear model and applying spectral analysis methods in the post processing phase.

SYSTEM LEVEL CHASSIS TESTING

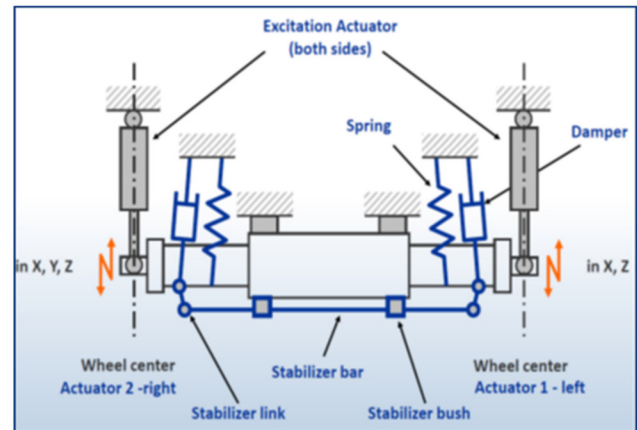


Figure 1. Schematic of the system level testing configuration (Courtesy of Trelleborg Vibracoustic)

To properly model the suspension response in the above mentioned frequency range a deep understanding of the chassis behavior is required. System level tests have been therefore executed using a suspension rig to fully characterize a reference axle.

As represented in Figure 1 in a schematic form, the axle is fixed to the rig and 2 hydraulic actuators excite it directly at the knuckles in vertical, longitudinal or lateral direction with a desired displacement. Forces are measured at the input location as well as at all the chassis interfaces with 3-dimensional load cells.

The tests include two main load cases:

- Low-frequency excitation at different amplitudes to highlight non-linear characteristics originating from suspension geometry, rubber bushings, friction, etc.
- High frequency excitation to characterize the main suspension load paths at small and higher amplitudes.

Some chassis components (e.g. stabilizer bar) can be deactivated during subsequent tests to determine its influence with respect to the response of the complete axle.

SYSTEM LEVEL MULTI-BODY MODEL VALIDATION

Taking advantage of the results of the chassis characterization described in the previous paragraph, a MB model of the suspension has been validated to match as close as possible the behavior

observed in the measurements. Starting from a base model, several enhancements have been made to the chassis component models to achieve correlation:

- *Chassis components:* all chassis components are implemented as flexible bodies by means of a Craig-Bampton modal reduction, accounting for as many normal modes as needed to cover the frequency range under investigation. To each body a proper structural damping ratio is assigned according to the material properties of the parts.
- *Rubber bushing and hydro-bushing model:* A custom bushing model has been developed and implemented to account for the quasi-static non-linear stiffness progression and the dependence of stiffness and loss angle on amplitude and frequency. This is shown schematically in Figure 2.
- *Springs and shock absorbers:* The suspension springs are represented by a linear one degree of freedom system model which is tuned to replicate the first axial mode of the coil springs. The shock absorbers are represented by a nonlinear relationship between force and relative velocity of the piston with respect to the tube, in parallel with a friction element that replicates the stick-slip behaviour of the damper.

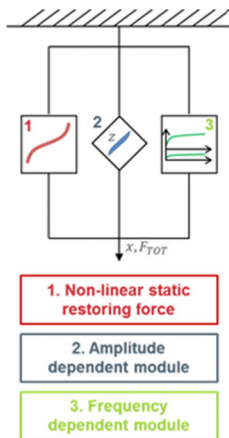


Figure 2. Bushing model modules

Figure 3 shows the level of correlation reached for low-frequency loadcases. As the amplitude of excitation decreases, the axle becomes more and more stiff. Due to the multiple amplitude dependent components implemented, the CAE response is very similar with the measured one

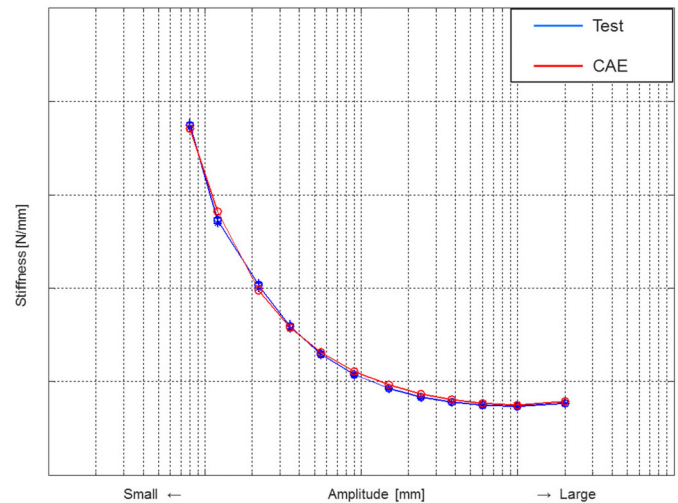


Figure 3. Vertical wheel centre stiffness as a function of the excitation amplitude-Test vs CAE

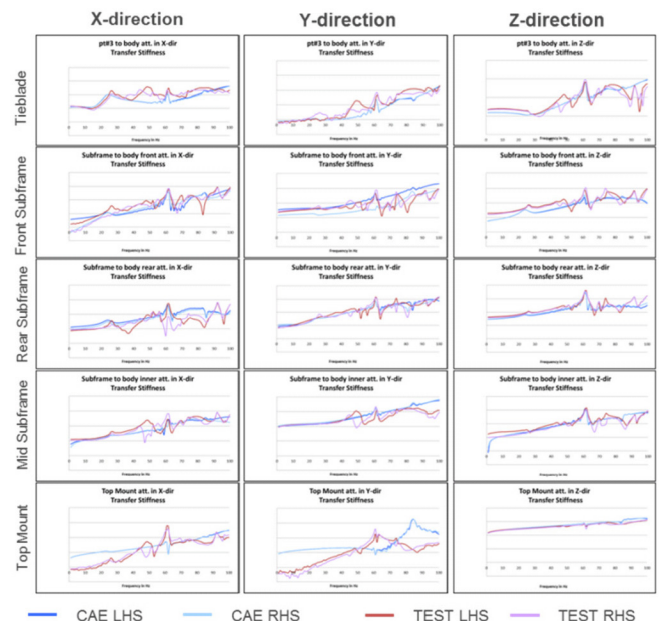


Figure 4. Transfer functions between force at the attachment points in the 3 directions due to a vertical input displacement at the wheel centre-Test vs CAE

In general, however, the model shows a very good agreement with the measurements both in quasi-static and dynamic conditions, in part due to the enhanced component models previously described.

EXAMPLE: INFLUENCE OF SUBFRAME ISOLATION ON LOAD PATHS INTO THE BODY

As mentioned previously, the forces that are generated at the contact patch by the interaction of the tire with the road and its irregularities are transmitted to the body - and eventually the occupants - via different load paths throughout the chassis components. The forces are initially transferred to the rim into the knuckle, then through the