

Optimization of Suspension Elastomeric Bushing Compliance Under Constraints of Handling, Ride and Durability

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ABSTRACT

Elastomeric bushings are widely used in the passenger cars to make the cars have an ideal vehicle Noise, Vibration and Harshness (NVH) performance. However, elastomeric bushings also influence on the vehicle handling, ride and the durability performance of each component in the vehicle suspension system. It is relatively easy and cost effective to change the compliance of the bushing components compared with other method because they are made of elastomeric materials. The design of an elastomeric bushing is really a big challenge. One of the main difficulties comes from the different target compliance is wanted according to the handling, ride and durability demand at each different orientation (indicated by X Y Z) of the bushing. In this paper the following procedure was used for optimization of suspension elastomeric bushing compliance. Firstly, a detailed multi-body model was built including the nonlinear bushing effects and lower control arm flexibility. Secondly, a series of virtual proving ground experiments are evaluated, such as step input and steering wheel releasing test, and impulse input test for handling, road cleat test for ride comfort, random input running test for the durability performance of components in the vehicle suspension system. Finally, a multi-objective optimization algorithm is taken as the appropriate framework for the definition the compliance of the bushing components of the addressed multi-objective optimization problem. The proposed optimal design process for elastomeric bushing compliance made it easy to evaluate and improve the vehicle handling, ride and the durability performance of the vehicle suspension components in the vehicle development process.

INTRODUCTION

Virtual design and test techniques based on computer aided engineering (CAE), such as advanced dynamic simulation, finite element analysis (FEA) etc, have been applied as a replacement of the physical prototype approach in various fields, especially in vehicle design and development. The design of vehicle handling, ride and durability, which are important performances of vehicle qualities, require relatively numerous processes and much time. By CAE performance parameters of suspension system can be guaranteed in conceptual design stage, thus bringing about a considerable reduction in cycle time of the overall R&D process.

Handling and ride have been well studied in a series of papers, such as the SAE technical paper 970091, 970092, 970097 and 970098, which mainly discussed the evolution of the Chevrolet Corvette C5's ride and handling characteristics. [Reference \[1\]](#) describes how vehicle level targets for vehicle dynamics performance were defined. [Reference \[2\]](#) described how these vehicle targets resulted in specific subsystem attributes for suspensions, tires, and steering system. The third paper in the series ([Ref. \[3\]](#)) described the execution of the front and rear suspensions in production hardware. The last paper ([Ref. \[4\]](#)) deals with the final development tuning of those suspensions, as well as that of the tires and steering system.

Durability analysis with dynamic simulation and finite element has become vehicle structural fatigue research development trend. A practical procedure for automotive CAE durability analysis using the random vibration approach is presented in [\[5\]](#). Two example results of automotive structures, a rear axle assembly made of metals, and a headlamp assembly made of plastics, are given to demonstrate the approach and their applications. Sudhakar

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Medepalli and Rajesh Rao [6] used computer simulations to predict road loads which are critical for the design of an automobile for durability and difficult to obtain this information in the early phases of the design process when no prototype vehicles exist and one has to rely on historic data. They outlined one such process and validate it by comparing the predicted loads to measured loads and pointed out the importance of including component flexibility information in the models. H. Riener, D. Peiskammer and W. Witteveen [7] dealt with the comparison of two different approaches of fatigue life prediction. Fatigue of a passenger car's front subframe is computed by means of modal stresses and by means of the common quasi-static approach. The full vehicle multi-body model was built in the software ADAMS. The durability analysis was performed with the software FEMFAT. The final results show that modal stress recovery is more accurate in prediction of structural elastic deformation impact.

The next section describes multi-body dynamic model for virtual proving ground test and finite element model for lower control arm. In current study, handling and ride are regarded as targets, while durability is as constraint and bushing compliance is as variables. All of them are introduced in detail in the third section. After that, the process of fatigue life analysis based on MSR(Modal Stress Recovery) is shown, including component modal analysis, generation of displacement course, fatigue life analysis based on MSR, and fatigue analysis result. Analysis tools and optimization method are then introduced. The results and conclusion of the paper are at last presented.

VEHICLE ANALYSIS MODEL

MULTI-BODY DYNAMIC MODEL FOR THE VIRTUAL PROVING GROUND TEST

A virtual proving ground test was conducted to calculate the dynamic loads applied to the components. A passenger car MPV with a McPherson strut front suspension subsystem and a twist-beam rear suspension subsystem was used, as shown in [Figure 1](#). A commercial sofeware, ADAMS/Car, was used for the modeling and dynamic analysis. The parameters of the vehicle are listed in [table 1](#).



Figure 1. Full vehicle multi-body model

Table 1. Full Vehicle Parameters

Parameter Name	Symbol	Unit	
vehicle total mass (full loaded)	W	kg	2080
full weight of front axle	Wf	kg	990
full weight of rear axle	Wr	kg	1090
wheelbase	s	mm	2695
front track	tf	mm	1480
rear track	tr	mm	1485
height of full loaded center of gravity	h	mm	720
the distance between front axle and the center of gravity to the axles	a	mm	1412
the distance between front axle and the center of gravity to the axle	b	mm	1283
Steering wheel diameter	hd	mm	385
road friction factor	μ	-	0.8

The vehicle multi-body model includes seven subsystems: front suspension subsystem, rear suspension subsystem, steering subsystem, braking subsystem, powertrain subsystem, tire subsystem and body subsystem. The body flexibility may be appropriate when predicting component loads, but for the simplification, the body is a rigid part in this model. The model also includes the nonlinear bushing effects and flexible lower control arm part.

The subsystem topology is consistent with the actual situation of the vehicle. The components with complex dynamic characteristics, such as engine power characteristics, brake characteristics, tire etc., are all built based on experimental data supplied by manufacturers. Besides, the mechanical parameters of the bushing compliance, shock absorber damping, helix spring stiffness, powertrain mount and other elastomeric isolator characteristic, are obtained by the dynamic testing instrument Bose ElectroForce 3510 in our own lab. [Figure 2](#) shows the equipment and testing components in experiments.



(a). Bose ElectroForce 3510



(b). Helix spring testing



(c). Shock absorber testing



(d). Bushing testing

Figure 2. Experiment equipment and components testing

FINITE ELEMENT MODEL FOR LOWER CONTROL ARM

The lower control arm (LCA) of McPherson strut front suspension is “L”-shaped forging structure. There is long enough distance between its two hinge joints which connect with subframe, that it can transfer vertical and horizontal forces effectively. One side of the lower control arm connects with subframe through elastomeric bushings, while the other side connects with knuckle(or called upright) through spherical joints. The lower control arm, which is one of the main parts of suspension components, is used to transfer force and torque between wheel and vehicle body.

In order to improve analysis accuracy and reduce the number of freedom, LCA is partitioned with hexahedral (see [figure 3](#)).



Figure 3. Finite element model of the lower control arm

DYNAMIC MODEL VALIDATION

When using the CAE methodology to build a model as a source of the simulation study, a reliable model needs to be validated first. It's very important and necessary to reproduce the dynamic of a real prototype. A common way is to test the real prototype to run some maneuvers and record the response data, then simulate the corresponding virtual prototype model to do the same analyses, and compares the responses, usually time signal.

To proceed with the model validation, a real prototype of the testing vehicle was equipped with a gyro located on the vehicle body to measure the lateral acceleration, yaw angular velocity and body roll angle, a steering wheel sensor to measure the steering wheel rotation angle and torque, a longitudinal and transversal speed sensor to monitor and measure the vehicle velocity, and an accelerometer placed on the outboard of the driver seat track to measure the Fore/Aft and vertical vibration. In this study, maneuvers were tested: a step steer input and then steering wheel releasing, a pulse steer input, and a road cleat input. The former two validate the handling responses and the last one validates the ride comfort. [Figure 4](#) shows some pictures of the tests in the proving ground.



(a). Steering wheel sensor



(b). Gyro



(c). Accelerometer



(d). Triangle road cleat



(e). Optical speed sensor

Figure 4. Proving ground testing

The model was run under steering wheel angle open-loop control and vehicle velocity closed-loop control, the comparisons are shown in [Figure 5](#).

[figure 5 here](#)>

The interaction between tires and road has a great influence on vehicle simulation, *Ftire* was chosen as the tire model, which is appropriate to be simulated well in both handling and ride application environments. As can be seen in the results, the both curves (model and measurement) showed good agreement. At this point, the validated model could be used in the following analysis process.

HANDLING, RIDE AND DURABILITY OF THE VEHICLE

HANDLING AND RIDE INDEX

The handling and ride analysis of full vehicle is a complex problem which refers to many different kinds of maneuver situations. Three experiment tests that provide both handling and ride information are prescribed:

Handling: Step steer response test (SSR) and then cornering steer release test (CSR)

Drive the vehicle at about 100 km/h in a straight line. Starting from the 0°/s yaw velocity equilibrium condition, apply a steering step input within about 0.2 second to a preselected value and maintain at that value after the measured vehicle motion variables have reached a steady state. ([Ref\[21\]](#)) After a while, then suddenly release the steering wheel, while maintaining the velocity constant and placing the vehicle in free control, so that its response is determined by its dynamic characteristics. ([Ref\[22\]](#))

Handling: Impulse steer response test (ISR)

Drive the vehicle at about 100 km/h in a straight line. Starting from the 0°/s yaw velocity equilibrium condition, apply a triangular waveform steering-wheel input to cause the amplitude of lateral acceleration level at about 0.4g. The steering wheel angle input pulse width is about 0.4s. The velocity is maintained constant. ([Ref\[21\]](#))

Ride: road cleat input running test (CIR)

The triangle cleat on the road for wheel bump event has 60mm height and 400mm hemline length as shown in [Figure 6](#). The forward velocity of 50km/H is applied for simulation as nominal value.

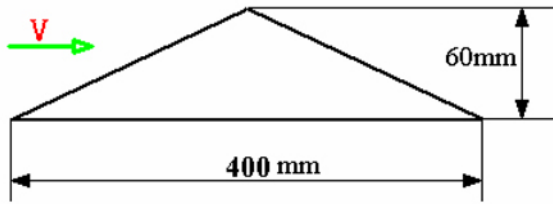


Figure 6. Triangle bump

Table 2. Targets for handling and ride index

	Test	Target
Handling	SSR	Response time of lateral acceleration $T_{a\dot{\gamma}}$
	CSR	Value of second peak of yaw angular velocity after steering-wheel release $\dot{\psi}_{\max 2}$
	ISR	Resonant peak level $f_{d,(\dot{\psi}/\delta_{\dot{H}})}$
Ride	CIR	Max absolute value $A_{x_{\max}}$ & $A_{z_{\max}}$

In table 2, these important characteristics or optimization targets are interpreted as follows:

SSR response time of lateral acceleration $T_{a\dot{\gamma}}$

The transient-response data reduction shall be carried out such that the origin for lateral acceleration response is the time at which the steering-wheel angle change is 50 % complete. This is the reference point from which lateral acceleration response time is measured. The lateral acceleration response time is thus defined as the time, measured from this reference, to first reach 90 % of its new steady-state value.

CSR value of second peak of yaw angular velocity after steering-wheel release $\dot{\psi}_{\max 2}$

According to section 9.3 of ISO 17288-1:2002 (Ref[22])

ISR resonant peak level $D_{\dot{\psi}/\delta_{\dot{H}}}$

Resonant peak level of yaw angular velocity related to steering wheel angle pulse input.

CIR max absolute value $A_{x_{\max}}$ & $A_{z_{\max}}$

The max absolute value in the acceleration response time history of the fore/aft and vertical vibration at the outboard of the driver seat track.

DURABILITY

A durability analysis can consist of a static (or quasi-static) durability analysis, or a modal stress recovery (MSR) durability analysis, depending on the impact of the dynamic effects of the structure on the fatigue life. H. Riener, D. Peiskammer and W. Witteveen [7] have proved that modal stress recovery is more accurate than quasi-static durability analysis, thus using MSR in this current study. The analysis process is presented in detail in later section.

In this study, the lower control arm of the front suspension is the durability analysis object. In order to simplify optimization problem, fatigue life is considered as the constraint. The requirement for the optimization is that the fatigue life of the front LCA does not fall under the 90% of its initial value.

COMMON DESIGN VARIABLES

Bushings in a vehicle suspension system reduce the NVH received from road surface while the vehicle is in motion. Bushings have great impact on handling, ride and fatigue life of vehicle.

Six important elastomeric bushings are chosen as design-variable (see Figure 7), which are front bushing and rear bushing in lower control arm of front suspension, and strut upper bushing of front suspension, and bushing connected to body in rear suspension twist-beam, and the upper and lower damper bushing of rear suspension. Each bushing is characterized as six directions: three translational stiffness (two radial and one axial) and three rotational stiffness (two conical and one torsional).