

New Analysis Technology for Automotive Chassis Sensitivity of Steering Wheel Vibrations Induced by Tire-Wheel Imbalance or Brake Torque Fluctuation

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ABSTRACT

A simulation model for analyzing vehicle chassis sensitivity for the vibrations caused by tire-wheel imbalance and brake torque fluctuation has been developed. Nonlinear elements, such as detailed steering power assist system, frictions and nonlinear bushings are taken into consideration. The simulation model is extensively correlated with laboratory excitation tests in both frequency and time domain. The model is then used to analyze the mechanism of chassis sensitivity, and as a result, contributing chassis vibration modes and influential chassis factors are identified. The analysis results are further correlated with vehicle on-road tests with specification changes in critical factors. Reliable correlation between simulated and measured trends is achieved.

1. Introduction

Chassis vibrations induced by tire-wheel imbalance and brake torque fluctuation are phenomena that affect overall vehicle driving comfort. For purposes of this paper, the vibration of the steering wheel and the chassis system caused by tire-wheel mass imbalance is called “steering shimmy” or “shimmy”, while the vibration induced by braking torque variation during a brake maneuver is referred as “brake judder” or “judder”. Steering shimmy and brake judder are the combined effect of the transfer path characteristics of the chassis system and the excitation force input from the tire-wheel or brake system.

In the typical transfer path for steering shimmy and brake judder, the excitation forces are initially transmitted into the suspension through the front wheel spindles, exciting the vibrations of the suspension system. As the primary path from the suspension to the steering system, a majority of the vibration energy is transferred to the steering rack through rigid body motion. As the secondary path, vibration energy is also transmitted to the steering gearbox through structural flexible body motion. The vibration is then transferred to the steering wheel

through the steering system. For a certain level of excitation input, the vibration can be either amplified or attenuated along the transfer path from the road wheel to the steering wheel, depending on the characteristics of the chassis system.

In the automotive industry, some efforts have been made in analyzing the transfer path of steering shimmy and brake judder^{(1)-(4), (7)}. However, many of the existing analysis tools or methods can not produce satisfying results, in terms of nonlinear characteristics, due to oversimplified modeling. Some of the factors which are often overlooked include detailed power steering system, system frictions, and the nonlinearity of bushings, as well as detailed correlation with experimental measurement.

To better understand the transmission mechanism of the chassis system and to help develop design strategies for reducing steering and chassis vibrations, we set out to develop an effective analysis technology. For analytical purposes, the excitation forces can be decoupled and evaluated separately from the chassis response. Excitation can be measured directly on the vehicle, i.e. using an instrumented road wheel, or on a component level using a brake dynamometer or similar equipment. The measured excitation forces can then be used as an

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excitation input to the CAE models of chassis response.

In this research, the following specific activities are undertaken:

- (1) A multi-body simulation model is developed for a nonlinear, coupled front chassis system which uses a MacPherson strut suspension and hydraulic power steering. Detailed steering and suspension systems, bushing-isolated sub-frame, frictions of joints, the displacement dependence of bushings, and a simple model for the tire and wheel are included in the modeling. The primary path, that is, the rigid body transfer path, is mainly considered.
- (2) The model is validated by laboratory excitation tests under various levels of excitation force in both frequency and time domain.
- (3) The simulation model is used to analyze the transmission mechanism of vibrations in the chassis transfer path. Contributing chassis vibration modes are discovered and important chassis factors are identified.
- (4) The analysis results are further correlated with vehicle on-road tests with specification changes in some critical factors. Comparisons of simulated and measured trends of steering shimmy and brake judder are illustrated.

2. Modeling Approach

A multi-body simulation environment is used to build the simulation model, due to the low-frequency nature of steering shimmy and brake judder phenomena and the software capability of assembling different sub-systems. Custom model templates and sub-systems are created to capture critical elements affecting vibration behavior⁽⁸⁾.

2.1. Model of Front Chassis System

A multi-body-simulation model for the nonlinear, coupled vibration characteristics of the full front chassis is created by the inclusion of the sub-system models of steering, suspension, and a virtual test rig, as illustrated in Fig. 1.

A custom model template is built for the power steering system. Mechanisms such as rack-pinion gear, steering column, universal joints, a rubber coupler and steering wheel, are modeled first. The model is then made to represent hydraulic power-assisting system (HPS) or electric power-assisting system (EPS), as needed. As an example, HPS is only referred hereafter. The power-assisting system is modeled as a flow-source-valve-cylinder system^{(5), (6)}. The control valve consists of two pairs of orifices, of which the flow cross-sectional areas are functions of the angular displacement of the control valve. Fluid inertia and compressibility are considered in the system connection pipes. Compressibility of the fluid in the steering cylinders is taken into consideration. Possible backlash in the steering rack-pinion gear is also modeled.

During the modeling of the suspension system, a

MacPherson strut type of suspension is considered. The stabilizer bar, sub-frame and powertrain mass and inertia properties are also included. The sub-frame attachment can be selected between bushing-isolated and rigidly-mounted on the vehicle body by a switching parameter.

A virtual test rig is created which can accommodate suspension preload from braking or drive torque, the mass and stiffness of the tire-wheel assemblies, as well as multiple degree-of-freedom excitation forces to the spindles. Friction-free bearings are modeled and included to support the tires on the ground. Measured excitation forces are used to define the excitation forces.

2.2. Model Parameters

As one of the nonlinear elements of the chassis system, friction is important for low-frequency vibration phenomena. During this study, frictions in the steering and suspension systems are included in the simulation model, such as:

- (1) Friction between the rack and the gearbox housing.
- (2) Friction torque between the gearbox input shaft and the gearbox housing.
- (3) Friction torque between the pinion and the gearbox housing.
- (4) Friction torque between the input shaft and the pinion.
- (5) Friction torque of steering column universal joints.
- (6) Friction torque at the ball joint between the knuckle and the lower control arm.
- (7) Friction torque at the ball joint between the knuckle and the spindle.
- (8) Friction torque at the ball joint between the knuckle and the tie rod.
- (9) Friction torque at the ball joint between the tie rod and the rack.
- (10) Friction torque between the drive shaft and the spindle.

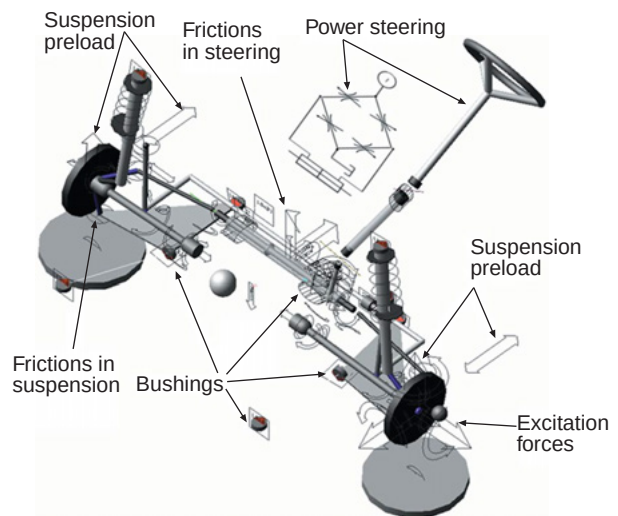


Fig. 1 Multi-body-simulation model of front chassis system

In order to ensure numerical simulation stability and replicate a realistic friction profile, a particular velocity-dependent friction law is used to describe the frictions. Figure 2 illustrates a comparison of the velocity-dependent friction and Coulomb friction. Obviously, Coulomb friction preserves the discontinuity at null relative velocity, and the discontinuity will cause unstable solution in simulation. Meanwhile, Coulomb friction is against physical phenomena where kinetic friction is usually lower than static friction. The proposed velocity-dependent friction profile allows the static friction to exponentially decay to the kinetic one, while guarantying continuity at zero relative velocity. The kinetic friction is defined as 70% of the static friction (a percentage commonly accepted for engineering friction assumption), and the static friction is determined as the measured break-through level.

Steering gearbox mounting bushings and suspension bushings are characterized under appropriate displacement. Stiffness and damping coefficients for the bushings are determined by experimental measurement for various frequencies and displacements. For a suspension hydro-bushing, both displacement and frequency dependence must be considered. However, for a conventional rubber bushing, within the normal frequency range of shimmy and judder phenomena, the stiffness and damping coefficients vary significantly with displacement but do not change greatly with frequency. In such a case, displacement-dependent bushing model is appropriate.

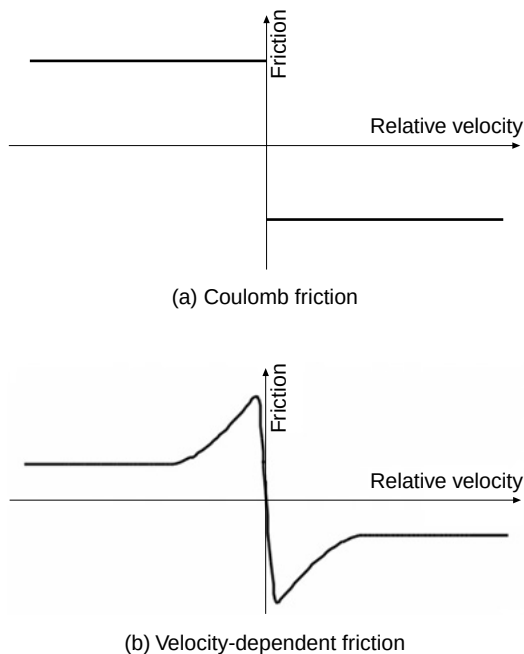


Fig. 2 Velocity-dependent friction profile used in simulation and its comparison with Coulomb friction

In order to understand the major contributions of excitation forces and quantify vertical and longitudinal components of preload for steering shimmy and brake judder, vehicle tests are performed on a SUV to measure the six-degree-of-freedom forces acting on the front wheels. During the tests, steering shimmy is reproduced by driving the vehicle at a speed at which maximum steering wheel vibration is detected with a known unbalanced wheel, while brake judder is duplicated by decelerating the vehicle from high speed at a certain level of deceleration with an induced front brake torque variation.

Let us define the coordinate system of the forces (F_x , F_y and F_z) and moments (M_x , M_y and M_z) acting on the front wheels as that X -components are in the vehicle longitudinal direction with positive toward front for the left wheel and toward rear for the right wheel, Y -components in the lateral direction with positive toward outbound of the vehicle, and Z -components in the vertical direction with positive toward up.

Figure 3 shows the force measurement results for steering shimmy. It is found that the peak of steering shimmy appears at around 14Hz-16Hz for the vehicle under test. It is revealed that F_x , M_y , M_z and M_x are the major contributions for the excitation input of steering shimmy. Similar results are obtained for brake judder phenomena. In addition, suspension preload forces and

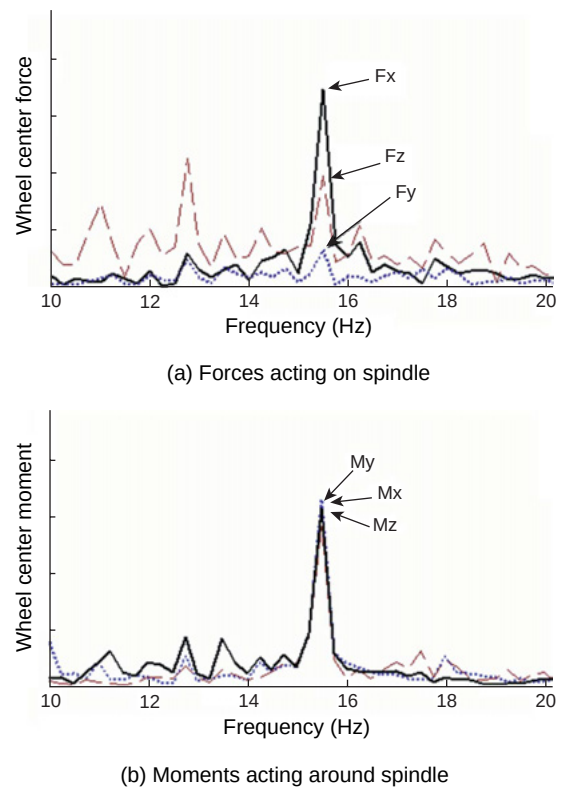


Fig. 3 Excitation forces for steering shimmy

vehicle weight transfer during brake judder measurement are detected by examining the measurement data in time domain.

The experimentally determined excitation forces, such as F_x , M_y , M_z and M_x for shimmy mode, are then inputted into the simulation model through the definition of a six-component general force at the spindles. Since the purpose of this study is to analyze the chassis sensitivity of steering shimmy and brake judder from the spindles to steering wheel, it is appropriate to excite the chassis system at the spindles during simulation. The generation mechanism of the excitation forces and the transmission characteristics of the brake system and tires are not required and discussed here.

3. Validation by Excitation Test

To validate the front chassis model, a laboratory excitation test is needed to reproduce the vibration behavior of the steering wheel and chassis systems during shimmy and judder modes.

There are several existing excitation test methods used by the automotive industry. In these existing methods, the vehicle under test is commonly excited through tire contact patch, which results in limitations such as the difficulty in applying suspension preload, the limitation to single-directional excitation, the need for understanding tire characteristics to derive the transmissibility of chassis system from test, and the variation of excitation input.

A new excitation test method is designed here to address these issues. The basic test capabilities with the new method, as shown in Fig. 4, are as follows:

- (1) Vertical and longitudinal preload to front suspension.
- (2) Multi-directional excitation input at spindle.
- (3) Direct measurement of chassis transmissibility from spindle to steering wheel.
- (4) Stable excitation and repeatable results.

During the excitation testing, the brake is locked for brake judder mode testing, while unlocked for steering shimmy mode testing. The validation work is mainly

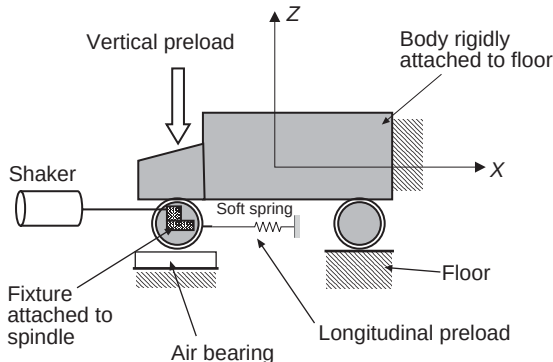
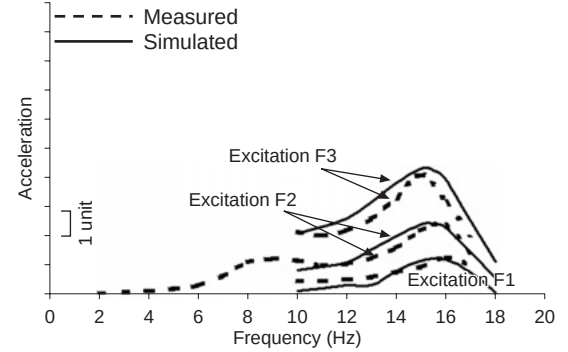


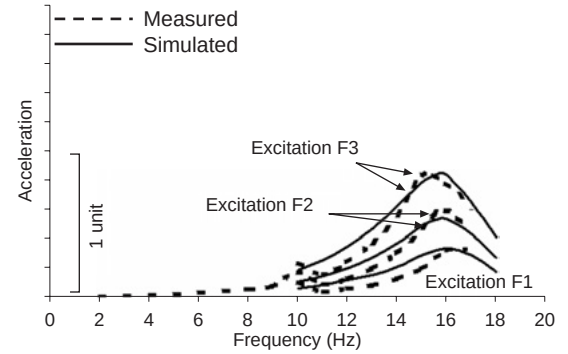
Fig. 4 Laboratory excitation method

focused on the rigidly-mounted sub-frame condition.

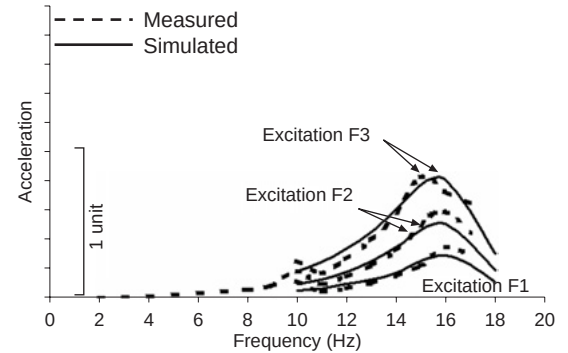
The simulation model is validated by frequency sweep tests under various levels of excitation amplitude. Figure 5 shows the simulated and measured accelerations at steering wheel and other different locations under three different levels of excitation input, for steering shimmy mode in frequency domain. A very good agreement is shown between the simulation and measurement of chassis vibrations under various excitation forces, in terms of both peak amplitude and peak frequency. Similar agreement



(a) Steering wheel



(b) Steering rack, lateral



(c) Knuckle, lateral

Fig. 5 Simulated and measured accelerations for shimmy mode under different excitations (frequency domain)

is revealed for brake judder mode, as illustrated in Fig. 6. Those results confirmed that the simulation model can predict the peak amplitudes and peak frequencies for both steering shimmy and brake judder with high accuracy.

Time waveform data at judder peak frequencies for three different excitation input forces are shown in series in Fig. 7. It is seen that the simulation can yield waveforms very similar to the measured ones. Meanwhile, the nonlinear vibration characteristics of the chassis system are revealed on the non-sine-waveform

response of the steering rack and knuckle to a pure sine waveform force input. As shown in Fig. 7 (c) and (d), these nonlinear characteristics are also predicted by simulation very well. Similar results are achieved for steering shimmy mode.

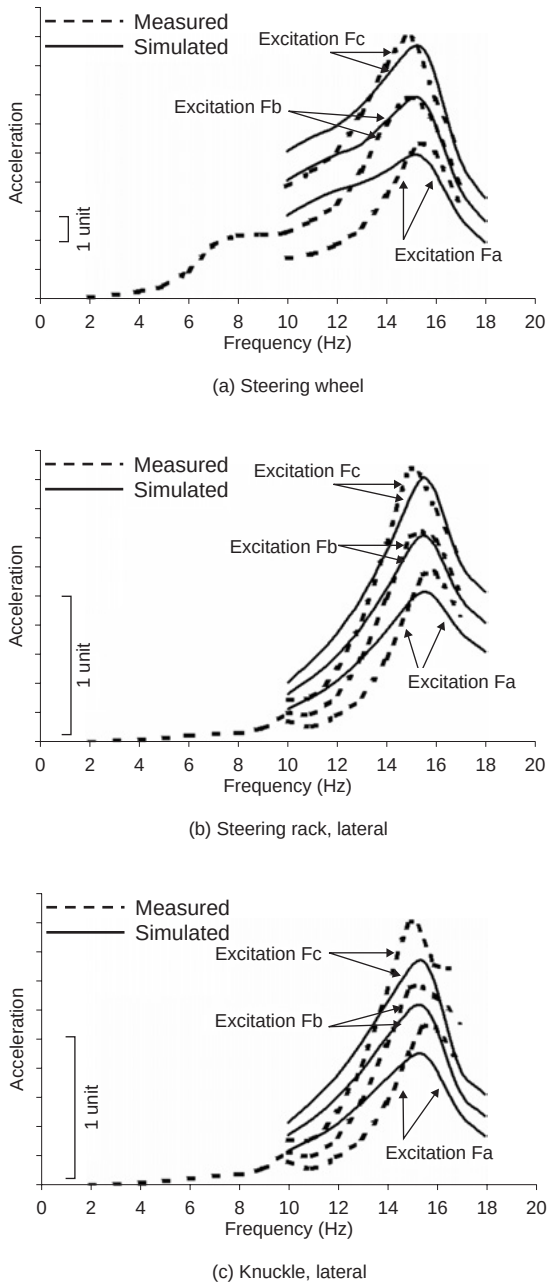


Fig. 6 Simulated and measured accelerations for judder mode under different excitations (frequency domain)

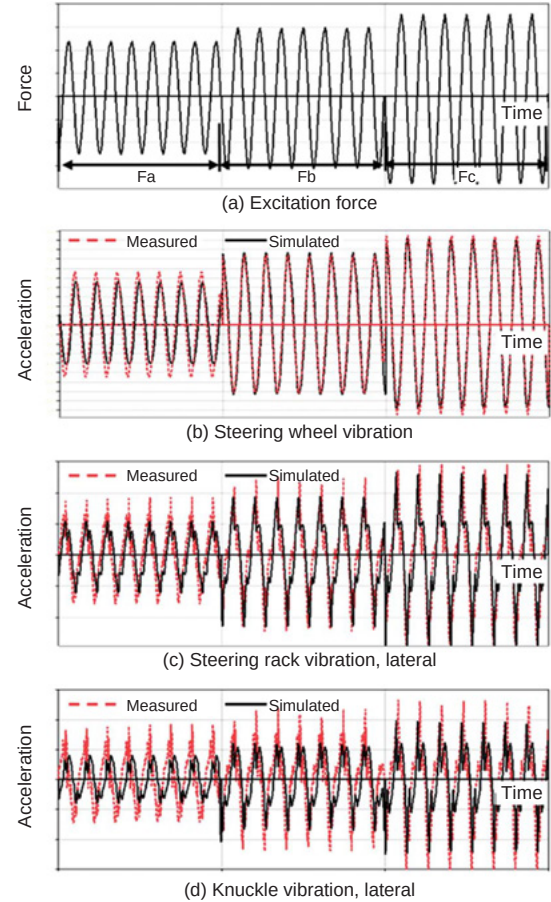


Fig. 7 Simulated and measured acceleration time waveforms for judder mode under different excitations

4. Mechanism Study

4.1. Influential Chassis Modes

The simulation model is used to identify the dominant chassis vibration modes and how they are coupled. This is done by a linear analysis around the equilibrium operating point. As the result, the following two critical chassis modes are discovered in the frequency range of interest for both steering shimmy and brake judder modes⁽⁹⁾:

- (1) Suspension fore-aft out-of-phase mode.
- (2) Steering wheel rotational mode.

The overall chassis sensitivity is the combined effect of these two dominant vibration modes. Modal coupling

phenomenon may occur between those two modes, if they are not separated appropriately. That is, if the frequencies of the modes are too close, the vibration will be amplified twice due to the resonances in both suspension and steering systems at the similar frequencies. This kind of modal coupling mechanism will eventually cause very high level of vibration at the steering wheel. Figure 8 shows the simulation results for the following three different conditions, respectively:

- (1) Independent response of the steering mode.
- (2) Independent response of the suspension mode.
- (3) Overall response of the two chassis modes.

It can be seen that in the case of the independent response of either the steering mode or the suspension mode (this can be achieved by moving the other mode far away from the frequency range of steering shimmy/brake judder), the steering wheel vibration is very low. However, when both of the chassis modes exist, the level of steering wheel vibration becomes much higher, due to the modal coupling mechanism. Building on this understanding to lower the chassis sensitivity of steering shimmy and brake judder, design emphasis can be put on the reduction of the peak amplitude of suspension fore-aft out-of-phase mode, the increase of the peak frequency of the suspension mode, or the reduction of the peak frequency of steering wheel rotational mode.

4.2. Influential Chassis Factors

Now that the chassis model has been validated, it can be used to perform a virtual design of experiment (DOE) to efficiently identify what factors have the greatest influence on the peak amplitudes and frequencies of steering shimmy and brake judder.

Two objective functions for the DOE process are defined to capture the effect of factors on steering shimmy and brake judder. Objective 1 is defined as the maximum amplitude of steering wheel vibration, while Objective 2 is referred as the frequency at which steering wheel vibration peak occurs, during a frequency sweep run. The frequency range is set wide enough to cover frequencies

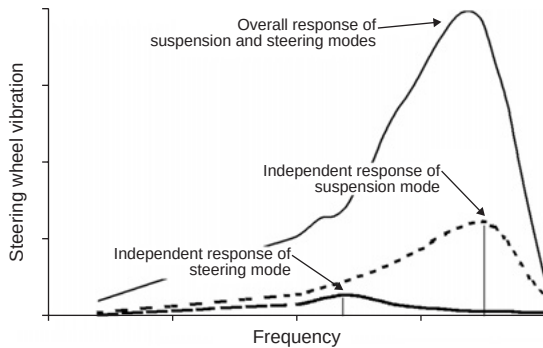


Fig. 8 Coupling mechanism of chassis modes to generate final vibration at steering wheel

of possible factor combinations. Quadratic response surface method is applied to create the design matrix for the simulation runs and to fit polynomials to the results of the simulation.

Based on the virtual DOE analysis, influential factors are identified. Among the influential factors are the locations of suspension geometric points, the mass and inertia of the suspension and steering components, characteristics of the mounting bushings, frictions in the chassis system, pressurized area of the steering cylinder, and steering ratio.

Figure 9 shows the ranked effect, or relative importance, of top influential factors for the peak value of steering wheel acceleration for shimmy mode and judder mode, respectively, while Fig. 10 illustrates the most influential factors for the peak frequency of steering wheel acceleration. It is observed that due to the similarity of chassis transfer path for both steering shimmy and brake judder, the factors exhibit similar importance rankings for both modes.

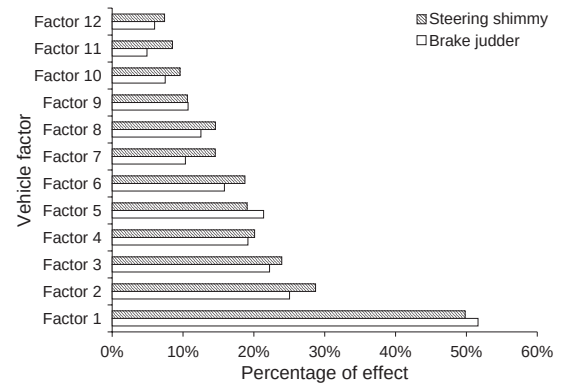


Fig. 9 Top influential factors and their effect on maximum amplitude of steering wheel acceleration when their specifications change from -10% to 10%

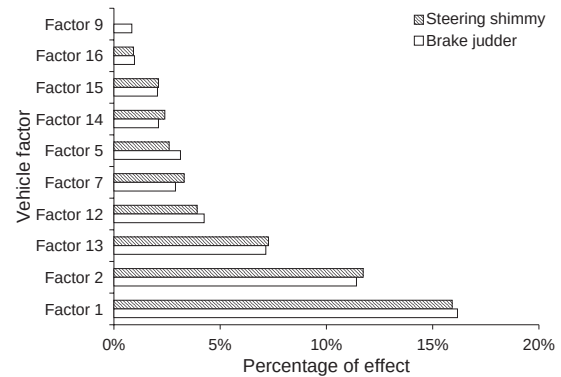


Fig. 10 Top influential factors and their effect on peak frequency of steering wheel acceleration when their specifications change from -10% to 10%

5. Correlation with Vehicle Tests

The simulation results are further confirmed with vehicle on-road shimmy and judder tests. Several top influential factors are selected, and test parts are accordingly designed to deviate from nominal specifications. The relative changes of steering shimmy and brake judder with the change of specifications of those factors are measured by vehicle on-road tests and then compared with simulation results. Based on the repeatability of on-road tests, the target for the correlation between test and simulation is set to 80%.

Figure 11 shows a correlation between measured and simulated percentage change of shimmy and judder due to the change in part specifications. It is shown that the correlation between the simulated and measured results is within 80% for almost all factors considered, for both steering shimmy and brake judder modes. Therefore, it

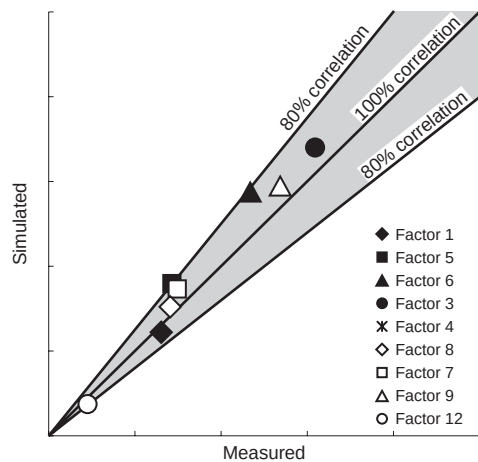
is demonstrated that the simulation can predict the effect of specification changes with good accuracy.

6. Conclusions

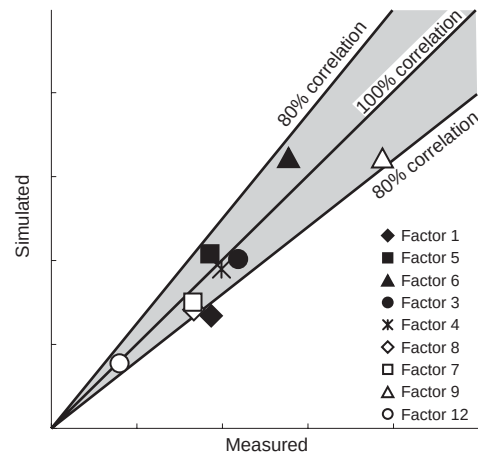
A new analysis technology for analyzing vehicle chassis sensitivity for the vibrations induced by tire-wheel imbalance and brake torque fluctuation has been established. A multi-body simulation model was created for a coupled front chassis system consisting of power steering, suspension, bushing-isolated sub-frame, tire-wheel, and a virtual excitation rig. Nonlinear elements, such as detailed power-assisting system, frictions and nonlinear bushings have been taken into consideration. The model has been extensively validated by laboratory excitation tests.

The simulation model was then used to analyze the transmissibility mechanism and quantify the relative importance of chassis factors. Contributing chassis vibration modes and influential chassis factors were efficiently identified. To further verify the simulation results, vehicle on-road tests were conducted to measure the effect of specification changes of key factors on shimmy and judder. Reliable correlation has been demonstrated between simulated and measured results.

The simulation tool developed here can be used to help quantify the effect of proposed specification changes through simulation during vehicle development, and has made it possible to enhance development efficiency and vehicle performance. In addition, the knowledge gained here about chassis modal coupling and relative importance of chassis factors are useful to establish design enhancement strategies for steering shimmy and brake judder in future models.



(a) Percentage of steering shimmy change



(b) Percentage of brake judder change

Fig. 11 Correlation of simulated and measured change in steering shimmy and brake judder with change of factor specifications

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