A study of Vehicle – Road Interaction Using Abaqus Co-Simulation Approach

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Abstract:

Early prediction of road loads is vital in vehicle development program to allow maximum flexibility for optimum design of structure. In this paper, Abaqus co-simulation based approach is explored and successfully applied to study vehicle-road interaction. The co-simulation approach provides user with tremendous flexibility in modeling vehicle system. The power of explicit and implicit codes can be unleashed to extract maximum benefits out of them. This combination allows user to simulate complex elements of system without putting over burden on system requirements. The ease with which highly non-linear elements like tires and bump stoppers can be represented, free use of substructures, selective sub-cycling are the vital features of this study.

Understanding role of suspension in road load signature is equally important. Early inputs in this regard to suspension designer provide him an opportunity to manage road loads effectively. In this exercise, sensitivity of various suspension elements towards road loads is studied.

In this paper, both these aspects are discussed. It is demonstrated that how changing few suspension parameters can affect road load signature of vehicle. Currently this finding is validated subjectively. Objective validation is planned in the future. The study is planned to extend further to cover various suspension types and vehicle configurations to come up with design rules for suspension design.

Introduction

Vehicle – road interaction has been area of interest in automotive field for long time. Knowing wheel loads and accelerations is vital in vehicle design process as it governs durability and NVH performance of the vehicle. Being able to measure or predict wheel loads early in design cycle provide opportunity to optimize and fine tune design in true sense. Huge amount of efforts were spent by automobile engineers in the quest of understanding this aspect. Both experimental and virtual techniques are being developed and undergoing continuous improvements day after day.

Experimental technique, though accurate, require full scale prototype hence usually provide inputs late in the design cycle. One can make use of these inputs in fine tuning the design but little opportunity is available in design cycle at latter stages to optimize the deign, hence one can not make most out of the available inputs.

People working in virtual domain confronted with different set of challenges. Virtual techniques can be applied early in design cycle, but achieving reasonable quality of results, calibration of virtual models and managing system resource requirements are major challenges to face. Various approaches based on implicit and explicit solution techniques are available and applied with some success by various OEMs. Currently there are mainly two approaches being used in the industry - using multi-body dynamic analysis technique with abstract representation of tires and second one is to use finite-element solvers with physical representation of tires. Both of these techniques have their own advantages and limitations. MBD techniques are very efficient but they require extensive testing of tires to calibrate tire model, where as tire calibration is relatively simpler in FE based technique but they were computationally quite expensive. Abaqus co-simulation approach provides elegant answer for this problem. Efficiency of FE based approach can be increased significantly with co-simulation technique.

To understand role of suspension elements in vehicle-road interaction, event based simulations are performed. Findings of event based simulations are confirmed on small four-post patch test. This knowledge is applied to get insight in to why two different vehicles of similar specifications have significantly different road loads under same four-post inputs. In this paper, author's work in this area is discussed.

Complexities involved and techniques used to resolve the issues

Apart from the scale of the problem, one has to deal with various complexities involve in the analysis. In this section, various such issues are discussed along with techniques used to resolve these issues –

1. Efficiency of technique:

Two main aspects one has to consider in vehicle-road interaction simulation are to be able to capture vehicle's dynamic response to road irregularities with reasonable accuracy and managing highly non-linear behavior of few system elements. Unlike highly dynamic events like vehicle crash, vehicle's response to road excitations is controlled by suspension modes and few lower frequency structural modes. Hence it is sufficient to have abstract model of vehicle system which can represent only these modes accurately. Implicit time integration schemes in combination with sub-structures are highly efficient in such cases. Sub-structures allow abstract modeling of large systems. Solution degrees of freedom required to be retained are decided by physical connection points of sub-systems and additional modes to be retained for representation of few important lower modes. Sub-structure reduces solution degrees of freedom by significant amount without loss of accuracy (e.g. few million dofs in vehicle model can be reduced to few hundred dofs). While using substructures, linear assumptions are implied and only portion of vehicle system which is suppose to behave in linear manner can be abstracted as sub-structures. In this study, BiW, frame, suspension control arms/links and rear axle etc are represented as substructures.

There are few highly non-linear elements in the system like bump stoppers and tires. Capturing highly non-linear behavior of these elements (e.g. large scale tire deformations along with rapid changes in contact conditions with road) is difficult to handle using implicit solvers. They pose lot of numerical issues and they may demand very small time steps. Explicit time integration methods are known to be very effective and efficient in handling highly non-linear dynamic events. For the similar reasons, in the past, explicit approach was being used for such simulations. But solving entire problem using explicit technique is very expensive and tedious task as –

a. Explicit scheme is conditionally stable, and maximum time step one can use is governed by stability constraint. Smallest element in the model controls the maximum time step one can operate with. This puts lot of restrictions on model building.
b. Use of substructures is not possible/very limited. This means one has to work with full scale FE model of large vehicle system. Hence all the benefit of efficiency and scalability of explicit technique get offset by large solution degrees of freedom of full vehicle model.

Hence, highly efficient implicit method alone is not useful in handling the complete problem because of its limited ability to handle highly non-linear and dynamic behavioral aspects of problem. On the other hand, fully explicit method can provide solution but it is highly inefficient as one has to work with large full vehicle models. Because of these limitations FE based technique was not competitive enough with MBD based technique in the past.

New co-simulation approach provides elegant way to overcome these limitations. Vehicle system is partitioned so that large part of the model is handled as substructures using implicit technique and remaining small but highly non-linear part using explicit technique. Figure 1(a) and 1(b) show implicit and figure 1(c) show explicit partitions of the model used in the study.

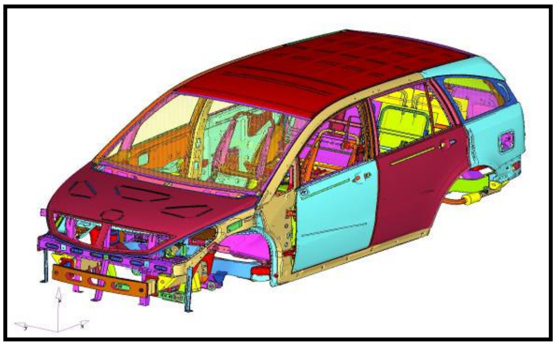


Figure-1(a): BiW and Frame model

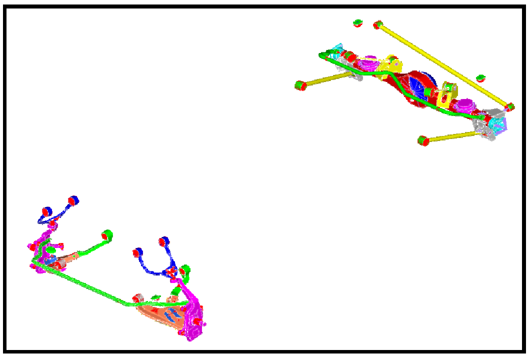


Figure-1(b): Suspension links, ARB, axle model

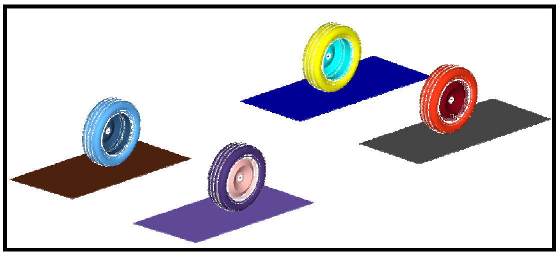


Figure-1(c): Wheels and four post model

Co-simulation approach improves efficiency of conventional full explicit approach significantly and provides lot of flexibility in building models. This method allows user to unleash benefits of both implicit and explicit solvers while avoiding their limitations. Table 1 shows some comparison of efficiency of co-simulation approach over full explicit approach.

Resource requirement for typical VPG simulation
(event of 5 sec)

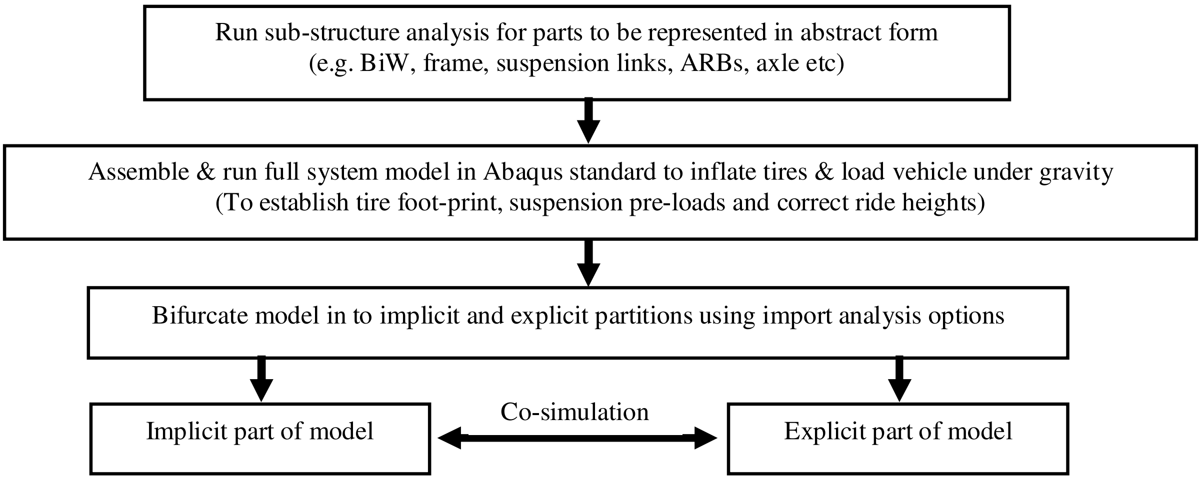
	Model size	No. of CPUs	Event time	CPU time
Typical crash simulation	~ 12,00,000 nodes	16	~ 0.15 sec	~ 20 hrs
VPG simulation with fully explicit approach with complete vehicle model.	~ 12,00,000 nodes	16	~ 5 sec	~ 600 hrs
VPG simulation with combined implicit-explicit approach with super-elements	~ 1,00,000 nodes	16	~ 5 sec	~ 5 hrs

Note – These estimates are indicative numbers.

Table-1: Comparison of solver time for event simulation

2. Model preparation and managing sequence of events :

Vehicle-road interaction analysis is actually a chain of simulations. Following flow chart show logical steps one has to follow during course of complete simulation cycle.



Sub-structures are prepared using Craig-Bampton method for parts like BiW, chassis, suspension links, anti-roll bars and rear axle. Apart from connection dofs, all the natural modes up-to 200 Hz are retained in sub-structures. Wheels, four post actuators/road and bump stoppers are modeled physically. Suspension springs, dampers and bushes are modeled using SPRINGA elements. Various joints between suspension links, chassis, wheel etc are modeled using CONNECTOR and JOINTC elements.

Complete model is assembled and analysis is performed under gravity loading to establish correct static wheel reactions, suspension preloads, tire foot-print and bump stopper ride heights. Figure 2 (a) and 2(b) deformed states of complete model before and after gravity loading.

Model is bifurcated into implicit and explicit parts using import analysis options. Co-simulation between these two models is set using sub-cycling option. Sub-cycling allows user to use different time incrimination schemes for implicit and explicit runs, allowing implicit solution to run with much bigger time steps which further improves the solution efficiency. Wheel centers are selected as Interface nodes for implicit and explicit parts.

3. Preparing tire and bump stopper models :

Tire and bump stoppers are highly non-linear elements in of the suspension system. Accuracy of simulation results largely depends on the representation of these elements in the model. Abaqus provides special features like embedded elements and REBAR reinforcements for tire modeling. Construction of tire is very complex and representation of all critical elements like body plies, belts, reinforcement cords etc is difficult task without abstract representation using embedded elements and REBARS. Figure – 3 (a) show typical tire construction. Figure – 3 (b) shows FE representation of tire.

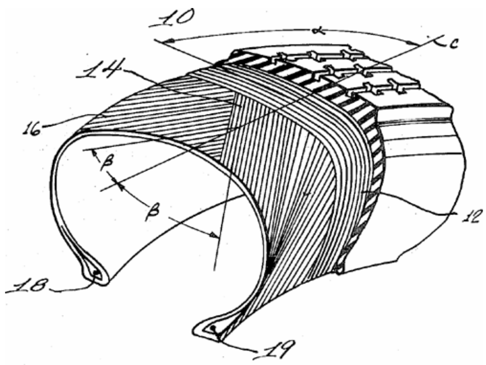


Figure 3(a): Typical construction of tire

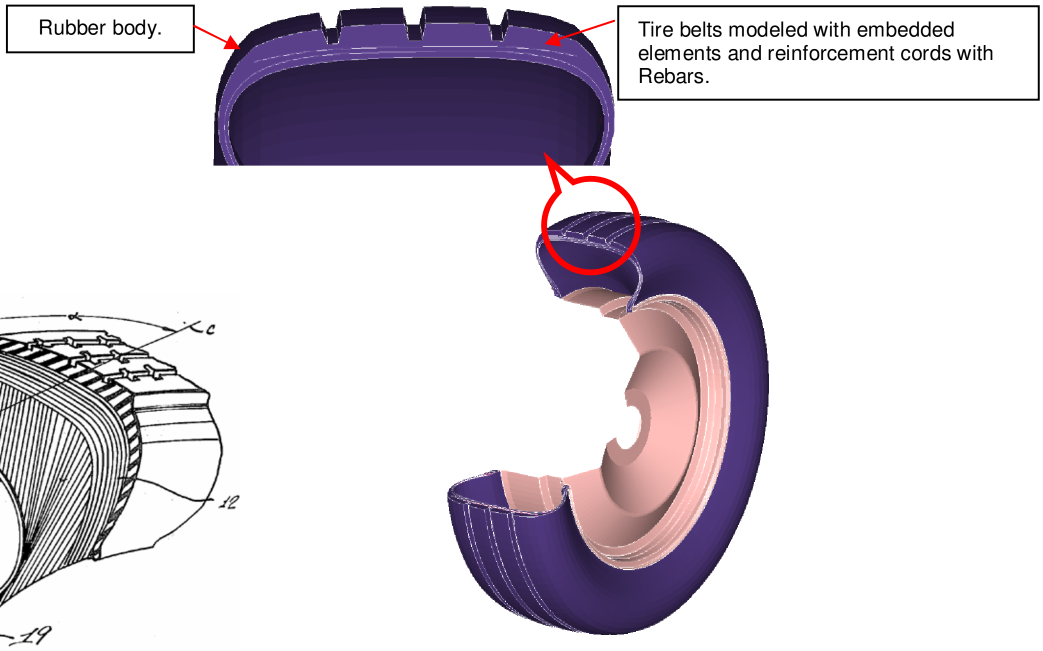


Figure 3(b): Tire FE model

Tire model is build with focus on having reasonable radial stiffness and damping representation under static and dynamic conditions rather than trying to capture all the minor details of tire construction.

Representation of bump stopper is done using Hyper-elastic material model for rubber part. Material constants are generated to match force-deflection behavior with experimental data. Physical model of bump stopper is used instead of simple non-linear spring representation to avoid issues arising due to complex motion of control arm resulting into awkward behavior of bump-stop spring and flipping of spring ends happening at extreme bump stopper compressions.

Event based simulations:

To understand significantly different road load behavior exhibited by two vehicles of similar specifications, sensitivity analysis is performed using simple road bump event. Variables used in the study are –

1. Obstacle dimensions (height and width)
2. Suspension spring stiffness
3. Un-sprung mass
4. Damper characteristics
5. Vehicle speed
6. Bump stopper ride clearance

Detailed comparison of results is made to study interaction within suspension-elements and sensitivity of road loads with selected variables to come-up with conclusions. Table – 2 shows all the variations used in the study –

Variables	Iteration 1	Iteration 2	Iteration 3	Iteration 4	Iteration 5	Iteration 6	Iteration 7
Obstacle height (mm)	**50**	80	80	80	80	80	80
Obstacle width (mm)	**100**	110	110	110	110	110	110
Spring stiffness	Reference	Reference	**+ 20 %**	Reference	Reference	Reference	Reference
Un-sprung mass	Reference	Reference	Reference	**- 15 %**	Reference	Reference	Reference
Damper stiffness	Reference	Reference	Reference	Reference	**+ 10 %**	Reference	Reference
Vehicle speed (kmph)	30	30	30	30	30	**40**	30
BS ride height	Reference	Reference	Reference	Reference	Reference	Reference	**+ 10 mm**

Table 2: Simulation parameters used for the study

1. **Results for iteration 1 and 2:** It can be observed that wheel force is increased with obstacle height as expected and one can also observe significant rise in bump stopper activity. There is marginal change in the spring and damper activity during the event. Refer figure 4(a) for details. Increase in forces on chassis is evident in simulation with bigger obstacle size. This can be linked directly with the higher bump stopper activity as change in spring and damper behavior is very small. One can also visualize small phase lag in highest wheel force and highest force transferred to chassis. Refer figure 4(b) for details.

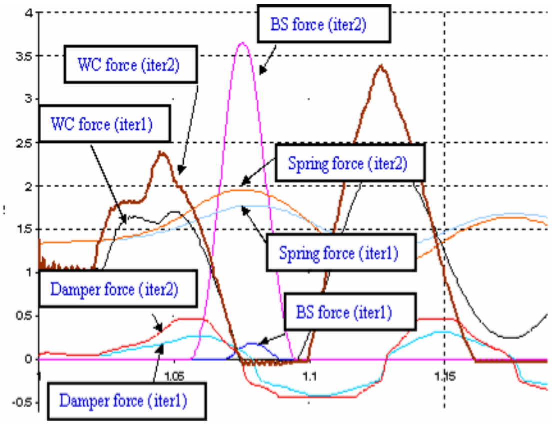


Figure 4(a): Wheel force and suspension element force phasing

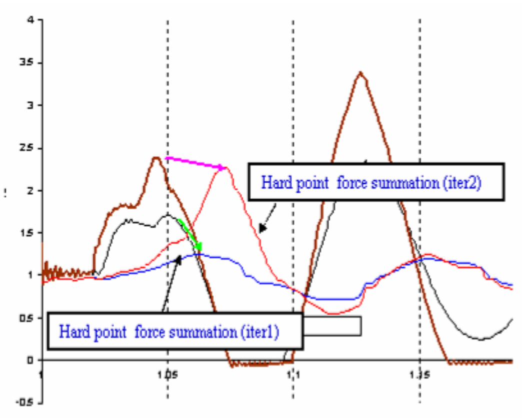


Figure 4(b) : wheel and chassis force comparison

Results for iteration 2 and 3: Magnitude of *W*heel force remain unaffected by 20 % higher spring stiffness, damper activity remain unaffected where as bump stopper activity reduced drastically. Spring force increased as expected. Refer figure 5(a) for details. Reduction in forces on chassis is evident with stiffer spring. Which may sound odd, but if one can appreciate small reduction in body to wheel displacement due to stiffer spring resulting in reduced BS activity and also the fact that higher initial BS ride clearance on account of stiffer spring results in reduction in transmissibility. Two facts are coming out of this comparisons– Wheel loads are more or less governed by wheel inertia and force on chassis is dominated by BS activity. Refer figure 5(b) for details.

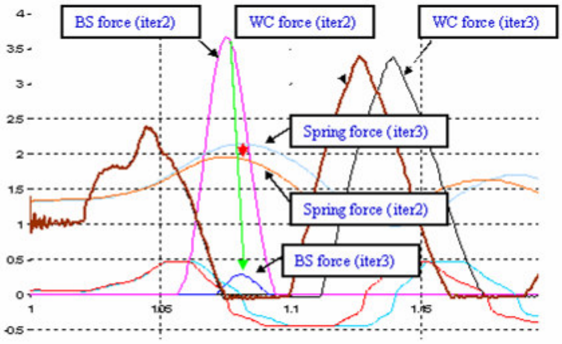


Figure 5(a): Wheel force and suspension element force phasing

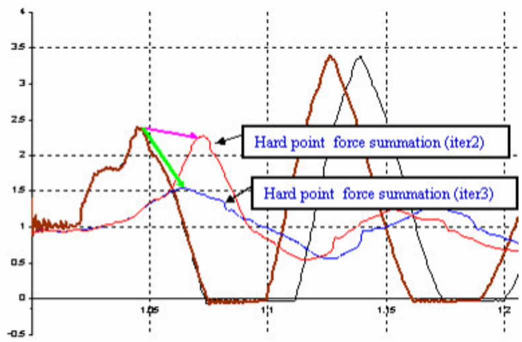


Figure 5(b) : wheel and chassis force comparison

Results for iteration 2 and 4: Wheel force reduced for lower un-sprung mass configuration again emphasizing on dependence of wheel loads on wheel accelerations and un-sprung mass. Spring and damper activity remain unaffected where as bump stopper activity reduced by small margin. Refer figure 6(a) for details. Force on chassis is same in spite of lower wheel forces on account of unaffected spring, bump stopper and damper activity. Refer figure 6(b) for details.

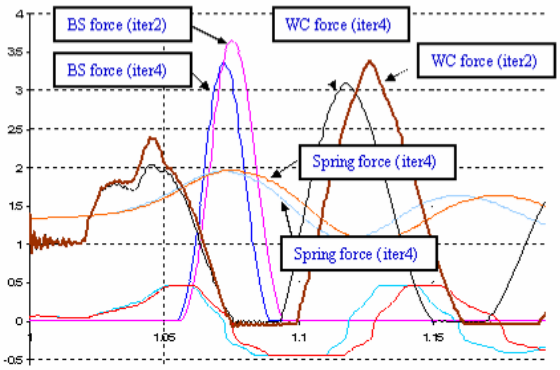


Figure 6(a): Wheel force and suspension element force phasing

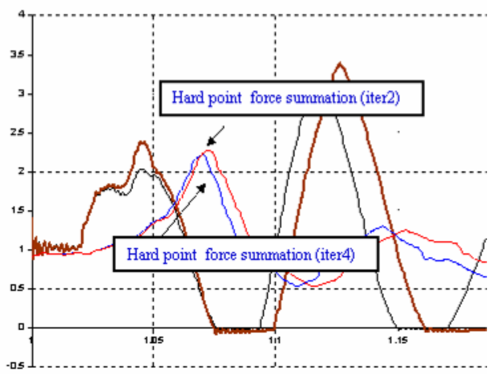


Figure 6(b) : wheel and chassis force comparison

Results for iteration 2 and 5: Wheel forces and spring force remain unaffected. Damper force increased slightly where as bump stopper force reduced slightly. Force on chassis also practically remained un-affected. As upper cut-off force of damper was not changed in this exercise, very little effect of damper is seen on behavior. More realistic variations in damper behavior are required to study effect of damper characteristics in more detail. This is planned in future.

Results for iteration 2 and 6: Wheel force increased with higher vehicle speed again confirming its dependence on wheel accelerations. Rapid force built-up in damper is seen before it trips at cut-off force. Spring force remains unaffected whereas slight increase in BS force is observed. Refer figure 7(a) for details. Higher wheel force does not result in to higher chassis force again re-emphasizing on two separate mechanisms controlling wheel and chassis loads. Refer figure 7(b) for details.

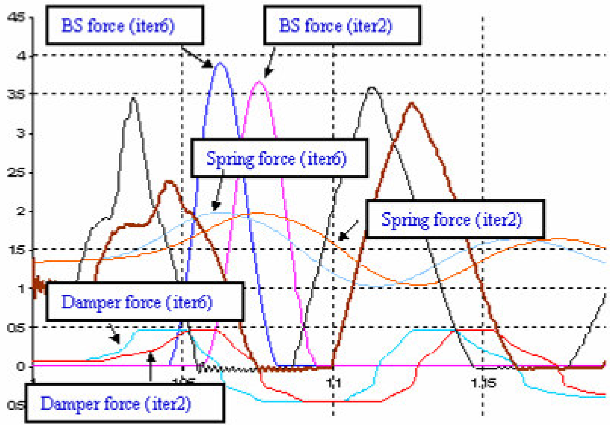


Figure 7(a): Wheel force and suspension element force phasing

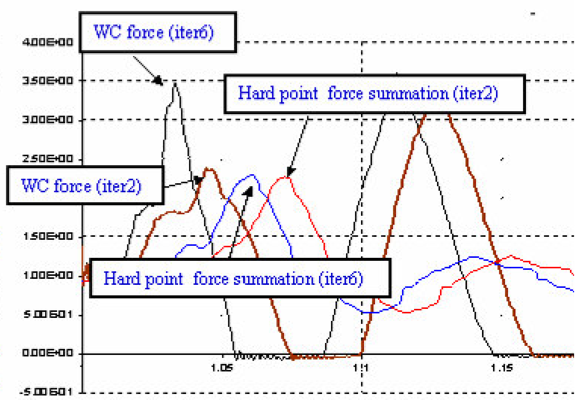


Figure 7(b) : wheel and chassis force comparison

Results for iteration 2 and 7: Wheel force along with spring and damper force remain unaffected. Bump stopper force reduced by significant amount as expected. Refer figure 8(a) for details. Force on chassis is reduced on account of reduced bump stopper activity. Refer figure 8(b) for details.

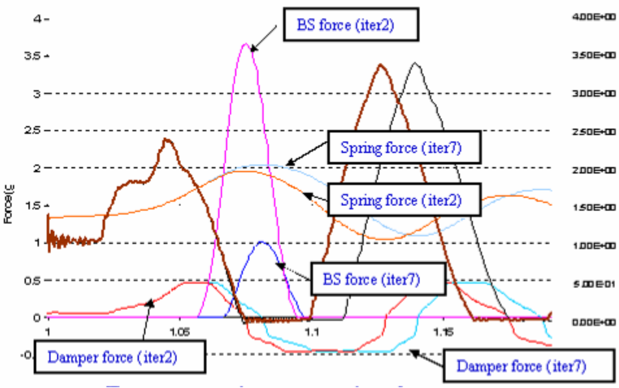


Figure 8(a): Wheel force and suspension element force phasing

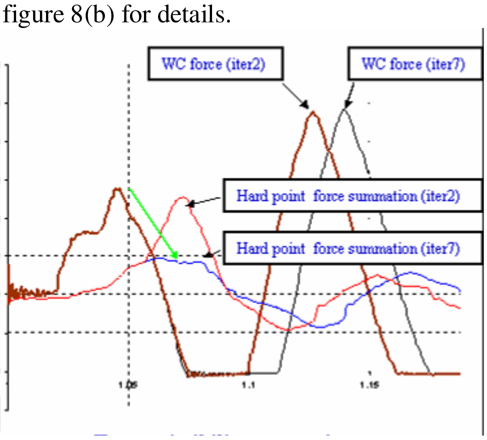


Figure 8(b) : wheel and chassis force comparison

Another important observation is that when wheel lands on ground (indicated by second peak on wheel force graph), there is little change in chassis force irrespective of wheel forces in the various iterations.

In the suspension used in this study, two distinct mechanisms that govern wheel loads and chassis loads are evident. Wheel loads are dominated by wheel accelerations and effective un-sprung mass of the vehicle. Suspension elements like spring and bump stopper have little influence on wheel loads. Where as chassis loads are mainly controlled by damper, spring and bump stopper. Among these elements, bump stopper characteristics and ride clearances are the most sensitive parameters which affect the chassis loads. It is also important to notice that derivation of static load cases based on wheel loads may not be correct and may lead to over conservative estimates in some cases or otherwise as well.

Patch test on four-posts:

To reconfirm that the conclusions drawn based on event based simulations are valid under random transient inputs four–post patch test is carried out. Results are compared with measured data on two vehicles having similar specifications but significantly different load signatures. Table – 3 show differences in two vehicles.

Parameters	**Model A**	**Model B**
Wheel rates	Reference	+ 4 %
Spring stiffness	Reference	+ 20 %
BS ride clearance	Reference	+ 15 mm
Un-sprung mass	Reference	- 30 %

Table – 3: Differences in vehicles under study

Figure 9 (a) and (b) show comparison of wheel and chassis forces respectively.

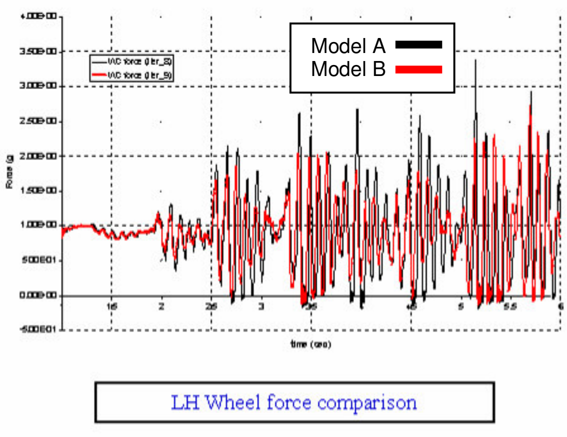


Figure 9 (a)

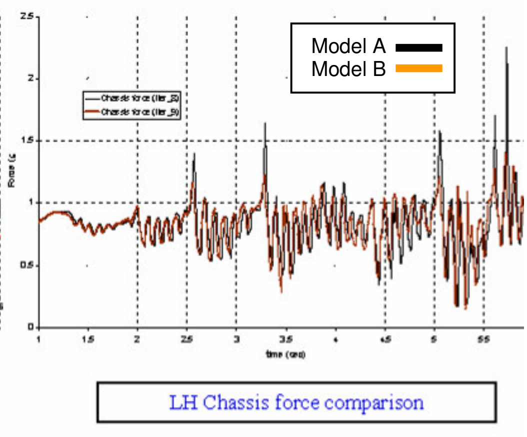


Figure 9 (b)

"Model B" shows lower wheel and chassis forces as compared to "model A" in spite of having higher static wheel rate. This can now be explained as lower wheel forces due to lower un-sprung mass and lower chassis forces due to lower bump stopper activity. Bump stopper force comparison is shown in figure 9 (c).

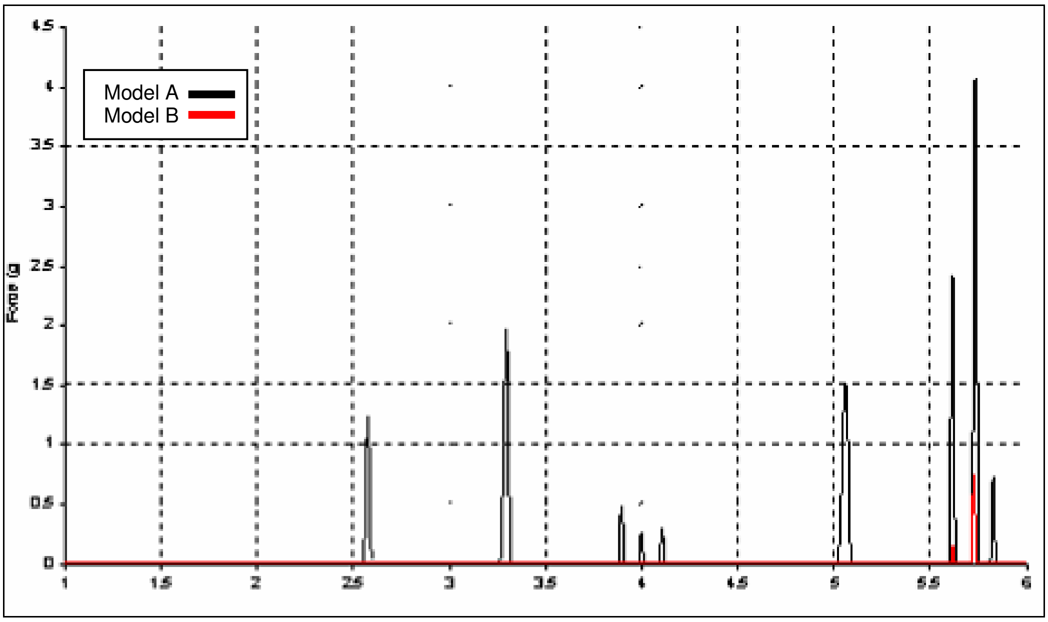


Figure 9 (c): Comparison of bump stopper forces

Experimental measurement also showed that model B produce much less wheel loads as compared to model A. Figure 10 show the plot of wheel loads level crossings. Driver seat accelerations in Model B are also significantly lower which can be attributed to lower chassis forces. But more objective correlation is required to confirm the findings.

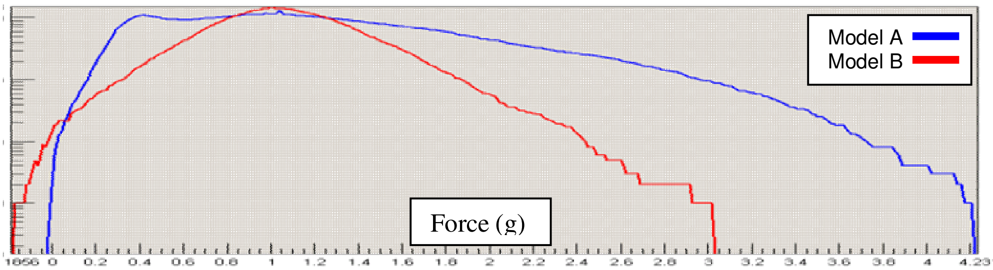


Figure 10

Future scope:

Detail experimental validation exercise is planned to achieve objective co-relation of simulation results. More comprehensive study with various other variables like tire pressure, damper cut off settings, different types of obstacles/roads, and variations in component flexibilities and many more along with different suspension types is planned in future. After successful validation, this method will be used to provide early design inputs to structural analysis.

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