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METHODOLOGY OF VEHICLE OPERATIONAL LOADS ASSESSMENT DURING VEHICLE DEVELOPMENT PROCESS FOR FURTHER CHASSIS AND BODY-IN-WHITE STRENGTH AND DURABILITY ANALYSIS

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ABSTRACT

Various approaches for vehicle operational loads assessment during vehicle development process are exemplified in an SUV vehicle development process. These approaches are combined in one general step-by-step design methodology for simulating the structural behavior in vehicle dynamics, which generalizes the open source information and modern software possibilities. Such methodology serves as a guide for including influence of operational loads into the complicated vehicle development process and helps to assess these load values on different development phases: from concept, where the main supporting structure of the vehicle (BiW or frame) is not yet approved, but first approximation of operational loads is already required, to the further phases, where taking the main supporting structure compliance into account can play an important role in vehicle behavior on the roadway. Getting more accurate operational loads values is important for further consistent chassis and Body-in-White operational loading simulations and effective design optimization such as strength, fatigue and vehicle behavior on the roadway optimization in a variety of different maneuvers.

***KEYWORDS:** Vehicle Dynamics, Operational Loads, Suv, Elasto Kinematic Suspension Characteristics, Modal Neutral Files, Vehicle Subsystems, Main Supporting Structure; Kinematic And Compliance, Full Vehicle Model & Full Vehicle Analysis*

Original Article

INTRODUCTION

Vehicle operational loads must be considered during each of the vehicle development stages to provide appropriate data for Body-in-White (BiW) and chassis strength, fatigue, and, especially, vehicle behavior evaluation and optimization in a variety of vehicle maneuvers on the roadway. Different approaches for operational loads assessment are analyzed in the paper as exemplified by the SUV development process.

The term ‘wheel loads’ implies the reaction forces that are transferred from the roadway to contact patches due to different vehicle maneuvers. ‘Operational loads’ are to be understood as the loads that are transferred from contact patches and suspension further to the vehicle main supporting structure, which is one of the vehicle’s subsystems. For the convenience purposes, it is agreed that vehicle construction is divided into different subsystems: steering, power train, brakes, wheels, main supporting structure (BiW or BiW/frame) etc.

At the stage of vehicle concept development the simplified methods of operational loads assessment are used, which are based only on suspension models with fixed points that connect suspension with the vehicle main supporting structure (so-called interface points) while the main supporting structure itself is not considered in this case. This structure (BiW or frame) is not yet approved at the first stage of development and has to be optimized in accordance with operational loads values in order to improve vehicle characteristics. For further development stages, when the main supporting structure is consistent, the accuracy of these approaches is insufficient to achieve trustworthy results. Operational loads values are of primary importance for further consistent simulations of chassis and BiW operational loading and effective vehicle design optimization.

Recently, due to the development of computer modeling, vehicle manufacturers have been improving the calculation methods and modeling procedures [1-7]. New approaches are based on leading hardware and software capabilities and a wide variety of available experimental data used to validate these codes. The modern simulation methods and software capabilities in the chassis design are well described in specialized literature and software manuals [8-10]. The authors of these works only describe the capabilities and methods, implemented in software products, but there is still lack a step-by-step design methodology. Unfortunately, most of the details regarding modern operational loads assessment approaches are sealed by automobile manufacturers and cannot be found in open sources and research literature. Thus, the information about the vehicle dynamics design is not complete and cannot be used properly as an a step-by-step guide for commercial automobile design.

Given such a background it is necessary to undertake research to define the details concerning models development and simulation processes. Most of the difficulties arise when more accurate operational loads prediction is required. The values that can be obtained using simplified approaches are overestimated and should be improved for the vehicle design optimization process when mass reduction, durability characteristics and vehicle behavior on the roadway are a matter of priority.

The target of this paper is to show a general step-by-step design methodology of operational loads assessment during vehicle development process, which generalizes the open source information and existing software possibilities for simulating the structural behavior in vehicle dynamics.

Further, in this paper the different approaches for operational loads assessment, used in the SUV development process, are considered. First, it is discussed how to estimate approximately the operational loads of a vehicle in a variety of standard maneuvers using only suspension models with fixed interface points while not considering the main supporting structure. Wheel loads for these simulations are taken from analytical calculations that are based on a primitive vehicle model with body point mass and suspensions, made only with a set of springs. The suspension concept is developed in accordance with this approach. During this stage, K&C tests contribute to optimizing joints, bushings, and other suspension parts' characteristics.

This is followed by a more complicated approach for operational loads assessment is described, which was also used in the development of the SUV. These simulations require a full vehicle model made of rigid subsystems for the sprung mass and flexible subsystems for un sprung mass. This approach helps to get more feasible operational load values due to the fact that vehicle description is completely different from that in the approach described earlier; this time a full vehicle model replaces of two separate suspension models on the test rigs. Here the full vehicle is tested on the virtual roadway, undergoing various road tests.

The last step considered in this paper is an advanced approach which takes into account not only flexible suspensions but also BiW and chassis subsystems as flexible bodies. This requires extra model generation, containing reduced stiffness matrices of the vehicle's main supporting structure, as well as the greater computational effort to evaluate the vehicle's maneuvering behavior with this approach.

Main Body

For clarity, the assessment of operational loads during the vehicle development process can be divided into four steps:

Step 1: At this step, the main supporting structure of the vehicle (BiW or frame) has not yet been approved, but the first approximation of the operational loads is required. Thus, a rough kinematic concept design of the vehicle suspension is created based on K&C tests in accordance with vehicle general requirements. Wheel loads are obtained using an analytical approach in static formulation with a simplified vehicle model. Wheel loads are applied to the kinematic suspension concept, modeled only with rigid parts, in order to assess their behavior while interface points are fixed. The main supporting structure itself is not considered in this formulation. As a result, the rough concept design of suspension parts is determined to the end of this step;

Step 2: It is dedicated to selecting the shape of suspension parts which concept design was determined on the previous step. For these reasons, suspension parts are considered as flexible parts. Here K&C tests help to approve obtained on the previous step suspension design and to adjust the appropriate suspension parts characteristics such as to perform safe and efficient suspension behavior in different maneuvers. So, as a result, the updated suspension model is developed with flexible parts, optimized to satisfy K&C and strength requirements;

Step 3: The main supporting structure of the vehicle is already approved before this step starts and full vehicle model in a dynamic formulation is considered. This model includes suspensions, steering system, main supporting structure and other components with their mass and inertia characteristics. This allows more realistic forces redistribution in suspension parts and displacements evaluation of the interface points due to main supporting structure declines in maneuvers. In this formulation all vehicle sprung mass subsystems are rigid and un-sprung mass subsystems are flexible. As an example of a full vehicle model considered in this paper, body-on-frame SUV is shown in Figure 1 (BiW is transparent for clarity). A range of virtual road tests are simulated based on this model to assess the operational loads due to different load cases;

Step 4: Full vehicle model on this step is built with elasticity characteristics of the main supporting structure for getting more accurate load redistribution in all vehicle subsystems. Subsystem compliance causes sufficient differences in vehicle behavior on the virtual road tests that must be considered on the further development stages after concept stage for achieving trustworthy results of the operational loads assessment and effective vehicle design optimization.

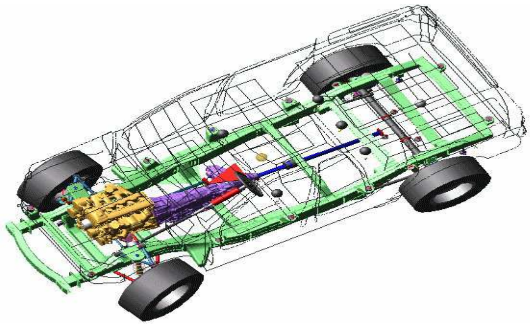

Figure 1: Full Vehicle Model of the SUV

All of these steps nowadays are made using leading computer-aided engineering software possibilities. There are various powerful multi-body system (MBS) solvers available on the market for predicting dynamic system behavior of different kinds of mechanisms, for example, MSC Adams, Simulia SIMPACK, LMS Virtual Lab. They became necessary instruments for the modern design development process and have different modules to simplify working on specific tasks for different industries. Concerning automobile industry, for example, such software has developed a road path tracking systems (in MSC Adams called "Smart Driver"), which control the implementation of the vehicle maneuvers during the simulations and help to keep maneuver characteristics such as vehicle velocity, cornering angle etc. unchanged due to physical phenomena. It is done with so-called standard maneuvers templates. These templates are based on vehicle "bicycle model" with a combination of PID controllers [11, 12].

Over the last years commercial MBS solvers have developed significantly and now can allow considering not only rigid bodies but also flexible ones (in MSC Adams it is done with Modal Neutral Files (MNF)) and nonlinear solvers (f. ex. MSC Adams Max flex) for dynamic problems. The cutting-edge growth area now is real-time solvers that can allow combining virtual simulations with an operation of the real objects, for example, to simulate vehicle behavior with engine characteristics, which are recorded from the real engine on the test rig in a real-time environment.

The first of the steps listed above for operational loads assessment starts with the wheel loads determination on the contact patches. This is made on primitive vehicle model containing body point mass with regard to mass distribution between axes and suspension made only with a set of vertical and torsional stiffness springs. In particular, this primitive model takes into account the only steady state motion of the vehicle and as a result gives only the reaction forces from the roadway. For further investigations the MBS model of suspension must be developed. This model allows estimating the loads redistribution on the suspension parts and its behavior in different maneuvers.

For the SUV discussed in the paper, the rear and front suspension MBS models with test rigs for K&C simulations are shown on the Figure 2 and 3. Standard K&C test series approximate a variety of vehicle maneuvers such as in-line acceleration, turning, suspension breakdown, etc. That kind of simulations can be easily done with commercial software templates. The suspension interface points that connect suspension with the main supporting structure are considered fixed. The main supporting structure itself is not considered in this formulation. This type of test rig is usually used for the first and second steps of operational loads assessment, considered in this paper.

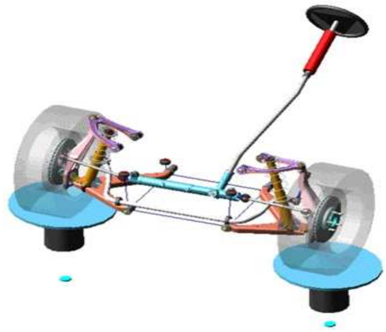

Figure 2: Front Suspension MBS Model View of SUV

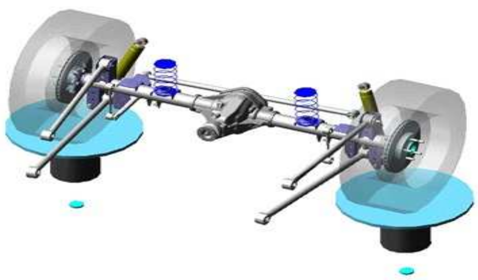

Figure 3: Rear Suspension MBS Model View of SUV

Firstly K&C simulations with all rigid parts of suspension are made. Then, according to the results, the first approximation of bushing stiffness characteristics is chosen. In this formulation– rough bushing characteristics and rigid suspension parts– a suspension optimization process are started to achieve the K&C targets of the vehicle by varying bushing characteristics and their position. According to K&C load cases, more than 100 kinematic and elasto kinematic suspension characteristics are investigated and matched to achieve the desirable behavior of suspension on the test rig. It is important that at the same time with suspension concept determination also the main supporting structure is being developed. Space design must be identified for the main supporting structure as well as for suspension and its envelopes. According to K&C test results, interface points position can be changed slightly on this step to improve suspension behavior, but it is highly important to synchronize these interface points displacements with the vehicle main supporting structure and other subsystems of the vehicle to check the possibility of such movements and update the structure simultaneously. At this moment it also must be controlled that the interface points on the main supporting structure have enough stiffness to ensure suspension durability and appropriate vehicle dynamics.

As a result of "step 1" procedures, a suspension concept is developed. After this, the suspension development process can move to "step 2" and the separate suspension parts optimization can be made (shape, mass, stiffness, etc.) according to previously defined wheel loads. Then further iterations of K&C virtual tests are made to check the suspension behavior with newly designed suspension parts and characteristics. As an example, front suspension behavior due to K&C opposite travel test for considered SUV is shown in Figure 4. On this step bushing characteristics and their positions can be adjusted. In this formulation, all the suspension parts are considered flexible. More details about building up the flexible model for MBS solvers can be found further in the paper. These simulations are made iteratively and it helps to design the suspension and the main supporting structure most effectively at each vehicle development stage.

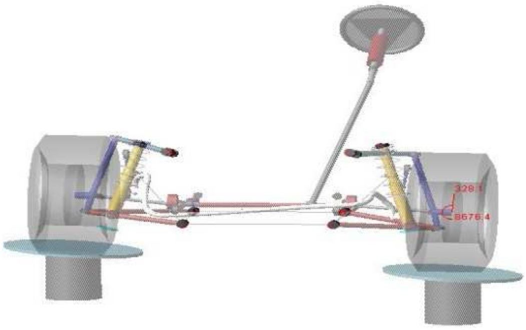

Figure 4: Interface Points during K&C Opposite Travel Tests

Moving further in the vehicle development process the “step 3” of the vehicle operational loads assessment can be started. For this step, the full vehicle assembly with all rigid subsystems for the sprung mass is required to be built in conjunction with the flexible subsystems for unsprung mass from the previous step. These simulations are called Full Vehicle Analysis (FVA) and are conducted in a dynamic formulation on a special virtual roadway test rig in opposite to steady state motion simulations considered in the previous steps.

FVA includes tests like step steer, lane change, steer continuous sinusoidal input, etc. This is advanced vehicle analysis after K&C tests and during these analysis vehicle parameters, such as another tire assignment for frequency response improvement or another dampers assignment for achieving desirable vehicle behavior, are tuned for vehicle targets achieving. The main K&C virtual series are usually finished till this period, but there can appear some serious changes, which can cause more subsequent K&C analysis for a redefinition of the suspension parameters and interface points’ location. As an example, such reasons could be the requirements from other subsystems for interface points adjusting, bushings adjusting or necessity of increasing main supporting structure stiffness.

FVA approach helps to assess operational loads more realistic than previously mentioned approaches. That is, in the analytical approach acceleration and mass are used to obtain the forces on the contact patches only. With the FVA approach, more accurate acceleration values can be achieved. For example, for analytical calculations acceleration in turning is set to 1,2g, while in FVA for considered in this paper SUV at an average only 0,8g can be achieved due to vehicle dynamics. With the acceleration values greater than 0,8g a slipping motion of the vehicle starts, which is shown further in the simulation results. The same kind of discontinues is occurring with other maneuvers such as suspension bump, acceleration, deceleration etc. The attention must be paid to the vehicle dynamic characteristics in different maneuvers and dampers functioning in FVA, which are absent in analytical formulation. This analytical formulation considers only steady-state motion and so the dampers are not under consideration. This means that the real load’s distribution in suspension is different to analytical results, especially for developed SUV’s rear suspension, because it has different interface points to the BiW for dampers and springs. This leads to the necessity of main supporting structure improvement.

But on further vehicle development stages, the accuracy of “step 3” approach can be not enough to satisfy the strict customer criteria for strength, durability, vehicle behavior on the roadway. In the “step 3” approach the main supporting structure subsystem is supposed to be rigid. Thus the influence of the real main supporting structure stiffness on the vehicle behavior in maneuvers cannot be evaluated on this step. This can have a significant effect on the real vehicle behavior and must be taken into account in further simulations.

During the road tests, wheel loads from the roadway are redistributed and transferred up to suspension. Through the interface points, these forces act on the chassis and BiW structures and the full vehicle model for "step 4" approach can show the critical design areas with high-stress levels, which must be considered in more details during further fatigue analysis of the vehicle. Taking subsystems compliance into account makes the assessed operational loads more realistic because of simulating the exact suspension and vehicle main supporting structure behavior in different maneuvers and conditions.

For the "step 4", the full vehicle model must be modified to include the elasticity characteristics of the main supporting structure subsystem. The main supporting structure of the SUV considered in this work is the frame and, optionally, BiW, which can also be considered as a flexible body for a complete picture of the vehicle dynamic behavior.

One particular aspect of this approach is much higher computational efforts than for the previous steps (hours compared to minutes on the same computing machine), so it is worthwhile to simulate only critical load cases in such formulation. Alongside this using vehicle model with elastic characteristics is reasonable after achieving the stiffness targets of the main supporting structure.

For creating the model for FVA at "step 4" it is required to build up extra model files, which contain the data of a flexible body. For MSC Adams [13]they are MNF and contain reduced stiffness matrices, inertia matrix, mode shapes, and frequencies. MNF is based on Craig-Bampton modal synthesis [14]. This kind of data is obtained from a linear Finite Element analysis and can be outputted in. mnf format from most of the commercial FE-solvers.

The full vehicle model considered in this paper SUV, containing mass-inertia and elasticity characteristics of the frame, is shown above in Figure 1.

For developed SUV this approach was applied to three load cases: suspension bump (4g), turning (0,8g) and cross-axling. As for example, deformed state of the vehicle during cross-axling simulation is shown in Figure 5. This load case assumes pushing up the diagonal wheels till the moment of taking-off one of the other two wheels.

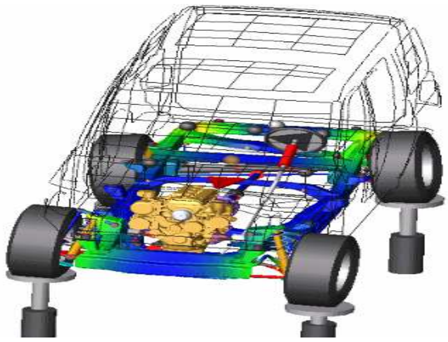

Figure 5: Deformed State Due to Cross-Axling of SUV

It should be noted that displacements, stiffness results and also the overall stress distribution, determined with this method, are trustworthy, but the stress values themselves may differ from the values obtained in strength simulations by specialized FE-solvers, such as MSC Nastran. For getting better stress results in MSC Adams for "step 4" formulation a more detailed description of parts with complex geometry is needed (using more intermediate interface points). This procedure is very complex and time-consuming. In this case, it is better to evaluate parts displacements/stiffness with MNF approach and for stress evaluation, it is better to use FE commercial code. In the same time, it is worth comparing the results of MSC Adams and FE solvers, especially concerning displacements and force redistribution on the design parts. This helps to make verification of MSC Adams flexible models.

To underline the difference between the results obtained using various approaches, there is a comparison of the steering wheel angle curve vs. lateral acceleration presented in Figure 6. These are the results for steady-state cornering maneuver with a constant velocity equal to 100 km/h. Three different vehicle model formulations are shown on this graph: "R+R" means rigid SUV frame and rigid BiW, "R+F" – rigid BiW and flexible frame, and "F+F" means both flexible frame and BiW. "R+F" and "F+F" results are almost coincident and are significantly different to "R+R" formulation. Oscillations on "R+R" results appeared at the moments when vehicle motion becomes unstable. This happens due to the fact that "bicycle model", implemented to the solver, does not take into account the frame's elasticity. At these moments software tries to compensate wheels' sliding and hold the vehicle trajectory according to the model input using Smart Driver module, which was described above. This is the limitation of rigid vehicle formulation, which is eliminated in flexible formulations.

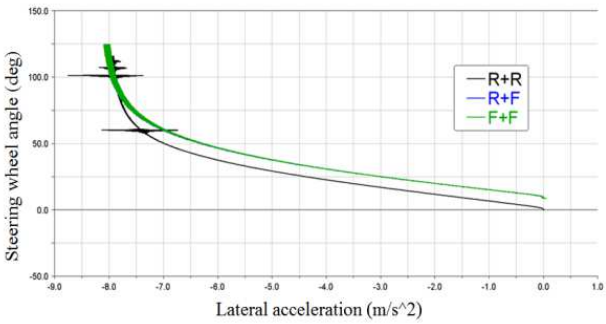

Figure 6: Steering Wheel Angle Curve vs. Lateral Acceleration for Steady-State Cornering, 100 Km/H

Another example of the "step 4" comparing to "step 3" results is provided on Figure 7. These are vehicle positions at different time moments due to step steer maneuver [15] at the constant velocity 100 km/h. This maneuver produces a 6 m/s^2 acceleration on the vehicle. Again, a significant difference between results can be seen. Because of considering the frame elasticity, there is different load redistribution to suspension parts that leads to different suspension operating and trajectory adjustment. According to Figure 7, differences between "R+F" and "F+F" for developed SUV are not significant, so reasonable results can be achieved with "R+F" formulation and it is no need to develop the flexible model of BiW, which is quite time-consuming.

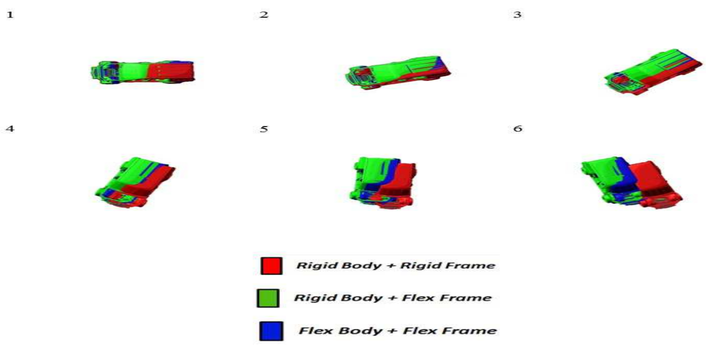

Figure 7: Vehicle Position at Different Time Moments Due to Step Steer (Left), 100 Km/h

At Figure 8 roll angle curve vs. time for step steer can be seen. For “R+R” formulation roll angle returns to zero value after process stabilizing, but for both flexible formulations there is a residual roll angle value. This happens due to the fact that the roll angle is calculated relative to the center of mass of the SUV. In the “R+R” formulation, after turning the steering wheel, the position and orientation of the center of a mass return to the initial position. In “F+R” and “F+F” formulations, the position and orientation of the center of mass are affected by the deformed state of the frame. After turning the steering wheel frame deformation remains due to centrifugal forces when driving in a turn. These features can affect the behavior of the vehicle significantly and must be evaluated properly.

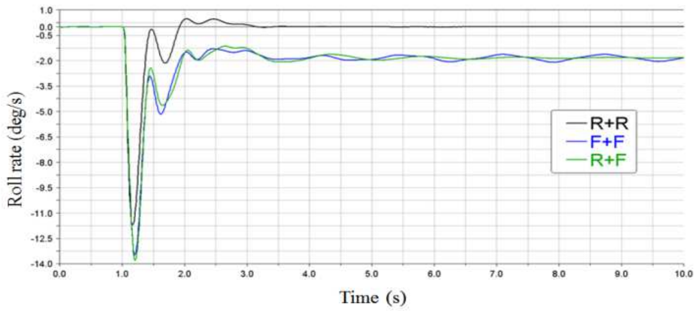

Figure 8: Roll Angle Curve vs. Time for Step Steer (Left), 100 Km/h

So the result differences for considered formulations can be seen clearly. Taking these factors into account on the technical and further stages of vehicle development can help to get more accurate simulation results, save manufacturer’s resources for vehicle development and also helps to shorten the time before the start of vehicle serial production. These benefits help automotive manufacturers to be most competitive and to offer the market their best-in-class solutions.

CONCLUSIONS

In summary, there are various approaches to virtual operational loads assessment. The choice of a specific approach to use depends mostly on the vehicle development stage. Each approach has its advantages and disadvantages and helps to assess the operational loads with different accuracy. The vehicle development process is closely related to operational loads assessment as far as they have a direct impact on chassis strength, fatigue, and, especially, vehicle behavior on the roadway.

The operational loads are calculated along with overall vehicle development process in accordance with a sequence of the steps 1 to 4 described in the article.

For further investigations, nonlinear elastic characteristics of the main supporting structure can be considered for achieving even more accurate operational loads values. Computational effort for such simulations would be even higher than for virtual tests of “step 4”, considered in this paper. So it is reasonable to simulate according to this approach only the most critical load cases that can have a significant impact on vehicle design solutions.

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