NVH ANALYSIS OF OFFROAD VEHICLE FRAME.EVALUATION OF MUTUAL INFLUENCE OF THE BODY-FRAME SYSTEM COMPONENTS

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I. INTRODUCTION

During the development of new vehicles, there is always required to work out tasks of improving the effectiveness of the design i.e. to find the best compromise between high level of safety, light weight, fuel efficiency, reliability and such customer demands as perfect handling, vibro-acoustic comfort, load carrying capacity etc. This article was made in frame of work under new body-on-frame SUV. Predecessor, the serial SUV, was used as the basis for analysis. In order to completely switch to digital design in the new paradigm of Digital factories, we built the Smart Digital Twin of serial SUV, that will allow to reduce the time and costs of the design stage in a future, by reducing the number of real tests. The New vehicle design based on the Smart Digital Twin, validated by results of our work, make us convinced the digital design will give us the results, that will match the acceptance tests.

The analysis objects, Smart Digital Twins of the Frame, Body and Trimmed body – are shown on figure 1. In a body-on-frame vehicles much attention paid to the frame as the main load-bearing part, which is a support for the powertrain, suspension and body.

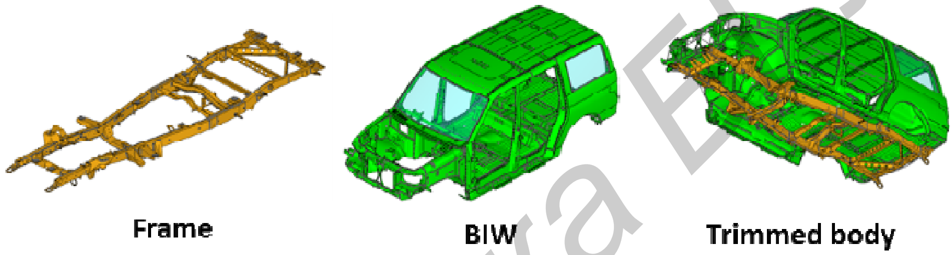

Figure 1 – Analysis objects

One of the design effectiveness key indicators is the static and dynamic stiffness. Static stiffness of the frame is majorly defined by its global torsional, vertical and lateral stiffness. The torsional and vertical stiffness of the frame plays big role in isolating PWT and road vibrations and load carrying capacity of the vehicle. The lateral stiffness of the frame is very important for appropriate handling and cornering ability of the vehicle.

Dynamic stiffness of the frame is majorly defined by its global eigenfrequencies and local dynamic stiffness (LDS), and also plays major role in isolating main sources of noise and vibrations. The eigenfrequencies of the frame should be as high as possible and be decoupled with frequencies of harmonic excitations of such systems as PWT, chassis, fans etc. The local dynamic stiffness should be high enough to provide good filtration of rubber mounts.

II. STATIC AND DYNAMIC STIFFNESS ANALYSIS OF BODY-ON-FRAME SUV

Because of the similarity of approaches, static stiffness is represented by the example of torsional stiffness, modal analysis – by the example of torsional modes, local dynamic stiffness – by the example of powertrain mounts only.

Static stiffness

To define the torsional static stiffness of the Trimmed body, frame is loaded with a twisting moment, and twisting angle of the BIW is evaluated (see figure 2). Torsional static stiffness of Trimmed body is the ratio of twisting moment (applied to the frame) to twisting angle of the BIW. This characteristic is included in the vehicle technical requirement list. Also it is useful for the structure analysis to make a plot of twisting angles along the BIW length, as shown in figure 3.

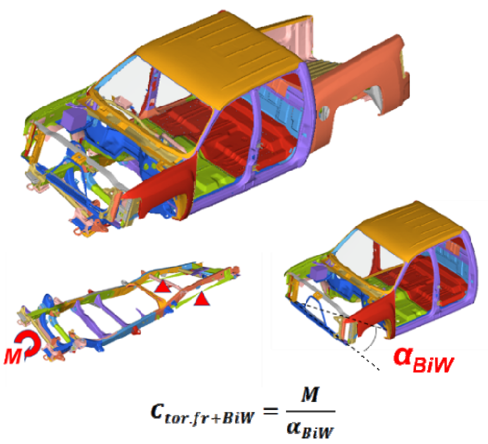

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Figure 2 – Torsional stiffness of the Trimmed body evaluation test scheme

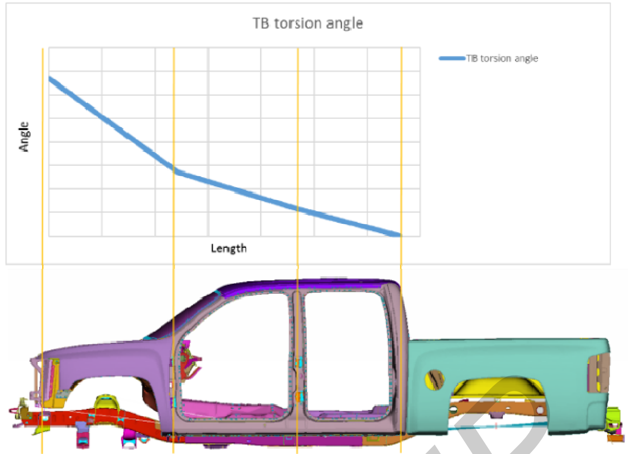

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Figure 3 – Example of the plot of twisting angles along the BIW length

We define lowest eigenfrequencies of trimmed body to decouple them with the eigenfrequencies of unsprung masses and the frequencies of engine idle vibrations (see figure 4).

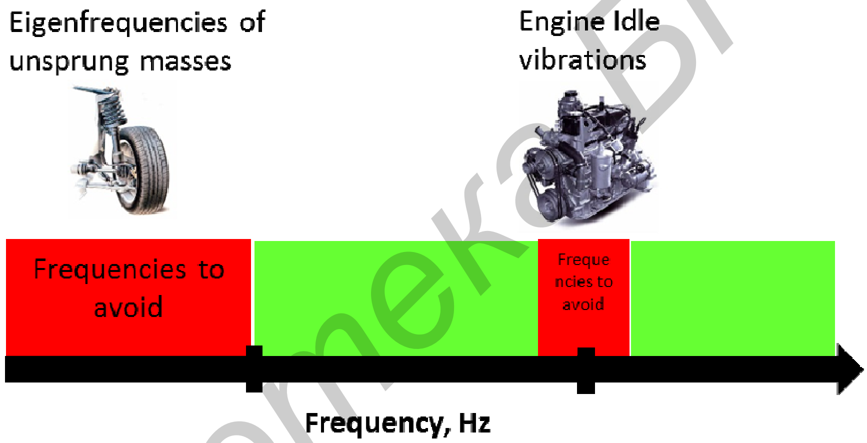

Figure 4 – Global eigenfrequencies to avoid scheme

Local dynamic stiffness

We evaluate the local dynamic stiffness of the rubber mounts installation zones to ensure good filtration. Local dynamic stiffness of Frame/BIW in the rubber mounts installation zones should be several times higher of the respective rubber stiffness.

III. SIMULATIONS RESULTS

Static stiffness results

Plot of twisting angles of the BIW (as part of Trimmed body) of the Serial SUV Smart Digital Twin is shown on figure 5. The point is mark the twisting angle which corresponds to the target stiffness. The dotted line – is the relative level below which all the twisting angles of the BIW as part of trimmed body must be. The continuous line – is the results of simulations. It is seen from the plot that the part corresponds to the twisting of the front end do not meet the target level. This is due to the concept of body-on-frame construction – the frame receives all the main loads, so the body do not need power elements such as longerons in the front end. This should be taken into account when assigning target stiffness for the frame and the body individually.

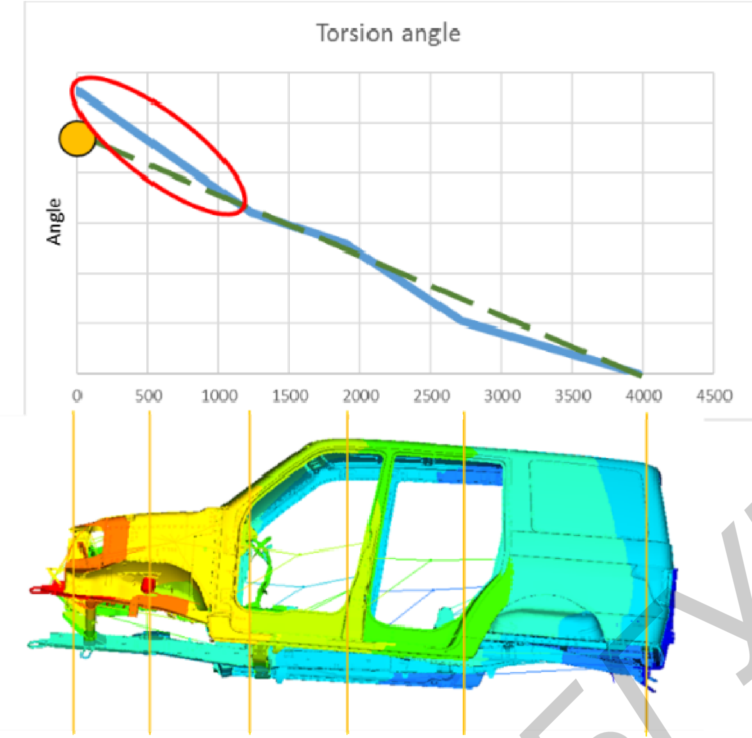

Figure 5 – Plot of twisting angle of the serial SUV Smart Digital Twin

Modal analysis results

Modal analysis of the simulation objects shows the following dynamics:

– lowest eigenfrequencies of the Frame as part of Trimmed body higher in comparison with the Frame individually by 10 Hz. It is explained by the change of boundary conditions (elastic contact with "heavy" body) at slight change in mass.
– lowest eigenfrequencies of the BIW as part of Trimmed body lower in comparison with the BIW individually by 7 Hz. This is explained by the minor change of boundary conditions (elastic contact with "light" frame) at significant increase in mass.

The dynamic of changes schematically shown on figure 6. The dynamic of changes should be taken into account, when assigning target values of the lowest eigen frequencies for the frame and the body individually, to eliminate the coupling between the frequencies of the Frame and BIW with each other and with engine idle vibrations on the Trimmed body.

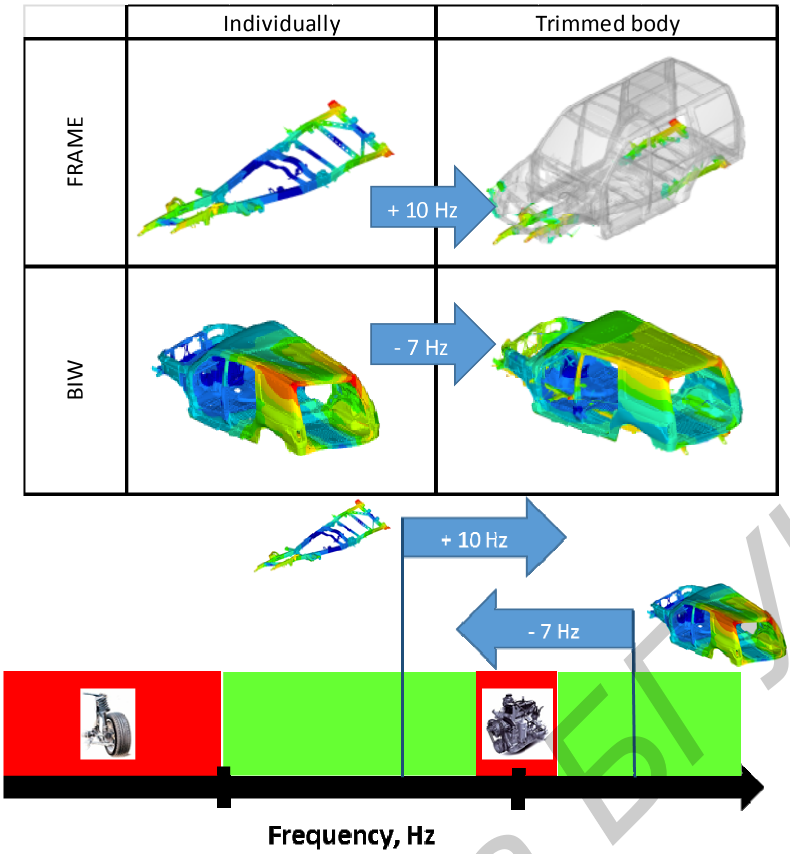

Figure 6 – The dynamics of eigenfrequencies changes at switching from the Frame and BIW individually to the Trimmed body

Local dynamic stiffness results

Local dynamic stiffness analysis results are shown in Table 1.

Table 1 – Local dynamic stiffness on the Frame and on the Trimmed body comparison.

Subcase	Units	Dir.	Difference between component in separately and in assembly, %	
			Low freq.	Mid. Freq.
Engine left mount	N/mm	x	-3.8	0.0
Engine left mount	N/mm	y	2.6	0.2
Engine left mount	N/mm	z	-0.7	0.2
Engine right mount	N/mm	x	-3.5	0.1
Engine right mount	N/mm	y	1.4	0.0
Engine right mount	N/mm	z	1.2	0.1
Gearbox mount	N/mm	x	1.4	-0.1
Gearbox mount	N/mm	y	-1.5	-0.5
Gearbox mount	N/mm	z	0.5	-0.2

IV. CONCLUSIONS

As the result the optimal target values of static stiffness of the Frame and BIW were selected to achieve the target static stiffness of the Trimmed body.

The optimal target values of lowest eigenfrequencies of the Frame and BIW were selected to eliminate the coupling between the frequencies of the frame and body with each other and with engine idle vibrations on the trimmed body.

From the results of LDS analysis we can see that difference between calculation on the frame and trimmed body is:

– maximum 3.8 % for the low frequency;
– maximum 0.5 % for the medium frequency.

In absence of full Trimmed body model it is allowed to make first estimations of LDS on the Frame. However, when the Trimmed body model is available it is needed to update the results, especially for low frequency range.

ACKNOWLEGMENTS

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INTEGRATED SYSTEM AS TOOL FOR IMPLEMENTATION OF SIMULATION- AND OPTIMIZATION-BASED DESIGNMETHODOLOGY

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I. INTRODUCTION

Today in the world design engineering practice, two approaches to design of structures can be distinguished. The first, more traditional, is that the designer on the basis of operational requirements and his experience invent a sketch of the structure, which then turns into a CAD-model. After that, a real prototype is created and full-scale tests are carried out. CAE calculations are also performed, mainly to verify those loading cases that have not been tested in full-scale tests. In case of dissatisfaction with some requirements, the model is returned to the designer, and loop is repeated.

With the traditional approach, in the world the engineers use other approach, based on mathematical modeling and optimization methods. Using this approach, called simulation- and optimization- based approach, allows you to get a design of structure under specified operating conditions as a result of the use of computer simulation and optimization. This approach is currently being implemented by leading Western industrial companies in the development of new aircraft, ground vehicles, etc. A key role in this approach is topological optimization, allowing for predetermined loads to predict the most effective material distribution in structure.

This work is devoted to the development of a tool for implementing the design approach based on mathematical modeling and optimization. The work includes the development of an integrated computer design and engineering system, a topological optimization module in the ANSYS APDL environment, and demonstrates obtained solutions.

I. TOPOLOGY OPTIMIZATION PROGRAM MODULE

The problem of topological optimization, in the classical formulation, is the problem of choosing the optimal distribution of material in fixed space. For each point of the body, we should answer the question whether there is material in this place or not. To look at this initially discrete problem as on the continuous problem we use the SIMP (Solid Isotropic Material with Penalization) method, which allows to associate the elastic properties of a material with an additional parameter called "density" [1]. In ANSYS APDL, this was implemented by assigning each finite element its own material. The further task is to achieve a minimum of the functional of external forces work, when the equilibrium condition of the system is satisfied. For this goal, we use the Method of Moving Asymptotes (MMA) [2]. To apply this method, it is necessary to define the partial derivatives (sensitivities) of objective with respect to design variables. In the case of minimizing the compliance of the system with a constraint on the volume, these derivatives are related to the potential deformation energy at each point. In ANSYS APDL, the procedure for finding sensitivities is consist in