Free and forced vibration analysis on finite element model of an off-road vehicle

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Abstract

In the present paper, the modal analysis on a full finite element model of an off-road vehicle. This vehicle was modeled in the CATIA software and then meshed in the HYPERMESH software. The free vibration analysis was conducted by the ABAQUS software. By applying an external displacement, the forced vibration analysis was also performed. As a result, natural frequencies and shape modes were extracted to detect critical regions. Then, some improvements were suggested to have better vibration behavior of the vehicle.

Keywords: *free vibration, forced vibration, modal analysis, finite element model, off-road vehicle*

1. Introduction

In two last decades, the modal analysis has been the knowledge for determining and improving dynamic characteristics of engineering structures. Nowadays, the design of complex structures has been based on high strength and low weight. For example, in automotive industries, a high effort has been carried out for reducing the body weight.

In this field of study, several researches have been presented by scientists. Most of modal analyzes were based on vibration models (masses and springs) of vehicles [1-4]. Some others were based on half-full [5-8] and full [9-10] models of vehicles. Sobieski et al. [11] minimized the weight of the vehicle body-in-white based on the weight, the crash and the noise, vibration and harshness (NVH) behavior. They utilized a finite element (FE) model with 390,000 degrees of freedom. Lam et al. [12] reduced the vehicle weight by modeling the structure in the NASTRAN software. They changed the material type and the reinforced plate thickness. Their constraints were NVH and crash behaviors. Kodiyalam et al. [13] had performed an optimization process based on the weight, crash and NVH behaviors. They used a FE model with 68000 elements and focused on bending and torsional modes in free-free conditions. Azadi et al. [14-15] performed a NVH analysis on a full FE model of a vehicle. They improved the vehicle structure based on a multidisciplinary optimization under NVH and weight constraints. They utilized design of experiment (DOE) methods such as the factorial approach and the response surface method (RSM).

In this study, we have focused on both free and forced vibration behavior of an off-road vehicle body. For this objective, a full FE model was analyzed in the ABAQUS software. Obtained results including natural frequencies, shape modes and forced vibration behaviors of the vehicle show in figures and tables.

2. FE modeling

An off-road vehicle (F800 Ford) was modeled in the CATIA software. All components of the vehicle (such as axles, springs, the chassis, the body, etc.) were assembled together. Then, it was meshed in the HYPERMESH software to have a full FE model. Element types were solid and shell with the second-order accuracy. All components were merged together by the multi-constraint point (MPC), as welding joints. The type of MPC in the HYPERMESH software was the rigid beam element (RBE2). It should be mentioned that the quality of meshing was controlled in the HYPERMESH software and the size of elements was investigated for the convergency of obtained results. The FE model can be seen in Figure

In addition, Figure 2 shows details for modeling the rear axle.

Generally, we used material properties of common steels for most of components. The elastic modulus and Poison's ratio was 205 (GPa) and 0.3, respectively. As we focused on the suspension system of the vehicle in this study, accurate material properties of springs and axles were applied in the ABAQUS software. Table 1 shows these properties (the elastic modulus, Poison's ratio and the yield stress of materials) and also the thickness of shell elements for mentioned components.

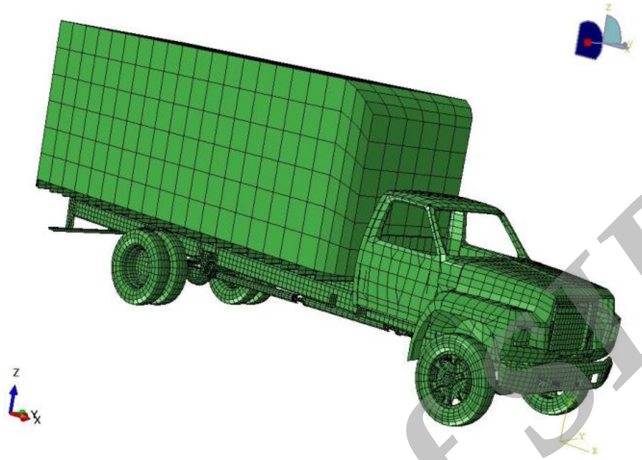

Fig 1: The full FE model of the off-road vehicle

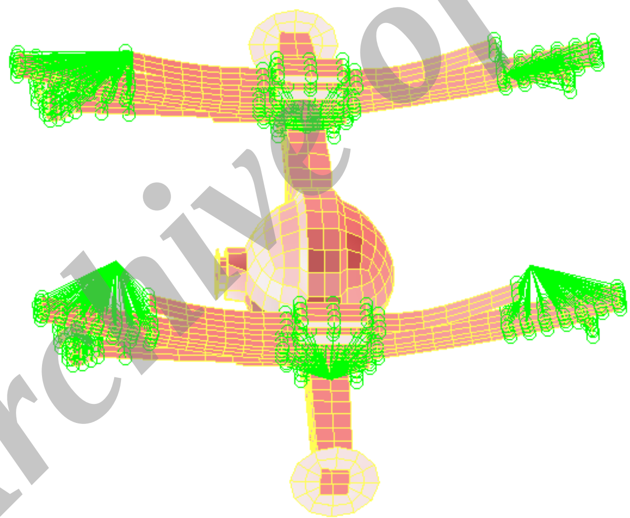

Fig 2: Details for the FE model of the rear axle

Table 1: Material properties for axles and springs of the vehicle

Components	Elastic modulus (GPa)	Poison's ratio	Yield stress (MPa)	Thickness of shell elements (mm)
Front leaf spring	200	0.3	1477	11.5
Front axle	205	0.3	700	19.9
Rear leaf spring	200	0.3	1477	12.4
Rear axle	205	0.3	700	22.0

Table 2: Natural frequencies of the vehicle in the free-free condition

Mode No.	Natural frequency (Hz)	Mode shape
1	0.0002	Rigid motion in longitudinal direction
2	0.0002	Rigid motion in lateral direction
3	0.0001	Rigid motion in vertical direction
4	0.0004	Pitch rigid motion
5	0.0003	Roll rigid motion
6	0.0002	Yaw rigid motion
7	2.2	Bending of vehicle structure
8	3.2	Bending of tires
9	4.9	Bending of tires and torsion of wheels
10	5.0	Torsion of wheels through vertical axis

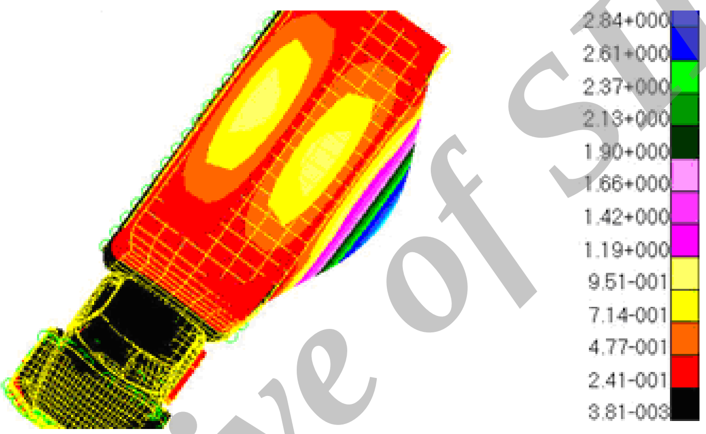

Fig 3: The model shape of bending of the vehicle structure at the frequency of 2.2 (Hz)

Results and Discussions

As a first result, the modal analysis for the FE model of the vehicle was performed at the free-free condition. In other words, we do not consider any constraints for the whole FE model of the vehicle. The objective is to verify the FE model. Natural frequencies of the vehicle can be seen in Table 2.

As it can be seen, six first natural frequencies were approximately zero. They were not exactly zero, since we use a numerical FE method to obtain natural frequencies. In addition, the FE model can be verified by obtaining six first natural frequencies. A component should have six first natural frequencies at the free-free condition. At this condition, it means that there were no separations between components. This demonstrates the verification of the FE model. Then, after these six first natural frequencies, the first un-zero natural frequency is 2.2 (Hz), related to the mode shape of bending of the vehicle structure. This mode shape is also shown in Figure 3.

The modal analysis for the FE model of the vehicle on the road was also performed. For this condition, all wheels were constrained to the road. It means they had no displacement and the vehicle was fixed to the road. Related results including natural frequencies until 30 (Hz) can be seen in Table 3. At this range of frequencies, zero to 30 (Hz), there were 14 mode shapes. The first natural frequency was related to the vehicle body through the vertical axis at the frequency of 5.1 (Hz). Some natural frequencies included a combination of shape mode, such as the third natural frequency. This mode was a combination of bending and torsion of the vehicle structure at the frequency of 7.8 (Hz).

The forced vibration analysis was also carried out. The objective was to find the displacement and the acceleration of the vehicle. For this purpose, a sinusoidal input with 1 (mm) displacement amplitude was considered on front and rear wheels. There was also an out-of-phase condition for loading. It means that the displacement amplitude has a 180-degree phase lag between front and rear wheels. Then, the displacement and the acceleration of the driver point were calculated in the FE model of the vehicle. Related results including the displacement and the acceleration versus the frequency, at the range of zero to 100 (Hz), are shown in Figures 4 and 5.

As it can be seen, there were some peaks (sudden increase in the amplitude) on the frequency response of the sinusoidal input. These peaks showed natural frequencies of the vehicle. For example, at the frequency of about 25 (Hz), one peak can be seen in Figure 4 for the displacement behavior of the driver point. According to Table 3, at the frequency of 25.7 (Hz), there was a natural frequency including a mode shape for bending and torsion of the vehicle body. This mode shape was near the driver point, which had effects on the frequency response of the vehicle. Such this behavior can also verify the FE model and results of the modal analysis. In other words, results of the forced vibration analysis had a proper agreement with results of the free vibration analysis in the FE model of the vehicle.

Some suggestions can be proposed to improve the vibration behavior of the vehicle. As shown in Figures 4 and 5, there were some peaks, which can be considered as defects in the vehicle. To improve the vibration behavior, the structure thickness can be increased and reinforced materials (with additional plates or beams) can be used.

Table 3: Natural frequencies of the vehicle in the fixed condition

Mode No.	Natural frequency (Hz)	Mode shape
1	5.1	Vehicle motion on vertical direction
2	6.1	Vehicle motion on lateral direction
3	7.8	Bending and torsion of vehicle structure
4	13.6	Bending of vehicle structure
5	15.8	Torsion of vehicle body
6	16.2	For/aft of vehicle body
7	19.5	Torsion of vehicle body
8	20.6	Bending and torsion of doors
9	22.4	Bending of driver room
10	24.1	Bending of vehicle structure
11	25.7	Bending and torsion of vehicle body
12	27.3	Bending of the vehicle body and bending of the front axle
13	28.6	Torsion of vehicle structure
14	30.0	Bending of driver door

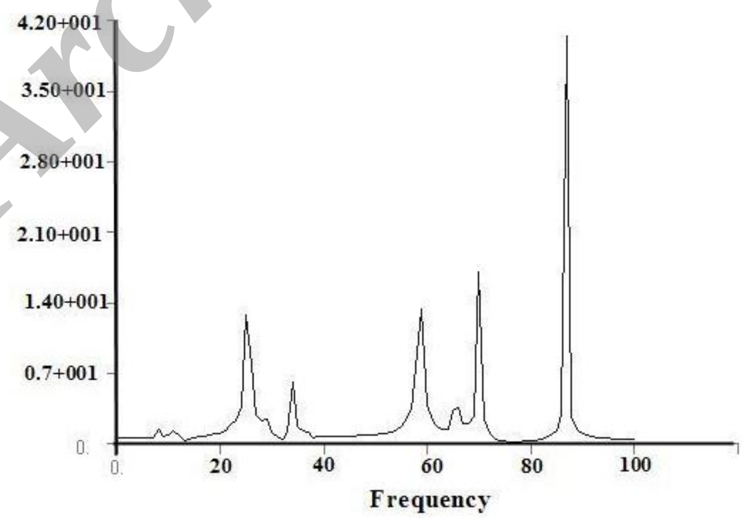

Fig 4: Results of forced vibration analysis including displacement versus frequency

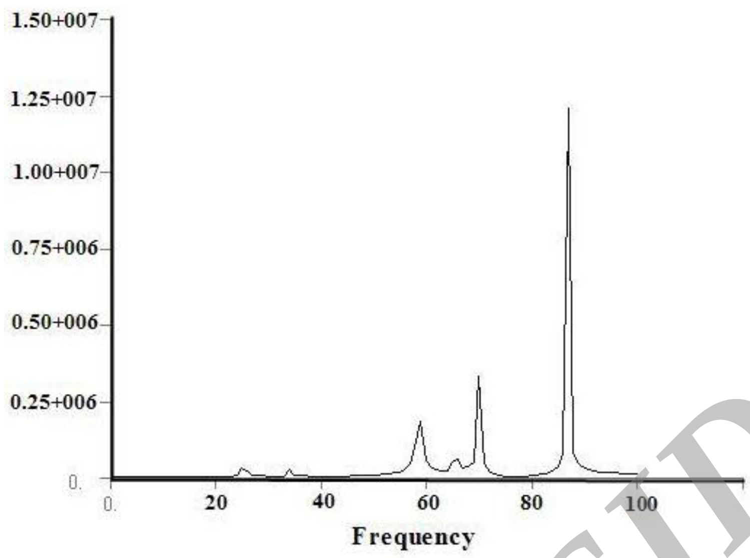

Fig 5: Results of forced vibration analysis including acceleration versus frequency

Conclusions

In the present paper, free and forced vibration behaviors of an off-road vehicle were calculated by the finite element method. In the free vibration analysis, natural frequencies of the vehicle structure were extracted. Then, by the forced vibration analysis, the displacement and the acceleration of the driver point were obtained. A good agreement was observed between results of free and forced vibration analyzes.

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