

Evaluation of engine-induced vibration levels on a Baja SAE frame: a pilot comfort improvement assessment using the Finite Element Method

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Abstract. The development of reliable virtual models to evaluate vehicle attributes, such as comfort, is crucial for the automotive industry to reduce time and costs on the product development process. The vibration levels generated by the engine can potentially cause uncomfortable effects on the passengers and compromise the vehicle operation when in resonance with other components, thus they should be carefully analyzed. The objective of this work is to numerically evaluate the vibration levels of a Baja SAE frame on the interface regions with the pilot and develop countermeasures to decrease these levels. To reach this objective, a finite element model of the frame was built, using shell elements. Modal and torsional stiffness analyses were performed to validate the model. Frequency Response Functions were generated to evaluate the vibration transmissibility from the engine to the seat and steering wheel fixing points, within the operational frequency range of the engine. To decrease the vibration levels, engine mount bushings and structural countermeasures were tested on the model to define the best solutions for the pilot comfort, comparing the frequency response functions amplitudes under different frame designs and bushings properties.

Keywords: Vehicle Comfort, Finite Element Method, NVH, Structural Dynamics, Baja SAE.

1 Introduction

Vehicle comfort is an attribute that depends on a set of variables in the human-machine interaction, going from the seat's softness to the vibrations experienced by the passengers, this last one being evaluated dedicatedly by the Noise, Vibration, and Harshness (NVH) teams in the industry. When it comes to a Baja SAE prototype, a single-seat off-road vehicle with a tubular frame, even though its project is less sophisticated, the vibrations transmitted to the pilot are still a concern.

According to Gillespie [1], there are multiple excitation sources to the vehicle vibrations, the principal being the road roughness and the internal sources, which are the rotative components such as engine and transmission. The engine, as the primary energy source of the system, has a major contribution to its vibrational behavior. While Yu, Naganathan, and Dukkipati [2] mention the importance of the engine mounts on supporting the operational loads and isolating the engine vibrations from the structure, Milliken and Milliken [3] observe the influence of the chassis stiffness on the vehicle's dynamic performance and reliability.

To analyze such a complex engineering problem in an efficient way, Alves Filho [4] suggests as an alternative the application of the Finite Element Method (FEM), a powerful numerical tool that allows fast modeling and evaluation of a vehicle structure. It fits perfectly to a Baja SAE team's purposes, once it allows the analysis of several designs within a short inter competitions period and with low costs.

2 Methodology

The object of this study is the 2020/21 prototype developed by the Carpoeira Baja SAE team, from the Federal University of Bahia. The frame design is based on the team's expertise and the restrictions imposed by the

category's regulation. The frame is built using a composition of steel alloys such as SAE 1020 and SAE 4130, with tubular thin-walled members. The powertrain specification is a four-stroke single-cylinder aspirated engine with 10 HP and weighing 24 kg.

The analysis scope was restricted to the frame and engine modeling only, due to the complexity of building a full-vehicle model. The FEM model of the frame was built using shell elements, recommended by Altair [5] for this type of geometry and application, and considering variables such as computational cost and available time. Fig. 1 shows both CAD and CAE models, with the frame geometry detailed, highlighting the main attachment points and interface regions between the structure and the pilot.

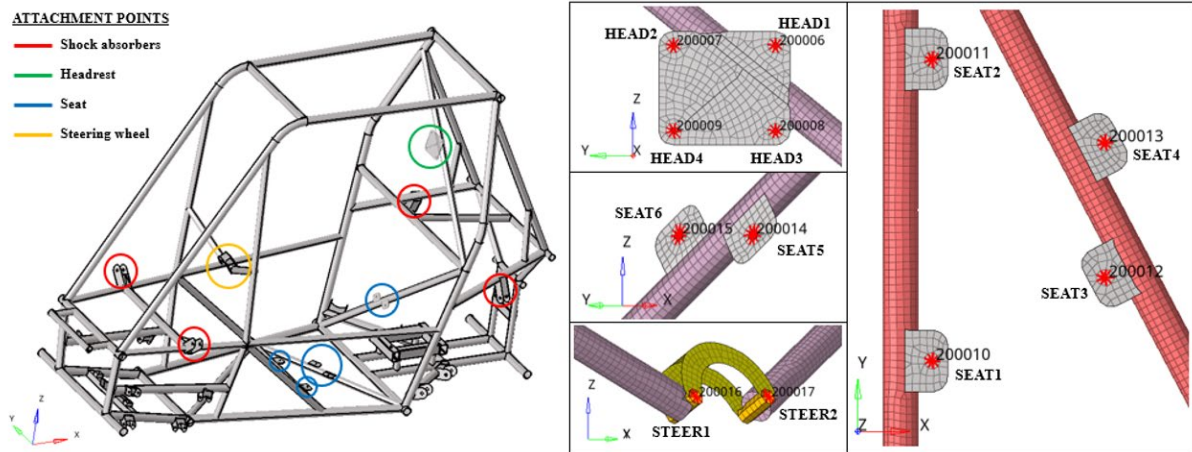


Figure 1. CAD model of the frame and FEM model detailed regions

The engine was modeled as a concentrated point mass located at the CG coordinates and attached to the structure with a rigid element (as further shown in Fig. 2a). A first round of model checks was performed by running modal analyzes in a free-free condition, with and without the engine mounted. To validate the FEM model, a torsional stiffness analysis was run, following the procedure suggested by Milliken and Milliken [3], where the rear-end shock mounts are constrained and torque is applied over the front-end shock mounts. The results were compared with the frame stiffness obtained experimentally by the team on an equivalent test procedure.

After the model validation, dynamic analyzes were performed in the model with the engine attached. Unit forces were applied on the engine CG at vertical (Z-axis) and longitudinal (X-axis) directions and a unitary moment was applied on the crankshaft roll direction (about Y-axis), defined by Gillespie [1] as the most important for the vibrational aspect. The response levels were measured at the points defined in Fig. 1, in terms of acceleration. The structural damping applied on the frame was 3%, based on De Silva's [6] work. Lastly, it was necessary to define the frequency range of the analysis, which depends on the engine main vibration orders (n). Brunetti [7] mentions the orders $\frac{1}{2}$ and 1 as the most dominants on a single-cylinder four-stroke engine and correlates its rotation (RPM) values with the frequency (f) on the eq. (1) that follows:

$$f = \frac{RPM}{60} n \quad (1)$$

Once the engine operates in a rotation range of 2500 to 4100 RPM, a frequency interval from 20 to 70 Hz was defined as critical for the dynamic analysis. With all the parameters set, the Baseline model was evaluated and the results registered for further comparison. The next step was to propose solutions for the vibrations problem that could fit the team's restrictions regarding time and costs. Two solutions were proposed: the addition of a structural reinforcement to increase the frame stiffness and the isolation of the engine by using elastomeric mounts.

Regarding the first solution mentioned, an extra tubular member was added to the structure roof to increase its torsional stiffness, in order to push the frame vibration modes out of the critical frequency range and prevent resonant behaviors. The second solution involved the addition of four elastomeric isolators on the engine attachment points, which were arbitrarily selected according to the engine packaging restrictions. Two mounts

with very distinct properties were selected in order to evaluate the vibrations under the influence of different materials and stiffness. Both countermeasures proposed can be seen in Fig. 2, which shows the modeling of the engine and the isolators details, as well as the roof reinforcement highlighted in red.

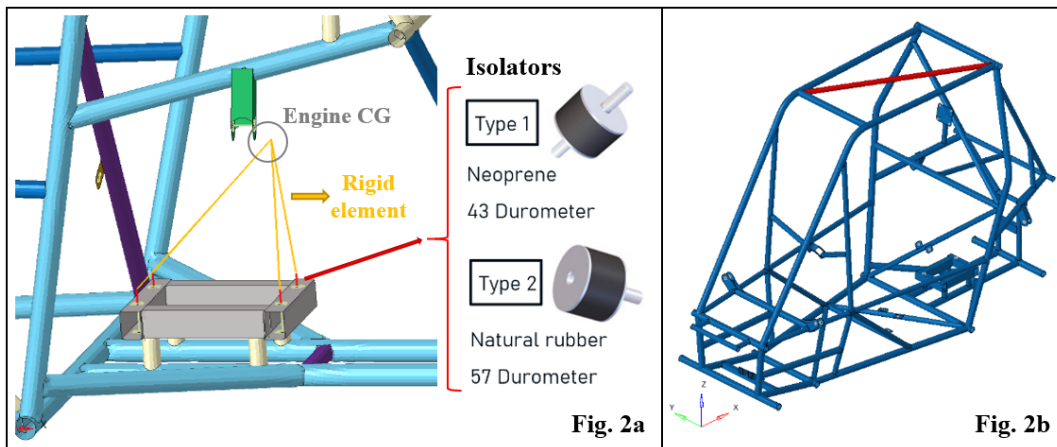


Figure 2. Countermeasures proposed: a) elastomeric engine mounts and b) roof reinforcement.

The isolators properties seen in Fig. 2a are complemented in Tab. 1, according to the data provided by the manufacturer. The RPM Mechanical Inc. [8] guidance is that the polymer static stiffness must be multiplied by a factor of 1.4 to reach its dynamic stiffness, which is the value that should be inputted in the models for the dynamic analyzes. Costa [9] corroborates with this consideration that elastomeric elements have a stiffness variation under a dynamic regime, while Das, Kumar, Banerjee, and Kundu [10] bring in their study the damping factor of the isolator’s materials.

Table 1. Mount’s properties

Elastomeric mounts	Type 1	Type 2
Compression static stiffness (N/mm)	109	1665
Compression dynamic stiffness (N/mm)	152.6	2331
Shear static stiffness (N/mm)	18	80
Shear dynamic stiffness (N/mm)	25.2	112
Damping ratio	0.56	0.3

Having all the variables set and using the previously defined boundary conditions, it was possible to run the dynamic analyzes and evaluate the countermeasures proposed. Four new model configurations were created, which are described in Tab. 2. The objective was to observe the countermeasure’s influence over the vehicle vibrational behavior individually (through proposals A, B, and C) and combined (through proposal D).

Table 2. Model’s details

Baseline	Proposal A	Proposal B	Proposal C	Proposal D
Baseline frame	Baseline frame Mount Type 1	Baseline frame Mount Type 2	Reinforced frame	Reinforced frame Mount Type 2

3 Results and discussion

For a better understanding of the vehicle’s vibrational behavior, the first step is to go through the preliminary results obtained from the torsional stiffness and modal analyzes. As shown, when it comes to the frame design, two different models are being considered: the baseline and the one with a roof reinforcement. The baseline frame model’s torsional stiffness was 1245 N.m/°, a value 6.7% smaller than the one obtained from the team’s

experimental test, 1335 N.m°. This difference is considered acceptable for the model stiffness validation and is justified by the considerable error margins of the experimental procedure and some slight adaptations applied on the virtual procedure. The reinforced frame model registered a torsional stiffness of 1464 N.m°, 17.6% higher than the baseline model.

As expected, the extra torsional stiffness had a significant contribution to the Global Torsion mode (GTO), which has increased from 51.6 Hz in the Baseline configuration to 67 Hz in Proposals C and D. However, this increment in frequency was not enough to move the mode out of the previously defined critical frequency range. The modal analyzes results can be seen on Tab. 3, which includes all the proposed models with the engine mounted. The mode shapes highlighted in red were identified within the critical frequency range.

Table 3. Modal analyzes results

Vibrational modes - Frame with engine										
	Base		Proposal A		Proposal B		Proposal C		Proposal D	
Mode	f (Hz)	Mode shape	f (Hz)	Mode shape	f (Hz)	Mode shape	f (Hz)	Mode shape	f (Hz)	Mode shape
1st	46.4	ELO	6.9	ELA	18.8	ELA	46	ELO	18.8	ELA
2nd	51.6	GTO	9.7	ELO	22.3	ELO	52.1	ELA	22.2	ELO
3rd	52.4	ELA	28.1	EVE	51	GTO	61.8	VB1	51.9	VB1
4th	62.7	VB1	51.3	GTO	53.4	VB1	67.1	GTO	67	GTO
5th	71.6	FTO	65.9	VB1	71.5	FTO	71	FTO	70.9	FTO
6th	93.2	LBE	71.5	FTO	80.7	VB2	95.6	VB2	80	VB2

Labels: ELO - Engine Longitudinal / ELA - Engine Lateral / EVE - Engine Vertical / GTO - Global Torsion
FTO - Front Torsion / VB1 - First Vertical Bending / VB2 - Second Vertical Bending / LBE - Lateral Beat

As seen in proposals A, B, and D, the presence of elastomeric isolators reduced the number of modes within the critical frequency interval, by decreasing the frequency values of the engine modes. The softer is the isolator, the lower will be the engine vibrational modes.

As the main goal of this research was to evaluate the comfort regarding the vehicle vibrational aspect, the approach selected was to analyze the vibrations on the frequency domain. Many Frequency Response Functions (FRFs) were generated for each one of the interest regions previously defined in Fig. 1, but only a sample of the results will be shown, that are the steering wheel attachments accelerations in the Z direction, generated by the three different load cases. Even though just a sample is being presented, it is important to emphasize that the results shown in Fig. 3 represent the behavior observed on all the analyzes performed.

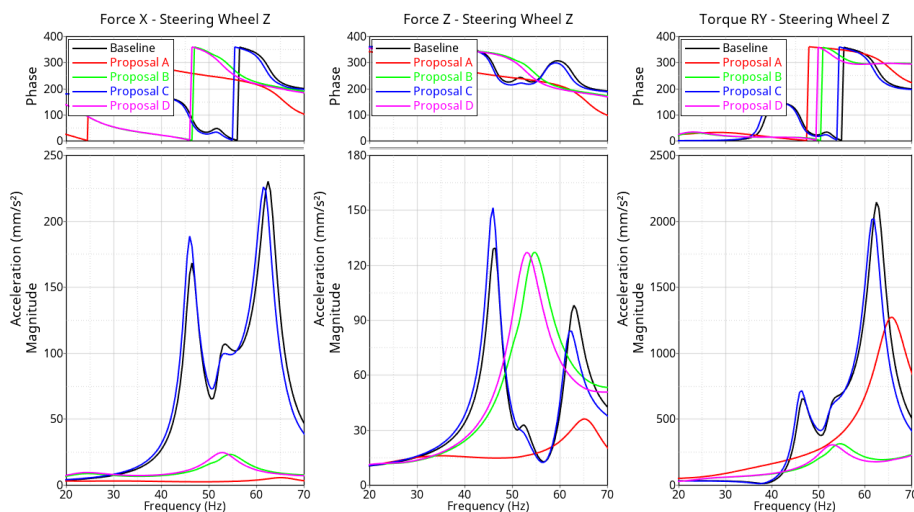


Figure 3. Inertance FRF's of the steering wheel at Z direction

The FRF's showed a significant reduction in the magnitudes of the vibrations on the models with the engine isolated when considering all the measured regions and load cases. However, relevant amplification zones were observed on proposals A, B, and D, showing that the isolator's application can contribute to the increase of the vibration levels at certain conditions. Another aspect observed is that the frame reinforcement wasn't successful

at decreasing the vibration levels, once the models Baseline and Proposal C had very similar behavior. To reinforce these conclusions and summarize the generated results, Tab. 4 was built using a conditional formatting applied column by column, just to compare the model's performances. Based on the data presented there, it's possible to confirm the significant vibration levels reduction with the use of isolators and even elect Proposal A as the best choice for this particular case, when taking only this set of analyzes as a reference.

Table 4. Compilation of the dynamic response peaks at the steering wheel

STEERING WHEEL ATTACHMENT POINTS									
Engine unitary loads	Force X			Force Z			Torque RY		
Response peaks (mm/s ²)	X	Y	Z	X	Y	Z	X	Y	Z
Base	161	133	230	105	95	129	1495	621	2143
Proposal A	8	1	5	5	10	36	202	171	1270
Proposal B	37	14	23	14	64	127	80	115	312
Proposal C	185	55	226	104	45	151	1647	290	2015
Proposal D	36	4	24	21	7	127	74	67	304

4 Conclusions

This paper presented a simplified assessment of the engine-induced vibrations transmitted to the pilot in a Baja SAE prototype. To measure the cockpit vibration levels, a finite element model was built and evaluated through static and dynamic analyses. Two different types of countermeasures were proposed to mitigate the vibrations within the engine's main operational range.

As seen in the results, the usage of elastomeric isolators on the engine mounts proved to be effective on vibrations reduction. However, at certain frequencies, acceleration peaks were observed at the measured points, indicating that the isolators were contributing to the amplification of the vibrations, which is a fact that demands precaution. The model with the softer isolator presented the best results in terms of comfort when compared with the baseline model, though it is important to point out that further analyzes (such as durability) are necessary to a proper engine mount selection.

The increasing of the frame stiffness wasn't effective on the vibration's mitigation, once it had influence only over the specific torsional mode. Thus, it wouldn't be a recommended action to improve the pilot's comfort. Beyond that, the adopted methodology was flexible and agile on the evaluation of the different proposed models, being suitable for the application in this competitive environment.

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