Structural Analysis and Optimization of a Baja Vehicle's Engine Support Using the Finite Element Method

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This work presents the structural analysis of the engine package in a BAJA prototype. Three distinct design configurations were evaluated: (1) the initial engine package; (2) the engine package without the firewall tube; and (3) the engine package without the vertical tubes. Simulations encompassed assessment of deformations, Von Mises stresses, combined tensile/compressive stresses, and buckling modes. All configurations demonstrated satisfactory performance, with high safety factors and stresses well below the yield strength of AISI 1020 steel, the material selected for its favorable mechanical properties, machinability, and widespread structural use in the automotive sector. Configuration (2) achieved a mass reduction of 1.04%, while configuration (3) reduced mass by 0.78%, both maintained the required structural integrity and performance for the intended application.
Keywords: Structural Optimization. Engine Package. Finite Element Method. ANSYS. Buckling.
Abbreviations: BLC, Overhead Lateral Cross member. CLC, Upper Lateral Cross Member. FABup, Fore-Aft Bracing Members. FEM, Finite Element Method. RRH, Rear Roll Hoop. SAE, Society of Automotive Engineers.

In automotive design, especially for off-road vehicles such as Baja SAE prototypes, every structural component required careful engineered to ensure performance, safety, and lightweight construction. In this context, the engine package served as a key structural connection, linking engine to the vehicle structure and must support not only its weight but also the dynamic loads generated during operation, such as vibrations and torques,as reported by Shigley and colleagues [1].

Therefore, optimizing this structure involved balancing strength and mass variations, to achieve a mid-term between durability and performance [2], often using softwares.

The structural analysis of mechanical components using the finite element method (FEM) has become established as an essential tool in engineering design development. This method enables the simulation of structural performance under different loading conditions, allowing the pre-validation of a concept, optimizing the time and resources required for physical testing as reported by Mac Donald [6]. Among the most relevant parameters evaluated in the simulations were deformations, Von Mises stresses, and buckling modes [3].

The Von Mises stress represented widely used criterion for predicting the failure of ductile materials, such as steel, and is based on distortion energy. For materials such as AISI 1020 steel, which exhibits yield strength of approximately 351.57 MPa, failure typically occurs when the equivalent stress reaches about 60% of that value, according to the safety criterion adopted [2].

Another critical aspect for automotive structures is buckling, a structural instability caused by compressive loads, whose occurrence depended on the component's geometry, boundary conditions, and material properties [4]. Buckling analysis in ANSYS employed critical load factors to predict the multiplier of the applied load required to reach the first buckling (unstable) mode.

Furthermore, structural fatigue is also considered in many assessments. Components subjected to cyclic loading were evaluated for service life, especially in automotive applications where vibrations and repetitive loads are constant [5]. In the present study, the observed stresses remained sufficiently low to ensure infinite life of the

Received on 17 October 2025; revised 21 December 2025.
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J Bioeng. Tech. Health 2026;9(1):32-37

structure, according to the Goodman Curve criteria which according to Norton [2] is conservative, but it keeps close to the experimental data.

Finally, structural optimization focused on reducing mass without compromising the integrity of the system. Weight reductions were particularly important in mobility projects such as off-road vehicles where performance and energy efficiency are directly affected.

The objective of this study was to conduct a comprehensive structural analysis of the engine package using ANSYS simulations, with the aim of optimizing its geometry and mass while preserving the mechanical integrity of the assembly.

The chosen software for the present study was ANSYS, because it has all the tools necessary for the analysis, and its web support.

Materials and Methods

The simulation modeled the engine as a concentrated mass of (32 kg) placed at its center of mass (Figure 1).

A torque of 30 N·m was applied at the engine package mounting points, with constraints imposed on the central differential supports, this torque was based on the engine's maximum torque. The material selected was AISI 1020 steel and the finite-element model used beam-type (1D) elements with a mesh size of 20 mm. The sequence of the steps are shown in Figure 2.

The engine package comprises four attachment points secured with bolts. In the simulation, the applied torque corresponded to the engine's maximum output and was applied at the output shaft location. Structural constraints were imposed

Figure 1. Engine package.

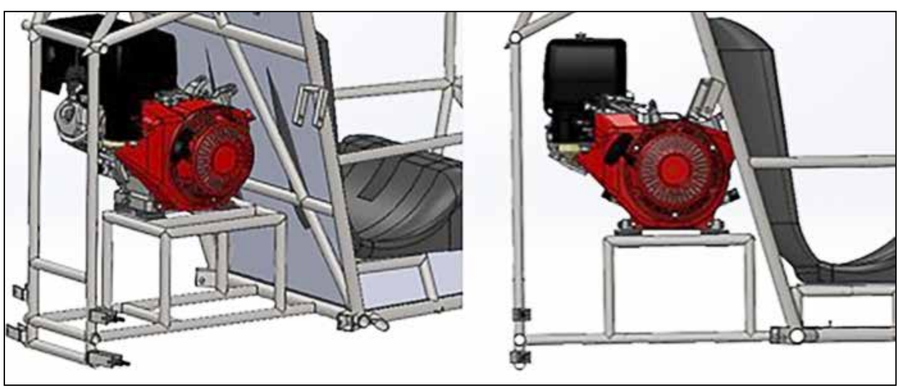
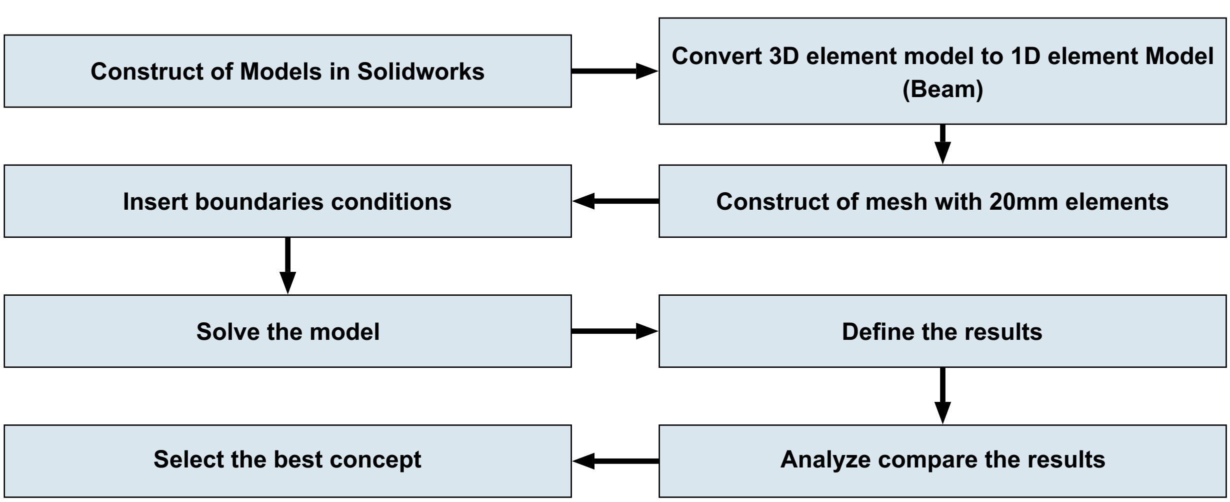
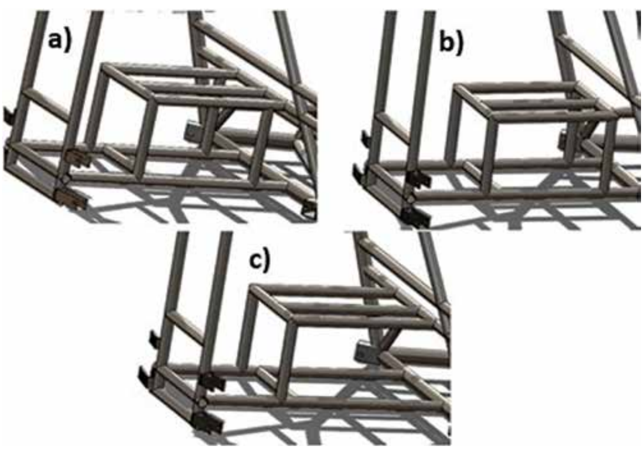


Figure 2. Visual representation of the method.



at the central differential attachment points, which are also secured by four bolts at a lower position on the chassis (Figure 3). This attachment location was selected because the differential serves as the primary component connecting the wheels to the vehicle structure; thus, the reaction forces resulting from the engine torque are transmitted through it.

Figure 3. Comparative view of the 3 engine package configurations: a) Engine Package 1, b) Engine Package 2, and c) Engine Package 3.



The three engine package was: Engine package 1, which has the anchorage in the firewall and in the bottom tubes. Engine package 2, which has the anchorage only in the bottom tubes, and the Engine package 3, which has the anchorage only in the firewall.

The analyses performed consisted of static and buckling studies aimed at evaluating the stress and deformation states across the different engine package configurations. The buckling analysis was used to identify the critical modes of structural instability. Critical modes were defined as those exhibiting the lowest buckling coefficient, i.e., the modes that indicate instability under the smallest applied load to the structure.

Results and Discussion

For this study, there were three results from each engine package. The total deformation, which shows how the body deforms based on the load. The Von Mises stress, that is a combination of the axial stress and shear stress, based on the Von Mises Ellipse. And the Buckling modes, which is the bending of a beam, or in this case, a chassis member, by compression, the number analyzed in this result is the buckling factor, that is a multiplier load that leads to the buckling mode.

Engine Package 1

The maximum deformation of the chassis was 0.00687 mm, concentrated in the upper-rear section of the chassis, specifically at the junction between the Rear Roll Hoop (RRH) and Overhead Lateral Cross member (BLC). On the engine package, the maximum deformation was 0.00598 mm, concentrated at the torque application point.

The highest stress was also located at the torque application point, being caused mainly by the bending tendency of the engine motion. The stress at this point was 7.615 MPa, resulting in a safety factor of 26.36 based on the criteria used by Robert Norton, thus there was a good margin for refinement of this chassis component (Table 1, Figure 4). The critical buckling mode occurred with a load factor of 9128, i.e., the torque would need to be increased by a factor of approximately 9.000 times to produce this buckling mode, indicating structural stability. The critical buckling mode developed in Upper Lateral Cross member (CLC) and in the Fore-Aft Bracing members (FABups). These values were used as the reference condition for the support selected for optimization.

Engine Package 2

In this model, the maximum deformation of the engine package was 0.00521 mm, an improvement compared to the first model. Conversely, because fewer attachments were positioned on the chassis, the chassis maximum deformation increased to 0.0083 mm, and was distributed across all upper chassis members. Stresses decreased slightly while remaining concentrated at the engine fastening point.

Figure 4. Deformation, Von Misses stress and critical buckling modes of engine package 1.

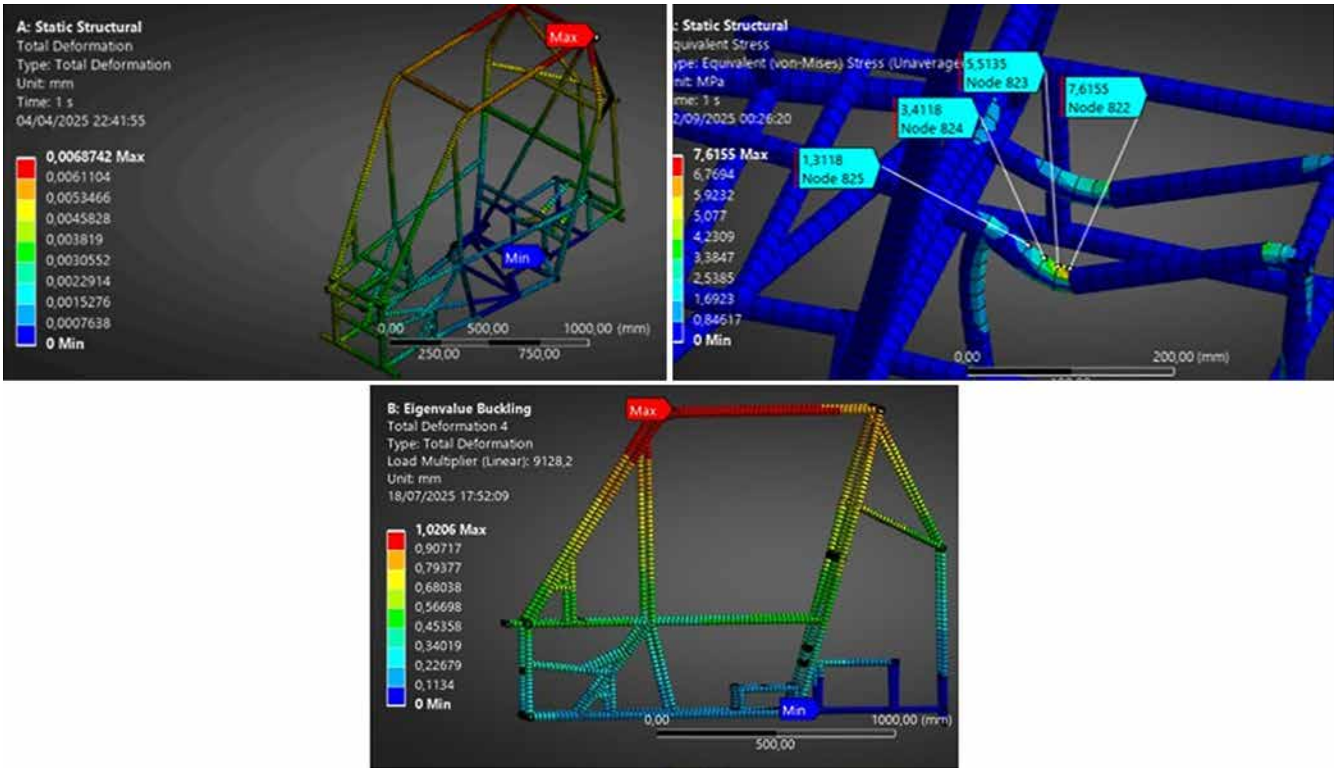


Table 1. Stress engine package 1.

Engine Package 1	
Stress (MPA)	**Node**
7.6155	822
3.4118	824
5.5135	823
1.3118	825

The peak stress reached 6.9778 MPa, corresponding to a safety factor of 30.13, as observed in Table 2, and Figure 5. The critical buckling load factor was lower than in the first model but still far from a critical condition (7049), which also indicates structural safety. This buckling mode occurs in the front portion of the engine package.

Engine Package 3

This configuration exhibited larger deformations both in the chassis, reaching a 0.08080 mm, and in the engine package, reaching 0.04958 mm, nearly one hundred times higher than the others package options. Despite this, the result could still be acceptable if a substantial mass reduction were achieved. Stresses retained a similar distribution, with the maximum at the engine fastening point; in this case the peak stress was 7.9 MPa, corresponding to a safety factor of 25.31 (Table 3, Figure 6). The critical buckling load factor for this configuration was the lowest of all, 5489, which was still well above the threshold for buckling failure but considerably smaller than the other cases. This buckling mode was

Table 2. Stress engine package 2.

Engine Package 2	
Stress (MPA)	**Node**
7,1244	517
5,2812	518
3,4400	519
1,6074	520

Figure 5. Deformation, Von Misses stress and critical buckling modes of engine package 2.

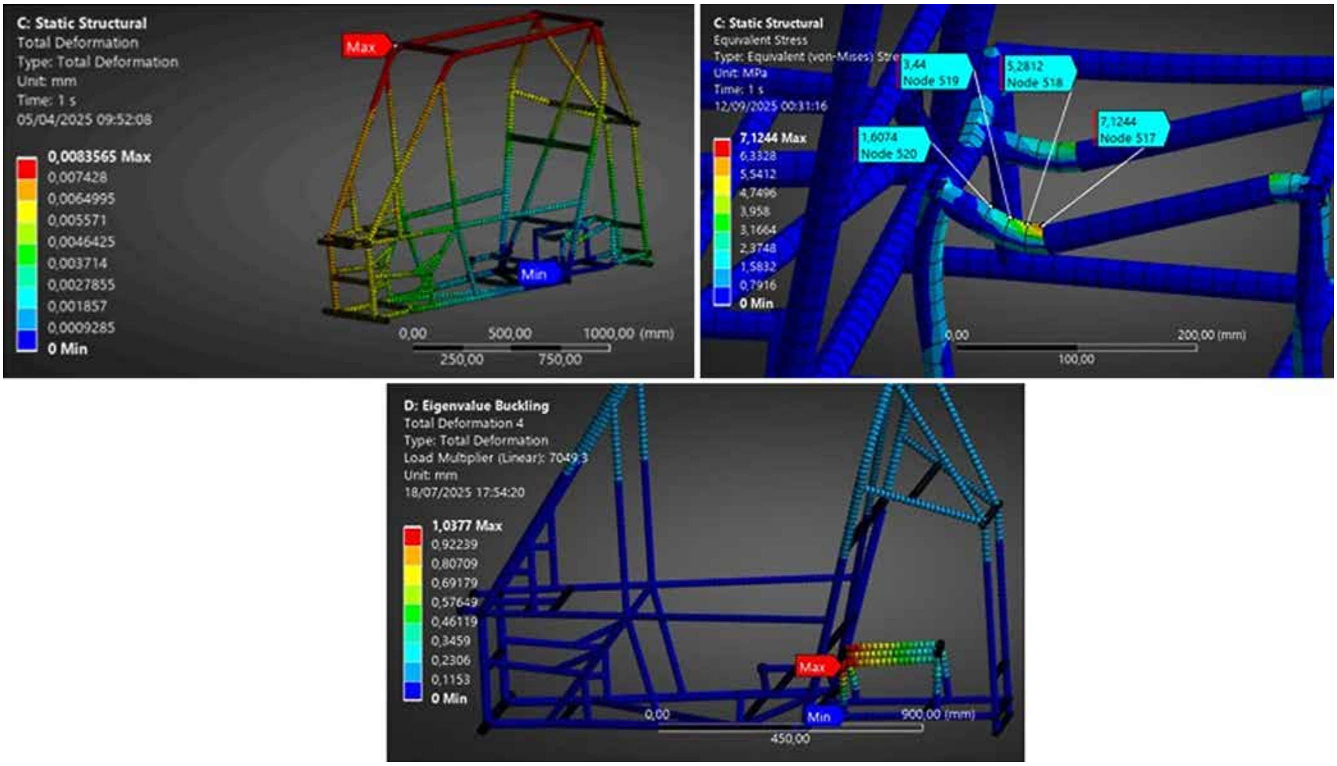


Table 3. Stress engine package 3.

Engine Package 3	
Stress (MPA)	**Node**
7,9777	818
6,7451	819
5,5196	820
4,3070	821

concentrated mainly in the rear transverse bar, due to the absence of rear support on the chassis.

Comparative Analysis of the Engine Package Configurations

Based on the results, a comparison was made of the total mass of each chassis configuration using Solidworks, and the configuration that presents the best balance between load performance and mass was selected. The percentage mass reduction is given relative to the initial model, which was Engine Package 1. Table 4 presents the comparison of the values. Therefore, engine package configuration 2 was selected for the vehicle, as it provided the highest percentage reduction in mass while resulting in only minor increases in stress and deformation. Furthermore, configurations 1 and 3, which include a mounting point directly attached to the panel against which the driver rests, would transmit higher vibration levels to the driver's body, thereby compromising the overall ergonomics of the vehicle.

Conclusions

The three configurations meet the mechanical strength and stability criteria. The second configuration (2) proved most advantageous, showing the largest mass reduction and the highest safety factor. However, when the assembly was evaluated together with the chassis, interference was observed between the engine package and the driveshaft arm; therefore, the package position must be modified and the simulations repeated accounting for the 4x4 drivetrain geometry.

Figure 6. Deformation, Von Misses stress and critical buckling modes of engine package 3.

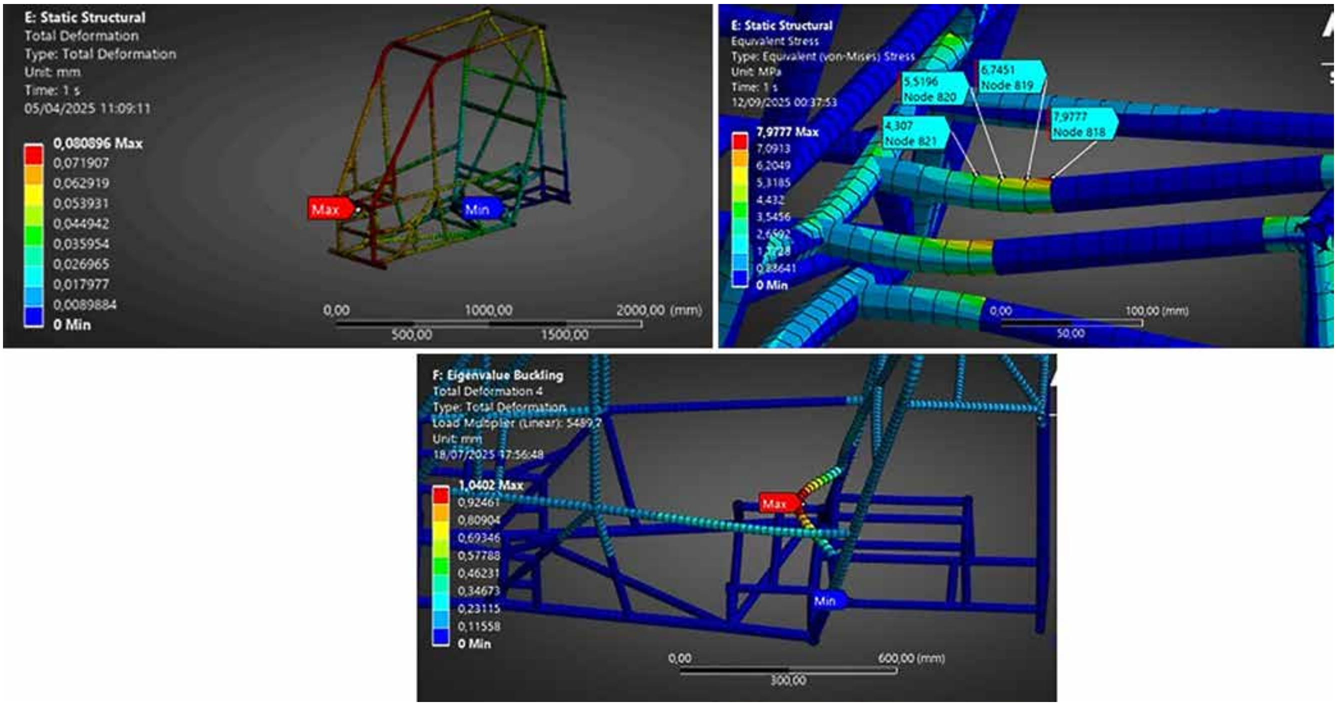


Table 4. Stress engine package 3.

Engine Package	1	2	3
EM def. (mm)	0.00598	0.00521	0.04958
Chassis def. (mm)	0.00687	0.00835	0.08089
Maximum stress (MPa)	7.6	7.1	7.9
Mass (Kg)	41.182	40.755	40.859
Reduction (%)	-	1.04	0.78

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