Analysis and Optimization of a Passenger Bus Frame Through Finite Element Software

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Abstract: The objective of this work is analyze and optimize a passenger bus frame using finite element software, in different static and dynamic conditions.

Through a static analysis, torsion and bending constants were extracted using boundary conditions different sets for each one of them. With a linear perturbation step, free and forced vibration modes of the system were obtained and a frequency response analysis was developed.

For dynamic analysis, boundary conditions were defined from dynamic automotive equations. Evaluated conditions were suspended weight, acceleration, braking, cornering, and cornering and braking together.

The bus body geometry as well as different suspension components was obtained directly from the CAD files. The degrees of freedom were also identified from the original drawing, either movements between the suspension components as between suspension and bus body elements. Most components were modeled by wire elements, and some others as shell, to which the mechanical properties of steel and different cross-sections were assigned. Additionally, connectors were used to model dynamic components such as air springs, shock absorbers and tires, whose behavior was described by characteristic curves. This modeling methodology allow us save time and computational capacity.

The most critical elements were found to be air springs fasteners and surrounding elements, which according to dynamic analysis carries a load elevated percentage of the load, while for curving conditions the torsion bars fasteners and nearby elements became critical components. After optimization process, structural frame weight was reduced 7.76% and, a decrease up to 50% of stress concentration at drive and auxiliary axles fasteners was reached.

Keywords: Design Optimization, Minimum-Weight Structures, Multi-Body Dynamics, Bus Structure Models, FEM Modeling.

1. Introduction

Due to the high costs associated to the development of new vehicle models, computer simulations of vehicle dynamics become more and more important in this process. While large multinational companies employ integrated design teams and the latest advances in computational technology and software using the economy of scale, many smaller companies are active in the design and construction of passenger buses according to local requirements and market specifications, principally in countries with emerging economies where usage conditions may significantly differ from the design specifications for Europe and the United States. To maintain a complete design team together with computational facilities and specialized software is often not feasible for such companies.

Vehicle dynamics, also called handling simulation, is only a small part of the design process, which means that it is uneconomic for most small companies to own the corresponding licenses for such specialized softwares since they only use them temporarily. Software used in structural analysis is much more common in the industry today and simulating vehicle dynamics with this kind of software could result in direct and indirect economic savings and new design possibilities. Due to the high license costs, the companies want to be able to do as many simulations as possible in software. Abaqus® seems to be a good alternative to the existing specialized software because it is possible to perform the desired handling simulations in a straightforward and flexible way.

2. Model development

2.1 Geometry generation

From the solid model, space position of all points that allow generation of each wire elements were defined, those wires are related to the center of each beam element and have a exact intersection with neighbor elements. With the space positions of all point, drawing of geometry was made in Abaqus® considering different independent sections, this in order to simplify the cross section and cross section direction assignment.

Connection between shell and wire geometries were made through connectors defined as weld. For static and linear perturbation analysis only a body bus frame was required, while for dynamic analysis mobile mechanisms that simulate the three axles suspensions were integrated.

2.2 Boundary conditions determination

2.2.1 Static analysis boundary conditions

First and second conditions were developed in order to obtain torsion and bending strength trough a static analysis. For torsion were applied two loads with the same magnitude ($50\ kN$) and direction but contraire sense at the frontal spring air fasteners ($F_3^{tr} = -F_3^{tl}$); fasteners of rear spring fasteners (drive and auxiliary axles) were encastred ($U_1 = U_2 = U_3 = UR_1 = UR_2 = UR_3 = 0$). For bending loads are applied in the same sense, magnitude ($10\ kN$) and direction ($F_3^{fd} = F_3^{fi}$); encastre nodes ($U_1 = U_2 = U_3 = UR_1 = UR_2 = UR_3 = 0$) are the same that torsion. Torsion and bending loads were calculated in order to maintain stress levels under $250\ MPa$.

Equations (1) and (2) represent torsion and bending constants respectively

$$K_t = \frac{M}{\alpha_t} \tag{1}$$

$$K_b = \frac{2FL}{\alpha_b} \tag{2}$$

Where

$$\alpha_t = \tan^{-1}\left(\frac{v_A - v_B}{W}\right)$$

$$\alpha_b = \tan^{-1}\left(\frac{v_A + v_B}{2L}\right)$$

W and L represent bus wide and long considering loads applying and fully constraint points; v_A and v_B are vertical displacement of the load application points; α_t and α_b are the torsion and bending angles.

2.2.2 Boundary conditions for linear perturbation

Third analysis seeks identify natural and forced frequencies of the system in order to avoid presence of any resonance conditions within the work frequencies threshold.

Through free vibration analysis the different vibration modes can be obtained, this mean, natural frequencies of the system. The only parameter defined is maximum frequency value up to which the analysis will be developed. This value is determinate by the working range of a four times Diesel engine governed to $2100\,rpm$, which correspond to $70\ Hz$ considering second firing order.

Additional to the maximum frequency value, for forced vibration analysis some boundary conditions have to be defined. In this case all spring air fasteners were pinned $U_1 = U_2 = U_3 = 0$; unit forces at the engine fasteners were applied in order to simulate the transmitted loads to the bus frame by engine operation.

With the free and forced modes, it is obtained the information required for the frequency analysis response of the bus frame. This process involves the selection of one or several nodes; this selection is made by interest sites and/or experience.

2.2.3 Boundary conditions for dynamic analysis

This analysis has as objective obtaining the behavior of the bus frame under different loading events that can occur during useful life cycle. With the addition, as an important part, of the mobile mechanisms that simulate each suspension of the three bus axles, it is considered that the model has sufficient elements to obtain an accurate dynamic analysis.

Engine, transmission and air conditioned system are represented as point mass with values of $1343\ kg$, $526\ kg$ y $250\ kg$, respectively. Those values were obtained experimentally.

After the bus frame weight was obtained, suspension systems, engine, transmission and air conditioned system were defined as a mass point located at the gravity center, this was calculated previously with a value of $14271\ kg$, which represent a maximum load condition.

Due to lack of information about speed and turning radius standards for a cornering condition, a speed of $40\ km/h$ and a turning radius of $14\ m$ are proposed, those values allow to get analytically load transference from one side of the bus to the other: applying this condition provide information about bus frame elements surrounding suspensions fasteners.

3. Finite Element Analysis

3.1 Torsion

Figure 1 shows displacement direction of the nodes due to boundary conditions defined for torsion analysis, at the mini-figure inside figure magnitude is also shown. Bigger displacements, just as predicted, are at the external side of the frontal axle. Notice that displacement resultants are plotted.

Images reported for those conditions (torsion and bending) a scale factor (12) is applied in order to provide a good idea of the bus frame strain.

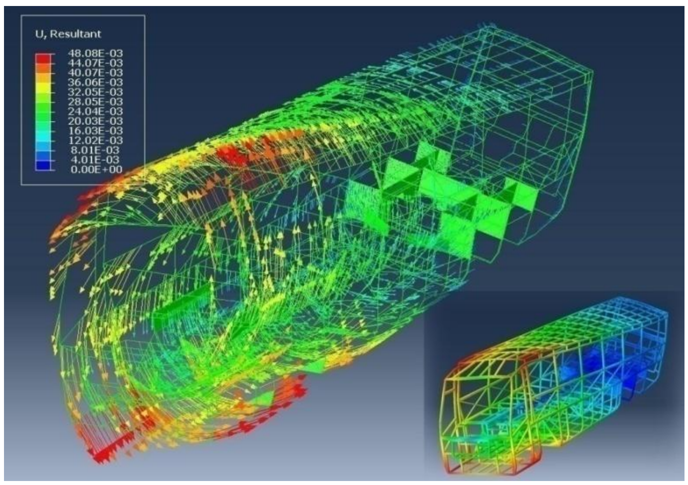

Figure 1. Displacements resultant, direction and magnitude [m], for torsion.

Stress distribution for torsion test is shown at figure 2, the most elevated values of stress are located at the external elements of the bus frame, this is, at the sides, roof, and base of the structure, mainly at the joints of the elements. Although there is a certain contribution from the central part of the bus frame, solicitations are mainly carried by external elements.

Inferior elements of the side frame reach $150\,MPa$ aproximately, located at joints. This value is nearby some other joints of the superior elements, reaching $170\,MPa$. However, the maximum ($200\,MPa$) value is located at a couple elements that work as joint between roof and sides.

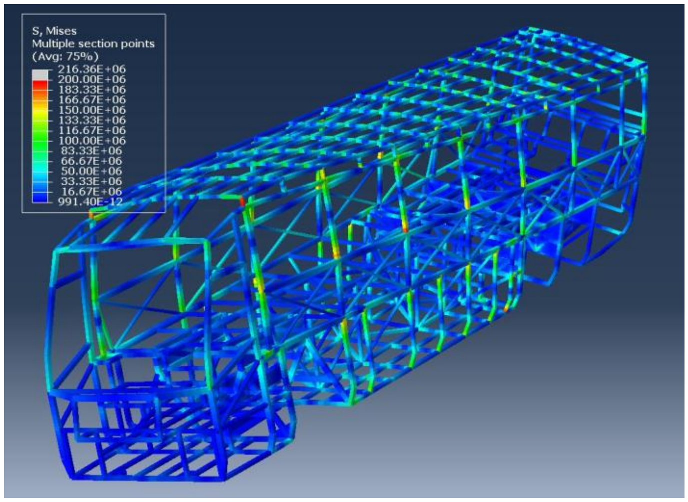

Figure 2. Von Mises stress distribution [Pa] under torsion condition.

3.2 Bending

Figure 3 shows that node displacements is located at front of the frame bus, displacement magnitude of elements near load application is $26\,mm$ aproximately. This value is used to calculate the bending constant according to equations 1 and 2. Maximum displacements have a magnitude of $35\,mm$.

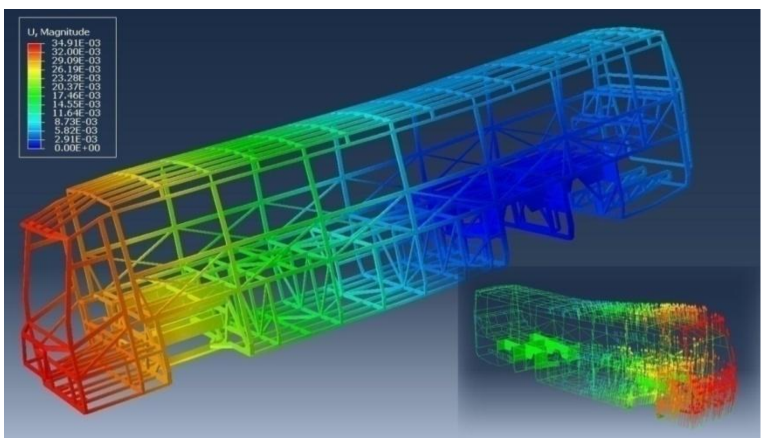

Figure 3. Displacements, magnitude and resultant [m], for bending.

Stress distribution (Figure 4) shows a maximum value, $176\,MPa$, at points where boundary conditions (encastre) were applied. Side frame elements, near to rear axles, and some elements surrounding suspensions fasteners present the highest stress with an average of $130\,MPa$.

From the two previous analyses torsion and bending constants can be obtained and are presented in table 1.

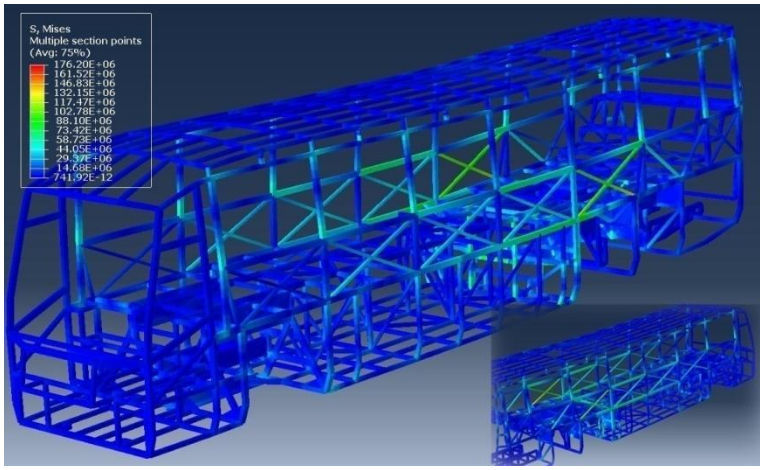

Figure 4. Von Mises stress distribution [Pa] for bending.

Table 1. Torsion and bending constants for initial bus frame (x10^{-6} [Nm/rad])

Torsion	Bending
2.466	28.229

Torsion and bending constants obtained from those analyses are taken as reference values for this work. Due to simplification of the system, actual values for this constants are suppose to be greater.

3.3 Frequency response analysis

Point mobility (Velocity-Frequency response function) was extracted a different frame location and is shown in figure 5. Based on the analysis output there are no frame resonance frequency at the working range.

Comparisons shown for free and forced vibration close to $700\, rpm$ (ralenti engine), do not match up, for figure 6 a) as well as 6 b) free vibration amplitude are concentrated at drive axle plates, while for forced vibration are concentrated at a transversal rear element; for 6 c) is the roof where high amplitudes appear for free vibration, this do not occur for forced vibration.

Vibration modes showed in figure 7 a) do not match up for free and forced vibration; in 7 b) roof and engine fasteners present similar deformation, which indicates that under this frequency an excessive vibration can occur in this region; in 7 c) again, vibration modes do not match up completely, however rear frame deformation still appearing.

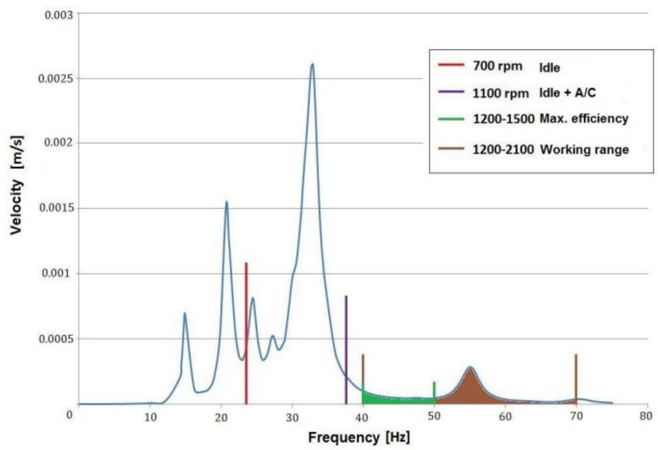

Figure 5. Velocity of node close to engine fasteners within working threshold.

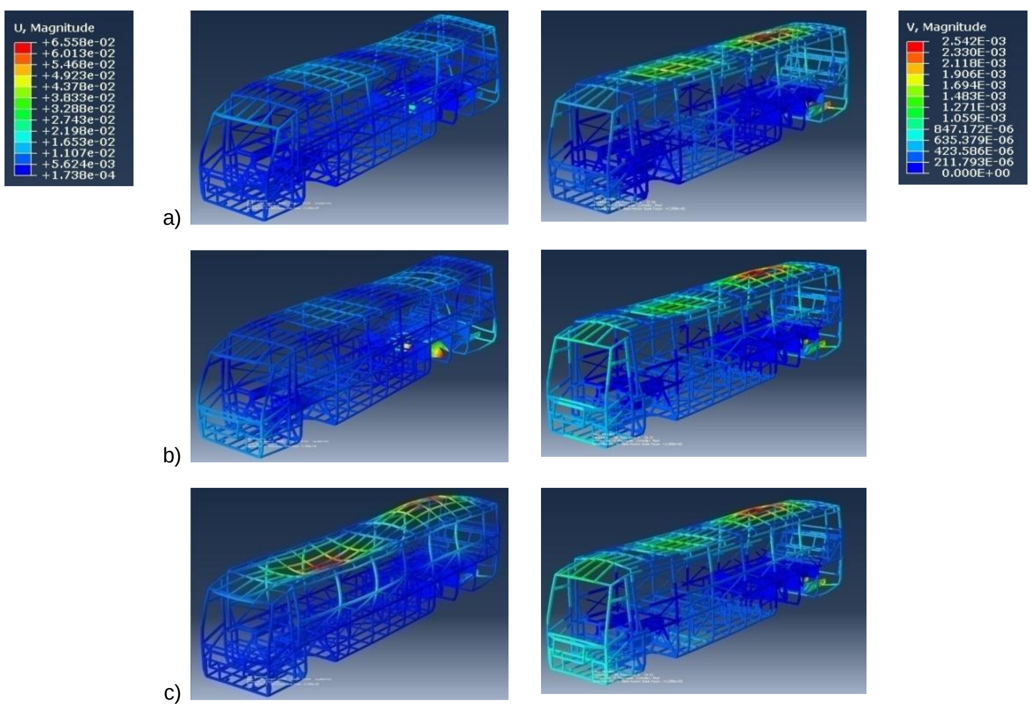

Figure 6. Vibration modes couples, free (left) and forced (right), corresponding with frequencies close to, 23. 33 [Hz]. a) 23.4 [Hz], b) 24.2 [Hz] y c) 24.4 [Hz]; displacement and velocity in [m] and [m/s] respectively.

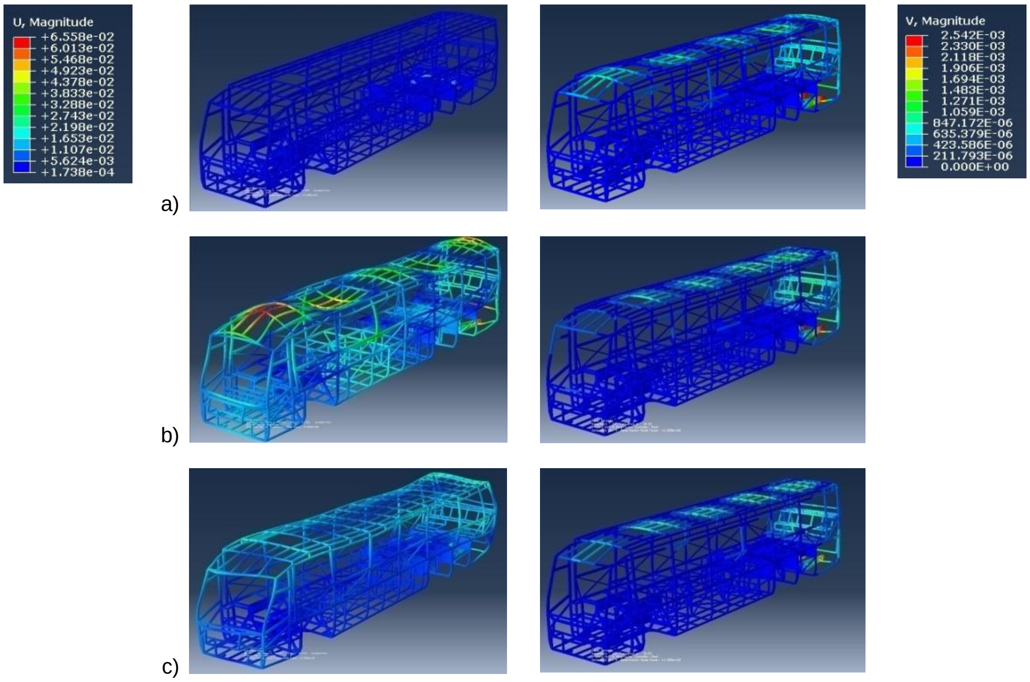

Figure 7. Vibration modes couples, free (left) and forced (right), corresponding with frequencies close to, 36.66 [Hz]. a) 33.6 [Hz], b) 34.5 [Hz] y c) 35.5 [Hz]; displacement and velocity in [m] and [m/s] respectively.

3.4 Dynamic analysis

3.4.1 Static weight, acceleration and braking

Stress distribution for static weight, acceleration and braking are very similar, so only one image is sufficient to obtain information. Under this conditions, maximum stress is located in one of the engine fasteners, with a stress value of $237\ MPa$.

Decreasing maximum graphic limit to $100\ MPa$, in figure 8 can be seen the work of other elements which stress magnitude is less than $70\ MPa$. Those levels of stress appear at sides frame, near drive and auxiliary axles, and slightly at front axle.

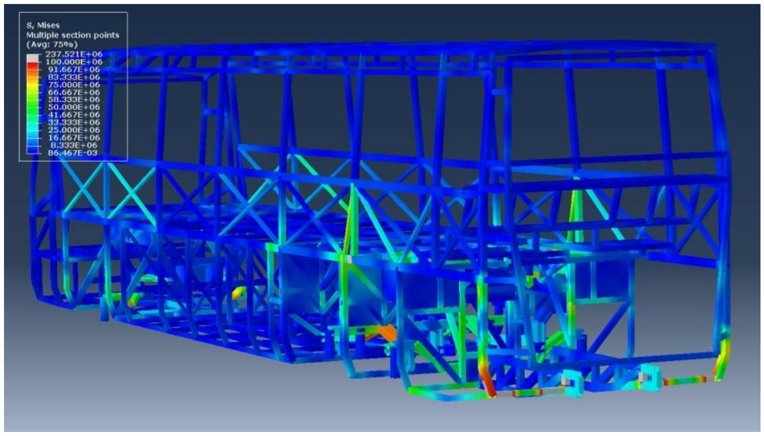

Figure 8. Von Mises stress distribution [Pa] under maximum load condition.

Since gravity center is located nearest rear axles, load transfer due to acceleration is smaller to those axles; then displacement of rear elements are smaller than for braking condition, where load transference is bigger, and hence, displacement of front elements is bigger. Those accelerations do not generate structural strains that involve substantial change in stress distribution.

3.4.2 Cornering

Getting relative displacements between four points in order to compare them with their equals at the torsion essay, it was found that maximum displacement is less than $0.4\ mm$, which is ten times smaller than displacement obtained for torsion analysis, accordingly the bus frame is capable of support severe solicitations.

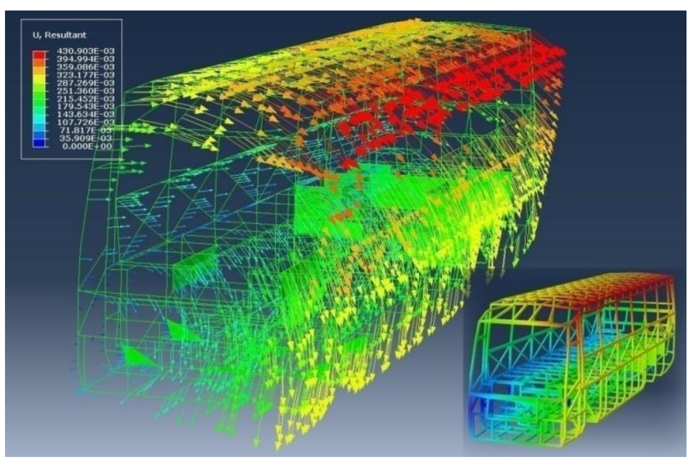

Figure 9. Displacements [m], direction and magnitude, under a cornering condition.

Is at rear section where the highest stress values are located. Figure 10 shows how some suspension fasteners reach elevated stress, even superior to $200\ MPa$. First three circles indicate elements that have connection with inferior arms of drive suspension (1), and superior (2) and inferior (3) arms of auxiliary axle. The punctual connectors can generate stress concentration at those points, and then, there is an error associated to punctual load application. For frame elements (4) and (5) it cannot be considered that stress concentration is due to connectors, however all of them give an opportunity to improve the bus frame.

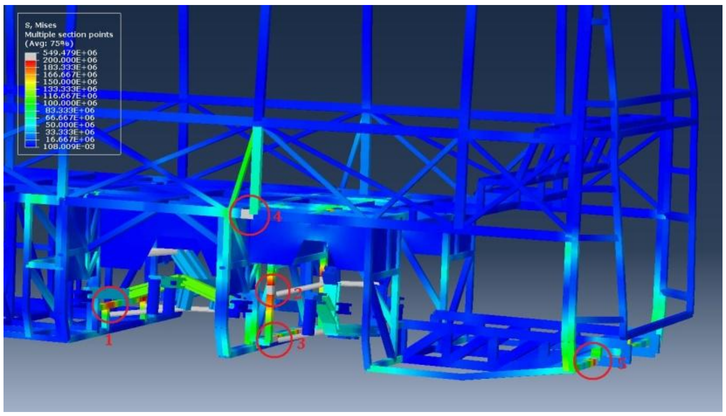

Figure 10. Von Mises stress distribution [Pa] at rear bus frame under cornering condition. Curve with 14 [m] radius at 40 [km/h].

Front axle surroundings has a stress distribution shown in figure 11, maximum visualization limit was fixed to $100\ MPa$. No one of the elements around front suspension fasteners has stress values which magnitude rises up to $75\ MPa$.

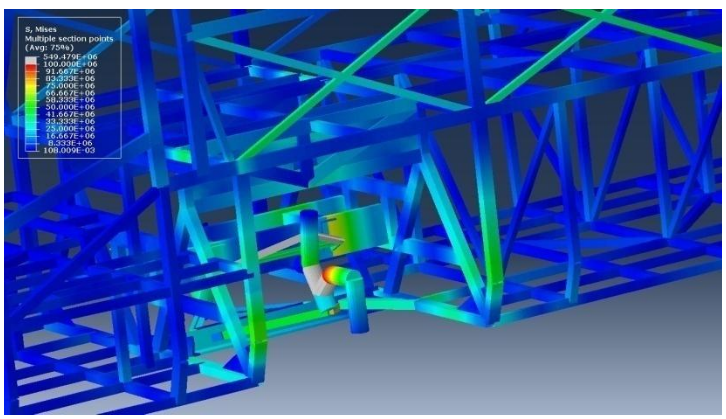

Figure 11. Von Mises stress distribution [Pa] at front bus frame under cornering condition. Curve with 14 [m] radius at 40 [km/h].

3.4.3 Road Condition

Figure 12 shows vertical displacements of front, drive and auxiliary axles fasteners, as well as gravity center, for a road condition (bump).

Front axle present stress levels under $100\ MPa$; drive axle do not show stress levels higher than $150\ MPa$; auxiliary axle has stress concentration at air spring fasteners, always under $150\ MPa$.

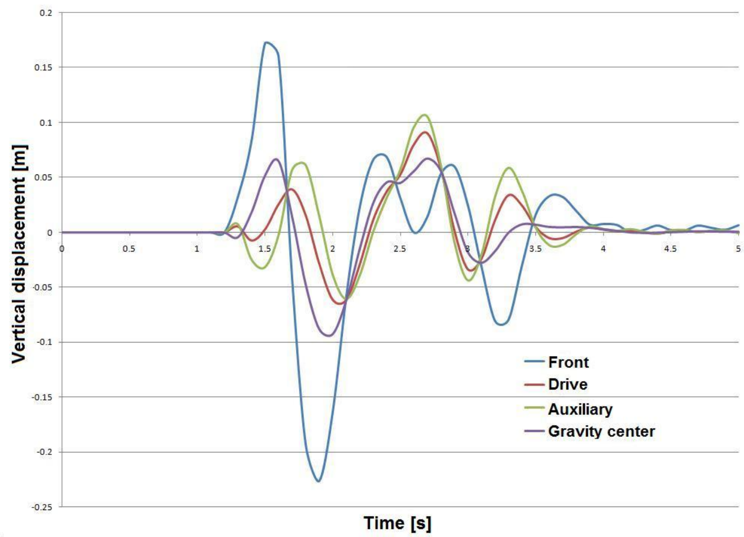

Figure 12. Vertical displacements of nodes close to each bus axles as well as gravity center under a road condition (bump).

4. Optimization

According to specific problem needs, bus frame optimization has as main objective mass reduction maintaining general strength and increasing it where it is required. In order to achieve the main objective, the thickness of some regions was changed repeatedly, increasing it or decreasing it depending on needs, additionally some pieces were deleted and some others added taking into account initial analysis.

Modification that contributed mainly to mass reduction was thickness reduction, this reduction were made at regions where stress levels were under $100\ MPa$, commercial thickness were always used. Addition of new elements to avoid stress concentration was required mainly at region surrounding drive and auxiliary suspension fasteners, as well as engine and transmission. Mass increasing results negligible after thickness adjustment.

After evaluation of different configurations, option with better results is presented. Bus frame mass went from $3451\ Kg$ to $3183\ Kg$, which represent a decreasing of 7.76%. Effects of modifications are shown at table 2, where initial torsion and bending constants are compared with optimized constants. Those values represent a decreasing of 6.2% and 3% for torsion and bending strength respectively.

Table 2. Torsion and bending constants for initial and optimized bus frame.

Strength x10^{-6} [Nm/rad]	Torsion	Bending
Initial	2.466	28.22
Optimized	2.311	27.38

The figure 13 is obtained at the region of interest considered for frequency analysis response; this figure shows the effect of modifications. Reinforcement elements added in the bus frame increase velocity for two peaks before $22\,Hz$. From 20 to $35\,Hz$ approximately, velocity magnitude is lower for optimized frame, after this frequency there is a similar behavior with a velocity magnitude bigger for optimized than initial bus frame.

Modifications and pieces addition, that increase region strength, do not result directly in an improvement for frequency analysis response. If it is certain that the highest velocity magnitude belongs to initial bus frame, there are some points where velocity is higher for optimized bus frame, pointing complexity to reach accurate results.

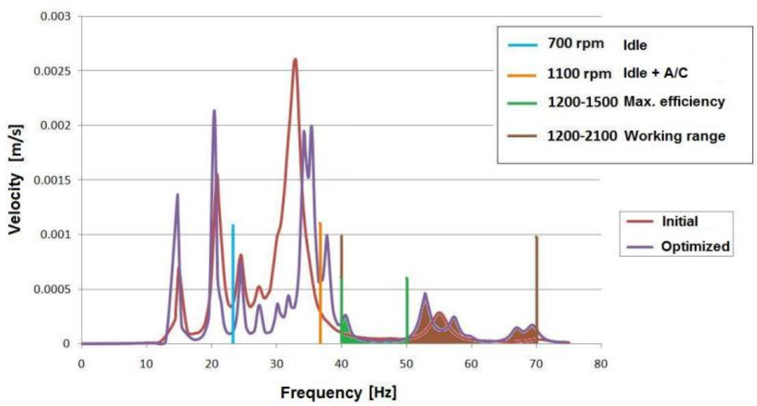

Figure 13. Velocity of a node close to engine fasteners within working threshold for initial and optimized bus frame.

After modifications at region shown in figure 14, a reduction of 50% in stress levels was achieved. Addition of elements with the function of distribute load concentrations in regions 1, 2 y 3 to adjacent elements had a remarkable effect taking Von Mises stress from $200\,MPa$ approximately to less than $100\,MPa$ at this three regions.

Point 4 shows an element that remained without modification; however stress levels decreased from $240\,MPa$ to $210\,MPa$, slightly higher than maximum limit defined for scale at the same figure. Deceasing of stress levels can be explained by addition of elements to nearby regions, elements that will absorb or redirect loads. Piece added at point 5 changes Von Mises stress levels from $200\ MPa$ to less than $60\ MPa$.

Stress distribution, corresponding to this region analysis under a cornering condition do not shows graphical differences with figures 10 and 11. Modified bus frame has no important differences for this condition specifically. Moreover, for cornering, road conditions applied to optimized bus frame do not show significant change with figure 12.

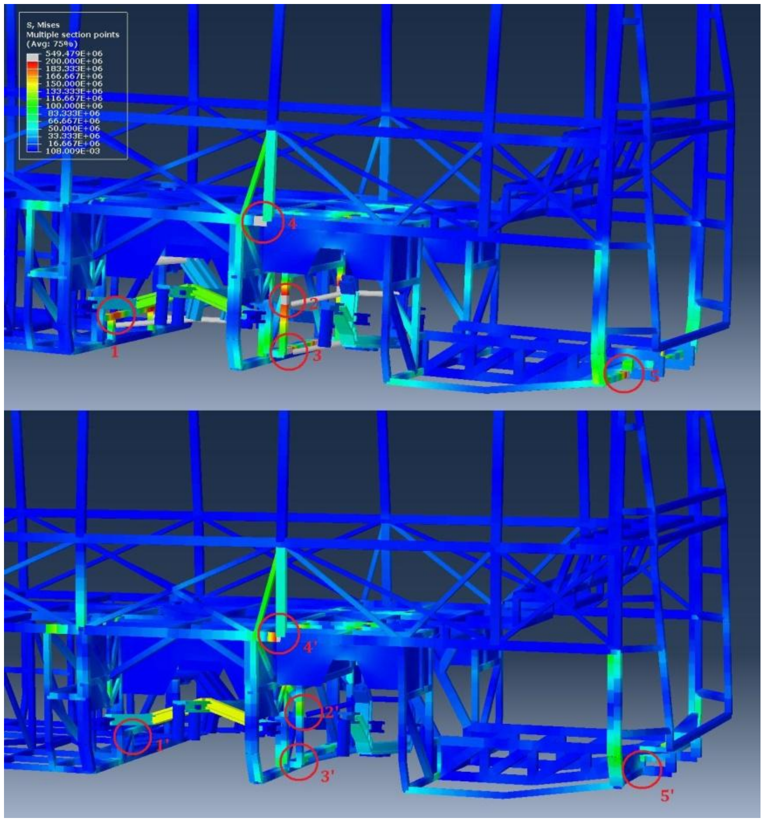

Figure 14. Von Mises stress distribution [Pa] at rear initial (up) and optimized (down) bus frame, under a cornering condition.

5. Conclusions

Once model was obtained a variety of analysis were made using different boundary conditions. Generation of different evaluations was made in an easy and efficient way. Enough information was contained into the model to achieve a full structural analysis.

Static analysis allowed obtaining two important constants considered in this kind of structures, torsion and bending. Through linear perturbation analysis, frequency modes of system were identified, giving data to generate a frequency analysis response considering a cyclic load related to engine working; additionally this analysis provided connection errors between structural elements. Dynamic analysis allowed exploring software capabilities including mobile mechanisms to the model. Simplified beam and plate elements required less computational resources.

Optimization process, which involved cross section changes and adding of new elements, was developed in an effective and simple manner, this gave the possibility of making different combinations until find a final bus frame presented here.

Modifications, those whom increased and decreased mass, resulted in a decreasing of 7.76%; this value results bigger than torsion and bending constants decreasing.

New small elements generated a decreasing over 50% at drive and auxiliary suspension fasteners, where some stress concentrations were found. For engine fasteners load distribution was improved with new plates, geometry required by design requirements.

6. Acknowledgments

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