Computational Analysis of Body Stiffness Influence on the Dynamics of Light Commercial Vehicles

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Abstract. This paper presents the computational analysis of the frame flexibility influence on the dynamics of light commercial vehicles. The goal is determine the handling performance influence of a lightweight aluminum rear frame pre-design and its comparison with the original steel body behavior. A bibliographical review points out FEM and Multibody computational integration as the best way of approaching the problem and a theoretical analysis highlights the main handling related issues originated from the flexibility of the body. Simplified FEM models are constructed using Hypermesh in order to obtain the static torsional stiffness and the vibration modes and frequencies below 100 Hz. This reduced model is included in the Multi-body simulation in Adams/Car, where four different full-vehicle models are built – rigid and flexible body versions of the original vehicle and of the aluminum design. To draw the conclusions, these models are submitted to three different dynamic maneuvers: constant radius curve, step steering, and pothole encountering. The results point out the importance of the body flexibility inclusion in the vehicular dynamic studies of light commercial vehicles and about the changes required to the aluminum frame in order to assure better dynamic performance of the vehicle.

Keywords: Vehicle Dynamics, Multi-body Simulation, FEM Analysis.

1 Introduction

The use of alternative materials to pursue mass reduction in the body [1, 2] and chassis components [3, 4] is a well established subject of research and development.

In the light commercial vehicles environment the mass reduction objective is once more decisive. The customer is usually more performance-driven and is very mindful of the cost and benefit derived from product improvement, since the objective of the vehicle is, at the end, a professional activity that should generate profit. It is then hypothesized that the adjunctive costs of an aluminum body could be quickly compensated by the additional load capacity originated from the body mass reduction [5].

The aluminum rear frame pre-design created by the company Vehicle Engineering and Design (VE&D), target of the present paper, is presented in Figure 1, side by side with the equivalent steel frame currently in production. The idea is to connect the aluminum rear frame to the normal production steel cabin, in order to create an alternative version of the light commercial vehicle. Besides the material change an independent bi-link suspension is adopted, in contrast to the original rigid axle leaf spring suspension.

The potential benefits intensely motivate the idea of designing a light commercial vehicle using aluminum as main material of the body, despite the fact that many issues must be solved in order to match the performance of the firmly established automotive materials.

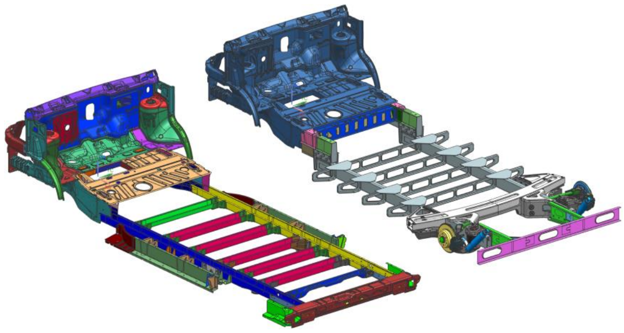

Fig. 1. Original steel frame (left) and aluminum frame (right)

One of the most widespread hypotheses used in vehicular dynamics textbooks is the one of rigid body. Given the high level of torsional and flexional stiffness achieved by nowadays passenger cars, this consideration usually leads to reliable results [6]. However, the assumption of rigid structure can be less appropriate for particularly compliant vehicles, such as convertibles or commercial vehicles [5].

The simpler way to consider the influence of frame flexibility during the design process is to reduce it to a concentrated parameter, such as a torsion spring. Nevertheless, it is evident that the response of a complex system as the body cannot be completely described only by a concentrated parameter. Comparing the rigid body model, the torsional spring model, a complete FEM model and the experimental results the conclusion is that the torsional spring model is an improvement to the rigid body model, but the FEM model is the one which provides the best correlation with the reality [7–9]. Many techniques can be used to describe the flexible behaviour of components inside the multibody environment and one particularly diffused method is the model reduction. As described by [10] the model reduction has the objective of reducing the number of degrees of freedom of the FEM analysis without detriment to the final results. Thus, a similar approach is used in this paper.

2 Simulation Models

2.1 FEM Analysis

The objective of this analysis is to estimate the static torsional stiffness of the vehicles' bodies, and to obtain their natural frequencies and vibration modes under 100 Hz. Bearing in mind that the automotive body is very complex and an analysis using all the details of it is not only burdensome in terms computational power, but also not worth in terms of results. Since a considerably general behaviour is pursued, a simplification is carried out. Surfaces representing the main forms and dimensions of the frame are meshed into shell elements of various thicknesses.

To calculate the torsional stiffness of the structure a set of constrains and loads is imposed. Exactly 6 degrees of freedom are blocked: X, Y and Z displacements in the front-right position; Z displacement in the front-left position; Z and Y displacements in the rear-left position. The rear-right position is destined to the load, in this case a 1000 N linear force in the Z direction. As far as the modal analysis is concerned, free-free boundary condition simulation are required, therefore just the collectors and interface nodes must be defined. The chosen collector is the CMSMETH (Component Mode Synthesis METHod). The specific synthesis method was the Craig-Bampton Method, and the condition of the simulation was the definition of all the vibration modes up to the frequency of 100 Hz.

Figure 2 shows the displacements after the torsional stiffness test.

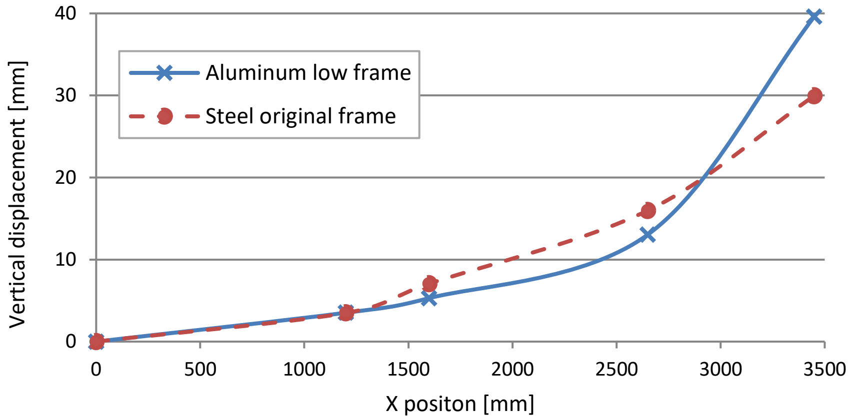

Fig. 2. FEM frame torsion simulation results

It is important to remember that the division of the body into sections has been done considering the features of the aluminium low frame body, in order to understand better the points with more or less torsional stiffness. To allow a relevant comparison, the same division was used also for the original steel body, even if no significant changes in the layout occur at those points. The X coordinates of the sections are: 0 for the front axle; 1200 mm for the end of the cabin; 1600 mm for the beginning of the frame; 2650 mm for the end of the frame; and 3450 mm for the rear axle.

The cabins, as planned during the modelling, present very similar characteristics. The aluminium frame has a higher stiffness in the bolted junction and, even with a slightly higher displacement during the third section, remains less compliant until the beginning of the suspension anchorage. With this last part, by far the weakest point of the frame, the aluminium rear body loses its advantage over the original steel one, and ends up with 24.4% less overall torsional stiffness. Tables 1 and 2 show the list of modes and their qualitative description - for both bodies - after the modal analysis.

Comparing the frequencies, it is possible to see that the two structures have similar behaviour when torsion or XZ bending are concerned, but the original steel frame has significantly higher natural frequencies for XY bending. Higher frequencies are normally associated to stiffer structures, and the closed profile of the cross beams provides a lower compliance when compared with the stamped open profiles of the aluminium solution. Furthermore, as predicted by the static analysis, the suspension anchorage of the aluminum frame is recognized as the weakest point of the structure and presents some independent vibration modes that are not present in the original frame.

Table 1. Original frame modal analysis results

Mode number	Modal frequency [Hz]	Description
1 - 6	0,0	Rigid body modes
7	22,5	Full body - 1st mode - Torsion
8	24,6	Full body - 1st mode - XY bending
9	26,0	Full body - 1st mode - XZ bending
10	33,0	Suspension anchorage - 1st mode - XY bending
11	69,0	Full body - 2nd mode - Torsion
12	86,5	Full body – 2nd mode - XY bending

Table 2. Aluminum frame modal analysis results

Mode number	Modal frequency [Hz]	Description
1 - 6	0,0	Rigid body modes

7	22,5	Full body - 1st mode - Torsion
8	24,6	Full body - 1st mode - XY bending
9	26,0	Full body - 1st mode - XZ bending
10	33,0	Suspension anchorage - 1st mode - XY bending
11	43,3	Suspension anchorage - 1st mode - Torsion
12	53,9	Suspension anchorage - 1st mode - XZ bending
13-15	around 54	Frame cross beams - various modes
16	61,5	Suspension cross beam - XY bending
17	77,5	Full body - 2nd mode - XY bending
18	80,0	Full body mixed mode
19-25	85-100	Cabin and suspension sheets - various modes

2.2 Multi-body Model and Test Maneuvers

The objective is to build four different assemblies, rigid and flexible body versions of the two solutions to be compared. Thus, the following subsystems must be made:

- Front Suspension (McPherson) and Rear Suspensions (Rigid axle in the Original vehicle and Bi-link in the Aluminum vehicle);
- Body (Rigid and two different Flexible files);
- Standard Steering, Powertrain, Brakes and Tires.

Once these models are completed, they are submitted to the following maneuvers:

Constant radius curve – this closed loop maneuver consists in bending a curve of constant radius while increasing the speed. To accomplish that the driver should control both throttle and steering wheel, while brakes and clutch remain untouched. The objective is to observe how the vehicle behaves in a wide range of different lateral acceleration conditions. In order to minimize the influence of longitudinal acceleration and transient behaviour, the velocity is slowly increased. Under those conditions, the most important considerations are related to under/oversteering behaviour, either on low lateral acceleration and limit slipping condition. This will allow observing the influence of frame stiffness in the roll gradient, lateral load transfer distribution and limit behaviour, as well as comparing the rear suspension topologies and frame solutions. Characteristics: Test time of 200 s; initial lateral acceleration of 0.1 g; final lateral acceleration of 0.8 g; and curve radius equal to 50 m.

Step steering – an open loop rapid change into the steering input is imposed and maintained until the end of the test, meanwhile the other commands are untouched, to highlight the transient response of the vehicle. The motivation is to analyse the response time, overshooting and stabilization of the vehicle for the different layouts, in order to analyse the influence of frame stiffness and suspension topology. In this case the frame influence is expected to differ more from the simple torsional spring behaviour, since the input is more likely to excite a wider range of torsional modes. Characteristics: Test total time of 4 s; step starts at 1 s; step duration of 0.2 s; steering angle of 90°; vehicle speed of 60 km/h.

Pothole encountering – not focused on the lateral dynamics, but on the vertical response of the vehicle. Maintaining constant the velocity and straight trajectory of the vehicle, it encounters a pothole in both left and right side. This test condition allows the analysis of comfort and safety during ride, and excites the flexible frame in a wide range of bending frequencies. Characteristics: Pothole starts at position 5 m; pothole length is 500 mm; pothole depth is 200 mm; total test time of 1.5 s; vehicle speed equals to 80 km/h.

3 Simulation Results

The aluminum rear suspension has a higher roll rate, which grants the aluminum rigid vehicle the highest overall rolling gradient and, therefore, the lowest roll angles throughout the test, as shown in Figure 3.

The performance of the bodies can be observed by looking at the aperture created when moving towards higher accelerations. Certainly the aluminum frame displays a lower performance, being further to its rigid reference than the original steel frame. This fact is aided by the higher stiffness of the aluminum rear suspension and also by its lower roll center. Moreover, all the trends seem to be straight lines but the first.

Starting with a slope similar to its rigid reference, the aluminum flexible vehicle tends to increase it during the test, crossing the original rigid line at 0.37 g and the original flexible one at 0.65 g. At the end, the original and aluminum flexible vehicles present similar performance when considering the overall roll, even if the frame contribution is significantly superior in the latter. The roll angle, however, is not the final reason to the lateral acceleration limit. The roll angle and the torsion of the frame affect, beyond many others, the LLTD and the inclination of the wheels.

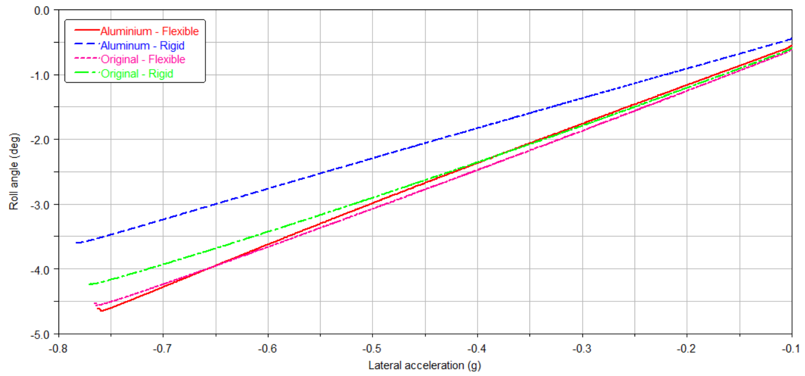

Fig. 3. Roll angle vs. Lateral acceleration - Constant radius cornering maneuver

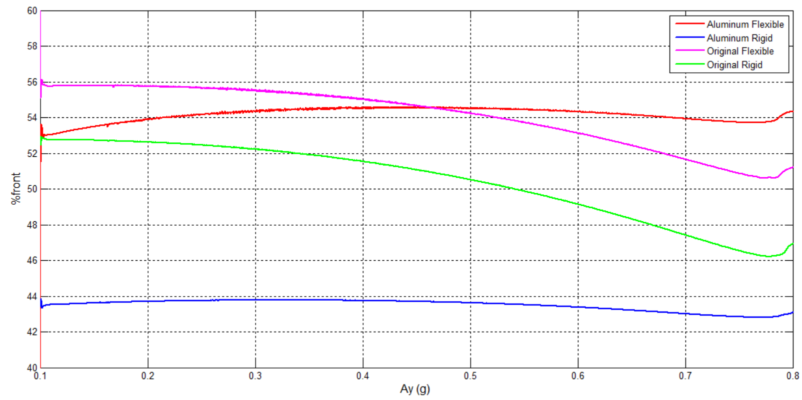

Fig. 4. Percentage of lateral load transfer in the front axle - Constant radius cornering

The mass distribution between front and rear is 65.1:34.9 in all cases; the static front roll rate is 123.9 N·m/°; the static rear roll rates 173.0 N·m/° for the aluminum suspension and 76.5 N·m/° for the original suspension; the static roll center heights are 11.9 mm for the front suspension, 41 mm for the aluminum rear suspension and 207 mm for the original rear suspension. The flexibility of the frame tends to approximate the behaviour of the elastic term of the LLTD to the one dictated by the mass distribution, in this case the higher the flexibility the closer it gets to the 65.1% of front load transfer. Indeed, looking at Figure 4, the flexible body vehicles lines are placed above those of the rigid references. Comparing the two layout solutions, the aluminum vehicle displays a steadier LLTD, while the original tends to transfer more load in the rear axle with higher accelerations. The flexible values of LLTD are comparable, even if the influence of the frame is more important to the aluminum one.

Once again in the step steer, depicted in Figure 5, the difference between rigid and flexible body assumptions is more clearly observed on the aluminum solution. After the delay time and the first linear section the flexible body vehicle's yaw rate grows faster than its reference's, reaching an overshooting peak at 1.3 s and proceeding with a couple of low amplitude oscillations around the S shaped curve of the rigid body.

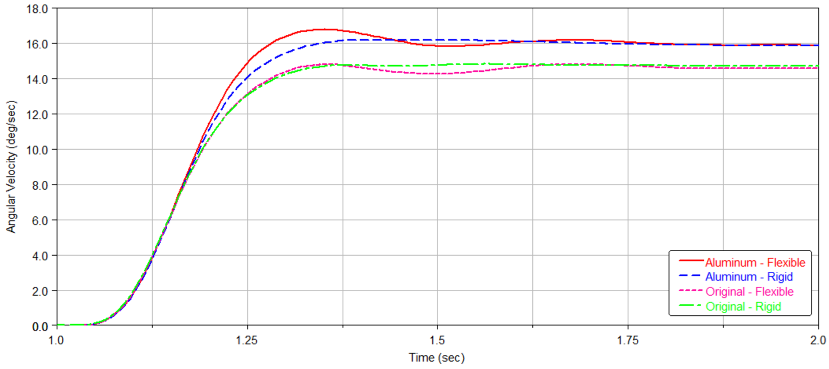

Fig. 5. Yaw rate evolution - Step steering

The frame is the only connection between front and rear axle and it will be responsible to transmit the control information received by the front wheels by the steering system. In other words, the response characteristics of the rear axle slip angle are dependent on the frame performance. The rigid frame model will have a prompt response without deformations, while the flexible one behaves like an elastic system which has a delay time and some compliance. The conclusion is that the flexibility of the frame will delay the response of the rear wheels to the steering input. This delay is going to reduce the lateral acceleration response, because in the beginning the summation of the lateral forces will be lower due to the lack of the rear component. Conversely, the absence of an instantaneous reaction of the rear axle is going to maintain the unbalancing of moments around the CG, rendering faster the variation of yaw rate in the flexible vehicles.

Regarding figures 6 and 7 it is clear that the front axle behaviour is scarcely impacted by the changes in the rear suspension or in the flexibility of the frame. The flat areas of the graph with zero tire forces represent the loss of contact between wheels and road, which is a situation to be avoided, given its danger.

The rear axle presents the biggest variations on the behaviour due to suspension and frame changes. The basic conclusion is that the original rear suspension presents a typical disturbed behaviour due to the elevated mass of the rigid axle. The aluminum suspension in the other hand has an excellent response, quickly eliminating the wheel load variations and remaining far from a second detachment of the rear wheels. When flexible bodies enter in the simulation the original vehicle suffer minor changes, maintaining the oscillation behaviour, but it is important to notice a brief separation of the rear wheels in the second period of vibration. The flexibility of the aluminum frame threatens the good performance of its suspension by inducing large magnitude vibration modes and failing to keep the wheels in contact with the road. The final comparison of the comprehensive vehicles results in a similar behaviour, as well as happened in other occasions. The original model vibrations have longer duration while the intensity of the oscillations is higher in the aluminum one.

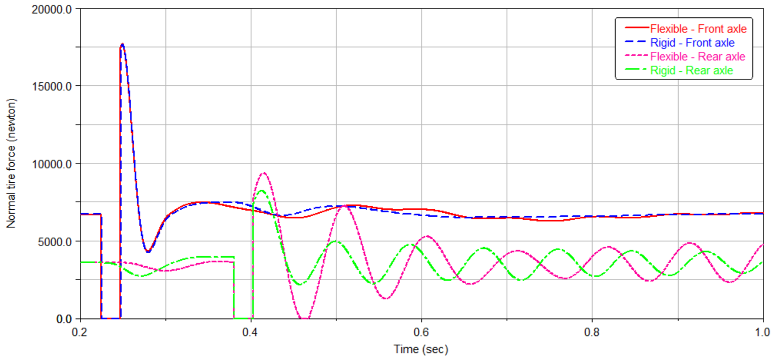

Fig. 6. Original configuration: tire normal forces - Pothole encountering

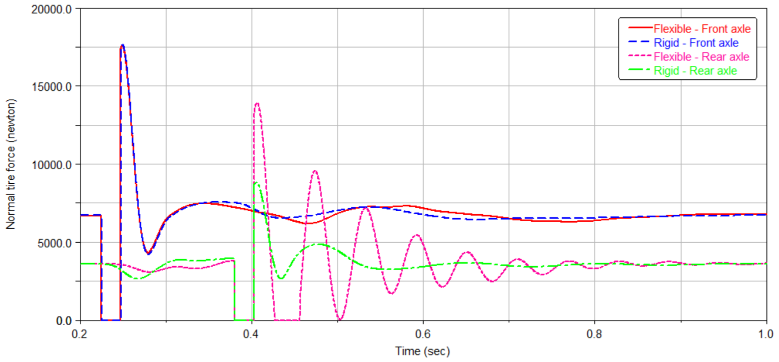

Fig. 7. Aluminum frame: tire normal forces - Pothole encountering

The pitch movements of the suspension are considered negative both for comfort and handling performances, therefore the suspension should be able to quickly eliminate this quantity and avoid large peaks during the obstacle encountering. Comparing the rigid body performances it is possible to observe that the first stage of the test produces similar results, mainly because the same front suspension is used. After the passage of the rear suspension onto the pothole, however, the original suspension obtain a better result, with a lower magnitude peak and promptly reducing the negative pitch, despite some small amplitude oscillations due to the rigid axle suspension motion.

The effect of the flexibility of the frame is, once more, negative to the performance of the vehicle, increasing the maximum values and retarding the damping effects. Both frames have almost the same consequences to the overall output: increased peak value during the front axle impact; steeper plunge during the transition phase; shifted forward and more intense negative pitch maximum; longer stabilizing period.

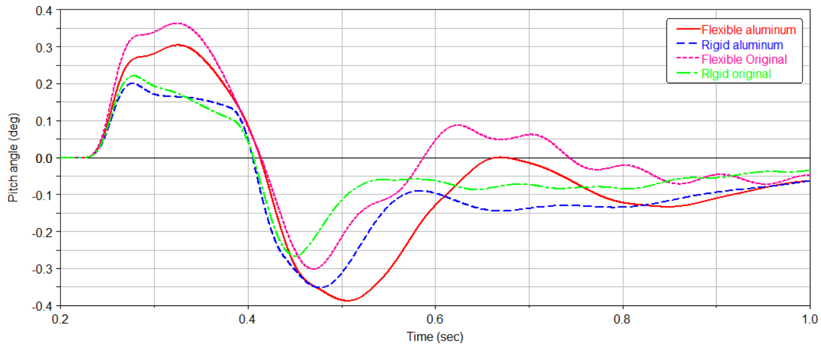

Fig. 8. Pitch angle evolution - Pothole encountering

4 Conclusions

The article used a combined Finite Element Simulation and Multibody Dynamics approach to assess the effects of an innovative commercial vehicle body structure in aluminum. The construction of the models and their computational burden proved to be reasonable and yet bring valuable insights about the behavior of the handling and comfort behavior expected. It is especially relevant to highlight the extend of the differences between rigid body and flexible body models in this context, showing irrevocably the importance of including this parameter early in the design phase.
When compared to similar approaches find in literature, the modal reduction presents a good compromise between simplicity and completeness, being much less time demanding than higher order structural models without reduction or co-simulation environments.
In the structural simulation, the aluminum frame presented an inferior structural behaviour comparing to the original steel frame. The static analysis found a 24.4% reduction of the overall torsional stiffness, for which the main responsible is the connection between intermediate frame and suspension anchorage and the suspension anchorage itself. In the modal analysis, the frequencies of torsion and XZ bending were similar, while XY bending presented lower frequencies in the aluminum frame. Six full body modes were found below 100Hz in the original vehicle and five full body and three modes related to the suspension anchorage were in the aluminum one.

In the multi-body analysis, the constant radius curve analysis confirms the tendency proposed by the theory; the frame flexibility shifts the TLLTD towards the CG position configuration, and being the static weight distribution 65.1:34.9 the flexible vehicles become more understeering and present lower maximum acceleration levels. The load transfer distribution changes introduced by the aluminum frame are in the order of 10%, while the original frame displays a 3.5% balance change. The final behaviour is, however, similar for the two vehicles, given the better performance of the rigid aluminum vehicle with bi-link suspension. The transient behaviour studied in the step steering manoeuvre brings up another issue to the analysis: the reduction of effectiveness of the damping system when a flexible frame is considered. The yaw rate of flexible models presents oscillations around the rigid model reference, the total force exerted by the dampers is lower and the stabilization time is longer for flexible frame vehicles. Comparing the roll angles, the contribution of the frame torsion is higher for the aluminum vehicle, but the overall value is quite similar. The pothole encountering test shows the response of the flexible body to a vertical impact. Once more the flexibility of the frame appears as a negative feature, even thought it must be pointed out that the peak forces/accelerations are reduced which could lead to a better comfort performance. When handling is concerned, however, the pitch angles and vertical load control of the wheels is worsen and due to the damping system impairment the stabilization time is higher. Moreover, the aluminum tends to have more accentuated peaks and slower stabilization while the original model suffers with oscillations of the unsprung mass of the rigid axle.

The final comparison between the two models reveals that the preliminary design of the aluminum frame grants a vehicle dynamic behaviour similar to the original layout. However, being equipped with a more sophisticated rear suspension topology, the project still has margin to improvement. The main point to be dealt with is the suspension anchor structural behaviour, especially its connection with the frame.

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