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A MULTIBODY-DYNAMIC MODEL TO STUDY THE INFLUENCE OF HAUL TRUCK SUSPENSION SETUP ON THE CHASSIS MECHANICAL BEHAVIOR

Alvaro E. Dantas
Marco Tulio C. Faria
Graduate Program in Mechanical Engineering, Universidade Federal de Minas Gerais
6627 Presidente Antônio Carlos Avenue, Belo Horizonte, MG, Brazil
aestanislaudantas@gmail.com, mtfariaufmg@gmail.com

Abstract. *The vibration generated by unpaved road roughness of mining sites can not only lead to health problems on the vehicle driver but also reduce the haul truck useful life. A crucial component in the design of these vehicles employed in the mining industry is the hydropneumatic suspension (HPS), whose physical parameters and setup variables must be carefully estimated in order to attenuate the vibration transmitted from the tire-soil contact through all vehicle subsystems. In this work, an analytical investigation is performed to assess the influence of some HPS parameters, such as nitrogen-oil initial ratio and tire pressure, on the mechanical behavior of the chassis of mining haul trucks. These parameters are selected due to the feasibility of adjustment to improve the vehicle ride dynamics. A multibody dynamic model using the commercial package ADAMS Car® is developed to estimate forces acting on the truck chassis for different setup parameters during operation on rough unpaved roads. Firstly, predictions are compared to experimental data to validate the suspension pressure behavior and to choose a proper road roughness model for the subsequent analysis. Mean and peak values of suspension pressure show good correspondence to field collected data. Later, the multibody model is tested on a double lane change maneuver with uneven road conditions to investigate the generated suspension loads for three different setups for loaded and unloaded truck. Multibody model indicates significant differences on the suspension loads for each setup. These forces combined with other components loading are imposed on a finite element model of the chassis to evaluate the influence of each suspension setup on the chassis stress fields. The results rendered in this work indicate that, from a static point of view, maximum stresses are slightly lower for softer setup when the truck is unloaded. For loaded truck, the results of maximum stresses do not show a clear pattern since other sources of load can contribute to the frame loading. However, looking at stress ranges counted by Rainflow method, overall results indicate that softer suspensions may induce improved frame lifetime, in contrast to more rigid setups. It is noteworthy to state that although decreasing the suspension stiffness can generate a positive impact on the chassis durability, further investigations about its influence on the driver's cabin vibration, lateral vehicle dynamics and sprung mass roll angle must be performed to evaluate the impact of different suspension setups in whole-body vibration (WBV) and handling performance.*

Keywords: *Mining haul truck, Hydropneumatic suspension, Multibody dynamics, Truck chassis, Ride dynamics*

1. INTRODUCTION

Haul trucks have a fundamental role in the mining process, carrying material in exploration sites with relatively low operational costs. These trucks commonly travel through unpaved roadways with severe irregularities due to high loads transferred by the tires and to inadequate maintenance. Consequently, such operation conditions subject the vehicle mechanical components to high cyclic loads and exposes drivers to high levels of WBV (whole-body vibration).

The compromise between the ride comfort and the mechanical components endurance is well known on the off-highway automotive industry. Many authors have studied the effects of road-truck interaction on the truck operating life. Kondo (1984) describes an exponential relationship between the frame damage and the road class determined by ISO TC108, showing that the payload ratio and the speed may also significantly result in reduced frame lifetime. Frimpong et al (2012) analyze the tire service life of haul trucks using a vehicle multibody model with flexible road implementation. A frame lifetime estimation is made by Mi et al (2012) using a procedure combining multibody dynamics and finite element method, on which the multibody model runs through a D-Class road roughness to generates output forces for the finite element model. Kansake (2019) also uses multibody dynamics to quantify the impact of payload variation and road roughness on truck dynamic forces providing a methodology to estimate forces for the haul road design.

Other authors have developed studies to enlarge the understanding about the dynamic behavior of mining haul trucks and hydropneumatic suspensions (HPS). Prem and Dickerson (1992) use a nonlinear analytical model to investigate the loads on suspension and tires during a steady state cornering maneuver. Later, Prem (1998) extends his previous work with a multibody model to obtain the truck transient response. Prem (1998) gives slip angles and suspension and tire forces on different types of maneuvers, including travelling on an uneven road. Ha et al (2021) use a full-vehicle model to study haul truck ride comfort for different road classes. Wu (2020) proposes a high precision methodology to model HPS and validates his results with a 11 DOF full-vehicle model and experimental data.

In the technical literature of mining haul trucks, there are few works describing the influence of the HPS parameters and the tire pressure on their dynamic response. Mining companies acquire a vehicle from the manufacturer with generic setups that may not be applicable to the operational condition of a specific site. If so, the company have two major options to enhance the truck's ride comfort and durability: (i) improve the haul road maintenance routine or (ii) adjust some vehicle parameters. The former option may be costly, specially at tropical countries where rainy seasons are longer and higher quality road materials and extensive drainage systems are required (Tannant and Regensburg, 2001). On the other hand, defining new suspension setups can be relatively easy and customizable.

This work presents a study of the influence of different suspension-tire setups on the behavior of a mining haul truck frame. Two setup parameters are chosen for this analysis, which are the initial HPS gas volume and the tire pressure. The selection of these two parameters is related to the feasibility of adjusting them on mining sites. Firstly, a multibody model is developed and validated with experimental HPS pressure data. The track considered in this analysis is a generic uneven road defined by Sayers (1988) methodology. Later, the vehicle is tested in a double lane change maneuver with 3 different setups for each of the dump body load cases (empty and full). The forces generated by the multibody model are the inputs for a finite element model implemented to evaluate the frame stress fields.

The vehicle selected for this analysis is a CAT 775G truck that presents a net weight of 47.5 t and is designed to carry a maximum payload of 64 t. It has two axles supported on six wheels, with four hydropneumatic suspensions and a diesel-powered engine.

2. MULTIBODY MODEL

The study is based on the full-vehicle multibody model of a mining haul truck presented on Figure 1. It has been developed using the computer package ADAMS Car® and comprises of suspension, body, steering, powertrain and tires assemblies. All relevant components, numbered from 1 to 17 on Figure 1, are described as follows.

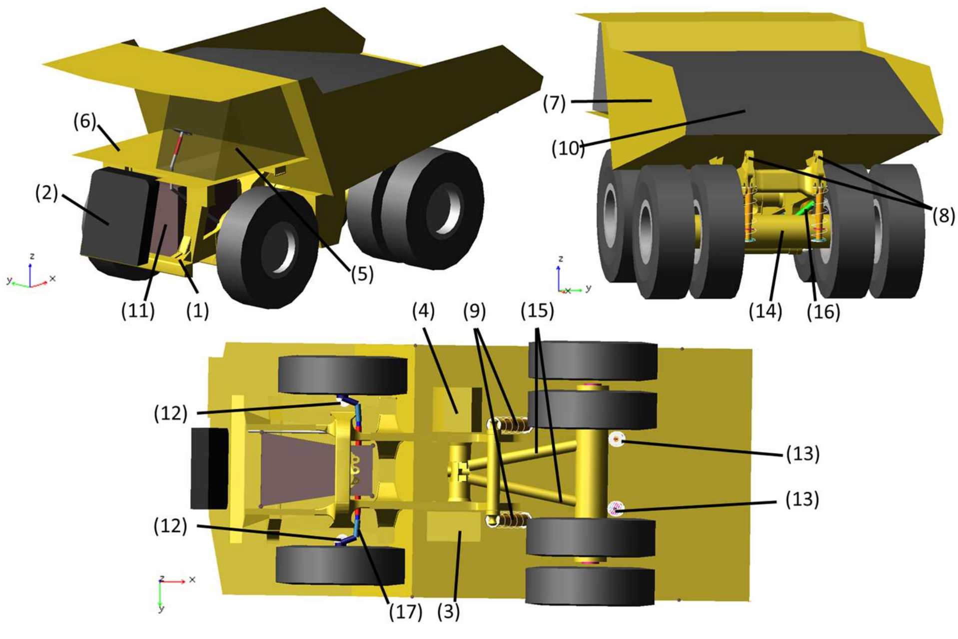

Figure 1: Multibody mining haul truck model.

2.1 Chassis (Body)

The chassis in the multibody model consists of the frame (1) and other relevant masses attached to it, such as the radiator (2), fuel and hydraulic system tanks (3)(4), driver cabin (5), front deck (6) and the dump body (7). For the vehicle dynamic simulation, the frame is considered as a rigid body and all attached components are assumed as rigid hardpoints with mass and inertia properties. The dump body is connected to the frame through a rear pinned connection (8) and a hoist cylinder (9), which is represented by discrete spring and damper elements. These elements present values of stiffness and damping coefficient (k=1.93E7 N/m and C=1.57E6 Ns/m) based on Kansake (2019). Payload (10) is considered a separated component rigidly attached to the dump body.

2.2 Powertrain

The selected truck is a rear-wheel-drive vehicle powered by a diesel engine (11). Its behavior characteristic curves, available on commercial catalogues, are used as input for the multibody powertrain system. Additional data, such as the final vehicle speed at each gear and differential and planetary ratios, are obtained from the vehicle specalog AEQ6350-02 (Cat, 2020).

2.3 Suspension and steering

The front suspension (12) features a sliding pillar geometry with independent axles, where the rod (moving part) is mounted rigidly on the hub-wheel set and is free to translate and rotate along the cylinder longitudinal axle. As illustrated on Figure 2, the cylinder is rigidly fixed to the frame through an end plate bolted joint.

Both right and left sides of the rear suspension (13) are interconnected through a rigid axle (14) that is pinned to the chassis through an anchor shaped trailing arm (15). The suspension rods are linked to the axle through pinned joints, such as the cylinder-frame connection. Rear suspension lateral forces are transferred to the frame through the Panhard rod (16), located between the rigid axle and the structure. The representation of these components is also depicted on Figure 2.

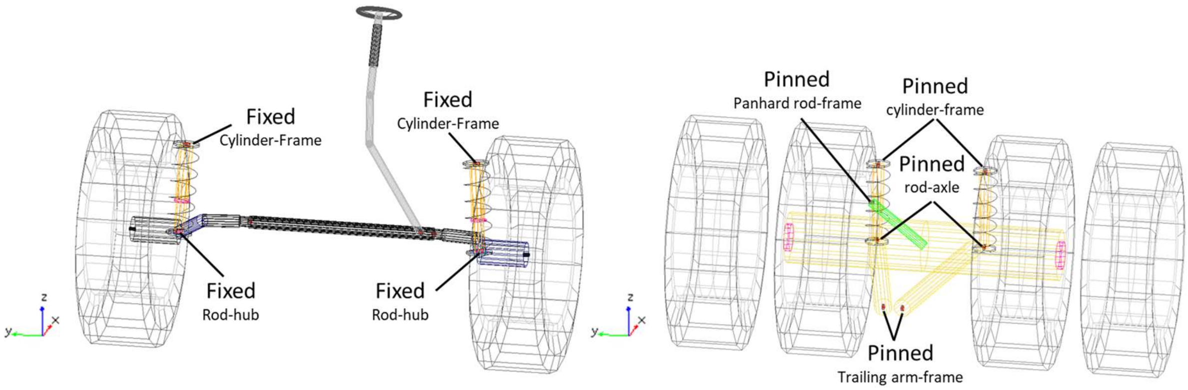

Figure 2: Front and rear suspension main connections.

HPSs located on the front and rear axles work with nitrogen gas and oil. This suspension is typical on heavy duty trucks due to its high damping capacity and non-linear stiffness, which allows better wheel load distribution for unloaded and fully loaded conditions. The forces in HPS comprise of the combination of elastic forces exerted by the gas (F_g) and dissipative force resulting from oil flow through the valves (F_h). F_g can be derived from the ideal gas equation,

$$F_g = P_0 \left(\frac{V_0}{V}\right)^r A_r = P_0 \left[\left(\frac{V_0}{V_0 + A_r\,(z_b - z_a)}\right)^r - 1\right] A_r, \tag{1}$$

where P_0 and V_0 are the initial gas pressure and volume when the rod is extended (nominal values are called P_{0N} and V_{0N} in this paper), r is the polithropic coefficient, A_r is the rod cross-section area, z_a and z_b are the rod longitudinal position at time instants a and b. HPS tends to behave as an adiabatic system at high frequency excitation ($r \approx 1.40$) and may be close to isothermal at low frequencies ($r \approx 1.00$). Since the study of this paper deals with excitations provided by road roughness that leads to high frequency forces on the suspension, the polytropic coefficient was estimated as 1.40.

Figure 3a and Figure 3b show force versus displacement curves for 0.85 V_{0N}, V_{0N} and 1.15 V_{0N}. Different amounts of initial gas volume can be achieved by increasing or reducing the nitrogen-oil ratio in the main chamber at constant gas pressure ($P_0 = P_{0N}$). It can be observed that the suspension stiffness is conversely related to V_0.

Figure 3c and Figure 3d show the dampening behavior produced by the oil flow through the HPS valves, which are based on the models for the dissipative force F_h presented by Long et al (2021) and Wu (2020).

The steering system (17) associated with the front axle is simply represented on the multibody model, since it is not expected that to play an important role in the analysis. Kingpin inclination, caster and camber angles are assumed as 3.5°, 3.0° and 0.0° as indicated by Loo (2003) for a similar haul truck.

2.4 Tires

Prem (1992) empirically states that haul truck tire stiffness may be estimated from its pressure through a polynomial regression. Applying this concept to this model, it has been considered that the 24.00R35 tires run with specified nominal pressure (TP_N) of 724 kPa. Figure 4 shows the curve of force versus displacement for three different tire pressure (0.90 TP_N, TP_N and 1.10 TP_N). Other tire performance parameters and coefficients are estimated from the available data of on-road truck tires. For the uneven road simulations, a contact 3D with enveloping properties is also implemented.

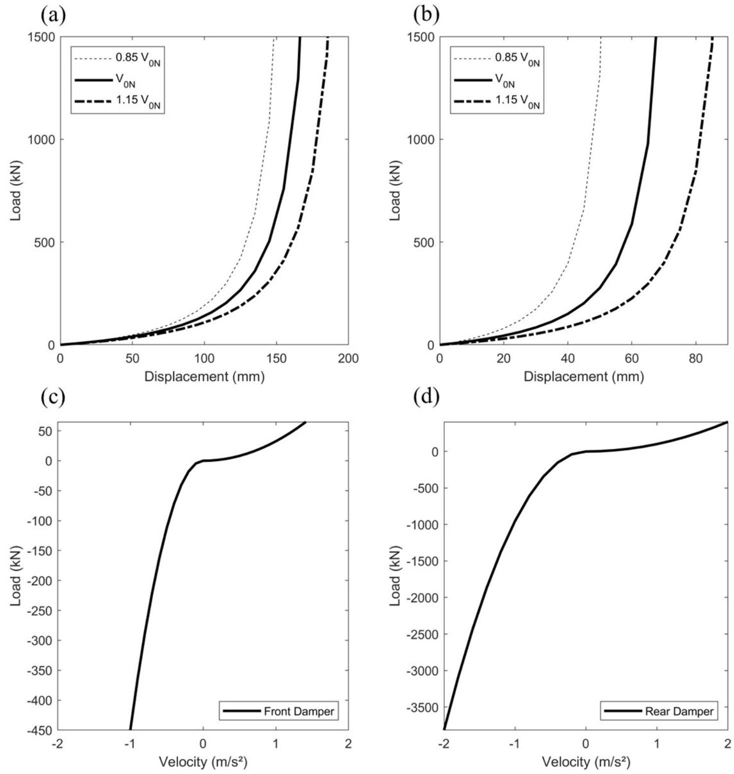

Figure 3: (a) Front suspension load-displacement characteristics; (b) Rear suspension load-displacement characteristics; (c) Front suspension load-velocity characteristics; (d) Rear suspension load-velocity characteristics.

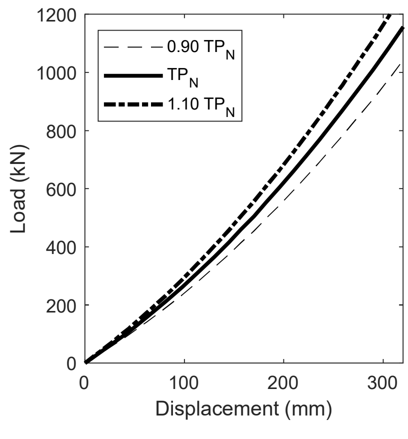

Figure 4: Truck tire load-deflection characteristics.

3. MULTIBODY MODEL VALIDATION

Available experimental data for the gas pressure in the four hydropneumatics suspensions of the selected truck are employed to validate the multibody model developed in this work.

3.1 Road roughness modeling

Daily operation of off-highway trucks involves unpredictable routes and obstacles that would require a high degree of knowledge about the road conditions for representation in computational simulations. Therefore, international standards for pavement roughness classification (ISO 8608) have proposed methodologies for roadway modeling using standardized parameters and classes based on spectral statistical functions (Power Spectral Density - PSD). Also, Sayers (1988) has proposed a mathematical model based on empirical data, which has been adopted in this paper due to its ease of implementation in ADAMS CAR. Sayers (1988) PSD function calculation is shown in Eq. (2).

$$G_d(\nu) = G_e + \frac{G_s}{(2\pi\nu)^2} + \frac{G_a}{(2\pi\nu)^4}, \quad (2)$$

where ν is the wavenumber or the inverse of the wavelength (λ^{-1}), G_d is a function of wavenumber, G_e is the white-noise elevation component, G_s is the white-noise slope (velocity) component and G_a is the white-noise acceleration component. To compare suspension pressure data obtained experimentally with the multibody model predictions, Eq. (2) is used to build the PSD function for values of G_e, G_s, and G_a equal to 10^{-6} m³/cycle, 0.0005 m/cycle and 0 $m^{-1}cycle^{-1}$, respectively. It can be seen in Figure 5a that this set of parameters gives a PSD similar to a D-class road from ISO 8608 (2016), including some additional high-frequency roughness.

Sayers (1988) indicates the use of Eq. (3) to convert the PSD function onto a road profile, where the amplitude is determined by $G_d(\nu)$ and the frequency interval ($\Delta\nu_i$) and phase angles (ϕ_i) are obtained randomly.

$$q(d) = \sum_{i=1}^{N}\left[\sqrt{\Delta\nu_i G_d(\nu_i)}\cos(2\pi\nu_i d + \phi_i)\right], \quad (3)$$

where d is the profile longitudinal distance, q is a function of d, N is the number of sinusoidal components used and i is the i^{th} component of the summation. Figure 5b exhibits the generated road profile for the left and right side of the road.

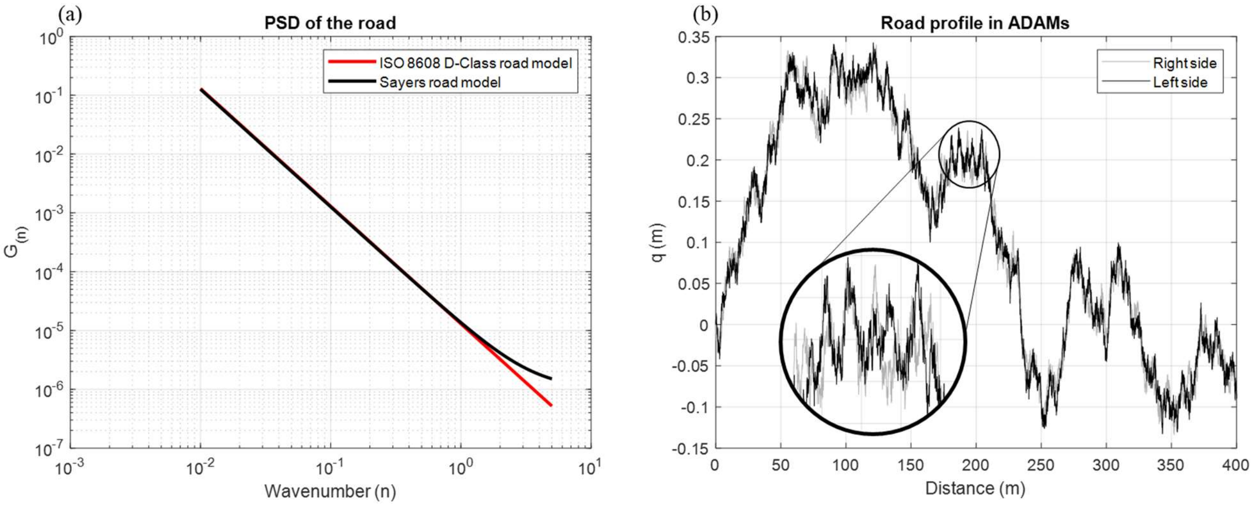

Figure 5: (a) Power spectral density curve and (b) road profiles used in the multibody analysis.

3.2 Data comparison

Experimental data of gas pressure on the four hydropneumatic suspension of four mining haul trucks have been collected during five days of continuous operation. These data are separated for two conditions – unloaded and loaded truck at constant speed. The first one refers to empty operation (0t) at a moving speed of 20 km/h, while the second one to maximum capacity operation (64t) at the same speed. Samples of 60 seconds from both conditions can be observed in Figure 6, where front and rear right suspension gas pressures are plotted against time.

For comparison, simulations were performed along a straight route at constant speed of 20 km/h using the ADAMS package. The estimate of the nitrogen pressure considers the track profile presented in Figure 5b and nominal values of nitrogen gas volume (V_{0N}) and tire pressure (TP_N). In Figure 6c and Figure 6d, an adjustment of 500 mm on the lateral (+Y, Figure 1) direction of the position of the payload center has been made to obtain a better correlation between field and computational data mean values. This adjustment is due to the asymmetry on the material loading.

Figure 6 indicates a good agreement on the values of suspension gas pressure obtained experimentally and numerically. Qualitatively, the overlapped curves indicate proximity in terms of amplitudes and mean values. Furthermore, the computed root mean square (rms) values of gas pressure indicate a maximum difference of 14.3% among experimental and numerical values.

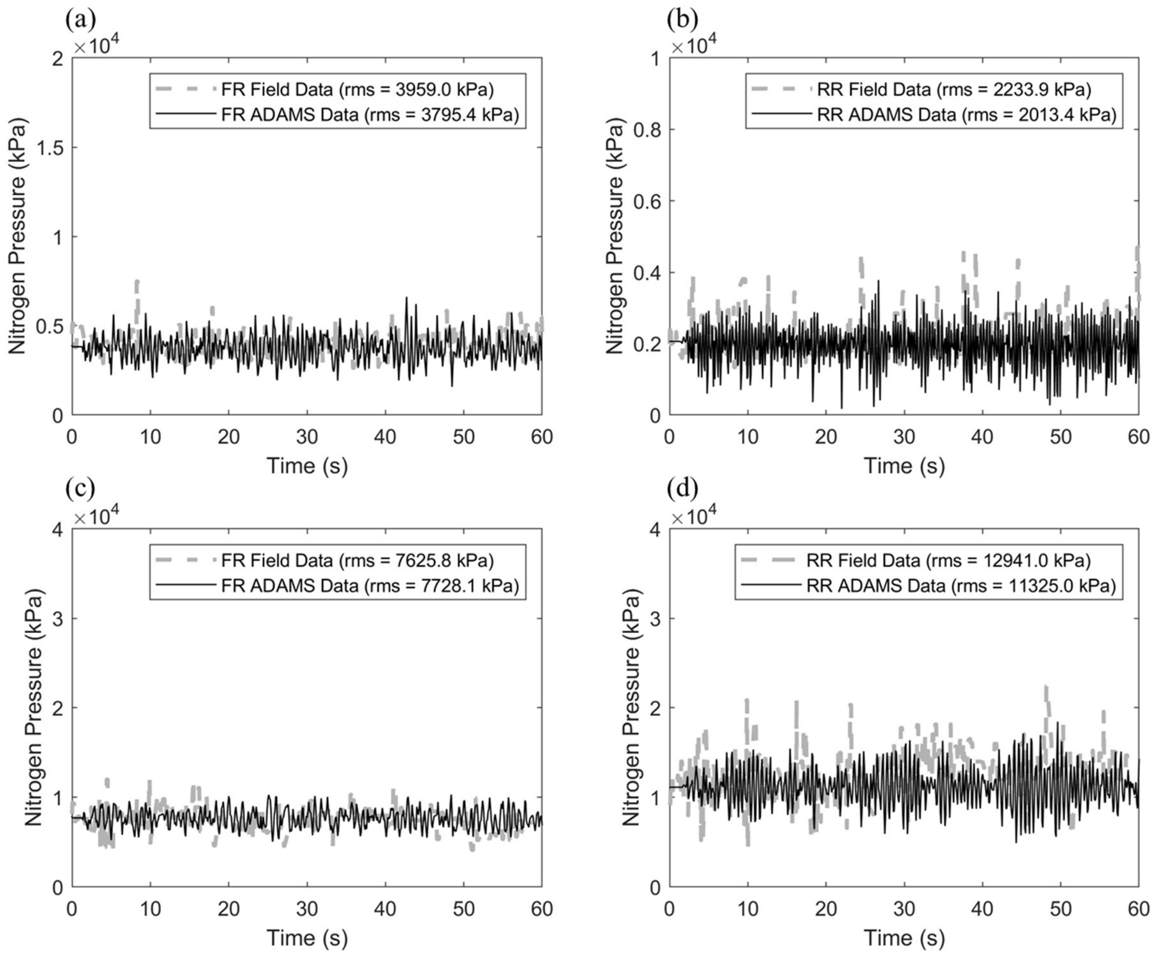

Figure 6: Comparative values of gas pressure from field measurements and ADAMS model: (a) Unloaded truck front right suspension; (b) Unloaded truck rear right suspension; (c) Fully loaded truck front right suspension; (d) Fully loaded truck rear right suspension.

4. CHASSIS ANALYSIS

4.1 Double lane change maneuver

To assess the effects caused by different HPS and tire adjustments, three setups are considered (Table 1) based on nitrogen initial volume and tire pressure described on sections 2.3 and 2.4. The first one, identified as 'A', gives a softer suspension, where V_0 is about 115% of the nominal value and TP is set to 90% of manufacturer recommendations. Condition 'B' is a more balanced setting with nominal values of $V_0 = V_{0N}$ and $TP = TP_N$. 'C' is set to 85% of the initial nominal gas volume and 110% of tire's nominal pressure, giving a more rigid setup. Each scenario was tested for loaded and unloaded truck, summing 6 total scenarios.

These cases are analyzed during a double lane change maneuver with parameters defined by ISO 3888-2. The first lane change occurs from time 12 to 17s and the second one from 21 to 26s. Road roughness shown in Figure 5b is employed in this analysis and truck speed is assumed as constant at 20 km/h, since that is a value close to the average speed observed at the mining area.

Comparatively, forces on rear suspensions are illustrated in Figure 7 for the setups depicted on Table 1. Table 2 presents the normalized standard deviation, called Dynamic Load Coefficient (DLC) by Prem (1998), calculated for each suspension and setup.

Suspension load plots and DLC values show that less rigid suspension setups (A and D scenarios) provides less force amplitude, providing suspension loads closer to the mean value (equal to the static load) and, therefore, lower maximum forces. Comparing to nominal setups, stiffer suspensions may increase the suspension forces amplitude in almost 14% for the empty truck rear axle (C) and 12% for the full truck rear axle (F). Thus, the relative difference from setups A to C and from D to F on rear suspension loads are about 27% and 24% for unloaded and fully loaded truck, respectively.

Table 1. Scenarios considered in the analyses.

Setup	Rate of the initial nitrogen nominal volume (V_{0N})			Rate of the nominal tire air pressure (TP_N)			Dump Body
	85%	100%	115%	90%	100%	110%	
A			X	X			Empty
B		X			X		Empty
C	X					X	Empty
D			X	X			Full
E		X			X		Full
F	X					X	Full

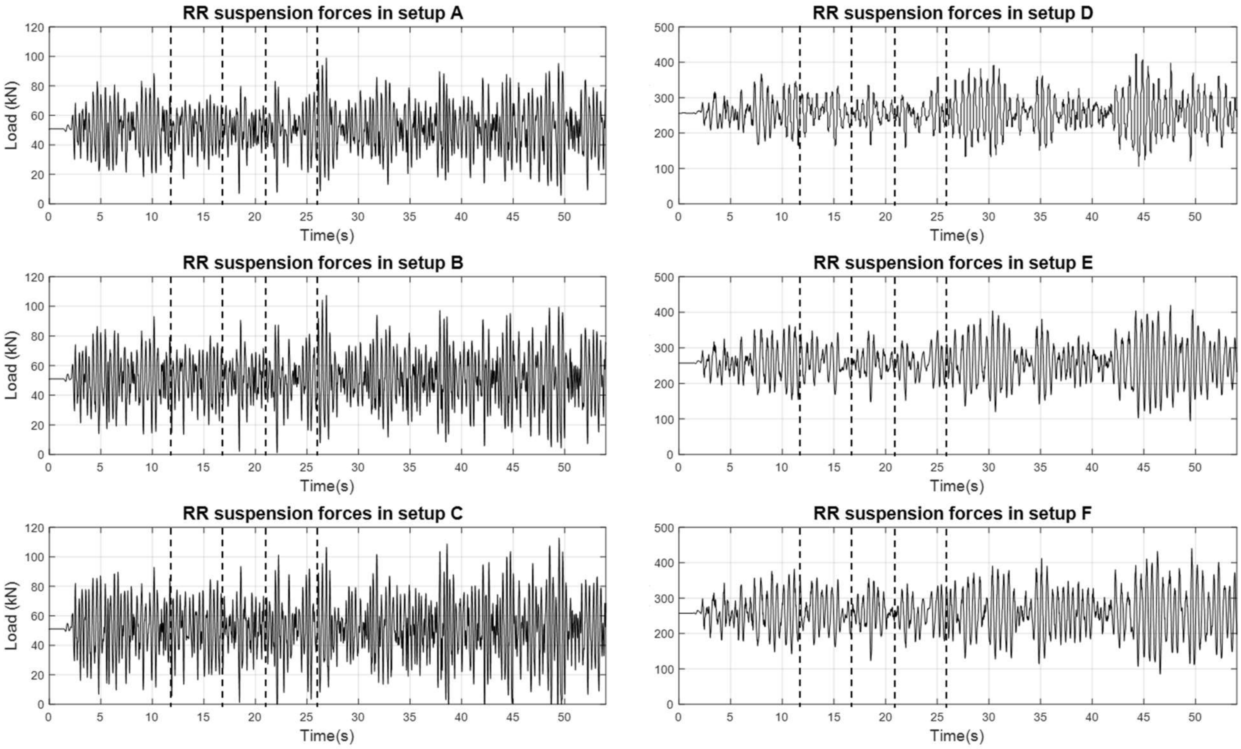

Figure 7: Rear right suspension loads in double lane change maneuver for each setup evaluated.

Table 2: Suspensions DLC values for each setup tested. (1)

Setup	Dynamic Load Coefficient (DLC)			
	FR	FL	RR	RL
A	0.193 (- 6.1%)	0.196 (- 8.2%)	0.288 (- 11.1%)	0.287 (- 11.2%)
B	0.206	0.214	0.324	0.323
C	0.221 (7.3%)	0.231 (8.0%)	0.367 (13.4%)	0.367 (13.6%)
D	0.146 (- 3.9%)	0.141 (- 4.6%)	0.179 (- 9.7%)	0.177 (- 10.0%)
E	0.151	0.148	0.199	0.197
F	0.157 (3.8%)	0.155 (4.9%)	0.220 (10.6%)	0.220 (11.8%)

(1) percentage values calculated with respect to nominal setups (B and E).

4.2 Finite element analysis

Forces obtained from the ADAMS model for the suspensions, trailing arm, Panhard rod and dump body (hoist cylinder and rear pinned connection) during double lane change maneuver are imported by the chassis finite element (FE) model (Figure 8) to quantify the influence of setups A to F on the frame stress levels. In order to save computing time, only maximum and minimum values of force at every 0.5 s are extracted from the multibody analyses.

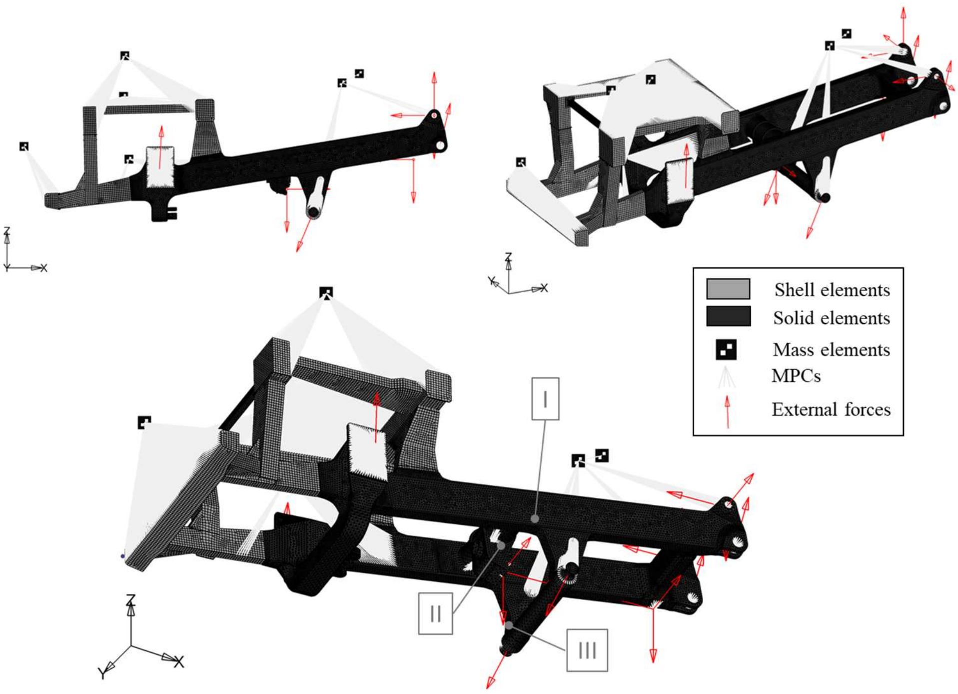

Figure 8: Mining haul truck chassis finite element model.

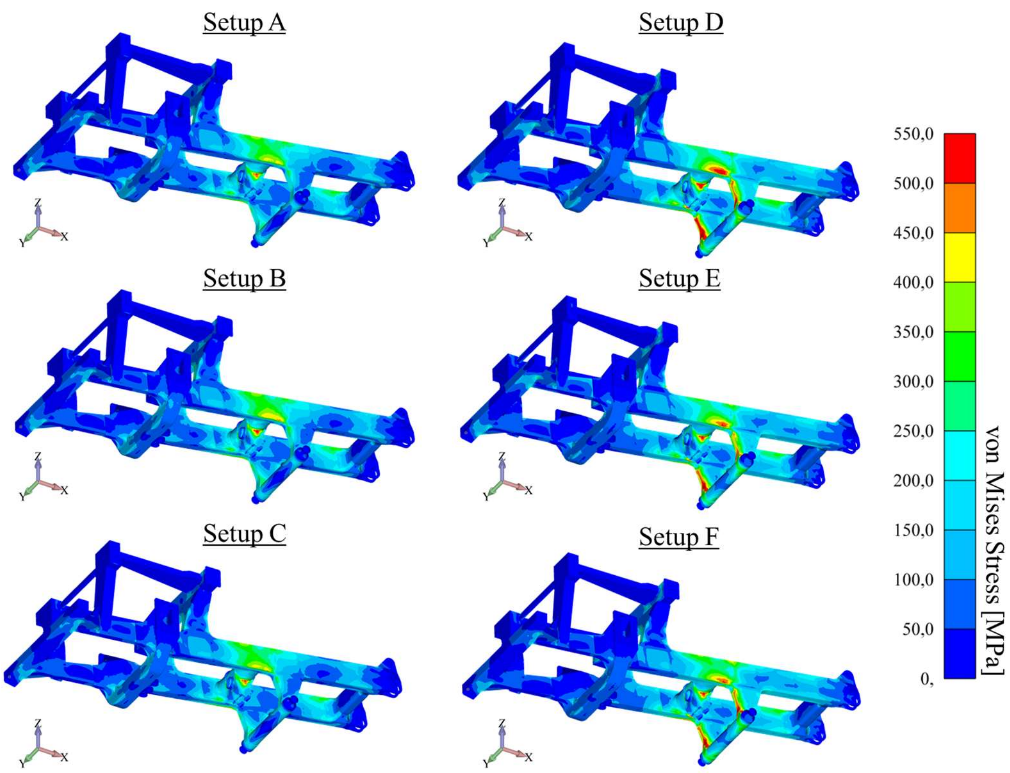

Figure 9: Fields of equivalent von Mises stresses on the FE model for each setup.

The FE model has been developed using the package FEMAP/NASTRAN (Siemens, 2016). Equivalent stresses computed by using the von Mises criterion can be observed in Figure 9. Table 3 indicates peak stresses obtained for the critical regions (regions I to III). Besides the maximum stress, the variation of the stress amplitude is evaluated to allow a fatigue life estimation. For the fatigue analysis, regions I to III are chosen since they are critical regions with butt welds and high tensile stresses on operation. Also, it is known that the presence of cracks due to fatigue in these regions are frequently reported.

Table 3: Maximum values of the equivalent von Mises stresses for each setup.

Setup	Peak von Mises stress [MPa]		
	Region I	Region II	Region III
A	468.9	469.4	264.6
B	500.1	501.9	282.4
C	513,6	515.3	291.4
D	557.1	656.1	604.8
E	618.5	720.7	615.1
F	497.9	581.0	597.8

The rainflow technique for cycle counting described by ASTM E 1049-85 (ASTM, 2005) is employed in the analysis. Figure 10 displays the cycle count for different stress ranges. There are about 65 to 75 cycles per maneuver.

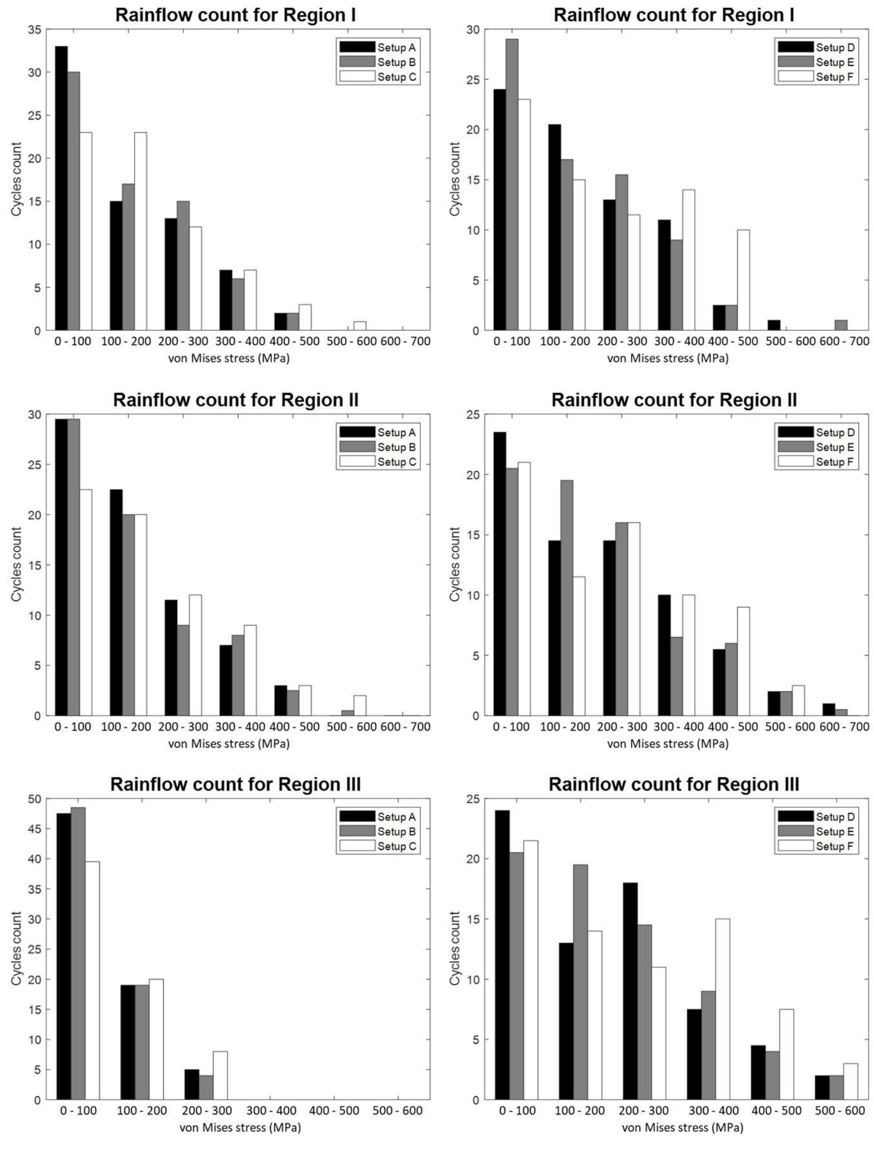

Figure 10: Rainflow count for different stress ranges.

Table 3 shows that the lower maximum stresses are estimated at the unloaded setup A (softer HPS and tires). The stress difference between setups A and C is around 10% for each region. On the other hand, the loaded setup cases indicate significant differences among maximum stress values. On regions I and II for rigid suspension (setup F), maximum stresses are 20% smaller than those of setup E. This result indicates the role played by the nitrogen gas pressure on the hydropneumatics suspension response. The rainflow cycle count indicates that higher stresses are generally observed at stiffer suspensions.

5. CONCLUSIONS

This paper describes a computational procedure that combines multibody dynamics and the finite elements method tools to evaluate the influence of tire pressure and initial nitrogen volume of a hydropneumatics suspension on the forces and stresses acting on a mining haul truck frame. From the obtained results, the following conclusions can be drawn:

1. Experimental values of the suspension gas pressure have shown a good correspondence with predictions rendered by the model. The haul road on which the measurements of gas pressure have been performed may be classified as a D-Class road (ISO, 2016), which represents unfavorable conditions for the truck operation.
2. Front suspension forces may increase up to 4-7% when stiffer setups (C and F) are used. When softer setups (A and D) are employed, the model results show a reduction of these forces in the range of 4-8%.
3. Rear suspension forces may increase up to 12-14% when stiffer setups (C and F) are used. If softer setups (A and D) are considered, it may reduce up to 10-11%.
4. At first glance, it seems that the stiffest suspension setup (setup F) is not the worst scenario for the truck frame in terms of stress levels. This result requires further investigation, maybe accounting for experimental data of suspension displacement and velocity, the oil viscosity on the suspension, and other variables in order to enlarge the understanding about the role played by the suspension nitrogen gas pressure. Furthermore, the finite element analysis accounts on other sources of input, such as the trailing arm and the hoist cylinder forces, which may have significant contribution to the stresses estimated for the truck chassis.
5. In the fatigue analysis, cycle counting shows that a more rigid suspension (setups C and F) may reduce the frame useful life. On the other hand, softer suspension setups can increase the fatigue life of the truck frame.

6. ACKNOWLEDGEMENTS

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