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Assessment of suspension damper failures on the dynamic response of a semi-trailer truck multibody system

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Abstract. This study investigates the dynamic behavior of commercial cargo vehicles, focusing on how suspension defects affect the performance and safety of semi-trailer trucks. It evaluates the impacts on handling and driver comfort—key factors for road safety and operational efficiency. A multibody dynamic model is used to simulate a semi-trailer truck subjected to ISO 8608:2016 road profiles and speeds, incorporating suspension failures such as shock absorber malfunction (zero damping). Acceleration power spectral density is applied to assess performance degradation, while the Vibration Dose Value (VDV) estimates potential health risks from prolonged vibration exposure. Results show that suspension defects significantly reduce vehicle stability and ride comfort, increasing driver fatigue and safety risks. Load variations also influence the vehicle's dynamic response. These findings highlight the importance of preventive suspension maintenance to preserve vehicle dynamics and ensure occupational health. The study contributes to improving commercial vehicle safety standards by minimizing accident risks and enhancing long-haul driver well-being through better vibration control.

Keywords: Suspension defects, vehicle dynamics, multibody model, semi-trailer truck safety.

1 Introduction

Damping systems are typically designed assuming full functionality of components like springs and shock absorbers. However, over a vehicle's lifespan, factors such as poor maintenance or lack of component replacement may compromise system performance [1][2]. The dynamic behavior and ride comfort of commercial freight vehicles depend strongly on both suspension condition and road surface quality [3]. According to the Confederação Nacional do Transporte (CNT) [4], in 2024, only 33% of Brazil's paved roads were rated "excellent" or "good," with the rest classified as "fair," "poor," or "very poor," based on 111,853 km of evaluations.

In Brazil, ISO 2631-1:1997 provides guidelines for comfort assessment, while ISO 8608:2016 classifies road roughness (A to E). Authors such as [5] define speed standards per road class, which in this study are adapted to actual conditions and CTB limits.

This work implements a full-vehicle multibody model with 24 DOF [6], simulating different road profiles and speeds per ISO 8608:2016. Scenarios with suspension failures are analyzed, with comfort assessed via ISO 2631:1997 and NHO-09:2013, focusing on dynamic effects and implications of shock absorber failure in the balanced suspension system.

2 Theoretical background

2.1 Dynamic model

The differential equations of motion of the model can be obtained based on the Lagrange equation, Eq. (1):

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_i}\right) - \frac{\partial T}{\partial q_i} + \frac{\partial U}{\partial q_i} + \frac{\partial D}{\partial \dot{q}_i} = Q_1 \qquad i = 1,2,\dots,n, \tag{1}$$

wich is function of the kinetic energy *T*, the dissipative energy *D*, the strain energy *V*, and the generalized forces *F*, associated with the generalized coordinates *x*.

This will be solved by the Newmark method. According to [7], the Newmark method is based on the assumption that acceleration varies linearly between two-time steps. Therefore, the convergence and accuracy of the obtained response will depend on the size of the time interval and the ability of the functions to accurately represent the acceleration within this interval.

$$\ddot{X}_0 = M^{-1}(F_0 - C\dot{x}). \tag{2}$$

In this work, the damping of the truck suspension systems is assumed to show nonlinear behavior, as it provides a better approximation to experimental data, as described by Eq. (2) presented by [6]. The suspension system under study is based on a bogie configuration, commonly referred to as 'balanced' because, due to your design, the weight is distributed across all axles.

$$F_c = \lambda[1 + \eta\, sgn(v)]|v|^n, \tag{3}$$

where respectively, λ is the scaling factor, η the asymmetry coefficient, *n* the characteristic damping exponent, and *v* the velocity of the damper shaft.

2.2 Road profiles

ISO 8608:2016 standardizes the characterization of road surface roughness profiles based on the power spectral density (PSD) of vertical displacements. The PSD is expressed as a function of spatial frequency and, when represented on a logarithmic scale, follows a decreasing linear trend within the range of 0.011 to 2.83 cycles/m (equivalently, 0.063 to 17.7 rad/m), under the condition of constant vehicle velocity.

The model employs a reference PSD, denoted as $G_0(n_0)$, and a reference spatial frequency $n_0 = 0.1$cycles/m. In this study, the exponent *w* was adopted as 2. The relationship between the PSD and spatial frequency is defined by the following equation:

$$G_0(n_0) = G_d(n_0)\left(\frac{n}{n_0}\right)^{-w}. \tag{4}$$

Annex C of the standard provides the classification of road surfaces according to the level of roughness or irregularity. The road profiles are categorized in alphabetical order, with Class A representing the smoothest (highest quality) surface. The subsequent classes indicate progressively higher levels of degradation, Figure 1.

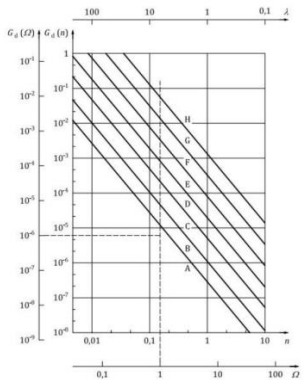


Figure 1. Classification of road surface profiles [ISO 8608, 2016]

2.3 Ride comfort and occupational health

ISO 2631-1:1997 is the primary international reference for evaluating the effects of whole-body vibration on

occupational health. The fundamental parameter adopted by this standard is the frequency-weighted root mean square (RMS) acceleration (a_w, in m/s²), which represents the quadratic mean of the vibratory acceleration over time, reflecting the energy transmitted to the human body during exposure.

The standard establishes that exposures of varying durations and intensities can be compared using energy equivalence equations, based on the weighted RMS acceleration. The equations below (Equations 3.26 and 3.27 from the ISO standard) allow for this comparison:

$$a_{w1} \cdot \sqrt{T_1} = a_{w2} \cdot \sqrt{T_2}, \tag{5}$$

$$a_{w1} \cdot {T_1}^{1/4} = a_{w2} \cdot {T_2}^{1/4}. \tag{6}$$

The resultant exposure acceleration (*aren*) is computed as the vector sum of the RMS accelerations along three orthogonal axes, weighted according to the standard. In Brazil, daily exposure limits are set by NHO-09, issued by Fundação Jorge Duprat Figuedo de Segurança e Medicina do Trabalho (Fundacentro). This standard defines normalized parameters for an 8-hour workday:

• *aren*: normalized RMS acceleration representing total daily exposure.
• VDVR: combined vibration dose value (VDV) across the three axes.

$$VDV_j = \sqrt[4]{\int_0^T \left[a_{wj}(t)\right]^4 dt} \tag{7}$$

3 Methodology

3.1 Vehicle model

A 24-degree-of-freedom multibody model was used to represent a cargo vehicle combination (CVC) composed of a 6x4 tractor unit (CM) and a tandem triple-axle semi-trailer. The model assumes point tire–track contact without separation, small component displacements and rotations, and linear stiffness for tires and suspension, while neglecting anti-roll bar and suspension geometry effects.

The degrees of freedom are distributed to capture the system's key dynamic behaviors: the cabin, tractor chassis, and semi-trailer each have three DOF (vertical, pitch, and roll); each of the six axles has two DOF; the driver's seat has one vertical DOF; and the bogie suspension has two DOF, allowing vertical motion and rotation around the trunnion. This nonlinear component follows the formulation in [6].

The model is excited by road profiles defined by ISO 8608:2016. Following [5], simulations consider specific vehicle speeds based on values from the literature, adapted to Brazilian conditions and legal limits [8], across five road classes (excluding speed bumps). Geometric and mechanical parameters (l_1 to l_9) are adopted from [6], with the exception of the sprung mass m_t, which varies across the case studies.

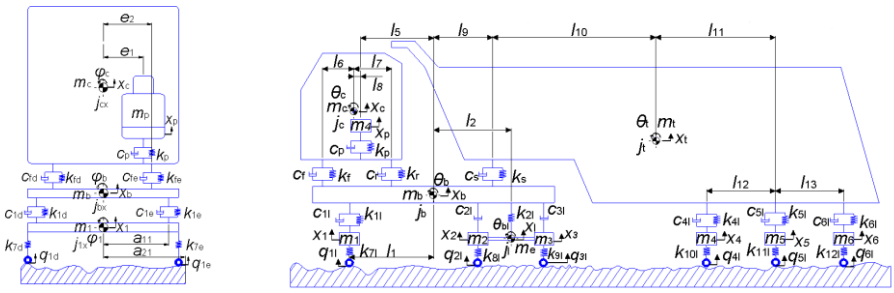


Figure 2. Truck and Semi-Trailer Model with 24-DOF

3.2 Road profile construction

The road profile was generated in MATLAB in accordance with the ISO 8608:2016 standard, as described in Section 2.2, using five levels of roughness, as shown in Figure 1, corresponding to Classes A through E. A fixed random seed was used in the generation of random numbers to ensure consistency of the profiles.

The left and right sides of the road surface are correlated, such that the wheels on each side experience

different excitations, allowing for the perception of body roll effects at the driver's seat. Figure 3 presents the time-domain road profile for a Class D roughness level, showing the surface traversed by the truck's tires, where the differences between the left and right tire paths can be clearly observed.

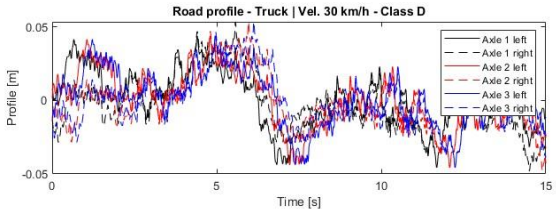


Figure 3. Road profile for Class D | Truck

4 Analysis and discussion of results

4.1 Load impact on acceleration by road class

Figure 4 show the driver's acceleration levels for E road class and their corresponding nominal recommended maximum speed. The load on the semitrailer varied across four conditions: unloaded (6.5 t), half load (17 t), full load (34 t), and 15% overload above full capacity.

For road profiles A, B, and C, was observed that the unloaded condition exhibits higher acceleration peaks at certain moments, and this tendency becomes more pronounced for roads D and E. This behavior is further analyzed in Figure 6, which shows that the weighted RMS acceleration values are higher in the unloaded condition, except for road class A.

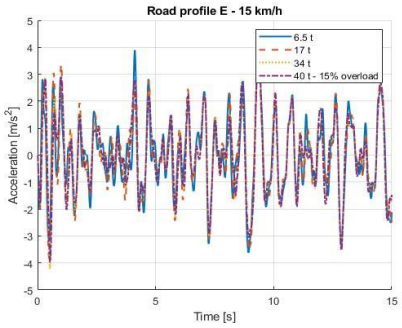


Figure 4. Acceleration for road profile E

4.2 Damper malfunction impact on acceleration by road class

The damping failure simulations in the balanced suspension of the tractor unit revealed that the third axle is the most critical in terms of acceleration levels experienced by the driver. One possible explanation for this behavior lies in the distance l_2, shown in Figure 1, which corresponds to the distance from the tractor unit's center of mass to the GDL of the balanced suspension. Simulations were carried out for road profiles A through E, using their respective speed limits, considering failure cases on both the second and third axles.

Figure 5 shows the acceleration profile obtained for the case of failure on the left third axle under road profile E, following the same pattern presented in Section 4.1. Figure 6 presents the RMS acceleration levels at the driver's seat, where an increase in values can be observed for nearly all loading conditions. Another behavior evidenced by the RMS acceleration values is the degradation in the half-load condition (17 t) under road profile E.

The acceleration spectrum in Figure 5 also shows a trend of increasing peaks, which explains the higher average (RMS) acceleration levels in Figure 6. Moreover, there is a increase in the contribution of unloaded and lightly loaded conditions to the peak values in the acceleration time history, including instances of phase shift.

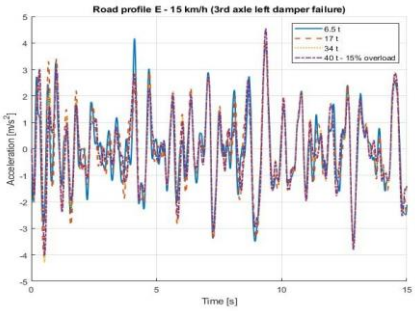


Figure 5. Acceleration for road profile E with simulated damper failure

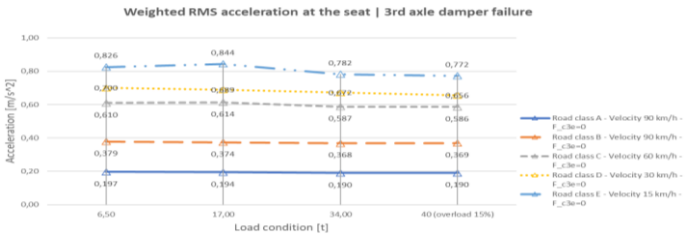


Figure 6. Weighted RMS acceleration for road class – velocity with simulated damper failure

4.3 Dynamic behavior using fault scenario simulations

As [1] suggests, Poincaré sections can aid as a useful tool for analyzing dynamic systems and comparing different scenarios. The Poincaré sections presented below aim to provide a simplified visualization of the chaotic behavior characteristic of systems of this nature.

Figures 7 and 8 show the phase planes for the left wheels of axles 2 and 3 of the tractor unit. The slight overlapping of the trajectories over time (indicated by the color gradient) suggests a tendency toward quasi-periodic behavior (steady-state), where each loop corresponds to an oscillation cycle. This behavior aligns with what is observed in the Poincaré sections in Figures 9 and 10. The higher density of points in Figure 9 matches the overlapping seen in Figure 7 — indicating that suspension 2 is better damped than suspension 3, even though no intervention has yet been applied.

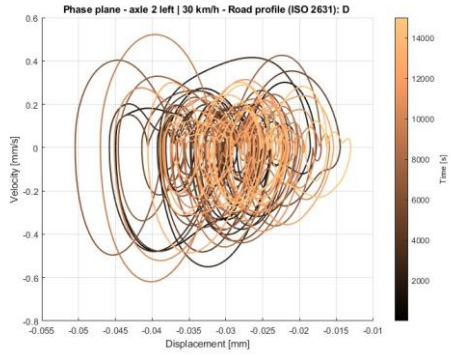


Figure 7. Phase plane – axle 2 left truck

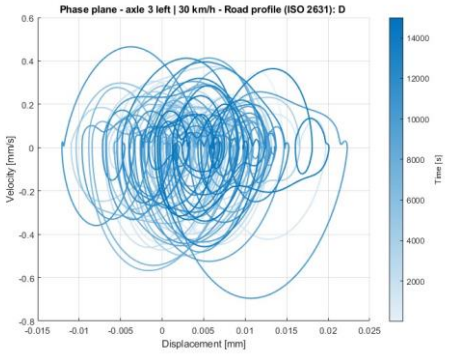


Figure 8. Phase plane – axle 3 left truck

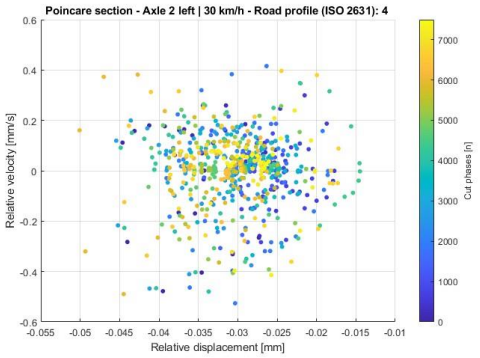


Figure 9. Poincare section – axle 2 left (truck)

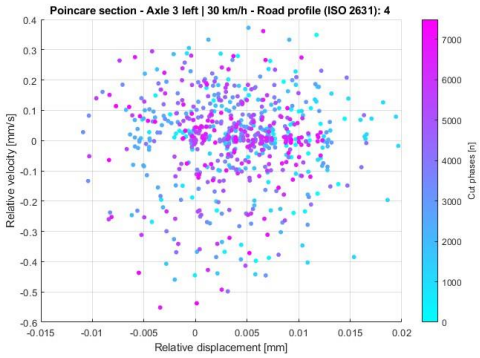


Figure 10. Poincare section – axle 3 left (truck)

From Figure 10 onward, the system's behavior changes drastically, as expected. The "scattered" or "shifted" appearance in the phase planes can be attributed to the transient behavior induced by the malfunction of the damper on the left wheel of axle 3. What is particularly noteworthy is the mutual influence between axles 2 and 3 — a characteristic behavior of bogie-type (non-independent) suspension systems.

Once again, the Poincare sections shown in Figures 13 and 14 support the observations from the phase planes. The dispersion of points characterizes the symptoms of chaotic behavior triggered by the loss of damping on the left wheel of axle 3. Each point in the Poincare section represents the system state at a specific time of the oscillation cycle. It can be observed that, despite varying velocities, the displacement values repeat, reinforcing the presence of structured yet chaotic dynamics.

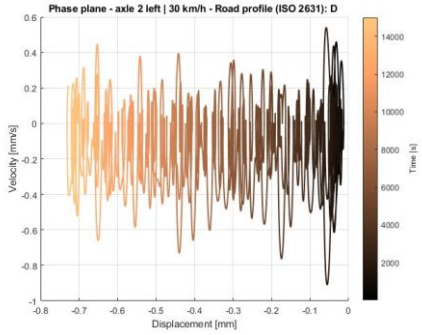


Figure 11. Damper failure in axle 3 left

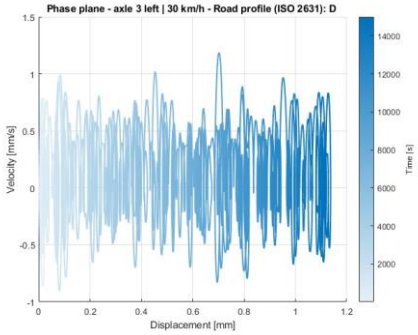


Figure 12. Damper failure in axle 3 left

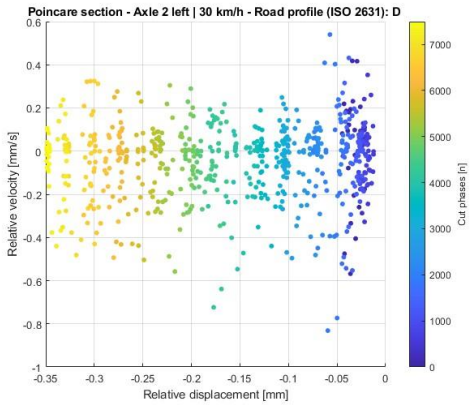


Figure 13. Damper failure in axle 3 left

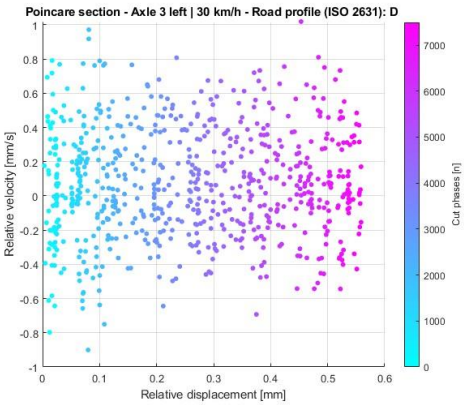


Figure 14. Damper failure in axle 3 left

5 Conclusions

This study was driven by the need to understand the dynamic behavior of freight vehicles in the Brazilian fleet, which plays a strategic economic role. These vehicles still rely on human operators, making their comfort, occupational health, and safety — as well as that of nearby personnel — strongly dependent on system performance, particularly the suspension [9].

Simulations were conducted using ISO 8608:2016-standardized road profiles to evaluate a 24-degree-of-freedom (DOF) vehicle model. Results showed that while individual track profiles did not present critical health risks, prolonged exposure was a key factor — underscoring the importance of journey duration in vibration assessments.

However, suspension faults in the tractor unit led to acceleration levels exceeding NHO-09 thresholds, especially on class E roads and at certain speeds. Damping failures also caused load redistribution, increasing mechanical stress on other components and potentially reducing their lifespan.

It is important to note that the results refer to the DOF of the driver's seat — the endpoint of multiple mechanical and dissipative systems that filter tire–road excitations. The absence or failure of these elements may severely compromise vehicle safety.

Future work should focus on experimental validation to improve model accuracy, as well as the integration of real Brazilian road data. A more comprehensive model is also proposed, incorporating yaw dynamics, detailed tire damping, and tire–road interaction, to support analyses of both longitudinal and lateral dynamics. Additionally, evaluating different loading configurations and CG positions is recommended to better understand their impact on vehicle stability and safety.

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