A FINITE ELEMENT ANALYSIS OF THE MECHANICAL-STRUCUTURAL BEHAVIOR OF THE CHASSIS OF OFF-ROAD DUMP TRUCK

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Abstract. This work deals with a finite element analysis of the displacement fields on the chassis of an off-road dump truck operating under various loading conditions. The numerical procedure is capable of predicting the stress and strain fields on the chassis subject to dynamic loads derived from different road roughness profiles during some typical maneuvers performed by a truck on mining sites. Experimental data of the suspension pressure measurements captured from four off-road dump trucks CAT 775G are used to reproduce the unpaved road conditions in this work. Some critical points of the chassis are selected to evaluate the cycling stresses, which are employed to perform a linear fatigue analysis of the chassis. The predictions of the stress fields permit to study the mechanical-structural integrity of the chassis of the off-road truck operating with and without dumping loads. The results rendered in this work depict the stress curves of some points of the truck chassis and estimates of its fatigue life for actual operating conditions, providing technical data for the maintenance program of this type of vehicle.

Keywords: Off-road trucks; Vehicle chassis; Finite Element Method; Fatigue analysis

1 Introduction

Off-road dump trucks are widely employed for material transport in coal and metal mines Recently, the development of numerical procedures for the mechanical behavior of mechanical systems has enabled the development of safer and more reliable cargo transport vehicles [1]. The advancements in the finite element method have enlarged the capabilities of all types of mechanical analyses, improving the accuracy and efficiency of procedures employed to predict the fatigue life of several mechanical systems [2].

The main purpose of this work is to analyze the mechanical behavior of off-road truck chassis using a finite element procedure, aiming at estimating the stress fields for studying their mechanical resistance under different loading conditions. A multibody dynamic model of the truck is employed to evaluate the behavior of the vehicle subsystems operating at several road conditions. The loads estimated by the multibody model are then use to study the mechanical response of the chassis, which is modeled by the finite element method. The Finite Element Method (FEM) has been widely adopted in the mechanical design of automotive components, such as lightweight structural design and the analysis of vibration characteristics of the vehicle subsystems [3].

According to Kamal [4], the study of mechanical behavior can be assessed applying a different set of mathematical methods allied to software tools that enable comprehensive analysis of the dynamic conditions to which vehicle mechanical components are subjected. In complement, Moaaz [5] provides an extensive review on the development and refinement of techniques for fatigue life assessment and prediction in heavy vehicle chassis.

The study employs an integrated simulation workflow combining multi-body dynamics (MBD) and finite element method (FEM) to evaluate the mechanical behavior of a heavy-duty off-road truck chassis under actual working conditions, following the work of Mi [6]. The methodology includes four main stages: dynamic load extraction, structural modeling, vibration mode validation and fatigue life estimation.

2 Finite element procedure

The vehicle frame, as the skeleton of the vehicle, plays a critical role on its mechanical behavior. Using ALTAIR Hypermesh® software to develop the FEM model, the chassis of a heavy-duty off-road truck features a ladder or box-type geometry for high torsional rigidity and optimized load distribution. The geometric design minimizes stress concentrations, while the material's constitutive properties enable the frame to endure extreme operational conditions without mechanical failure. Figure 1 depicts the three-dimensional geometrical model of the chassis, in which the manufacturing process for materials are described using different colors, as it is shown on the legend.

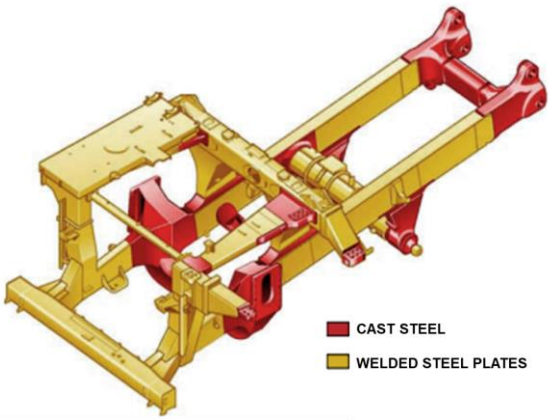


Figure 1 -Typical configuration of off-road truck chassis Source: Caterpillar [7]

The materials applied for the FEM model according to Gandra [8] on the physical components:

- ASTM A36 steel for welded steel plates components, including longitudinal beams and transverse stiffeners;
- SAE 1021 steel for cast elements, such as mounting brackets and suspension interfaces.

Table 1 - Material mechanical properties

Material	Modulus of Elasticity, E (GPa)	Poisson's ratio, ν	Yield Stress, S_Y (MPa)	Ultimate tensile Stress, S_{UT} (MPa)	Maximum elongation, Ɛmax (%)
A36	210	0.3	250	414	20
SAE1021	210	0.3	315	520	27

2.1 Boundary Conditions and Load Application

The boundary conditions represent as faithful as possible the actual constraints imposed by the body structure and loading behavior of the vehicle. The suspension load points, identified in the MBD model, were used as the primary locations for applying force vectors (including forces and moments in all axis). Figure 2 shows the application points of the load transferred from the suspension to the chassis.

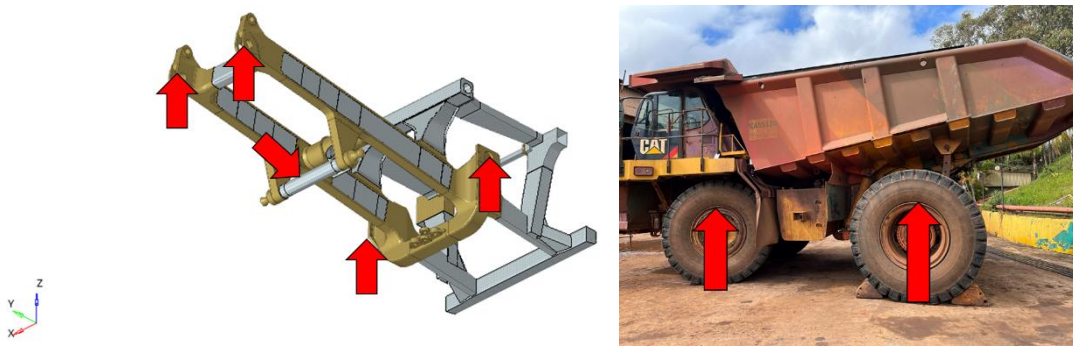


Figure 2 - Reference for load appliance in FEM model

To prevent rigid body motion and better approximate operational loading, constraints are applied to the truck body bed (dump body)—the region identified as having the highest reaction forces during load transfer events. These constraints were applied in selected translational directions, while preserving the chassis torsional flexibility to avoid over-constraining the structure, according to Fig. 3.

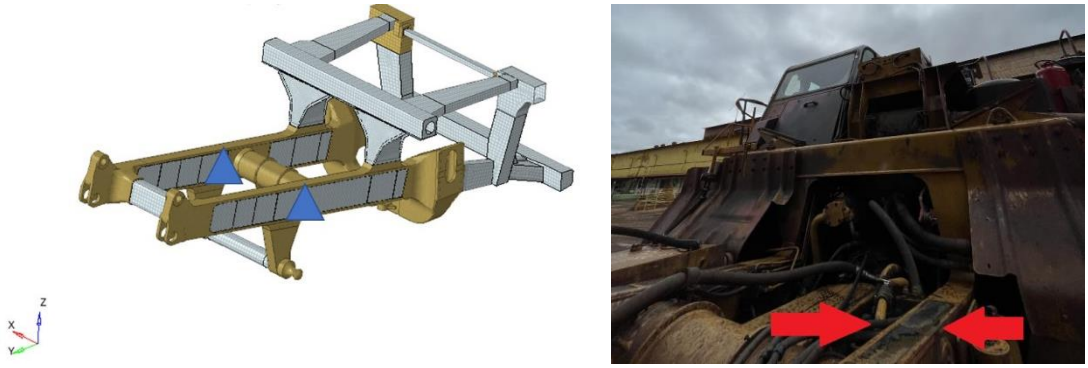


Figure 3 - Reference for Single Points Constraints (SPCs) in FEM Model

2.2 Determination of the Vibration Modes

The natural vibration modes of the chassis are estimated to assess the chassis dynamic to validate the FEM model in terms of mass and stiffness distribution. In finite element modeling of free vibration problems, the generalized eigenvalue problem is used to determine the system's natural frequencies and mode shapes. By discretizing the system into elements, the dynamic equilibrium of the eq. 1 is achieved, with remarks over the meshing discretization, as presented by Boffi [9].

$$[K - \omega_i{}^2 M]\phi = 0 \tag{1}$$

Where ω_i represents the natural frequencies of the model, [K] and [M] are respectively the stiffness and mass matrix of the model, and ϕ is the corresponding eigenmode.

- The first torsional mode was identified at 22.4 Hz, higher than the typical range for smaller truck chassis, which is bellow 10Hz [10];
- Bending modes (longitudinal and lateral) appeared between 25 Hz and 75 Hz, confirming the global stiffness integrity of the structure;
- These results validate the FEM model as a reliable base for subsequent fatigue evaluation.

Table 2 - Natural frequency modes of CAT 775G Chassis

Mode	Frequency (Hz)
1st	22.4
2nd	26.5
3rd	32.1
4th	39.9

Mode	Frequency (Hz)
5th	46.3
6th	51.8
7th	58.9
8th	70.9
9th	80.2
10th	82.1

Following the studies of Abd Rahman [11], the solution of the eigenvalue problem leads to the vibrational modes as described in Fig. 4:

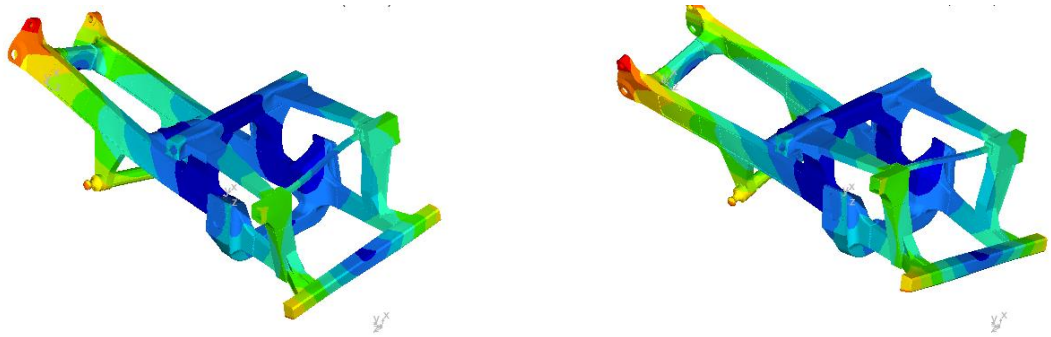
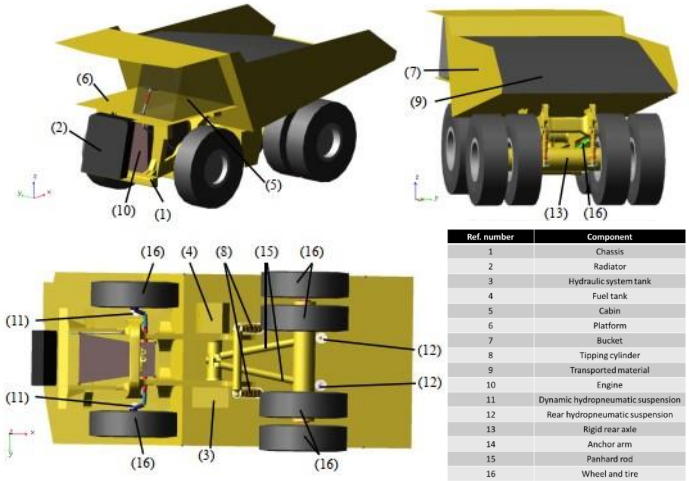


Figure 4 - 1st Natural frequency mode (22.4Hz) behavior of the FEM model

3 Multibody dynamics model and load extraction

The loading conditions applied to the finite element model were derived from a study conducted by Dantas [12]. In his study, a dynamic multibody model of the CAT 775G mining truck was developed using ADAMS Car®. The MBD model include main vehicle subsystems:

- Front and rear suspension systems (with actual bushing and damping behavior);
- Axle and wheel dynamics;
- Contact with irregular terrain.



Ref. number	Component
1	Chassis
2	Radiator
3	Hydraulic system tank
4	Fuel tank
5	Cabin
6	Platform
7	Bucket
8	Tipping cylinder
9	Transported material
10	Engine
11	Dynamic hydropneumatic suspension
12	Rear hydropneumatic suspension
13	Rigid rear axle
14	Anchor arm
15	Panhard rod
16	Wheel and tire

Figure 5- MBD model. Adapted Dantas [12]

Experimental data related to pressure in the four hydropneumatic suspensions, the transported material load, and the travel speed of three different CAT 775G trucks were obtained through the manufacturer's own monitoring system, known as the Vital Information Management System (VIMS). These data, recorded at a frequency of 0.5

Hz, cover a total duration of approximately 100 continuous hours (equivalent to 4.2 days) of uninterrupted operation. For the MBD model validation, the data collected in field from suspension pressure points for different road conditions were compared and validated, with an average difference of 14.3% [12], as presented in Fig. 6:

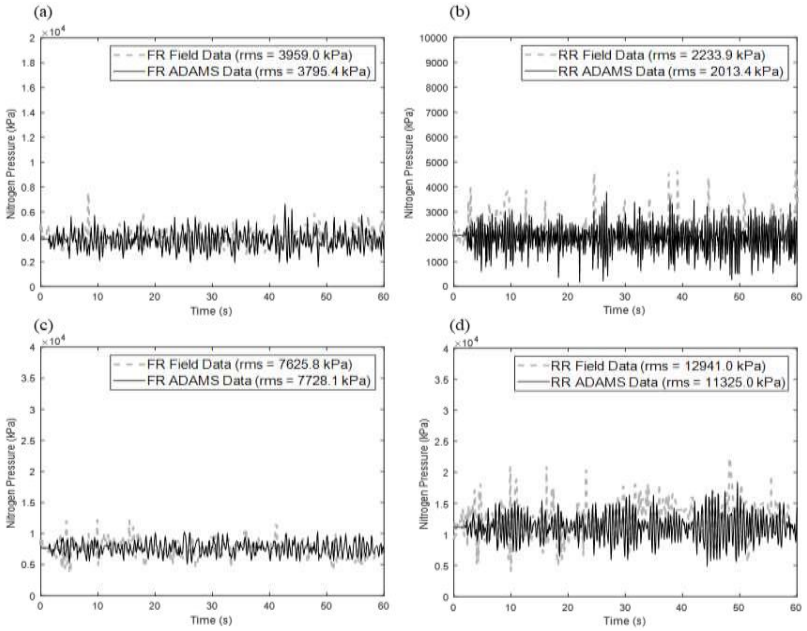


Figure 6 - Comparative values of gas pressure from field measurements and ADAMS model: (a) Unloaded truck front right suspension; (b) Unloaded truck rear right suspension; (c) Fully loaded truck front right suspension; (d) Fully loaded truck rear right suspension. Dantas [12]

Through simulations of off-road operational scenarios, the MBD model provided reaction forces at the suspension mounting points, including both transient and peak loading conditions representative of loading, unloading, and transport cycles. These loads were exported from ADAMS Car® and used as input loads in the FEM model.

The static analysis revealed critical stress distributions across the chassis under different operating conditions, including loaded and unloaded scenarios. Suspension forces and moments obtained from the MBD simulation (ADAMS Car®) were applied at the mounting points of the chassis. The stress concentrations were located in the transition zones between the main longitudinal beams and the suspension support brackets, with maximum von Mises stresses approaching 140 MPa, as described by Fig. 7:

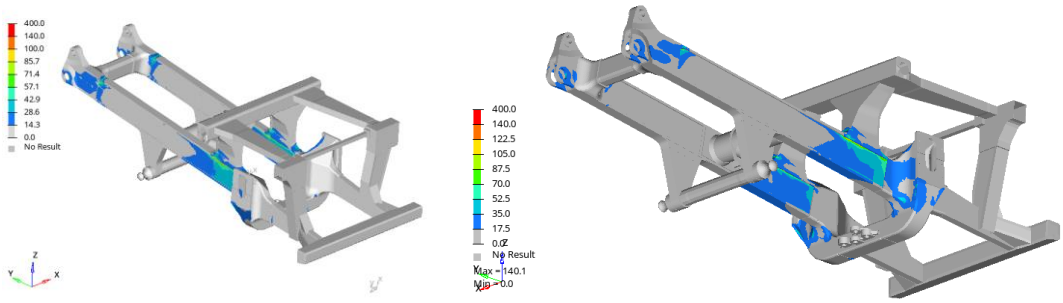


Figure 7 - CAT 775G chassis static analysis Von Misses Stress (MPa) behavior

4 Fatigue life prediction analysis

Following static stress analysis, a fatigue life estimation was conducted using ALTAIR® HyperLife, based on the stress-life (S-N) approach of cumulative damage fatigue [13], as described in eq. 2:

$$D = \sum D_i = \sum \frac{n_i}{N_i} \leq D_{Adm} \tag{2}$$

Where, D is total cumulative fatigue damage, D_i is the individual fatigue damage of each interaction, D_{Adm} is the admissible damage, n_i and N_i are respectively the number of cycles applied at a specific stress and number of cycles to failure at that same stress.
Fatigue input data included:

- Stress distribution from FEM static analysis under critical load cases;
- Operational load history collected via VIMS derived from operational cycles, distinguishing periods of loaded vs. unloaded operation, as defined in Dantas [12];
- Material fatigue properties sourced from standard databases for A36 and SAE 1021 steels;
- The input cycling study considered a 20year long vehicle usage.

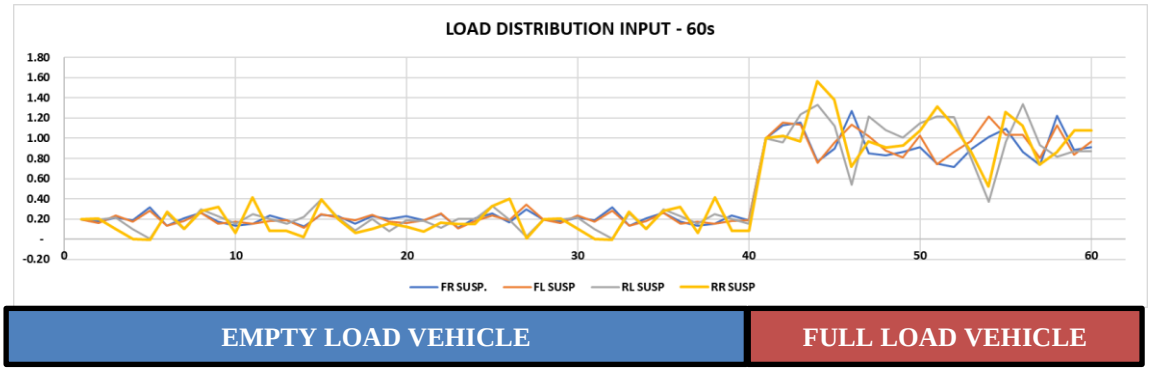


Figure 8 - Load Distribution input for each suspension point in FEM model

To replicate the cyclic loading conditions relevant to the fatigue life prediction of mechanical vehicle components, a load reconstruction methodology was employed using a 60-second input signal representative of operational duty cycles. In this signal, 65% of the time interval corresponds to the vehicle operating under unloaded conditions, while the remaining 35% reflects loaded conditions.

The fatigue evaluation considered variations in load amplitude during hauling (loaded) and return (unloaded) cycles, with results considering 20 years cycling input. Results revealed that almost all chassis structure exhibited a predicted fatigue life greater than 10^6 cycles, classifying it within the over expected life range for the defined loading. The worst results show a life expectancy of 2 years of work, at the edges of the load constraints, following Fig. 9:

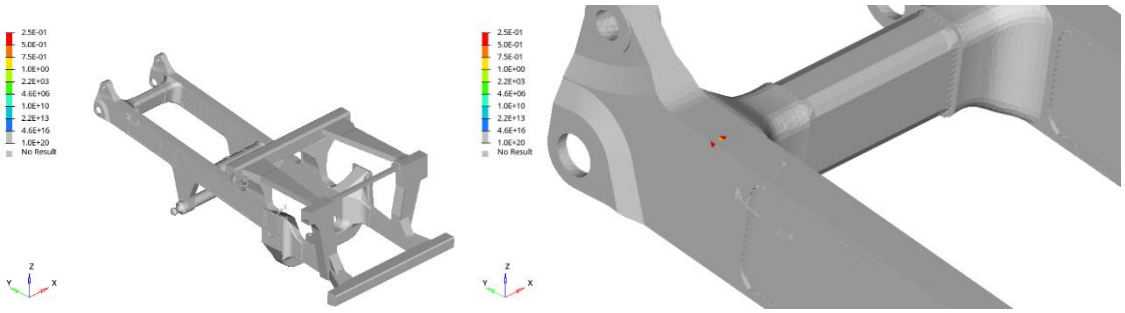


Figure 9 - Fatigue life prediction following cumulative damage theory - CAT 775G

5 Conclusions

The numerical procedure based on multibody dynamic and finite element models has been implemented to predict the stress distribution and the fatigue life of the chassis of an off-road dump truck. The stress analysis performed for static conditions indicates that the critical points of stress concentration are located in the transition zones between the main longitudinal beams and the suspension support brackets, with maximum equivalent von Mises stresses approaching 140 MPa. This value remains below the yield strength of ASTM A36 steel (250 MPa). Cast components modeled with SAE 1021 steel demonstrated structurally robust behavior, with stress levels concentrated around load-transfer interfaces, such as transverse supports. The overall stress distribution followed

expected structural load paths, supporting the model's mechanical validity under severe off-road conditions. This previous stress analysis demonstrates that the chassis design is performed considering that the stress levels must be below the allowable stress limits.

Furthermore, the linear fatigue analysis performed in this work indicates that most of the structural components of the chassis can present fatigue infinite life. Except for the connecting regions among suspensions and chassis, in which the fatigue analysis shows that the life expectancy can reach two years of operation.

Some challenges have been identified within the methodology applied that does not allow the model to fully represent actual life situations, such as:

1. Seam weld modeling used for the FE model. The actual stress concentration caused by the welded joints cannot be reliably predicted;
2. The distribution of material and load on chassis is non-continuous, hard to reproduce on computational model;
3. Simplifications taken to the FE model, such the very simplified models for the engine, cabin, and bucket, may lower the actual stiffness behavior of the system and induce non reliable results.

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