Automation of wheeled vehicles load bearing frames finite-element models loading procedure by using inertia relief method and vehicle multi-body dynamics model

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Abstract. The paper presents a technique for strain-stress calculation using finite element analysis for wheeled vehicle’s load bearing frames with the use of loads, obtained from full vehicle multi body dynamics models (MBD models). In this technique, boundary conditions need not to be defined, as the inertia relief method used. Frame and suspension parts are imposed in MBD model as flexible bodies to increase accuracy of loads calculations. The presented method is capable to automatize loading procedure of finite-element models, thus decreasing computational costs and results processing costs while analyzing numerous load cases. As an example, strength analysis of 6x6 articulated vehicle frame is presented. Proposed method compared to “classic” finite-element frame modelling technique, which use conventional loads formulation and fixed boundary conditions. In load case “standing on ground”, calculation results with the use of two methods demonstrate high convergence in zones, located far from suspension and frame mounting joints. For load case “hanging of second axis” results are significantly different, due to ability of MBD-model to capture behavior of suspension links, and calculate true vehicle frame movement in space in particular load case. Loading procedure of frame finite element model is automated, using script, which transfer loads from MDB-model.

1. Introduction

At the moment, methods for vehicle load bearing frames stress calculation, providing quantitative and qualitative convergence with experimental results, are developed and broadly described in literature [1], [3], [4], [6], [7], [6], [7], [6] . Most of these works use finite-element analysis (FEA) [9]. To obtain results, relevant to real strain-stress state of frame under load, it is necessary to create their complex finite-element models, that also takes into account suspension and other components, that contribute to combined system stiffness. In order to simplify the model, simplified rod-and-beam finite elements models of suspension and other vehicle systems usually used. This method has a number of disadvantages, which are, for example, difficulty in modeling of suspension kinematics; leaf springs deformed shape, stiffness properties of metal-rubber mountings, etc.

At the early stage of vehicle design concept development [9] in order to estimate properties of future product, one usually develop a multi-body dynamics model of entire vehicle. While virtual prototyping of different vehicle systems (suspension, transmission, etc.), these models are used as “virtual test benches”, examples of these test benches for suspension, steering and the entire vehicle are presented in [10], [11], [12], [14], [15], [16], [16], [16]. Using virtual prototyping, one can get general and complex
properties of the future vehicle (or its system). One of this particular results – load factors, acting in every joints of dynamic model.

Loads, obtained in this way, are applicable for finite-element analysis of load bearing frame with the use of inertia relief method. In inertia relief method, it is assumed, that body or system of bodies are in quasi-static balance state. To achieve quasi-static balance state of the model, equilibrant accelerations are evaluated to eliminate imbalance of loads and moments. Resulting system of linear and angular accelerations makes the sum of all the forces, acting on model, to be equal to zero. Here assignment of boundary conditions in the form of fixed constraints is not required. This makes it possible to transfer loads from MBD model to finite-element analysis of the frame with the use of automated procedures (scripts). Thus, it is possible to analyze a numerous of loading cases, comparing to the manual definition of boundary conditions and loads, that usually used in “classic” methods.

Thereby, the study of the paper is inertia relief method and full vehicle dynamic multi-body model use in automated loading procedure of vehicle frame finite-element model.

To illustrate method, paper describes developed vehicle MBD model with frame, equalizing beam suspension system with flexible parts and non-linear stiffness of metal-rubber mountings. Developed finite-element model of articulated 6x6 vehicle frame also presented. To compare “classic” and proposed method strain-stress state calculation techniques, two analysis with both methods are performed.

2. Algorithm for automatization of frame finite element model loading procedure

In order to perform stress calculation of vehicle frame with the use of the inertia relief method, it is necessary to follow block diagram on figure 1.

![Figure 1. Algorithm for automatization of frame finite element model loading procedure.](image)

Detailed analysis of methods for load adequacy checking and script generation are not presented, because these tasks are strictly specific to chosen software.

Static loads in vehicle frame and front/rear suspensions mounting joints are evaluated with the use of MBD model. Figure 2 shows general view of articulated vehicle MBD model. MBD model consists
of: front frame, front leaf suspension, connected to it, rear frame with equalizing beam suspension system, articulation joint assembly, connecting two frames and providing one degree of freedom: rotation of front and rear frames about vertical axis. Model includes steering hydraulic cylinders, modelled as rigid rods.

In the model, the following assumptions are made:

– parts of suspension system, except equalizing beam, are modelled as rigid bodies;
– joints are free of friction;
– wheels deformations are considered in the model of wheel interaction with supporting surface, wheel forces are applied in the center of the wheel.

3. Description of articulated vehicle multi body dynamic model

The MBD model of the articulated vehicle is shown on figure 2.

![Figure 2. General view of articulated vehicle MBD model: 1 – front leaf suspension; 2 – steering cylinders; 3 – rear suspension with equalizing beam; 4 – front frame; 5 – articulation joint assembly; 6 – rear frame.](image)

General view of front spring suspension model with two leaf springs and stabilizer is presented on figure 3, locations and types of joints, providing connection of parts, are presented on figure 4.

![Figure 3. General view of front leaf spring suspension model: 1 – axle; 2 – leaf spring; 3 – stabilizer; 4 – buffer; 5 – shock absorber.](image)
Figure 4. Joints location in the model of front leaf spring suspension: 1 – spherical joint; 2 – buffer springs; 3 – bushing; 4 – damper (shock absorber); 5 – bushing; 6 – cylindrical joint.

Leaf spring model description is presented in work [10].

Suspension stop buffers, modelled as a spring element with nonlinear stiffness, are mounted on the vehicle frame above leaf springs. Stabilizer consists of two rigid units, connected to each other by cylindrical joint with angular spring, whose stiffness corresponds to torsion stiffness of stabilizer.

Figure 5 shows general view of rear suspension with equalizing beam and two axles. Model includes six longitudinal reactive rods (two lower and one upper per each axle) and two lateral reactive rods, one per axis (figure 6).

Figure 5. General view of rear suspension with equalizing beam, two axles, longitudinal and lateral reactive rods.
All reactive rods connect axles with the frame by rubber-metal bushings, which are modeled by standard bushing elements. Equalizing beams are connected to the frame with revolute joint and can rotate about the lateral axis. Endpoints of equalizing beams are connected to axles with rubber buffers.

The feature of the developed model is the use of flexible equalizing beam in the rear suspension model. Flexible equalizing beam is defined by Craig-Bampton method, based on a specially developed finite-element model of equalizing beam (figure 7). Suspension stop buffers fixed on the frame model.

**Figure 6.** Rear suspension system model: 1 – front axle; 2 – rear axle; 3 – stop buffers; 4 – left equalizing beam; 5 – longitudinal reactive rods; 6 – right equalizing beam; 7 – lateral reactive rods.

**Figure 7.** General view of rear suspension equalizing beam finite element model for use as a flexible body in multi-body dynamics model.

Vertical, longitudinal and lateral reactions in tire contact patch with the ground are defined with specific model of wheel-ground interaction, and not described in this paper.
4. Load bearing frame finite element model description
For strength analysis of wheeled vehicle frame finite-element model is developed. The model is based on shell-type finite elements. The average size of shell finite element is 30 mm, total number of nodes of front and rear frames is 895973, number of elements is 564100. Bolts and rivets are modelled with short beam elements.

To compare two methods of frame strength analysis, two calculations are performed: 1) “classic” method with the use of simplified rod and beam suspension model and 2) inertia relief method with loads, calculated in multi body dynamics model.

Finite-element model of the frame is presented on figure 8. Full mass of the vehicle is defined with pointed masses, located in gravity centres of massive vehicle aggregates (figure 9).

Figure 8. Finite-element model of load bearing frame: 1 – front frame, 2 – front part of articulation joint assembly, 3 – rear part of articulation joint assembly, 4 – rear frame, 5 – simplified model of front suspension, 6 – simplified model of rear suspension.
Figure 9. Pointed masses, located in gravity centres of massive vehicle aggregates:
1 – cooling system; 2 – battery and receivers; 3 – engine; 4 – cabin; 5 – gearbox; 6 – pipes, fluids, electrical wires, etc.; 7 – exhaust system; 8 – mass of spare wheel and crane; 9 – cardan shaft; 10 – transfer box.

Joints of front and rear frames are modelled by rod-type elements with assisting RBE2-type rigid elements. Rigid element “RBE2*” allow linear relative translation along vertical axis (figure 10).
Simplified rod and beam models of front and rear suspensions are presented on figures 11, 12. Joints of front and rear suspensions with frame are modelled by RBE2** elements, that permits rotation around Y-axis degree of freedom. RBE3** element in front suspension model have free linear degree of freedom along X-axis. This allows longitudinal motion of leaf spring rear end relative to the frame.
**Figure 11.** Simplified finite-element model of front suspension.

**Figure 12.** Simplified finite-element model of rear suspension.
Finite-element model of rear frame is presented on figure 13. Payload is modelled by single point mass, located in the center of gravity of rear frame and distributed uniformly on upper longerons with the use of RBE3 element.

![Center of gravity](image)

Figure 13. Finite-element model of rear frame with payload, distributed along upper longerons.

Steering hydraulic cylinders in articulation joint assembly and their mountings to front and rear frames are presented on figure 14. Hydraulic cylinders are modelled by rod finite elements, connected to brackets of front and rear frames, via RBE2 elements and intermediate BEAM elements. Material properties of frame are presented in table 1.

Table 1. Material properties of frame parts.

| №  | Part name                        | Material | Elastic modulus, E, MPa | Poisson ratio | Yield stress $\sigma_y$, MPa | Stress limit $\sigma_m$, MPa | Breaking elongation, $d5$, % |
|----|---------------------------------|----------|------------------------|---------------|-------------------------------|-----------------------------|------------------------------|
| 1  | Longerons                       | 20GUT    | 200000                 | 0.3           | 480                           | 600                         | 24                           |
| 2  | Articulation joint assembly parts | 10HSND   | 200000                 | 0.3           | 390                           | 530                         | 19                           |
Two described method of strength calculation compared on two load cases, typical for 6×6 vehicle: 1) vehicle static standing on plain surface and 2) hanging of second axis. Many other load cases are analyzed as well, but they are not presented in this paper due to its limited volume of article. Also comparison of the stress-strain state is presented for the rear frame only. However, all conclusions made from this two load cases are true for all analyzed loadcases and other frame parts.

In first method, “classic” approach, boundary conditions and loads are presented in figure 15, 16 (where \( g = 9,81 \, \text{m/s}^2 \) – gravity acceleration).

Figure 14. Finite-element model of steering hydraulic cylinders.

Figure 15. Boundary conditions and loads for load case “standing on ground”. “Classic” approach to definition of loads and boundary conditions via simplified rod and beam suspension model.
While second axle hanging, rear suspension fits into stop buffer of third axle. Therefore additional beam element and RBE3*** element defined for this load case, which carry Z-axis load from third axle (figure 17).

In second method loads and reactions from suspension joints, calculated using multi body dynamics model, are applied in corresponding points of rear frame (figure 18). Automatization of loading finite element model achieved by specially developed script. This script transfers loads and reactions in key points of frame (joints, rod holders, mountings of hydraulic cylinders, etc.) from multi body dynamic model into finite-element model of the frame.
5. Strength calculation results
Calculated stress state of rear frame, obtained for two computational approaches, are shown on figures 20, 21 – load case “standing on ground”; figures 22, 23 – load case “hanging of second axle”. Results demonstrate, that both methods are qualitative convergent. Insignificant difference in results for load
case “standing on ground” is related to the influence of reactive rod forces, stiffness of rubber-metal joints and loads re-distribution in rear frame from suspension. For load case “hanging of second axle” calculation results have significant quantitative differences, caused by change of spatial orientation of vehicle and its frame (during hanging of second axle, vehicle tilts, resulting in redistributing of internal loads in rear suspension, articulated joint assembly). Figures 22, 23 shows stress distribution of the rear frame.

Negative feature of “classic” method is the necessity to modify suspension rod and beam model in every different load case: load case “hanging of second axis” demands additional elements, for stop buffers modelling at third axle. In other load cases, for example, “hanging of third axle” it is also necessary to modify finite-element model respectively. Modification of finite-element model for every single load case makes analysis of results more complicated: for example, the plotting of envelope stress distribution over number of different finite element models become very labor consuming. Inertia relief method is free of this disadvantage.

**Figure 20.** Equivalent von Mises stress, Pa, distribution on rear frame for “classic” calculation method, load case “standing on ground”.

**Figure 21.** Equivalent von Mises stress, Pa, distribution on rear frame for inertia relief method, load case “standing on ground”.
Figure 22. Equivalent von Mises stress, Pa, distribution on rear frame for “classic” calculation method, load case “hanging second axle”.

Figure 23. Equivalent von Mises stress, Pa, distribution on rear frame for inertia relief method, load case “hanging second axle”.

6. Conclusions

1. Strength calculation results of rear frame have qualitative and quantitative agreement for both methods in load case “standing on plain surface”. In load case “hanging of second axle” the results have qualitative agreement, but they have significant quantitative differences, because the multi body dynamic model capture the change of spatial orientation of the vehicle and its frame, general frame and suspension deformations.

2. “Classic” approach with simplified rod and beam suspension models does not provide accurate simulation of real suspension kinematics and real force distribution. Thus, it has significant influence on values and directions of loads in local areas of frame and suspension joints.
3. Automated loads transfer from multi body dynamic vehicle model into finite-element model of frame and inertia relief method makes it possible to analyze numerous load cases, which makes it possible to perform optimization tasks of frame more precise, which is crucial, for instance, for topology optimization tasks.

4. Inertia relief method for automated loading of finite element model makes it possible to use single finite element to analysis of full range of load cases. Thus, it is possible to build an envelope stress distribution over all load cases. This allowing significantly reduce difficulty of numerous results processing.

5. Inertia relief method shown in this paper allows to analyze front and rear frame of articulated vehicle, as well as other systems of vehicle, – separately from each other. This reduce computational time costs and results processing time costs significantly.

6. The proposed method makes it possible to divide front and rear frame analysis tasks between different developers to speed up analysis process. It is also decrease possibility of human error in calculation process.

7. Using “classic” method of loads and boundary conditions formulated via simplified suspension rod and beam models it is necessary to modify finite element models in every load case, also it is necessary to analyze entire frame model, with mass model and suspension models, which increase complexity of calculation significantly.

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