EXAMPLE OF EXPERIMENTAL VALIDATION AND CALIBRATION OF A FINITE ELEMENT MODEL OF A HEAVY VEHICLE

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Abstract

An example of the experimental validation of a finite element model of a selected heavy vehicle is presented in the paper. A truck tractor with a three axle single drop low boy trailer and the total weight of 20 tons was selected as a representative for the tests. The major goal of the conducted studies was to develop a well validated the numerical model of a heavy vehicle applicable for computer simulation of dynamic interaction between a vehicle and a bridge or road structure. Therefore, only several components of the vehicle, affecting the vehicle-road interaction, like wheels and the suspension systems, were modelled in detail. The rest of components were simplified and considered as rigid bodies. The finite element model included fully pneumatic tires and the suspension system modelled with discrete massless springs and dampers. Numerical analyzes were performed using the LS-DYNA computer code.

The validation and calibration procedure proposed in the current paper was carried out in two steps. In the first one, some parameters such as material densities, thickness of selected elements, were modified to achieve the correct mass distribution in the model based on the measured axle loads. In the next step the stiffness and damping parameters of the suspension system were evaluated based on the results of the experimental tests. The spring and damping coefficients at all axles were adjusted until the performance of the FE model closely matched that of the actual vehicle.

Keywords: *numerical simulations, finite element analysis, heavy vehicles, validation, suspension systems*

1. Introduction

One of the important issues for maintenance and design engineers is the magnitude of the actual dynamic loads exerted by heavy vehicles on roads and bridges. Experimental, analytical, and numerical studies were conducted to estimate the actual values of dynamic load allowance which represent dynamic overloading as compared with static loads [1, 2]. A need for consideration of a wide variety of heavy vehicle configurations suggests a selection of computer methods as the most efficient. Heavy vehicles are usually considered in such analyses as simplified systems of rigid masses connected through linear or nonlinear springs and dampers [3-6]. Besides the correct mass distribution, a model of a heavy vehicle should have an appropriate representation of the actual suspension system.

2. Determination of parameters for the actual suspension system

Characteristics of the vehicle suspension can be determined through experimental compression tests conducted either on an isolated suspension system or indirectly through field experiments carried out on the entire vehicle. The purpose of such tests is to determine a spring stiffness k and a damping coefficient c of the suspension system. The first method is expensive as it requires removal of the suspension from the existing vehicle or a purchase of a new one. In this method, the velocity of the piston of the shock absorber is measured and recorded as a function of the load applied. This relationship is usually non-linear, therefore it is simplified by piecewise linear functions. Idealized, perfectly fixed boundary conditions in direct suspension testing do not account for sometimes worned out and partially loose connections between the suspension and the vehicle. In addition, testing of a new suspension system will often result in different suspension characteristics as compared with those found in actual and used vehicles.

In the indirect method, the tests are conducted on the entire vehicle moving along predefined road surface profiles with different loads and at different speeds [7-9]. Typical data acquisition from such tests usually includes time histories of accelerations and relative displacements between selected points. Filtered experimental output data is analyzed and can be used for validation of analytical or numerical models. The first approximation of the suspension characteristics can also be obtained for some of the technical solutions using simplified formula developed by the automotive industry. Such formula allow for calculation of linear stiffness of leaf spring suspension based on dimensions of leaves and their number. The disadvantage of the indirect method is the difficulty in measuring dynamic interaction forces between suspension components or between wheels and the road surface.

The indirect method of the suspension testing is proposed and discussed in the paper. The heavy vehicle was driven with different loads and at different speeds (16, 24 and 32 km/h) on the selected, testing track including a standard speed-bump (Fig. 1). The shape and dimension of the speed bump allowed for obtaining the measurable range of vibration for representative points. The vehicle velocities were the most common among the tests described in the literature for such vehicles [7-9]. Moreover, lower velocity ensured the driver safety and preserved testing equipment and sensors.

Fig. 1. A speed bump used for the suspension tests

Deflections of the suspension components and accelerations in the selected points were recorded during the tests. Displacement gauges used during the tests were attached to the shock absorbers, as presented in Fig. 2. The accelerometers with a higher "g" range were glued to the axles, whereas ones with a lower range were attached to the frame and load deck of the trailer.

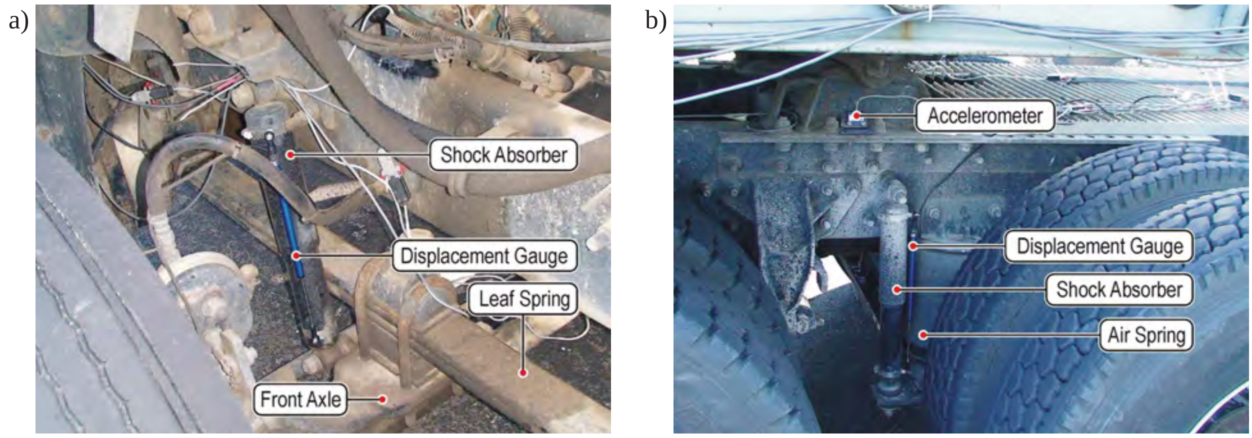

Fig. 2. Location of the displacement gauges and the accelerometers for the truck tractor: front steer axle (a), forward drive tandem axle (b)

3. FE model of the heavy vehicle

The truck tractor (MACK CH613 Series), with a three axle single drop lowboy trailer as shown in Fig. 3a, was selected for this study. A complete FE model of the tractor and the trailer (Fig. 3b) consists of over 25,000 finite elements. The HyperMesh Software from Altair was used as a pre-processor for modelling. In-situ measurements and data available on the manufacturers' websites were used for developing the complete FE model.

Fig. 3. The heavy vehicle selected for the tests (a) and its FE model (b)

Development of the suspension system and identification of its major components are presented in Fig. 4. Each suspension consists of the rotating horizontal axles located along corresponding non-rotating axles which are connected to the frame through vertical cylindrical joints. All axles are modelled as rigid bodies using beam elements. The rotation of an axle is implemented by a constraint option, referred to as a revolute joint [10]. The revolute joints were also used in the "fifth" wheel connection. They allow for the relative movement of the trailer in the vertical plane. The FE model was restricted to the vertical direction while no sideway motion of the trailer was allowed. Hence, the fifth wheel and the skid plate of the trailer were rigidly connected.

The vertical movement of an axle set is achieved by using the cylindrical joints and the special purpose massless discrete elements which simulate springs and shock absorbers (Fig. 5).

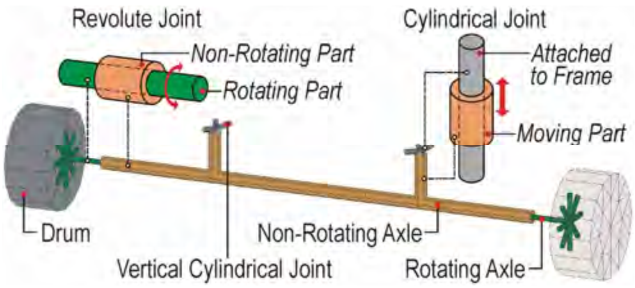

Fig. 4. Scheme of constrained joints applied in the FE model of the front suspension system

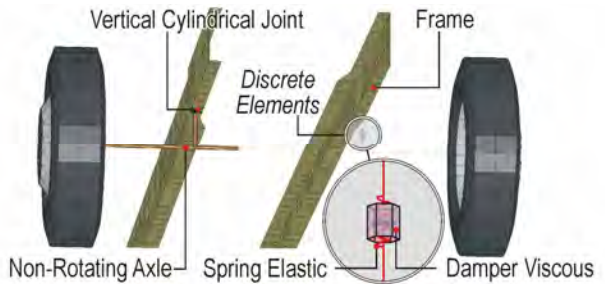

Fig. 5. Discrete elements in the FE model of the front suspension system

The FE model of each suspension system includes the axle, two springs, and two dampers. Other components of the actual suspension system such as the frame hangers, bushings, height control system, upper shock brackets and axle attachment hardware were lumped together as a sprung mass connected with the bodywork. The total weight of each suspension system was estimated based on the available data. The mass of entire axle was distributed between rotating and non-rotating parts represented by beam elements in the FE model.

The correct representation of wheels and tires is an important issue in modelling of the vehicle dynamic behaviour. Currently, the computational techniques allow modelling the wheels as rotating, pneumatic, three dimensional objects interacting with the road surface through a well defined contact algorithm. The burden of complexity of such an approach is addressed by the FE code used for the simulation. Such approach is actually easier and more reliable as it requires fewer simplifying assumptions. In the presented FE model, two types of shell elements were used for the wheel models – 3-node elements for the discs and 4-node elements for the rims and tires (Fig. 6). A simple pressure volume airbag option [10] was used for the FE pneumatic models of the tires. The values of pressure inside the airbags were set up according to the data provided by the tire manufacturer and can be easily adjusted in the FE model. The FE model of the tire consists of the sidewalls and the tread parts. Each of these components includes two coincident layers of 4-node shell elements. The first layer represents rubber-like material with average properties for rubber, whereas the second one (representing the cord) uses a material model for fabrics, with stiffness for tension only. The dimensions of the wheel FE were based on the data available from manufacturer websites; however some of them were modified in order to obtain masses similar to the actual wheels.

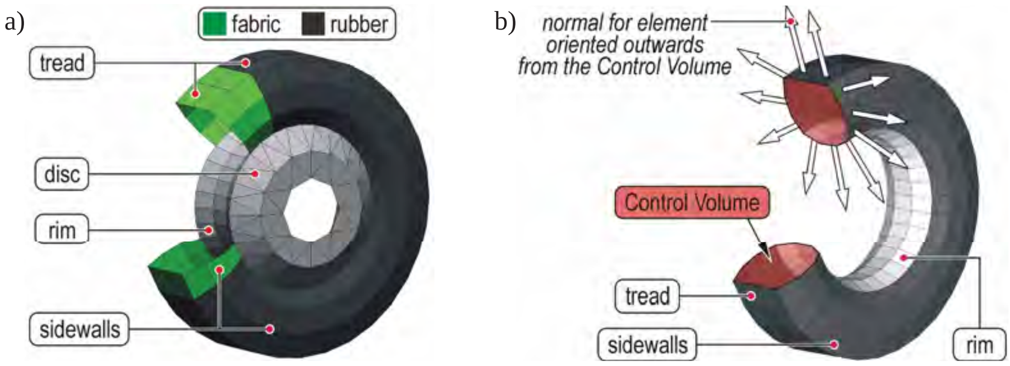

Fig. 6. FE model of the wheel (a) and components used in the airbag model (b)

As mentioned at the beginning, the main aim of this work was to determine the characteristics of the suspension systems of the selected heavy vehicle. Therefore, the complete FE model of the tractor and trailer was simplified, as detailed modelling of each component was not necessary to achieve the goal of this project. The simplifications included the bodywork where the driver cab, hood, fuel tanks, and engine were modelled as rigid bodies. Moreover, the FE models of the drive and steering systems were also simplified and they do not include transmission, transmission shafts and a few other minor components. These simplifications not only helped in reducing the size of the FE model and the CPU time needed but they also made all the calculations more stable and reliable.

4. Validation and calibration of the heavy vehicle FE model

Validation is the process of determining the degree to which a model is an accurate representation of the real world from the perspective of the intended uses of the model [11]. Calibration is applied when experimental data is used to modify a model and in this way to improve its predictive capabilities.

Two criteria were used for validation of the vehicle FE model. Validation began with checking the mass distribution in FE models on the basis of axle loads. The results obtained for FE models were compared with the values taken from measurements of the actual vehicle (Tab. 1). During the FE analysis, the model was dropped on planar rigid walls located under each axle. Such virtual tests allow assessing the normal load. Global damping with high damping factor was used to damp vibrations of the FE model in the initial phase.

Tab. 1. Comparison of the axle loads from FE analysis and measurements for the unloaded tractor-trailer

Axle No.	Axle Type	Axle Load (kN)		Relative Error (%)
		Measurements	FE Model	
1	Front Steer Axle	38.802	38.093	–1.83
2	Forward Tandem Drive Axle	26.253	26.373	0.46
3	Rear Tandem Drive Axle	29.546	30.319	2.61
4	First Trailer Axle	29.635	29.516	–0.40
5	Second Trailer Axle	33.818	33.866	0.14
6	Third Trailer Axle	38.713	38.633	–0.21
TOTAL		196.767	196.800	0.02

Since the FE model was lighter than the actual object due to above-mentioned simplification, its calculated mass had to be increased by changing densities of some materials and applying mass nodes in several points of the bodywork. No additional changes in the FE models were performed for the axles and for the wheels. The modifications allowed keeping an appropriate ratio between sprung and unsprung mass of the vehicles.

The second step of the FE model validation was to determine the spring stiffness and damping coefficient for each suspension system. Experimental suspension tests were simulated using the LS-DYNA code. This analysis allowed for validation of each suspension system and the complete FE model. The vehicle FE model was virtually driven over the modelled speed bump. The velocities of the vehicle FE model were compared with the speeds of actual objects during the tests. The vertical accelerations of the selected nodes and the change in length of the discrete spring elements were recorded as a function of time and compared with the time histories obtained from the experimental tests. The spring and damping coefficients at all axles were calibrated until the performance of the FE model closely matched that of the actual vehicle.

Comparison of the selected results obtained from the experimental tests and numerical analysis are presented in Fig. 7 and 8 – for the front suspension, in Fig. 9 – for the rear suspension system of the truck tractor. Selected results for the trailer suspension system are presented in Fig. 10.

A determinant used in this matching process was a correlation coefficient between two considered variables – experimental and numerical ones. It can be defined as follow:

$$r_{xy} = \frac{\Sigma(x-\overline{x})(y-\overline{y})}{\sqrt{\Sigma(x-\overline{x})^2\Sigma(y-\overline{y})^2}}, \tag{1}$$

where: x, y are variables, and $\overline{x}, \overline{y}$ are means of x and y. If correlation coefficient encloses between 0.5 to 1.0, it can be assumed that correlation between two considered variables is large [12].

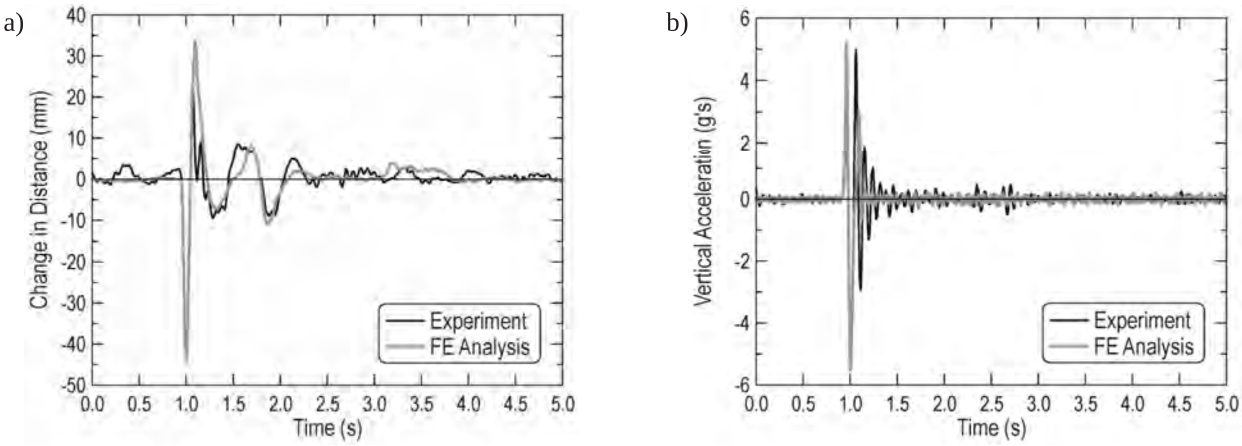

Fig. 7. Comparison between time histories obtained from the experimental tests and FE analysis for the front axle of the truck tractor (velocity of 24 km/h): change in distance between axle and frame (a), vertical acceleration of the axle (b)

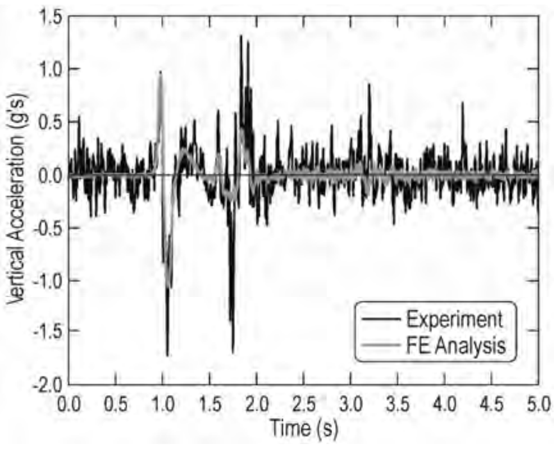

Fig. 8. Comparison between time histories of vertical acceleration obtained from the experimental tests and FE analysis for points located on the frame above the front axle of the truck tractor (velocity of 24 km/h)

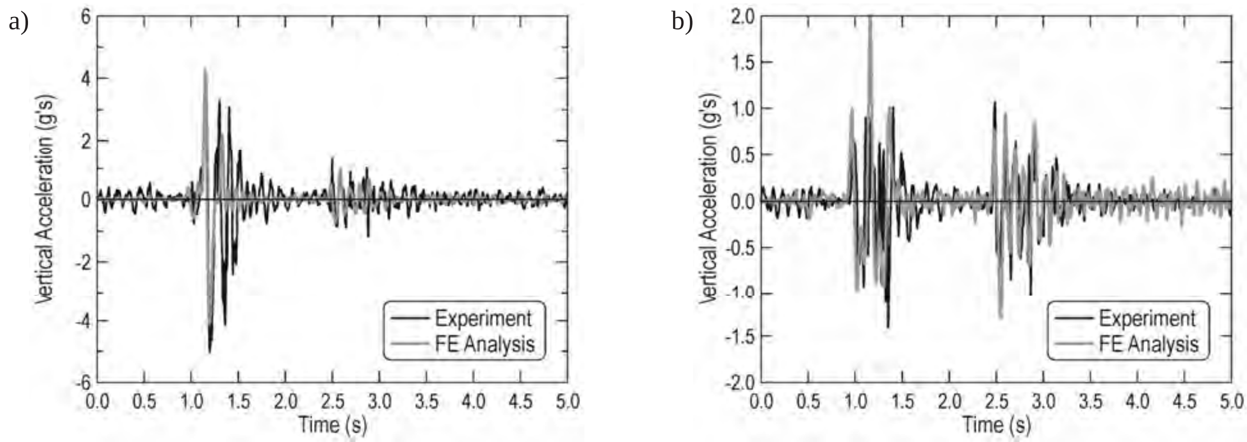

Fig. 9. Comparison between time histories of the vertical acceleration obtained from the experimental tests and FE analysis for the rear tandem axles of the truck tractor (velocity of 24 km/h): rear tandem axle (a), point located on the frame above the rear tandem axles close to the fifth wheel (b)

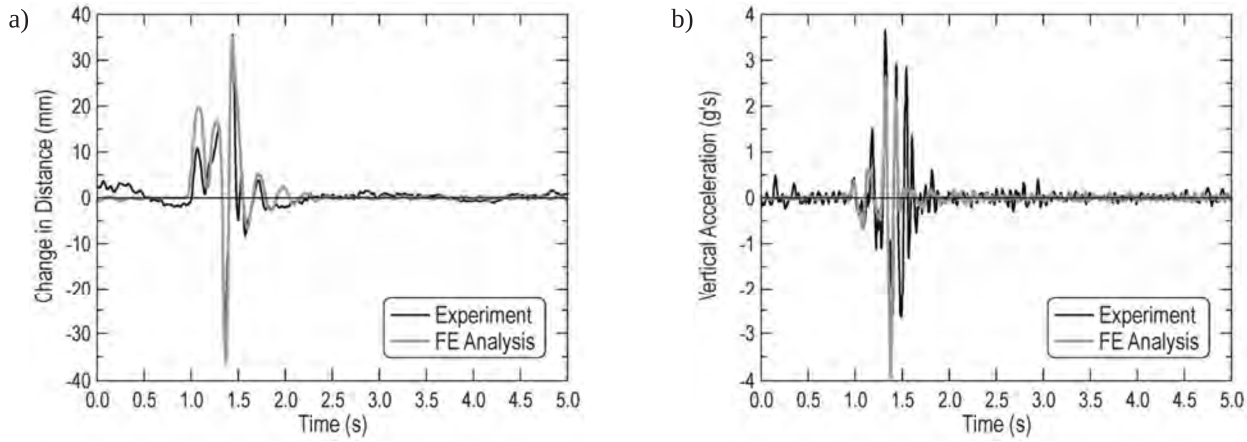

Fig. 10. Comparison between time histories obtained from the experimental tests and FE analysis for the third trailer axle (velocity of 24 km/h): change in distance between axle and the load deck (a), vertical acceleration of the axle (b)

The values of the spring stiffness and damping coefficients for each suspension system are provided in Tab. 2.

Tab. 2. The spring stiffness and damping coefficients for each suspension system in the FE model

Suspension System		Spring Stiffness k (kN·m^{-1})	Damping Coefficient c (kN·s·m^{-1})
Truck Tractor	Front	280	17
	Rear	305	22
Trailer		875	40

5. Conclusions

The most satisfying correlation of the result between FE analysis and the experimental test was obtained for the front suspension system. The only concern was a high range of noise recorded by the accelerometers located on the frame due to close proximity of the running engine. As depicted in Fig. 7a, the shapes of both curves – experimental and numerical one – are similar, however some minor differences are also visible. They could be the result of simplification of the suspension FE model, which did not include any buffers. A practical stroke of the front suspension in the actual vehicle was bounded by additional rubber buffers during the compression of the leaf spring but also limited while the spring was expanded.

Slightly worse correlation was obtained for the rear suspension system. It is worth to mention that the selected tractor was equipped with the air springs in the rear suspension system allowing for adjusting its height depending on an actual axle load. Such suspensions are more complex for modeling due to their non-linear responses. However, in a short operating range they could be modelled as linear springs.

The results obtained for the trailer suspension were characterized by large correlation coefficient. The biggest challenge was to estimate an appropriate value of the damping coefficient. The selected trailer was equipped with very stiff leaf springs without any dampers. However, additional discrete damping elements had to be applied in the FE model to reduce a high range of vibrations generated during driving over the speed bump.

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