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DETERMINATION OF MEASUREMENT PARAMETERS FOR VIBRATION ANALYSIS BY BUS

SZALAI Péter[1], HAJDU Flóra[2*], HORVÁTH Péter[3], PAPP Csenge[4], APAGYI Antal[5], BRINISSAT Marame[6], KUTI Rajmund[7]

[1] Department of Machine Design, Faculty of Mechanical Engineering, Informatics and Electrical Engineering, Széchenyi István University, Egyetem tér 1, H-9026 Győr, Hungary, e – mail: szalai@sze.hu
[2] Department of Machine Design, Faculty of Mechanical Engineering, Informatics and Electrical Engineering, Széchenyi István University, Egyetem tér 1, H-9026 Győr, Hungary, e – mail: hajdfl@sze.hu
[3] Department of Applied Mechanics, Faculty of Mechanical Engineering, Informatics and Electrical Engineering, Széchenyi István University, Egyetem tér 1, H-9026 Győr, Hungary. e – mail: horvpe@sze.hu
[4] Department of Machine Design, Faculty of Mechanical Engineering, Informatics and Electrical Engineering, Széchenyi István University, Egyetem tér 1, H-9026 Győr, Hungary, e – mail: papp.csenge@sze.hu
[5] Széchenyi István University, Doctoral School of Multidisciplinary Engineering Sciences, H-9026 Győr, Hungary, e – mail: apagyi.antal@ga.sze.hu
[6] Széchenyi István University, Doctoral School of Multidisciplinary Engineering Sciences, H-9026 Győr, Egyetem tér 1, H-9026 Győr, Hungary, e – mail: brinissat.marame@sze.hu
[7] Department of Machine Design, Faculty of Mechanical Engineering, Informatics and Electrical Engineering, Széchenyi István University, Egyetem tér 1, H-9026 Győr, Hungary, e – mail: kuti.rajmund@sze.hu

Abstract: From the point of view of field examinations, especially in the case of measurements done by instruments, it is crucially important to determine measurement conditions. Accurate identification of parameters enables accurate and certified measurement results to be achieved. In our work, we represent the determination of measurement conditions for harmful vibrations occurring during urban road transport of an Ikarus 55 type long-haul bus. This is a favourable basis for further vibration analysis regarding other buses.

KEYWORDS: urban road measurements, bus, parameter identification, harmful vibrations

1 Introduction

Vehicles with special structures, like buses, are built specifically for passenger transport. Long-distance passenger transport is carried out by long-haul buses, thus their structure and interior are designed to satisfy comfort as well. These vehicles have to withstand different types of harmful effects due to load, which should be examined continuously with strict regard to operational safety factors [1]. Such undesired effects are harmful vibrations, which can, on one hand, cause health damage to passengers, and on the other hand, damage the structural elements of vehicles [2-3]. With the available literature in connection with the topic, and with their analysis, we assessed, that similar measurements, examinations had been carried out mostly in laboratory conditions or some cases on test tracks. Field examinations had been done in a few cases and obviously not with the type in our work [4-9]. Field examinations are realized in a wide spectrum. The progress of examination, data analysis, and simulation results are highly influenced by pre-given examination parameters. In our work, we represent the parameter determination of the occurring vibrations in the Ikarus 55 type bus made available for us. Our article is structured as follows: firstly, we overview the theory of parameter identification and the chosen method, which is followed by the description of the measurement. In the next chapter, we represent a few methods to identify the parameters for suspensions in built-in

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vehicles. Finally, other investigation tasks are represented. With our research results, we wish to contribute to increasing the safety of bus operations.

2 Parameter identification methods

It is not an easy task to determine the measurement parameters, because the proper method should be determined in correspondence of the task to be carried out and to provide the necessary conditions for its application. Examinations and measurements can be extremely complex, thus several methods exist for parameter identification.

Plenty of methods, used in practice, are based on statistics. In the course of literature research, we assessed that statistics-based parameter identification had earlier been used to determine vehicle diagnostic parameters of buses (Maximum Likelihood approximation). Furthermore, it was also applied for vibration analysis in the cabin of heavy transport vehicles [1]. Literatures [10] and [11] describe a Maximum Likelihood-based approximation to identify the parameters of a bus according to ARMA and ARMAX models. Another publication [12] represents a mathematical model to determine the stiffness and damping of a suspension of a heavy transport vehicle based on the curve fitting method. A curve fitting method was used to obtain the least square error function between the vertical acceleration power spectral density of the simulated seat and the measured power spectral density. Publication [13] represents a new algebraic approximator to determine the unsprung mass of the suspension. The publication aims to improve the comfort of passengers in a simulation environment with a sliding mode operator.

In the case of parameter identification, it is rather practical to apply the network aspect of the system, from which, the in- and output of the system can be determined, for instance, based on measurement data, while the characteristics of the different elements of the system are unknown. The seven parameters of the Besinger-model that describe the non-linear characteristics of dampers, were identified by measurements. Similar methods for parameter and model identification can be used in other engineering fields, such as magnetic hysteresis modeling [14-15], to approach nonlinear systems linearly [16] or parameter identification of flywheels [17].

Our research task is to examine the vibration excited by the road during the operation of a long-haul bus, furthermore, to determine the necessary parameters for field measurements. To solve this task, we chose the least square method parameter approximation. The advantage of this method is that it can easily be implemented and the measurement data about time is instantly usable. This method is applicable to curve fitting and further vehicle analyses. During parameter approximation measurement data are available, and the input excitement signal, the equations, and varying values of the examined system are given. An optimization task shall be solved, whose objective function derives from the minimization of the summary of the least squares:

$$minimize \sum_{i=1}^{m}(y(i) - x(i)^2). \quad (1)$$

We applied computer solutions, we chose MATLAB as software and its optimization module, the OPTI program package. The solver was NL2SOL which is an algorithm based on the adaptive nonlinear least square method. We decided to choose the above-mentioned software because according to our experiences with such tasks, it is widely used.

3 Measurement

In Hungary, especially on minor roads, lots of surface unevenness and potholes can be found. At first, we discovered several minor roads in the agglomeration of Győr and examined the

structural faults, measuring the depth of potholes. The deepest pothole examined by us was 10 cm deep. At the same time, we measured the height of trim-stone of pavements, which was also 10 cm. Taking this into account, we decided to drive through a 10 cm high curbside with the bus, and in the meanwhile, we measured the occurring vibrations. Before introducing the measurement itself, we consider the introduction of technical specifications of Ikarus 55 as an inevitable part. The examined bus was manufactured by Ikarus Car-Body and Vehicle Factory in Budapest in 1957. This type was the first, that was built with a self-supporting chassis. This bus has dependent suspension on both front and rear, it is rear-wheel driven and its wheel configuration is 4x2. Springs are leaf springs on both suspensions, on the rear, there are 15 sheets of springs to withstand the permanent load, on the front the number of springs is 10. The engine of the bus and the transmission are located at the back of it, behind the rear suspension. To adapt to these, the rear has a special design, it is visible in the next figure.

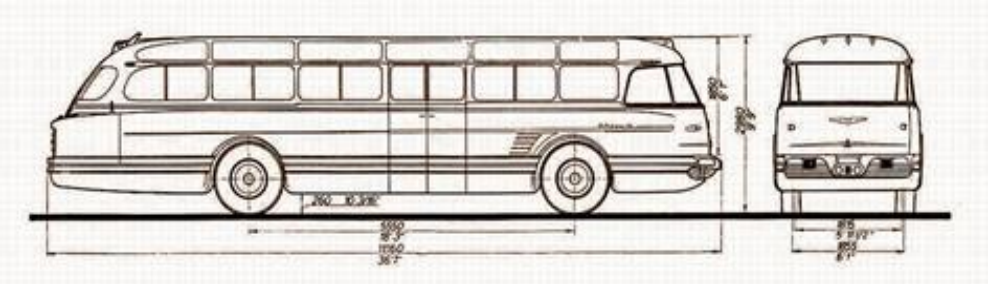


Fig. 1 Section of Ikarus 55 type bus (Source: Volánbusz Ltd.)

On both sides, lever arm shock absorbers provide the proper road holding. The drive is a Csepel D-613 type engine which is a pre-combustion chamber suction diesel, the 150-liter fuel tank is placed on the rear right side of the vehicle. The battery holder is fixed on the left side of the vehicle. It has a 5+1 ATRA type synchronous transmission, its highest speed is 78 km/h according to the catalog. The steering structure has no servo assistance. The deadweight of the bus, with filled-in liquids, is 9500 kg. The front suspension frame weights 500 kg altogether with the tires. The weight of the rear suspension frame with tires is 1220 kg. The vehicle is bottom-heavy in the base position, the division of mass is 2/3 to the rear axle and 1/3 to the front axle. By the boarding of passengers, the mass division gets even.

Before the measurement, the vibration sensors were placed on the suspension of the bus on the rear and front, on both sides, and in the passenger compartment. In the cabin, and on the structure, we have put sensors in the middle. This can be observed in the next figure.

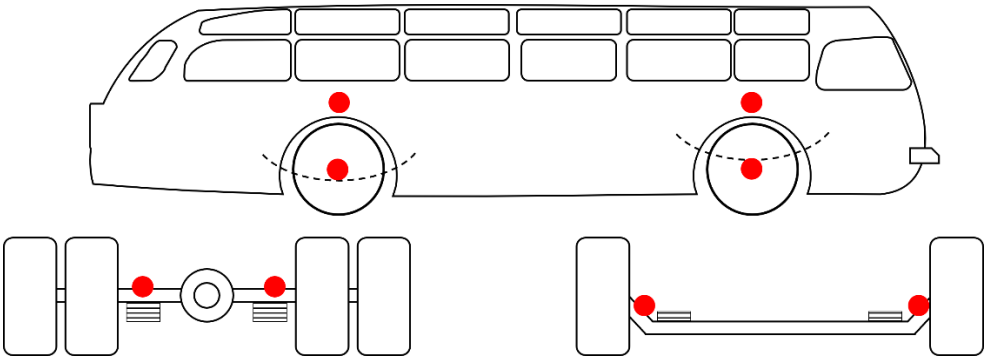


Fig. 2 Positioning of the vibration sensors on the vehicle (authors property)

Applied measuring apparatus: Brüel & Kjaer 3560-C Frontend, Accelerometer sensors, Pulse Labshop 18, and Reflex Core software.

3.1 Measurement process

Measurement was carried out on a solid covered, free from potholes, straight road track in the parking lot of the university. After fixing the measuring instruments on the above-mentioned structural elements of the bus, we parked in front of the curb, on which the bus had to be driven at an even speed. The running speed is defined as 5 km/h. After having the instruments in operation, the bus started and with the given speed run through the curb with both axes. The results were recorded and the measurements were repeated. Both measurements were valid. Measurement data were recorded in the case of the second experiment as well, we compared it with the first one, and validation was done. Results can be observed in the following figures.

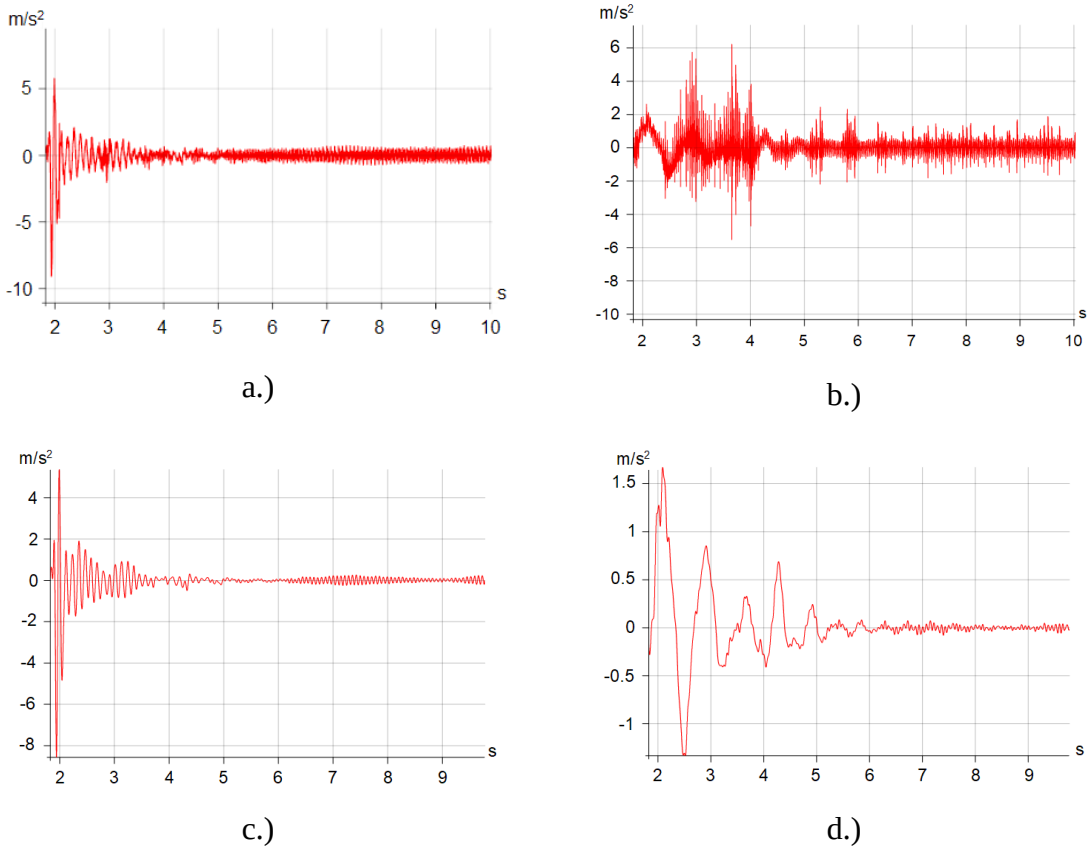


Fig. 3 Raw a.) – b.) and filtered c.) - d.) measurement data (authors property)

Measured signs had noise, so that, before identification, disturbance with no connection to the measurement had to be filtered. The noise filter was done by the Pulse Reflex Core program. The first step was to determine the frequency of the noise with Fourier transformation, after that, a low-pass filter was applied to filter unnecessary noises. Using the results remaining after noise filtering, we started to prepare a computer simulation. Filtered signs are represented in the next figure.

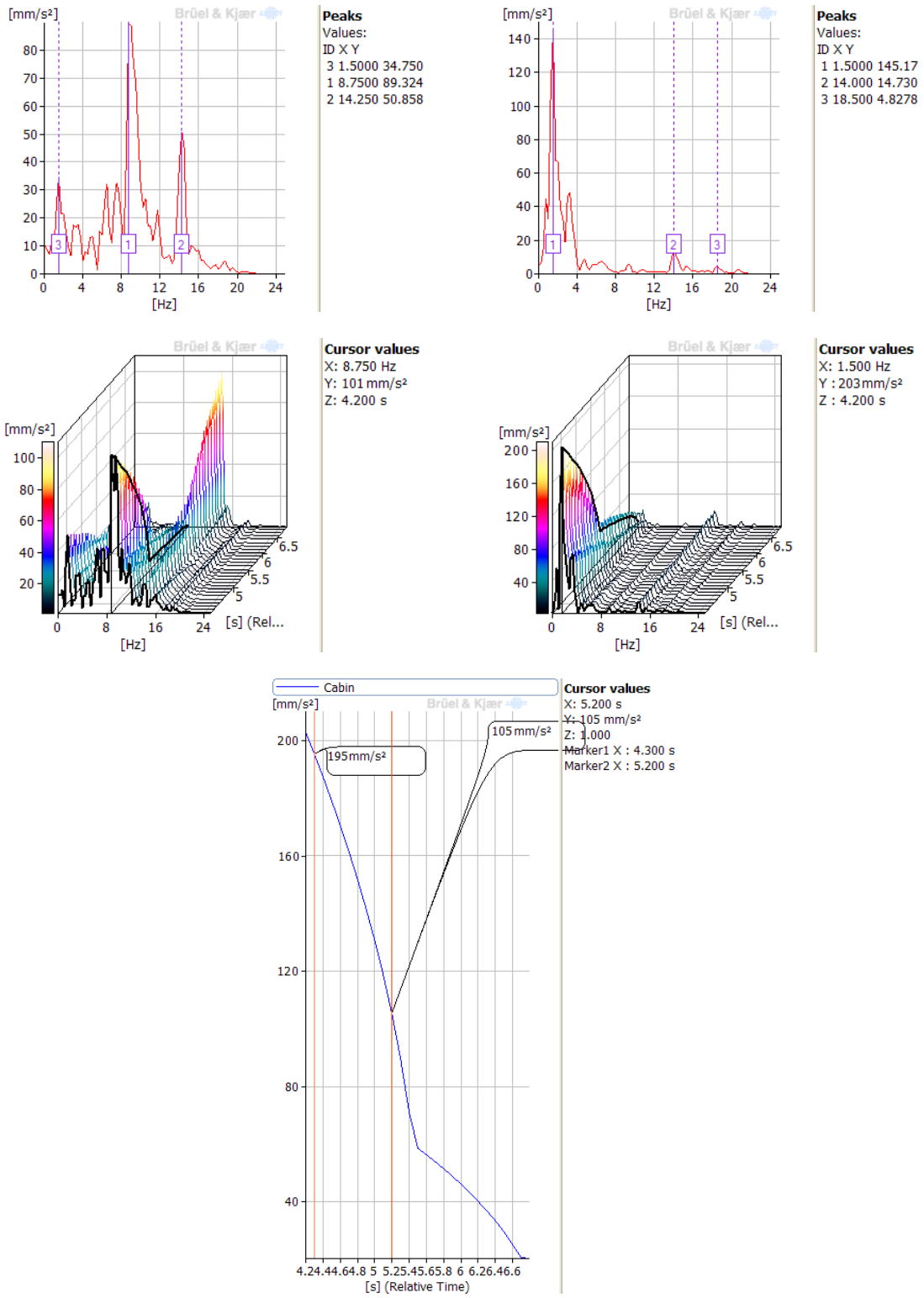


Fig. 4 Diagrams with filtered signs (authors property)

By analyzing the figure above, we can state, that the vibration spectrum is damped.

4. Parameter estimation of a bus

4.1 1st method: analytic calculations

For the following conducted calculations, we considered the technical parameters of the bus, according to beam theory the following formulas were used in our calculations [18].

$$k = \frac{8Enwt^3}{3l^3} \tag{2}$$

$$k_f = 113983\ N/m \tag{3}$$

$$k_r = 170974\ N/m, \tag{4}$$

where k_f is the front spring stiffness and k_f is the rear spring stiffness. Damping factors are:

$$\beta = c/c_c, \tag{5}$$

where the critical damping ratio is:

$$c_c = 2\sqrt{mk}. \tag{6}$$

Mass was divided into 1/3 and 2/3 ratios:

- front part: 3236.67 kg
- rear part: 6473.33 kg

The damping ratio was calculated based on measurement in the passenger compartment, which was 34.145%. From the damping ratio, the damping coefficients could be calculated:

$$c_f = 2366.36\frac{Ns}{m}, \tag{7}$$

$$c_r = 16394.6\ \frac{Ns}{m}. \tag{8}$$

4.2 2nd method: Determination of spring stiffness by finite element method

The stiffness of the leaf spring was determined also using the finite element method. For this purpose, the widely used ANSYS software was applied.

The first step was to select the mechanical model. In doing so, two simplifications were used. First, since the loads and supports are both in the same plane, the problem can be considered as a 2D generalized plane stress problem, where the thickness is not modeled but is taken into account as an additional parameter during the integration afterward when the stiffness matrix is calculated.

Another simplification is to utilize symmetry. Since the geometry is symmetric, it is advisable to model only half of the geometry rather than the whole geometry, thus significantly reducing the computational effort required. Of course, in order to utilize symmetry, the supports must also be symmetric. In reality, this is not fully achieved, as the leaf spring is connected to the chassis by a pin at one end and by a lever at the other end. However, this problem can easily be solved by slightly changing the boundary conditions, which do not affect the final result. The boundary conditions used are shown in Figure 5. Using the symmetry, only the left side of the spring was modeled, restraining the displacement in the x-direction, while allowing the displacement in the y-direction of the right-hand edges of the leaves (illustrated by the roller supports placed along with section BC). The nodes on the inner surface of the master leaf's eye were rigidly connected to node A in the center of the eye, giving it three degrees of freedom, with y-directional displacement restrained and free x-directional displacement and rotation about axis z.

To determine the stiffness, unit displacements were applied to all nodes of the BC section, so the model contained only kinematic boundary conditions.

Since the individual leaves are in contact and may slip on each other, it is important to define an appropriate contact between the leaves. For this purpose, we used the ANSYS Workbench's so-called "No Separation" contact. In this case, once the contact is detected, the target and contact surfaces are tied up for the rest of the analysis. The slide is possible, but the nodes in contact are bonded to the target surface in a normal direction.

Taking advantage of the relative simplicity of the geometry, we used a structured mesh with 8 node quadrilateral elements, with 2 rows of elements per leaf. Convergence testing shows that this size is sufficient to obtain an accurate result, i.e. reducing the element size does not significantly affect the final result.

For the determination of the stiffness, the reaction force corresponding to the kinematic load applied on the BC section was retrieved from the program (Figure 6), the numerical value of which is exactly equal to the half of the spring stiffness, due to the half geometry.

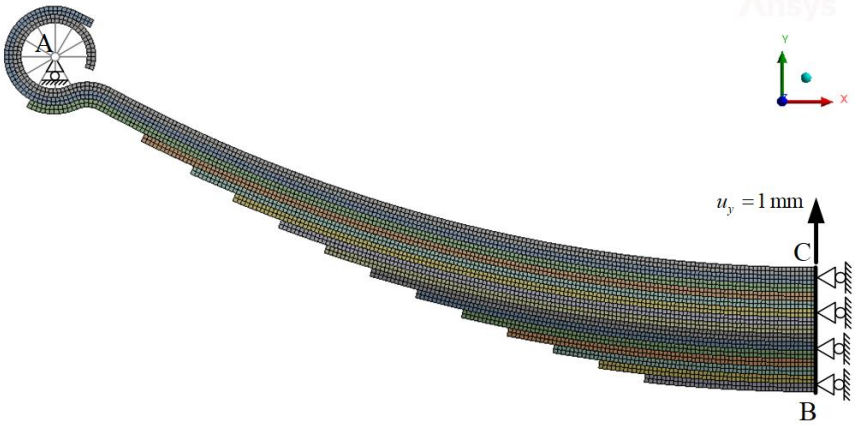


Fig. 5 Mechanical model of the rear leaf spring

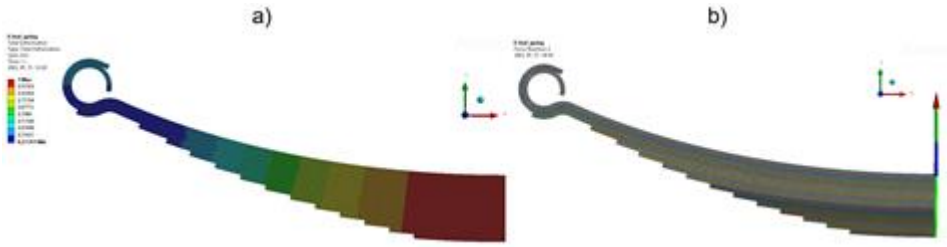


Fig. 6 a) displacement field of the rear leaf spring b) the observed reaction force belonging to the unit displacement

For the spring stiffnesses determined as detailed above, the following values were obtained:

- front leaf springs: $k_f = 134.4$ N/mm
- rear leaf springs: $k_r = 243.1$ N/mm

4.3 3rd method: quarter vehicle model

To check the parameters coming from analytic and finite element methods and for fine-tuning, we also created the quarter vehicle model of the bus (Figure 7) [19]. With the help of the model, we were able to form the acceleration-time functions, that can be compared with the functions coming from the measurements.

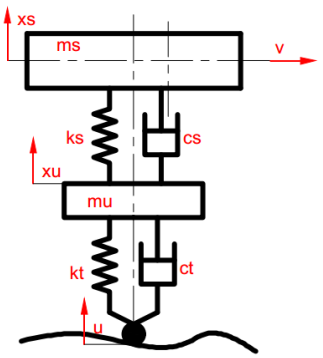


Fig. 7 Quarter model of the vehicle (authors compilation)

By the application of the quarter vehicle model, the following formulas were used:

$$m_s\ddot{x}_s = c_s(\dot{x}_u - \dot{x}_s) + k_s(x_u - x_s)\,, \tag{9}$$

$$m_u\ddot{x}_u = -c_s(\dot{x}_u - \dot{x}_s) - k_s(x_u - x_s) + k_t(u - x_u) + c_t(\dot{u} - \dot{x}_u). \tag{10}$$

The input obstacle was modeled as the function of the front suspension when it drove through the curb (Figure 8).

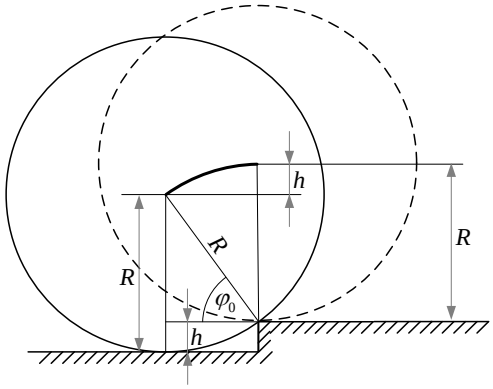


Fig. 8 The input obstacle model

According to Figure 8, the function of road excitement is written as follows:

$$u = R\,sin(\varphi_0 + (v/R)t) - (R - h), \tag{11}$$

where

$$\varphi_0 = asin\left(\frac{R-h}{R}\right). \tag{12}$$

During the identification, we used up different models available in the research field. In the first approach, we used a linear model. We used the analytic values for the damping coefficient, while for spring stiffness we used the results coming from the finite element method. In the case of tires, we used data from the literature, according to this, the stiffness is106 N/m, while the damping is 150 N/(m/s) [20]. According to simulation results, the damping effect is not significant, by its omission, the acceleration values are not significantly modified. Considering stiffness, using the values taken from the literature, the acceleration values were higher than the measured values, so the stiffness of the tire was modified to 800000 N/m. The acceleration values (Figure 9) from the linear model are similar to the results of the measurement. However, their time running is slightly different, which gave the reason to conduct a nonlinear model. In the case of linear and nonlinear models, the characteristics of applied spring and damping are

summarized in Figure 10. Nonlinear-originated acceleration-time functions can be found in Figure 11.

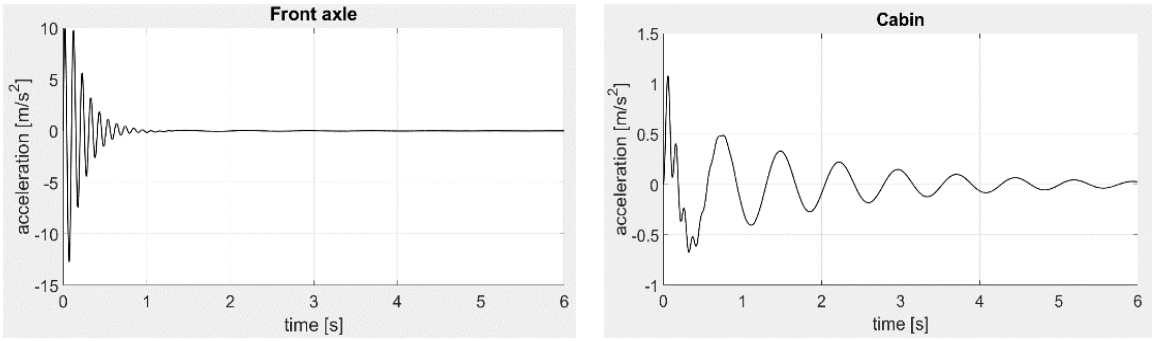


Fig. 9 Acceleration-time function of front axle and the cabin based on linear model

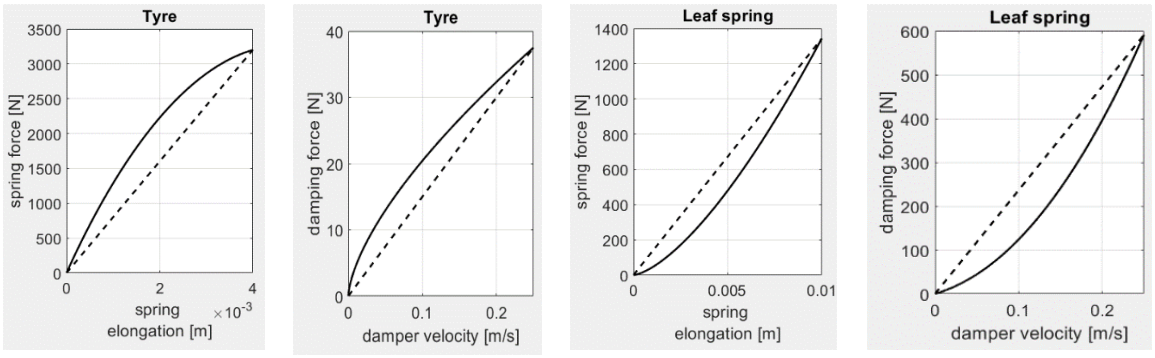


Fig. 10 Spring and damping characteristics

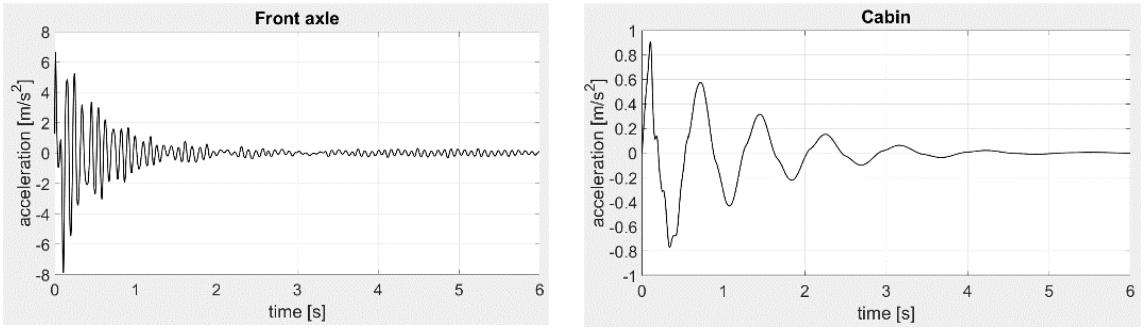


Fig. 11 Acceleration-time function of front axle and the cabin based on nonlinear model

5. Summary of results

Based on the quarter vehicle model, we can state that the analytically determined leaf spring damping, the finite element method determined leaf spring stiffness, and the literature-based tire stiffness/damping are a quite applicable basis. In the case of tires, the stiffness value coming from the literature, was too high compared with our experience, so we decreased it in our simulation.

The first, linear model provided a slightly higher maximum acceleration value for the axle and a slightly lower maximum acceleration value for the cabin than the measured value, and the variation in time was also different from the measured data.

Regarding the nonlinear model, we could state, that the best fitting is ensured by progressive characteristics leaf spring and degressive characteristics tires. The acceleration values of the front axis show strong similarities with the measured values.

In cases of both the measurement and the nonlinear model, 8 complete oscillations could be observed in the second after the excitation. The maximum acceleration value is also about 8 m/s^2. In the case of the structure, the simulation yields a lower maximum acceleration value and, in the measurements, a permanent sinusoid sign was experienced. This is assumed to originate from the excitation force of the engine.

CONCLUSION

By comparing the results of the quarter vehicle model with the measured results, we can conclude that we have obtained a good approximation of the vehicle stiffness/damping parameters. By comparing the methods, we can conclude that the analytic solution and finite element method have great advantages, for example, there is no need for expensive and time-consuming measurements. Geometric data from the catalog is only needed. As we have seen, the finite element model provided slightly higher stiffness values that are closer to reality, so - although the creation of a finite element model is more complicated than using analytical formulas - we would consider using this approach in future works. For the stiffness of the tire, we found that in our case the stiffness is slightly lower than the literature value. From nonlinear models, it can be stated, that the leaf spring has progressive, while the tire has degressive stiffness and damping characteristics. However, we should note, that these are approach models, in which, certain external effects are omitted. As we analyzed the measurement data, it turned out, that an unanticipated sinusoid effect occurred. This is an important conclusion since the consequences of this could not have theoretically been detected. Attention should be paid to the determination of eigenfrequencies as well. It was concluded that field measurements are essential for parameter identification in order to create accurate models and simulations that can be used to achieve practical results. Learning from the experience gained in the current study, we plan to carry out more precise measurements for parameter identification in the future.

Our research work, along with the represented results, promoted the aim of setting further research work. Further tasks, that are planned to be carried out in our next research: parameter identification in the determined frequency range, to ensure efficiency in other methods implementation, and the validation of the model based on measurement data.

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