Modal Analysis of the Vehicle Model

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Abstract The article deals with the numerical solution of modal analysis of a simple model. It is a system of rigid bodies resiliently mounted and bound. The solution was done in the Ansys simulation program. The article describes how to build the program. Further, some of the results of the actual frequencies and shapes of the symmetrically loaded system are shown. The results served to refine the mathematical model that solves the vertical oscillation of the symmetrically or asymmetrically loaded model with different kinematic excitation. The numerical solution of vehicle model vibration was done in MSC Adams. The results of the vertical vibration measurement of the vehicle model are also given in the article. After adjusting the boundary conditions of the numerical solution, good agreement between experimental and numerical solution (more than 90%) was achieved.

Keywords Vehicle · Experiment · Modal analysis · Vibration

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1 Introduction

Modal analysis is most often performed to determine modal parameters (own frequencies, own shapes or modal damping) without reference to the theoretical model. Finding modal parameters to compare experimentally obtained data with corresponding data obtained using FEM or other theoretical methods. The aim here is to verify the theoretical model, which serves to predict the dynamic behaviour of the model system—the vertical vibration of the vehicle model.

Oscillation is a phenomenon that can be used in many technical applications. However, it can also be an uncomfortable compilation, which can lead to equipment malfunctions as well as environmental degradation. In order to exploit the favorable effects of oscillation and to suppress it adversely, we have to understand the problem of oscillation. It is not easy to master oscillation. Explanation of a number of intuitively unexpected behaviours of oscillating systems often requires relatively demanding mathematical calculations. Some of these are analytically manageable; others require a numerical approach [1, 2].

Solution of oscillation of a spatially resilient housing with consideration of varied influences can be applied in various areas of technical work, e.g. in the elastic storage of machines and machinery [3–5], in the suppression of the influence of unacceptable vibrations and impacts, the transport of special consignments, the oscillation of production machines, Foundations (forming and machine tools, production lines, etc.), in the investigation of oscillation of sprung parts of rail and road vehicles, oscillation of chassis, etc. [3, 6].

The mechanical system consists of elements interconnected. Elements can be carriers of kinetic energy (mass) or potential (springs). These elements have the ability to change mechanical energy in heat (shock absorbers, energy dissipaters). If each element of the system has only one of these functions, we speak of a system of simple (discrete) elements. Otherwise the system may be composed of elements, each of which has simultaneously two or three functions (acts as a mass, a spring and damper). It can be said that real bodies always have all the properties and that simple elements are mere models having the property that is most manifest in the real body [7].

When analyzing the effect of the asymmetry on the vertical oscillation of the body array, it is necessary to distinguish three basic cases of asymmetry with respect to the geometric symmetry axes. These axes are determined by two mutually perpendicular axes of symmetry of the gauge and the wheelbase of the vehicle and intersecting in the geometric centre of the mechanical system

– unbalance of mass distribution with respect to geometric symmetry axes, centre of gravity position, directions of main centre axes of inertia
– the asymmetry of the geometry of the distribution of the elastic and dissipative elements of the joints of the individual bodies of the system and their mechanical properties, the stiffness of the springs, the intensity of the viscous damping, provided the linear bonds of the individual variables and the small displacements and the turning of the parts of the system

– asymmetry of kinematic excitation, for example in road or rail vehicles, the unevenness field of the road surface, Tracks defining kinematic excitation of the system at the wheel-road contact point.

These kinds of asymmetries may exist separately or together. For real objects (such as road or rail vehicles, storage of machines, etc.), the third case occurs most often.

Nowadays simulation programs make it easier for us to work and it is advisable to use them to predict the behavior of mechanical systems or structures when they are loaded. Once the numerical model has been assembled, the results obtained must be verified (e.g. with experiment, modal analysis, etc.) and in the case of nonconformity the numerical model can modify or change the boundary conditions. Almost all commercially-used FEM programs allow fast and reliable analysis of their own frequencies and their own shapes and harmonic and transient analysis with symmetric matrices.

2 Simplified Body Model

The model system (Fig. 1) consists of a steel plate, which is resiliently mounted on four screw springs. To achieve geometric asymmetry, two steel weights were placed on the base plate. These weights were placed in different combinations on the base plate. The investigations are carried out in the case of a symmetrical arrangement (using weights) and for five cases of asymmetrical arrangement (see Table 1). In an unbalanced arrangement of the weight, the centre of gravity of the system is shifted to the point (displacement e_x, e_y), where T is the centre of gravity of the system, C—the geometric centre of the plate. In the case of a symmetrical arrangement ($C = T$).

3 Modal Analysis of Mechanical System in ANSYS Program

Modal analysis was solved using ANSYS. Almost all commercial FEM programs allow fast and reliable analysis of their own frequencies and their own shapes and harmonic and transient analysis with symmetric matrices. In solving modal analysis for multiple types of plate layout, a macro (text file) was created by which the used and modified commands were applied to all types of tasks, changing the positions of individual bodies. The solution process consists of the following steps:

(1) General program settings

- Exiting all previous tasks using FINI commands and deleting all data from the database/CLE so that values are written to an empty value field

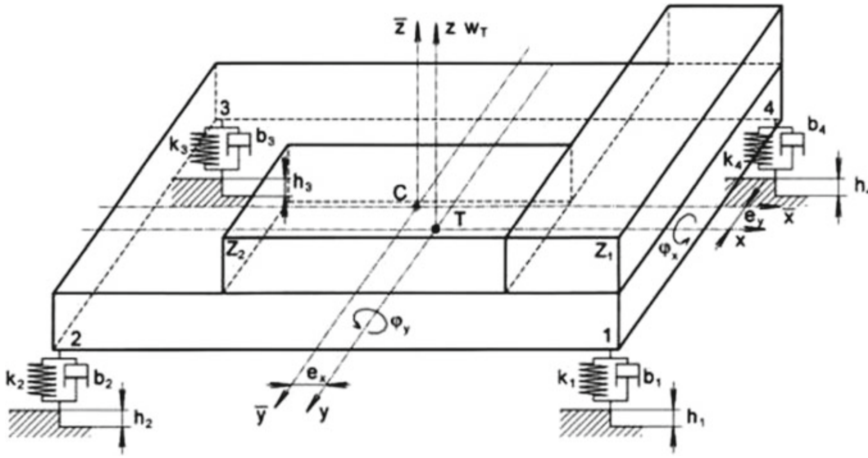


Fig. 1 Scheme of system—asymmetrical arrangement of extra weights

- Entering a file name, command/FILENAME, NAME, 1 and creating output file/OUTPUT, NAME, LOG
- Inputting the analysis type (modal) - ANTYPE, MODAL and setting the modal analysis specification by MODOPT, LANB, 12. This command sets Block Blocking Lanczos' modal analysis and the number of system frequencies we find. You can also set the frequency range to search for frequency, etc. Using the command MXPAND, 12, you set the number of custom shapes that the program renders.

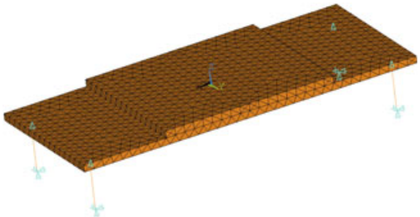
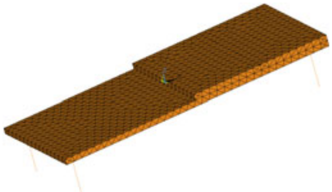
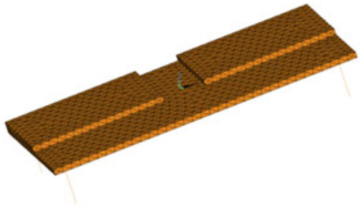
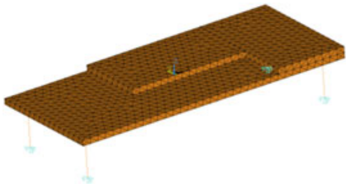
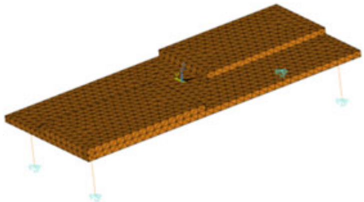
(2) Creating a geometric model

- For simplicity, the input dimensions, density and density constants were initially specified to be easily identified
- Geometry was entered through points K, POINT NUMBERS, X, Y, Z, A, NUMBER OF POINTS and volumes by command VA, NUMBER OF AREAS
- Springs were entered as rod members with a measured stiffness value.

(3) Definition of material properties

- Entry of modulus of elasticity—MP, EX, NUMBER MAT. GROUP, VALUE with a value of 2.1 × 1011 Pa
- Poisson number entry—MP, PRXY, MATERIAL GROUP NUMBER, VALUE (value was entered 0.3)
- Density input—MP, PRXY, MATERIAL GROUP NUMBER, VALUE that has been re-calculated according to the dimensions and weight of the individual plates of the experimental model.

Table 1 Variants of loading model system—ANSYS program

Variant I	
	
Variant II	Variant III
	
Variant IV	Variant V
	

(4) Definition of boundary conditions

– The boundary conditions were applied to the spring end attachment using the DK command, POINT NUMBER, REDUCED DEGREE OF CHARACTER, VALUE. All degrees of freedom have been removed, i.e., shift ux, uy, uz and rotation φx, φy, φz at all four points.

(5) Setting network parameters, network generation

– Defining element types. For the boards, the element type SOLID92 was chosen, which is a three dimensional 10-node quadrangle, and the element CONBIN14 as a 3D element (ET, NUMBER ET, ELEMENT TYPE)
– Division of geometric entities - determination of network density. Element size was entered 0.02 m by ESIZE command, 0.020 for volumes. For the springs, the number of ESIZE elements was specified, 1
– Network generation was performed first for volumes (VMESH, 1), and then elements corresponding to the springs (LMESH, NUMBER OF LOT) were

generated. Then, the stiffness of the springs (REAL, NUMBER OF CONFORMITY SUCCESS) was entered.

(6) Start solving a task

– The solution was run by the SOLVE command.

(7) Analysis of results

– The results were evaluated in postprocessing, where the respective own frequencies and shapes were obtained. For each variant of the geometric arrangement of the weights, models of the modal analysis system in ANSYS were compiled—see Table 1.

The results of modal analysis were custom shapes. They contained solid forms of their own, deformed own shapes in a given frequency spectrum, and high own shapes that may contain residual effects [2].

Custom frequencies are positive, but your own frequencies may be zero (or near zero). The zero own frequencies correspond to the self-styled shapes.

The results of the values of the frequencies of the given storage system are shown in Table 2. The first six shapes always match the solid shapes (highlighted in yellow in the table). Other shapes are already own shapes.

The following figures show some instances of custom shapes in a symmetrically arranged system. These shapes can be rendered from ANSYS even for other storage options (Figs. 2, 3, 4, 5).

Table 2 Cases distribution of load on the vehicle and the location of wedges

Variant I		Variant II		Variant III		Variant IV		Variant V	
Set	Time/Freq.	Set	Time/Freq.	Set	Time/Freq.	Set	Time/Freq.	Set	Time/Freq.
1	0.0000	1	0.0000	1	0.0000	1	0.0000	1	0.0000
2	0.0000	2	0.0000	2	0.0000	2	0.0000	2	0.0000
3	0.00014	3	0.000288	3	0.0000	3	0.0000	3	0.000738
4	0.20725	4	0.20167	4	0.19973	4	0.20123	4	0.20741
5	0.26201	5	0.26238	5	0.28770	5	0.26999	5	0.25459
6	0.37564	6	0.34989	6	0.31746	6	0.34360	6	0.34457
7	281.49	7	176.45	7	179.87	7	230.41	7	176.89
8	416.65	8	274.03	8	312.26	8	320.71	8	300.02
9	554.38	9	577.15	9	680.97	9	545.93	9	658.24
10	614.42	10	692.67	10	727.09	10	630.06	10	767.90
11	1001.5	11	936.91	11	1067.0	11	975.73	11	1056.7
12	1001.9	12	1044.4	12	1077.6	12	1131.5	12	1090.2
13	1190.2	13	1049.3	13	1222.2	13	1158.9		

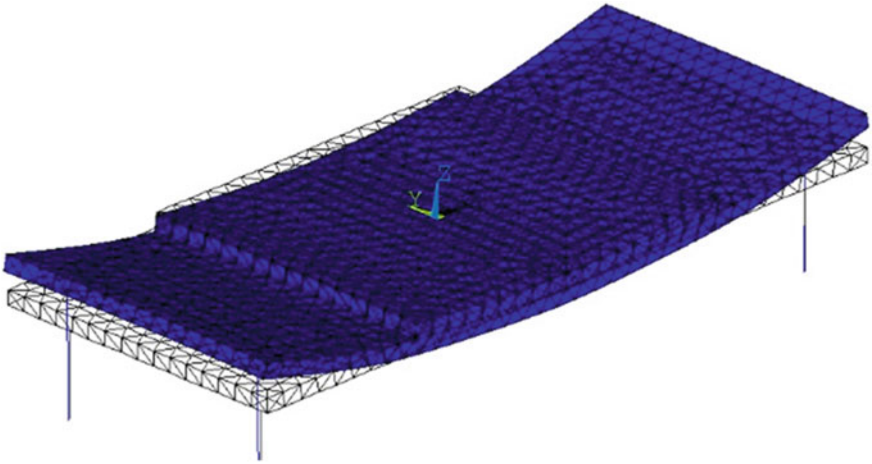


Fig. 2 The first natural modes—long symmetric bending

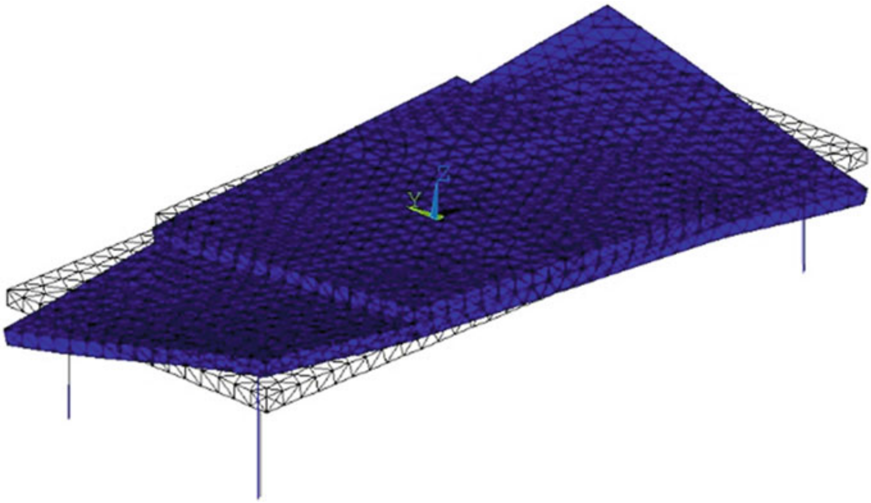


Fig. 3 The second natural modes—torsional

4 Verification of the Numerical Model

The kinematic excitation of the model system was performed by a single jump (Heaviside function) h = 5 mm. For our investigation, six variations of jumps (see Table 2), one case of symmetrical excitation (all springs jump at once) and five cases of unsymmetrical excitation were created. For unbalanced excitation (variants A to E), one, two or three springs jump in different combinations. The numbering of the jump

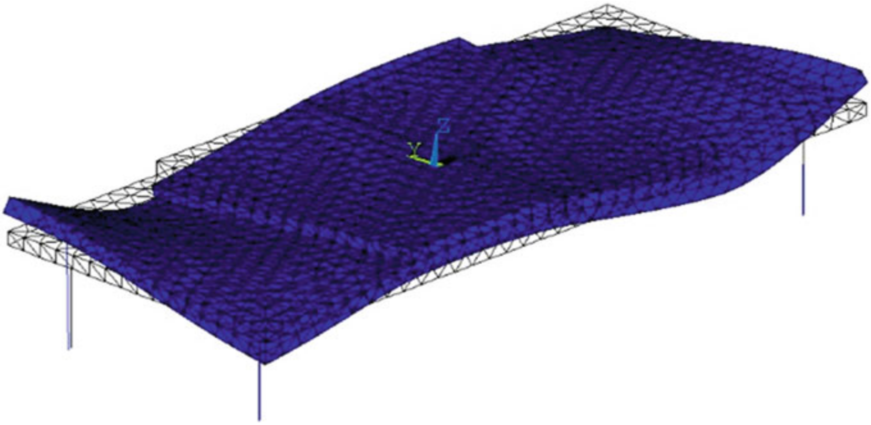


Fig. 4 The fifth natural modes—combinational of to bending and torsion

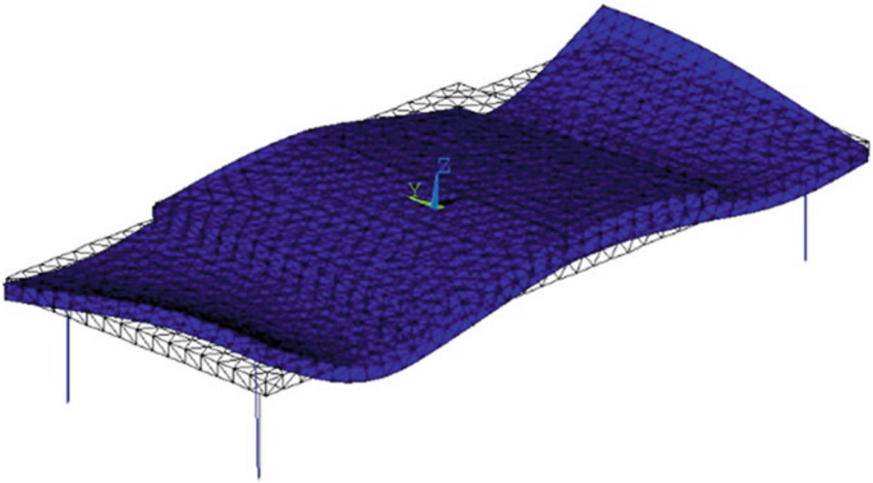


Fig. 5 The sixth natural modes—bending

springs in the Table 3 is used according to the markings that can be observed in Fig. 6.

The oscillation plate or vertical displacements were measured by inductive proximity sensor type Hottinger VA-50-T with the tip with a range of 0–50 mm. On the board, 3 sensors were installed at the mounting points of the springs to the plate. The signal is transmitted to the bridge amplifier.

Amplifiers were stored in DeweRack and the signal was sent to the computer where it was evaluated in LabVIEW.

Table 3 Options of excitation

Variant of generation	Designation of jump springs
A	3
B	2, 4
C	3, 4
D	2, 3
E	2, 3, 4
F	1, 2, 3, 4

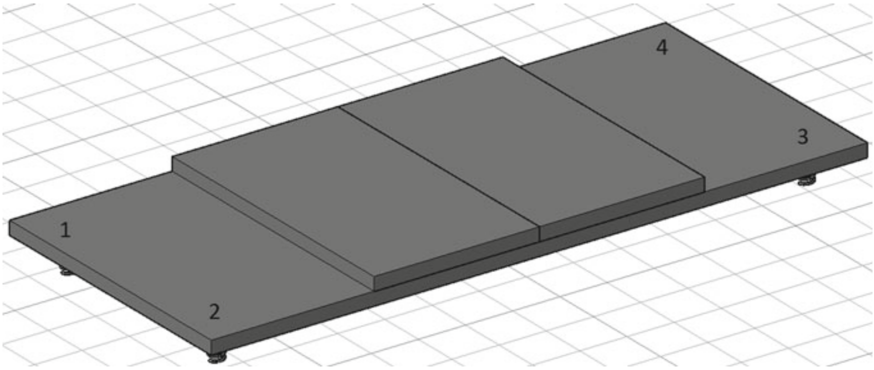


Fig. 6 Marking the springs

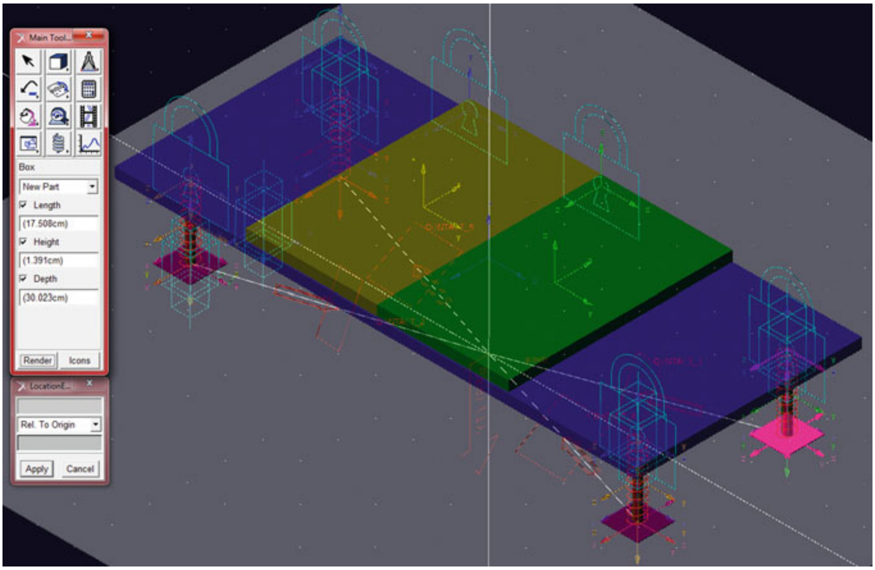


Fig. 7 Model of mechanical system in ADAMS program

The ADAMS simulation program was used to solve the vertical oscillation of the mechanical system. The boundary conditions were verified based on the results of the modal analysis. A vehicle model for a numerical solution is shown in Fig. 7.

The numerical model was verified based on experimental results and with the help of a modal analysis performed in ANSYS program.

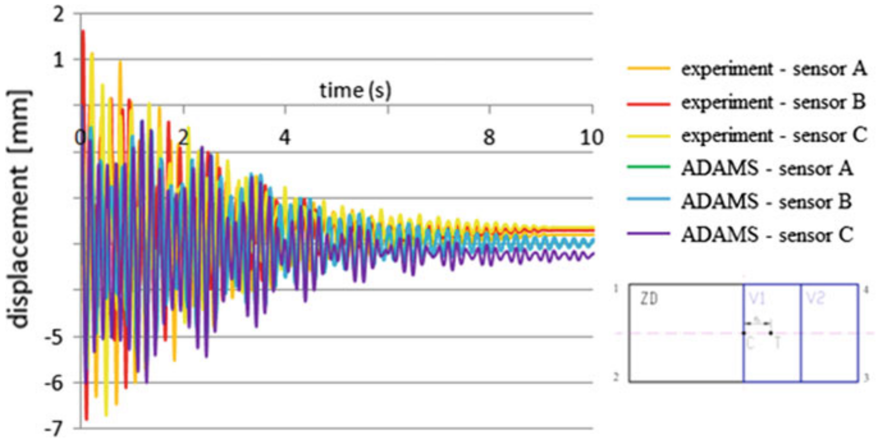


Fig. 8 Comparison of experiments and simulations—Variant II, spring jump 2, 4

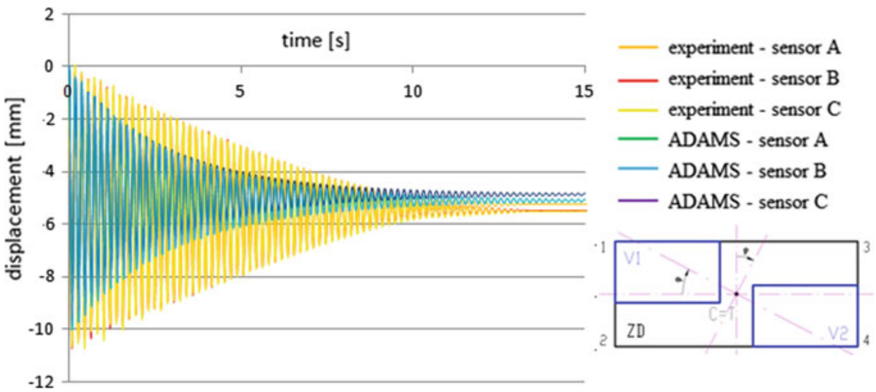


Fig. 9 Comparison of experiments and simulations—Variant V, spring jump 1, 2, 3, 4

5 Results of Numerical and Experimental Solution

The results of the experimental and numerical solution are shown in one graph (Fig. 8), which corresponds to the same asymmetry and the same kinematic excitation. Measured values can be seen in Table 4 (Fig. 9).

Table 4 Comparison of measured values—Variant I, excitation 1, 2, 3, 4

Time (s)	Deflection—experiment (mm)			Deflection–ADAMS (mm)			Deviation (%)		
	Sensor A	Sensor B	Sensor C	Sensor A	Sensor B	Sensor C	Sensor A	Sensor B	Sensor C
0.1	−10.0616	−9.95541	−10.7018	−9.718	−9.718	−9.718	3.417	2.387	9.195
0.2	−0.73339	−0.77808	−0.83246	−0.737	−0.737	−0.737	0.468	5.303	11.489
0.3	−9.0429	−9.37508	−8.98949	−8.631	−8.631	−8.631	4.553	7.935	3.986
0.4	−2.11115	−2.39702	−1.57624	−2.115	−2.115	−2.115	0.160	11.785	34.150
0.5	−7.72173	−7.71211	−8.20076	−7.046	−7.046	−7.046	8.748	8.634	14.078
0.6	−3.29182	−3.43563	−3.70745	−3.782	−3.782	−3.782	14.904	10.094	2.022
0.7	−5.94391	−5.99111	−6.21872	−5.375	−5.375	−5.375	9.566	10.278	13.562
0. 8	−5.62972	−5.74897	−5.19092	−5.404	−5.404	−5.404	4.012	6.003	4.102
0.9	−4.44125	−4.43621	−4.37971	−3.877	−3.877	−3.877	12.708	12.608	11.481
1	−6.83575	−6.74155	−7.1934	−6.721	−6.721	−6.721	1.683	0.309	6.571
1.5	−1.64279	−1.47048	−1.85209	−2.248	−2.248	−2.248	36.863	52.900	21.396
2	−7.13823	−7.11376	−7.47742	−6.015	−6.015	−6.015	15.732	15.442	19.554
2.5	−6.45325	−6.41336	−6.656	−6.067	−6.067	−6.067	5.989	5.404	8.852
3	−2.82345	−2.69319	−2.91667	−3.457	−3.457	−3.457	22.437	28.359	18.524
3.5	−6.46692	−6.37333	−6.69278	−5.497	−5.497	−5.497	14.993	13.744	17.861
4	−6.22883	−6.20924	−6.38423	−5.648	−5.648	−5.648	9.319	9.033	11.527
4.5	−3.81287	−3.74581	−3.87705	−4.135	−4.135	−4.135	8.442	10.384	6.647
5	−5.75266	−5.67293	−5.89792	−5.237	−5.237	−5.237	8.967	7.687	11.209
5.5	−6.02783	−6.00112	−6.11655	−5.388	−5.388	−5.388	10.616	10.218	11.912
6	−4.74764	−4.67235	−4.80064	−4.527	−4.527	−4.527	4.642	3.105	5.694
6.5	−5.23942	−5.17864	−5.32782	−5.107	−5.107	−5.107	2.532	1.388	4.149
7	−5.73315	−5.69894	−5.77531	−5.229	−5.229	−5.229	8.788	8.240	9.454
7.5	−5.27259	−5.23867	−5.32578	−4.734	−4.734	−4.734	10.216	9.634	11.112
8	−5.15746	−5.1086	−5.2277	−5.046	−5.046	−5.046	2.168	1.233	3.483
8.5	−5.4736	−5.43879	−5.52398	−5.136	−5.136	−5.136	6.172	5.572	7.028
9	−5.41505	−5.38676	−5.46677	−4.851	−4.851	−4.851	10.417	9.946	11.264
9.5	−5.30187	−5.24868	−5.34621	−5.016	−5.016	−5.016	5.394	4.435	6.179
10	−5.39749	−5.35074	−5.44838	−5.079	−5.079	−5.079	5.906	5.084	6.785
average							**8.922**	**9.898**	**10.831**

6 Conclusions

The paper describes the numerical solution of the mechanical analysis of the mechanical system in ANSYS simulation program. The result of the modal analysis of the mechanical system was its own shapes. Custom shapes contained gestures of their own shapes, deformed own shapes in a given frequency spectrum, and high own shapes that may contain residual effects. Experimental measurement of the modal analysis was performed on the laboratory model. Experimental results served to verify numerical model data and its further modification, refinement. The numerical vehicle model for measuring vertical oscillation was compiled in ADAMS. This model is used to predict the dynamic behaviour of the model system - the vehicle model. When comparing the results obtained were found good agreement between experimental and numerical solutions. The numerical model of a vehicle can be used to describe and predict the behaviour of a given system for the general case of loading and its kinematic excitation. More than 90% match between numerical and experimental measurements was achieved.

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