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COMPUTER AIDED SIMULATION ANALYSIS FOR COMPUTATION OF MODAL ANALYSIS OF THE FREIGHT WAGON

This paper deals with theoretical and practical aspects of modal analysis, which belongs to the tests for design of new rail wagons. The paper describes the process for calculating the eigenfrequencies of the wagon with non-standard design and dimensions and strength analysis of the modified construction of bogie type Y25. This computer aided simulation analysis significantly reduces the time to verify the static and dynamic evaluation of rail vehicles.

Keywords: *Modal analysis, eigenfrequencies, design, wagon frame.*

1. Introduction

Modal analysis is a relatively young field of dynamics and in industry started to be used in the 80s of the last century. Late inclusion into practice is associated with the development of software and hardware for finite element method. Modal analysis can be applied in theory, such as computational method or at practical level, such as real experimental measurements of mechanical structures. The modal parameters obtained from experimental analysis in engineering practice are often compared with the modal parameters obtained from computational methods [1]. The resulting modal parameter analysis include:

- eigenfrequencies of the construction,
- mode shapes,
- modal damping of the construction.

The great advantage of the mentioned simulations is that the entire development process of rolling stock is so accelerated, leading to a reduction in overall costs. Simulations and subsequent optimization of the vehicle structure is made before production of the vehicle itself. This leads to minimizing the number of unsatisfactory results conducted on a real vehicle. This may, in such a stage of development lead to delays and increased costs [2].

Computational models of vehicles and their components are more or less simplified compared with the actual ones. This simplification is seen when comparing the results from real tests.

2. Application of modal analysis

Modal analysis method can solve many technical problems encountered in the design, manufacture and operation of mechanical systems or parts. It is also used in the analysis of adverse events of mechanical systems, such as excessive noise, deformation, vibration, damage and so on.

Ride properties significantly influence vehicles mechanical systems [3, 4, 5] dynamic behaviour. We can theoretically predict the movement of the wheelset in the track by means of the wheelset and track geometric characteristics [6] analysis. Geometric characteristics define the rail / wheel profiles contact couple geometrical relationship. The shape of the contact couple crucial influences the size of the contact patch and contact stress between wheel and rail [7] value. This creates loading and excitation forces acting inside vehicle and track systems [8, 9]. The analysis of the mechanical systems dynamics may be analyzed by means of various methods [10].

Reasons for using modal analysis:

- Comparison of data obtained from experimental measurement on the prototype with the corresponding data obtained from finite element method. Optimization of the analytical model, which will be used for further calculations and simulations. This optimized model is free of errors, which were caused by poor application of boundary conditions.
- With the resulting eigenfrequencies unsafe operating conditions can be determined, which are not allowed. If the eigenfrequencies and frequency of excitation are equal, the resonance occurs. This reduces operating life, increases noise and could damage the construction.
- With the resulting mode shapes of vibrations we can determine the places of maximum errors. Subsequently, it is possible to make structural modifications (editing geometry, adding additional elements, changing material characteristics, etc.), which eliminate dangerous vibration.

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- The resulting modal parameters are used to diagnose faults and places of operation.

3. Modal analysis solved by finite element method

The most common type of the dynamic calculation is modal analysis, which determined mode shapes, eigenfrequencies and modal damping of mechanical systems [11]. These parameters provide us with basic information on the dynamic behaviour of mechanical systems.

At present, the modal analysis of mechanical systems is performed in computer programs that operate on the principle of finite element method. The most commonly used programs include ANSYS, ADINA, COMSOL and others.

Using modal analysis by finite element proceeds as follows:

1. Create geometry of the analysed construction.
2. Define material properties (density materials, Poisson's ratio, Young's modulus of elasticity of material).
3. Define the boundary conditions for the creation of computational model.
4. Create a mesh of finite elements (Fig. 1), which consists of a suitably chosen element and its final size (smaller mesh, longer calculation).

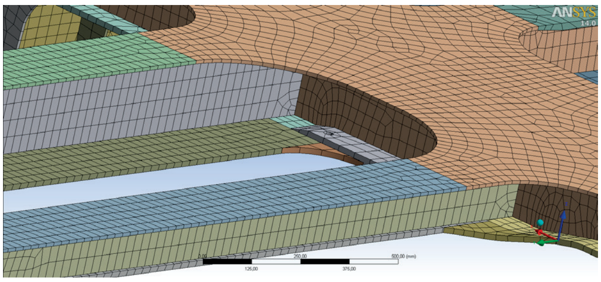

Fig. 1 The mesh of finite elements in the model of wagon

5. Set the solver which contains a suitable computational algorithm. Select the frequency range and number of wanted modes of vibrations in mechanical constructions.
6. Export modal parameters of the analysed construction.

4. Model eigenfrequencies computations

4.1. Model description

- CAD model – freight wagon for intermodal transport in Europe – WEL-WAGON [12, 13, 14],
- the dimensions of FEM model and CAD model – 1:1,
- spatial 3D elements (automatic meshing) – (15 – 30) mm [12, 13, 14],
- standard gravity in axis z – g = 9.8066 m/s^2.

4.2. Material model

- engineering steel S355J2C+N,
- minimum yield value 355 MPa (323 MPa in an immediate close distance of the weld),
- material – homogenous, isotropic, linear and elastic,
- mechanical properties – Young modulus of elasticity E = 210 000 MPa, Poisson`s ratio μ = 0.3.

4.3. Utilised software

ANSYS software allows engineers to construct computer models of structures, machine components or systems, apply operating loads and other design criteria and study physical responses, such as stress levels, pressure, etc. [15].

4.4. Boundary conditions

a) boundary condition in the place of A–D (4 slides), (The coordinate system is oriented in accordance with Fig. 2. The wagon is supported in the spots of slides. The knots in the slides spots are interconnected with spring elements having stiffness of 0.57 N/m),
b) boundary condition in the places E and F (2 hemispherical bogie pivots) - Table 1.

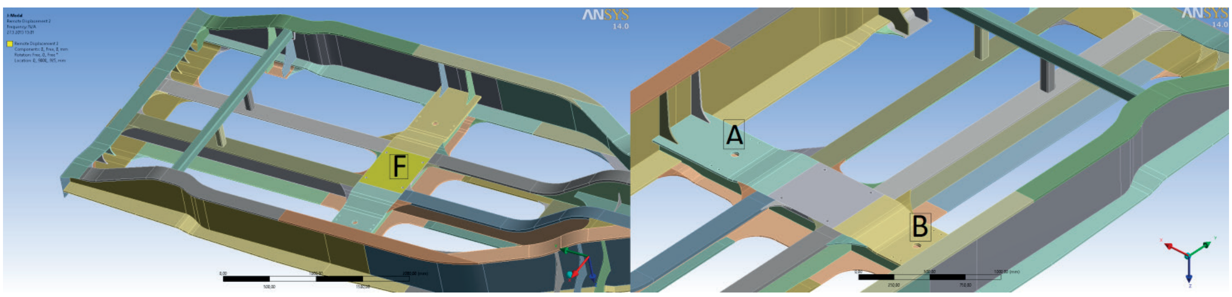

Fig. 2 The places for hemispherical bogie pivots and slides in the model of wagon (the figure shows only 1/2 of wagon, because the model is symmetric)

Boundary condition in the hemispherical bogie pivot Table 1

Ball point No. 1	Condition	Ball point No. 2	Condition
Fx	LOCK	Fx	LOCK
Fw	LOCK	Fw	LOCK
Fv	LOCK	Fv	FREE
Rx	FREE	Rx	FREE
Rw	FREE	Rw	FREE
Rv	FREE	Rv	FREE

LOCK – fixed, FREE – free to the directions of movement or rotation. F_x - displacement in the x direction, F_v – in the v direction, F_w – in the w direction. R_x - rotation in the x axis, R_v – in the v axis, R_w – in the w axis. The program ANSYS used to determine the direction of the symbols x, v, w instead of x, y, z) [16, 17].

4.5. Computation

The bogie for wagon design is the same as the Y25 Lsd1 wagon design and the used suspension has the kinked characteristic curve in the point of 6.63 t/axle [2 and 3]. Spring rigidity: outer spring 0.5076 N/m, inner spring 0.8244 N/m [18].

Wagon spring stiffness up to 6.63 t/axle.
(acts the outer spring only) c = 8121827 N/m
Wagon springs stiffness over to 6.63 t/axle.
(act both of springs) c = 21312264 N/m
Unsprung mass of 1 axle:

$$m_R = m_{wheelset} + m_{axlebox} + m_{springs} \quad (1)$$

$$m_R = 1072 + 240 + (7.2 + 15.45) * 4 = 1402.6\,kg$$

$$m_a = TARA - 4 * m_R = 22000 - 4 * 1402.6 = 16390\,kg \quad (2)$$

The natural angular frequency can be computed by:

$$\omega = \frac{1}{2\pi} \cdot \sqrt{\frac{c}{m_a}} \quad (3)$$

Input parameters for the calculation are given in Table 2.

4.6. Results

It is clear from the analysis that the third loaded wagon eigenfrequency (Fig. 3) is close to the third loaded wagon suspension eigenfrequency (the difference is 0.95 Hz). For further development of the wagon, its ride tests in operation performance are needed. The structural design modification for 3-rd eigenfrequency from the loaded wagon suspension is also needed. Modification can be done by using structural parts respectively assemblies and subassemblies (shape, material thickness, etc.).

5. Strength analysis of the modified construction of bogie type Y25

Bogie Y25 is equipped with a single suspension with duplex coil springs with kinked characteristic curve, a wheel guiding device of axle guard without clearances and with friction dampers with a special construction [19]. Transverse suspension is partially achieved through flexi-coil spring effect (clearance 2 x 10 mm). The frame of the wagon is usually associated with a bogie through the hemispherical bogie pivot (radius 190 mm) and its centre of 925 mm above the track at a weight of 20 t. The bogie

Input parameters for computation Table 2

State of load	Mass [kg]	Suspension stiffness [N/m]	Eigenfrequency [Hz]
Empty -1t	15390	8121827	3.66
Empty	16390	8121827	3.54
Empty +1t	17390	8121827	3.44
Loaded	80390	21312264	2.59

Comparison of the calculated value of eigenfrequency and eigenfrequency of suspension is presented in Table 3.

Comparison of the results for empty and loaded wagons Table 3

Eigenfrequency - number	Calculated value of eigenfrequency [Hz]	Eigenfrequency of suspension [Hz]
Empty wagon		
1	5.01	3.54
2	5.59	3.54
3	9.08	3.54
Loaded wagon (2x40´ containers)		
1	1.52	2.59
2	1.87	2.59
3	2.1	2.59

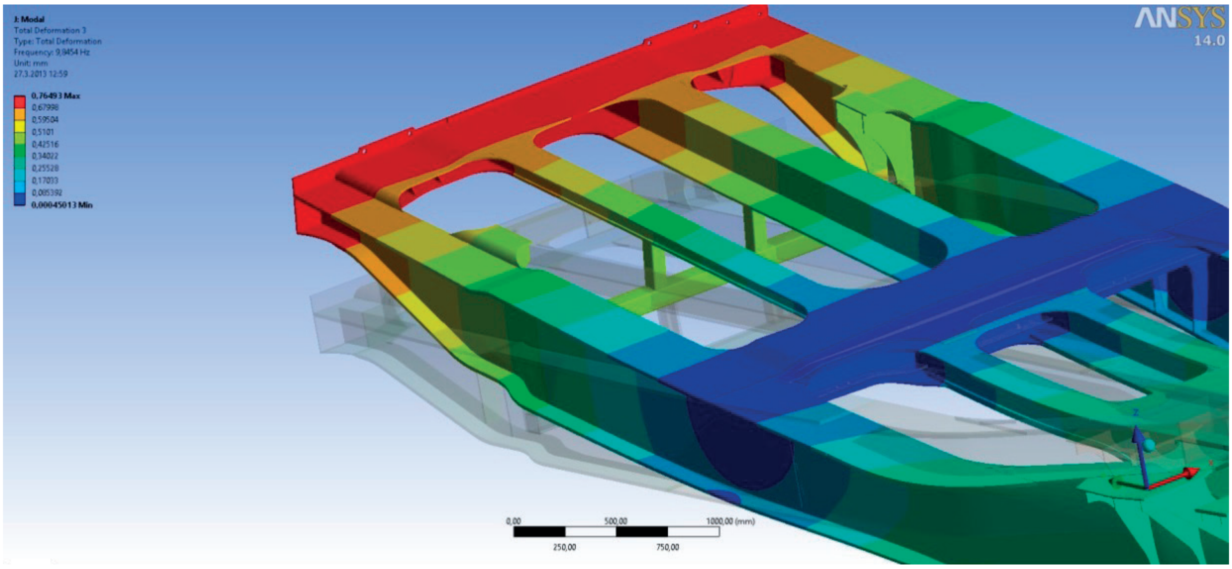

Fig. 3 Visual display of shape shifting in the third eigenfrequency in program ANSYS

was originally designed for the load of 20 t/axle and maximum speed of 100 km/h with a wheel base of 1800 mm [20]. During the development these parameters were upgraded.

At present most bogies are designed for 22.5 t axle load and the maximum speed increased to 120 km/h. Bogies weight is usually from 4.5 to 5 t. Wheel diameter is 920 mm and the wheelbase is 1800 mm. The overall width of the bogie frame is 2440 mm, width at the centre of the axle boxes is 2000 mm for 1435 mm track gauge. 3D model of said bogie frame is shown in Fig. 4.

The modification brought about expansion of the bogie frame Y25 to 1520 mm (Russian gauge) in such a way that the geometry of the frame was changed in the cross-section (+36 mm). The width in the middle of the axle box after modification is 2036 mm. Axle dimensions are given in Table 4.

The diameter of the wheel on the axle is 957 mm. The axle load also increased from 22.5 t to 23.5 t. Due to the rougher climate it is intended with material labelled S 355J2 + N (11523) which has a yield point of 355 MPa and tensile strength 490/630 MPa. It is believed that the weight of the bogie with the adjustments is raised to about 5 t.

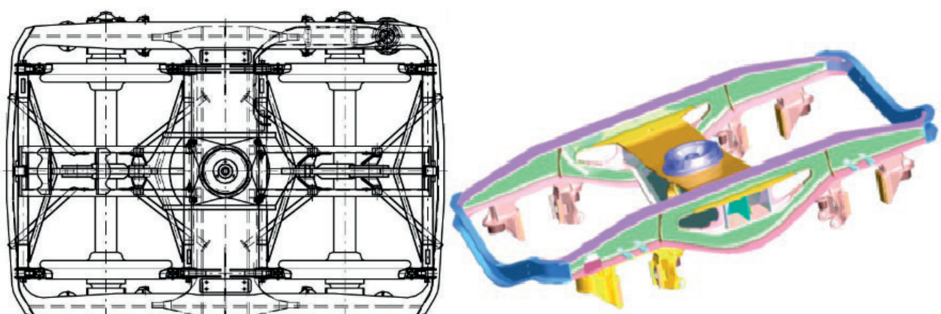

Fig. 4 2D and 3D model of frame bogie in program ProEngineer

Dimensions of PM3 axle — Table 4

<table>
<tr><th colspan="11">Dimensions in mm</th></tr>
<tr><td rowspan="3">Type of axis</td><td>d1</td><td>d2</td><td>d3</td><td>d4</td><td>d4</td><td>d5</td><td>d5</td><td rowspan="3">R1</td><td rowspan="3">R2</td></tr>
<tr><td>+0.052</td><td>+0.2</td><td>+2.0</td><td rowspan="2">Nominal value</td><td rowspan="2">Tolerance</td><td rowspan="2">Nominal value</td><td rowspan="2">Tolerance</td></tr>
<tr><td>+0.025</td><td>+0.12</td><td>-0.5</td></tr>
<tr><td rowspan="2">PM3</td><td rowspan="2">130</td><td rowspan="2">165</td><td rowspan="2">197</td><td rowspan="2">180</td><td rowspan="2">-1.0</td><td rowspan="2">200</td><td>+0.045</td><td rowspan="2">100</td><td rowspan="2">25</td></tr>
<tr><td>+0.015</td></tr>
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Boundary conditions – hemispherical bogie pivot Table 5

Hemispherical bogie pivot	Displacement	Rotation
The direction of the longitudinal axis of the bogie (global axis x)	ux = R	ϕx = R
The direction of the transverse axis of the bogie (global axis y)	uv = F	ϕv = F
The direction of the vertical axis of the bogie (global axis z)	uw = R	ϕw = F

Boundary conditions – axle guard Table 6

Axle guard	Displacement	Rotation
The direction of the longitudinal axis of the bogie (global axis x)	ux = F	ϕx = F
The direction of the transverse axis of the bogie (global axis y)	uv = R	ϕv = F
The direction of the vertical axis of the bogie (global axis z)	uw = F	ϕw = F

5.1. Computation

The object of the calculation is the strength test of freight bogie frame through FEM analysis. For the calculation of the analyzed parts of the bogie through finite element the program ANSYS was used. They are used as rectangular, four-node ”shell” elements. The size of elements in the area under consideration is 5 to 10 mm. The frame is stored in a cylindrical tube with the stiffness equivalent to the stiffness of the suspension. Boundary conditions are created so as to be applicable to all burdensome conditions. Analysis is performed in the linear region. Distortion analysis results due to the simplification mentioned in the introduction are practically negligible [21, 22]. Consideration is being given to the fact that the material is linearly elastic and isotropic.

5.2. Boundary conditions

Reactions in the longitudinal direction (x-axis) are captured in the nodes lying inside the hemispherical bogie pivot (Table 5).

Reactions in the transverse direction (y-axis) are captured in nodes of slides (Table 6).

5.3. Elements

SHELL181 Element

SHELL181 (Fig. 5) is suitable for the analysis of thin and medium thick shell structures. This is a four-node element with six degrees of freedom at each node: in the direction of axis x, y, z and rotation x, y, z. SHELL181 is suitable for large rotation or high stress. Changes in the shell are counted in nonlinear analysis [15].

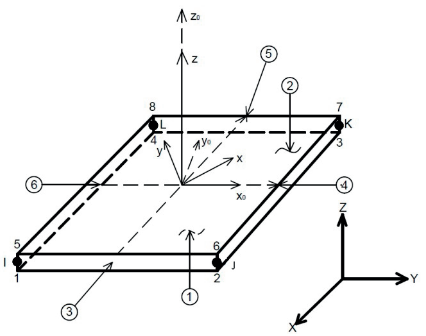

Fig. 5 Four-node element with six degrees of freedom at each node

5.4. Load conditions

Load conditions are as follows:

- **Load condition 1** - consists of vertical force in a hemispherical bogie pivot - 824 kN.
- **Load condition 2** - consists of vertical force in a hemispherical bogie pivot - 429 kN, vertical force on the slides - 107 kN and transverse forces on the hemispherical bogie pivot - 90 kN.

Schematic view of the load condition is shown in Fig. 6. Each component of the solid model was exported as a separate part. Contact links were created among the components (type “bonded”), which simulate the welded joints.

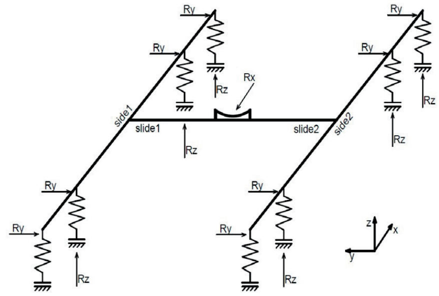

Fig. 6 Schematic view of load condition

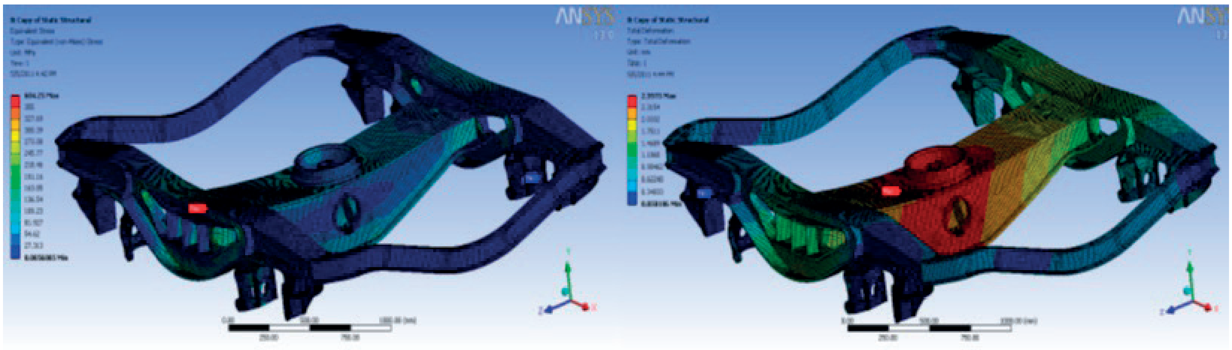

Fig. 7 The behaviour of stress and deformation in program ANSYS

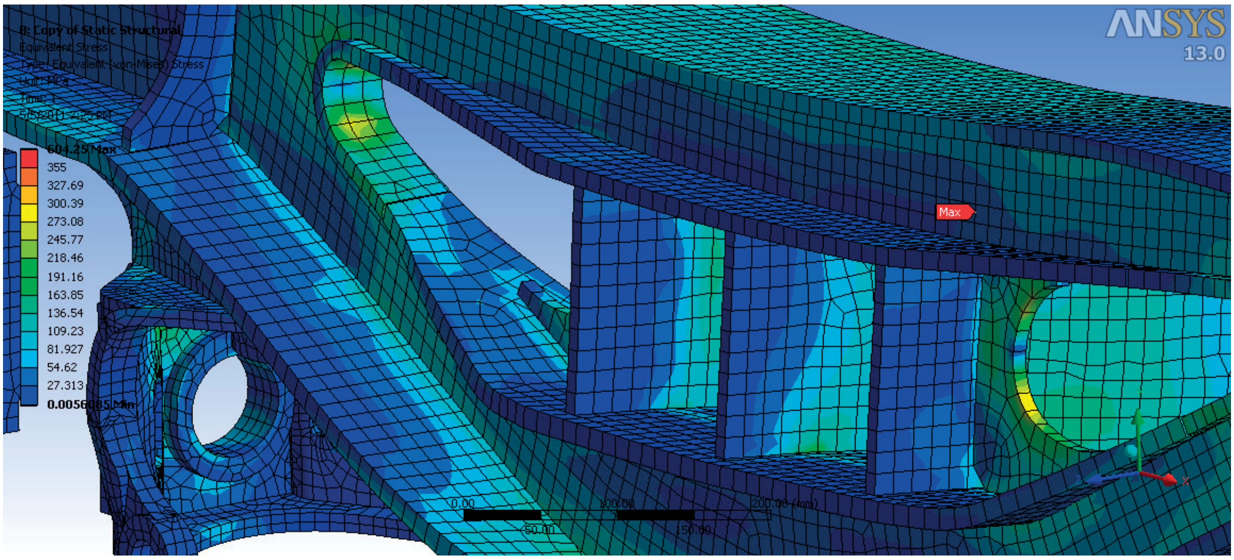

Fig. 8 Location on the model with greatest stress

5.5. Results

Due to the limited scope of this paper, only the results to the load condition 2 are presented. As mentioned, in this case vertical and transverse forces operated in the hemispherical bogie pivot and the slides as well. After the loading of construction in certain places the peaks incurred of stress. These deficiencies can be remedied in several ways. For example, the shape or thickness reinforcements are modified or another type of material is used.

The behaviour of stress and deformation (displacement) is shown in Fig. 7.

Figure 8 shows the location of the greatest stress.

6. Conclusion

Computational simulations are now an integral part of the development process of rolling stock. They allow a more detailed analysis of the behaviour of the vehicle as a whole or its individual parts. Therefore, it is possible to better optimize the design of rail vehicles and prevent potential problems in the operation, which would require increased costs.

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