DYNAMIC ANALYSIS OF RAILWAY PLATFORM CHASSIS MODEL

Karol Chłus, Wiesław Krasoń

Military University of Technology
Department of Mechanics and Applied Computer Science
Gen. Sylwester Kaliskiego Street 2, 00-908 Warsaw, Poland
phone: +48 22 6839226, +48 22 6839654,
e-mail: kchlus@wat.edu.pl, wkrason@wat.edu.pl

Abstract

The aim of the paper is to build a numerical model of a railway wagon, to perform statics calculation as well as to compare the calculations results with the experimental results of measurement of a real railway wagon maximal deflection and to conduct dynamic analyses. The examined railway wagon was designed to transport interchangeable containers in ACTS (Abroll Container Transport System). The essential matter of such reloading is placing the container on a special rotating platform which enables horizontal reloading of the load onto the truck. In order to perform strength and dynamic numerical analyses with a finite element method (FEM), a 3D coat-beam model of the wagon were prepared. The discreet FEM model was executed with the aid of MSC.Patran preprocesor and for calculations MSC Nastran program was applied. The obtained results were verified by the experimental results of measurement of real railway wagon maximal deflection. The correctly verified model was applied both for modal analysis of free vibration and time dependent dynamic analysis. The influence of dynamic load on construction strength was examined. The loading model was selected so as to represent hypothetical simultaneous loading of the containers on the carriage frame. It turned out that the most strenuous element during the simulation of loading the containers onto the frame was the element of the vertical reinforcing plate which is in the direct contact with the pivot support of the carriage.

Keywords: *mechanics, strength of construction, FEM, dynamic analysis, railroad wagon*

1. Introduction

Rail-road combined transport nowadays constitutes an alternative for car transport. In intermodal transportation systems, convenient and effective loading systems play a significant role [6, 7]. They are based mostly on the principle of horizontal loading (ACTS system). ACTS system [1] is created by three fundamental elements: a carriage equipped with rotating platforms, a truck with suitable load capacity, interchangeable containers.

The present paper presents selected elements of strength numerical investigations methodology of the undercarriage frame-loading platform of the carriage body system. There were presented the numerical models of an exemplary rail carriage operating in ACTS system – Fig. 1. The carriage is equipped with a flat frame cooperating with two bogies and three mobile platforms rotated in respect to central knots (Fig. 1) in order to simplify the process of loading and unloading of the interchangeable containers. After loading the containers the loading platforms are blocked for the time of transportation between stations. The frame system with three blocked platforms of the carriage in the transportation configuration was the object of the detailed numerical and experimental investigations.

Static calculations of the frame-carriage loading platforms system were executed in the option of allowed loads. The calculations results were compared with the experimental results. The frame model with the properly modelled supporting knots on the bogies as well as rotating platforms were adapted, after verifying its accuracy to dynamical tests [9].There were carried out the numerical investigations of frequency and the free vibrations modes of the system of the frame with rotating platforms blocked in the transportation position of the carriage. The results were applied to develop the parameters of dynamic characteristic of the examined carriage system. The selected results from the dynamic simulation of the carriage load test were presented.

Fig. 1. Exemplary carriage platform for intermodal transportation in ACTS system

2. Numerical FE models

A FE numerical model of the frame-loading platforms system of the carriage was carried out in MSC Patran environment [2]. At the first stage of works, the model of the carriage frame was constructed. The frame was modelled with the use of the 1D and 2D shell-beam elements (Fig. 2.).

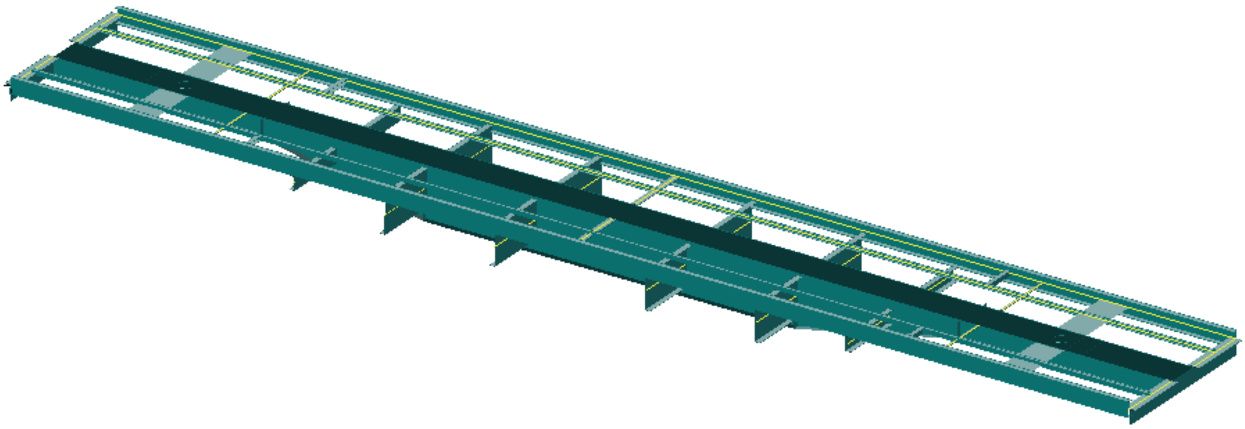

Fig. 2. Carriage frame model

At the next stage of works, FE models of carriage rotating platforms were developed. The platforms were modelled with the use of 1D elements. The view of the single loading platform with a modelled rotating knot was shown in Fig. 3.

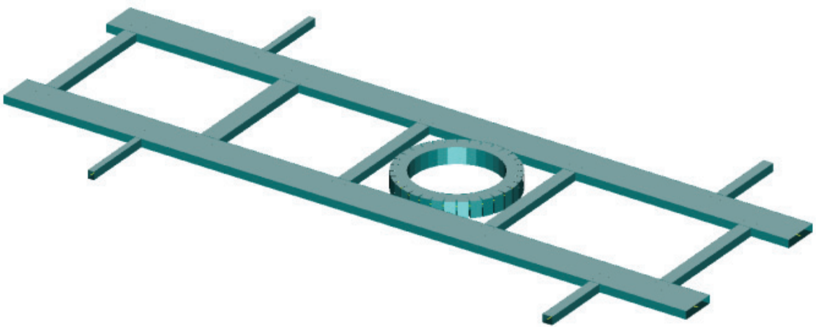

Fig. 3. Model of carriage rotating platform

The models of rotating platforms prepared in this manner were integrated with the model of the carriage frame. The completed model of the system is composed of of the main frame and three platforms adequately arranged on the frame main plate, as it is shown in Fig. 4.

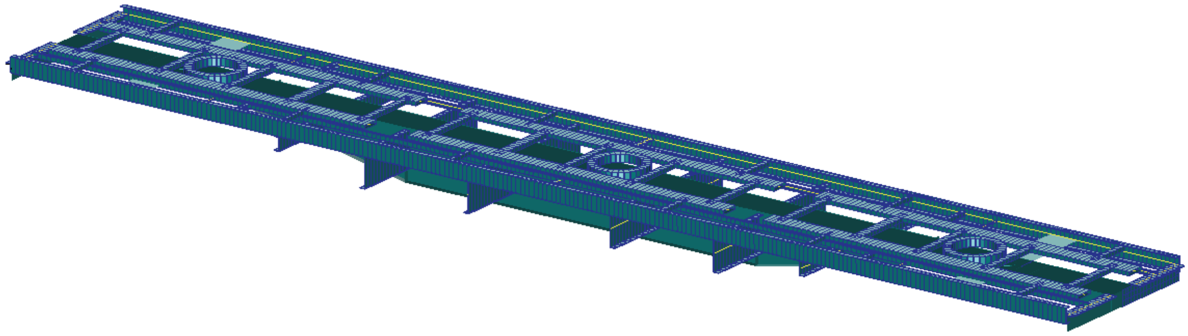

Fig. 4. Full model of the carriage platform: frame with three rotating platforms

In the numerical model, there were introduced the boundary conditions corresponding to the support of the carriage frame on the pivot supports and on the bogie side supports symmetrically on the both sides of the model. The side supports between the chassises and the frame of the examined carriage were modelled with the use of the spring elements with adequately selected stiffnesses. These elements were introduced in the places of occurring the contact between the transverse beam of the carriage frame and the bogies, as it was presented in Fig. 5. These elements were placed symmetrically on the both sides of the carriage. The stiffness of the single spring element used to represent the side support in the model equals $k= 4.1 \cdot 10^7$ N/m.

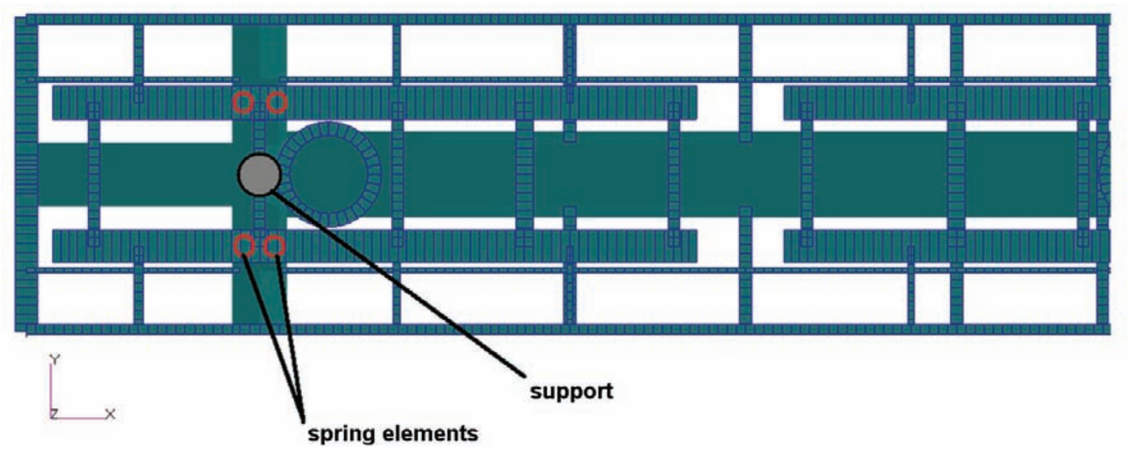

Fig. 5. Arrangement of spring elements

The joints between the subassemblies and the contact conditions were defined with the use of MPC elements [3, 5]. The parameters of cross sections corresponding to the real ones were ascribed to all the elements of the model. The model of material was defined as steel with standard strength parameters for constructional steel. There was assumed a material linear-elastic model with the following parameters:

- Young modulus $E = 2 \cdot 10^{11}$ Pa,
- Poisson ratio $\nu = 0.3$,
- Density $\rho = 7.8 \cdot 10^3$ kg/m$^{3.}$

3. Numerical analysis of the static loading test

At this stage of research, the conditions of the loading test carried out on the real carriage were numerically modelled. There were considered two options of loading. First of them assumed that the model was only influenced by the deadweight of the frame-loading platforms system equal to 195 kN. The other variant of loading assumed that the all-up weight of a completed carriage with the load on three platforms equalled to 710 kN. This is the loading option corresponding to the weight which was used at the extreme measurement of carriage deflection (carriage deadweight along with the weight of three containers loaded with the fine coal). The external loading from the containers with the load was modelled in the form of constant loading put on the elements of cross-bar of the three rotating platform (Fig. 6).

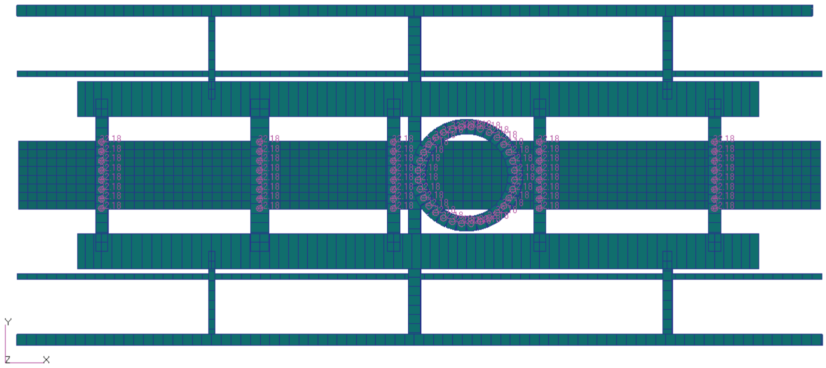

Fig. 6. Loading in the model of rotating platform (view on the rotating platform and carriage frame)

3.1. Comparison of results

There were carried out the static analysis with the boundary conditions representing the loading test. The results obtained from calculations were compared with the experimental results of the deflection of the above mentioned carriage.

The first measurement was performed while the carriage was loaded only with the deadweight. The second measurement of the deflection was carried out after complete filling of the three loading containers of the carriage with the fine coal. The containers were fastened in the transportation position on the carriage rotating platforms. The relative deflection was calculated through subtracting the value obtained in the first measurement from the value obtained in the second measurement. It was verified that the maximum value of the relative deflection for the examined platform equalled to 14mm. The deflection determined in such a manner will be used for comparison with the results of numerical analyses and for the evaluation of the accuracy of the applied numerical models.

In the numerical investigations representing loading tests, there were obtained the following deflection values:

- the first option of loading: 21.2 mm,
- the second option of loading: 6.1 mm.

Therefore the value of relative deflection is equal to 15.1 mm. The maximum stresses reduced according to H-M-H hypothesis [4] determined in the second option of the test equal to 168 MPa. Fig. 7 presents the lines of deflections determined experimentally and calculated with the use of FEM. The described above values are compared in Tab. 1. On the base of the obtained results it can be concluded that numerical models and applied boundary conditions correctly represent the real structure of the examined system. The models can be applied to other simulation of operations of the frame-carriage loading platforms system.

Tab. 1. Comparison of results for the individual models

Comparable value	Experiment	FEM
Deflection [mm]	14	15.1
H-M-H stresses [MPa]	$\frac{Re}{x}=\frac{350}{2}=175$	168

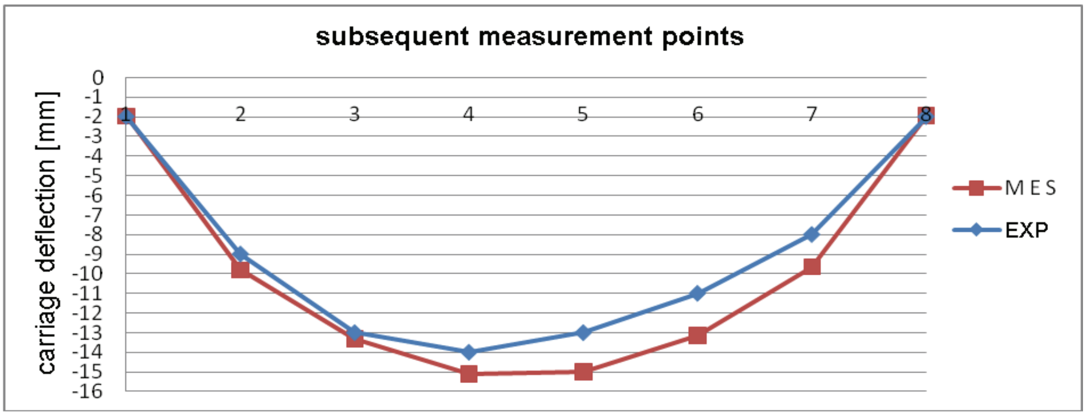

Fig. 7. Carriage deflection at the subsequent measurement points

4. Dynamic analysis

The verified FEM model of the frame-carriage loading platforms system was modified through taking into account describing the selected dynamic properties of the structure. Such a model was next applied in modal analysis [5, 10, 11]. There were taken into consideration the constraints corresponding to the option of supporting and configuration of the system previously analysed in the scope of statics.

Free vibrations of the undamped system are analysed through solving the system of homogeneous equations (1):

$$[B]\{\ddot{q}\}+[K]\{q\}=0\,, \tag{1}$$

where:
{q} - vector of node displacements,
[K] - matrix of stiffness related to displacement,
[B] - matrix of inertia related to acceleration.

In the analyses of this type, neither the damping influences nor the interaction of external loading interacting on the model are taken into consideration. The numerical analysis results in determining the frequency of vibrations and further free vibrations modes of the frame-carriage loading platforms system.

Figure 8 presents the first four modes of vibrations of the examined system of the carriage as well as free vibrations frequencies corresponding to them. The first recorded mode of vibrations is dominated by torsion of the central section of the system in respect to the longitudinal axis of the carriage model. The second mode presents pure bending in the longitudinal plane of the system. For the third and fourth frequency of free vibrations, the complex modes of system deformation were obtained. Due to the manner of modelling the contact between loading platforms and the carriage frame (mutual interaction of the subassemblies was consider only on the direction of the vertical freedom degrees OZ) deformations of the rotatable platforms, independent on the frame chassis occur for two last vibration modes. The obtained results were used to determine the models parameters in the next stage of the analysis of frame-carriage loading platforms system.

In ‘Transient Response Analysis’ [3] structural response is computed by solving a set of coupled equations using direct numerical integration. In the equations (2) describing this stage of the research on the carriage, there are additionally considered damping and external forces changeable in time.

$$[B]\{\ddot{q}\}+[C]\{\dot{q}\}+[K]\{q\}=\{P\}\,, \tag{2}$$

where:
C - damping matrix related to velocity,
P - vector of forces.

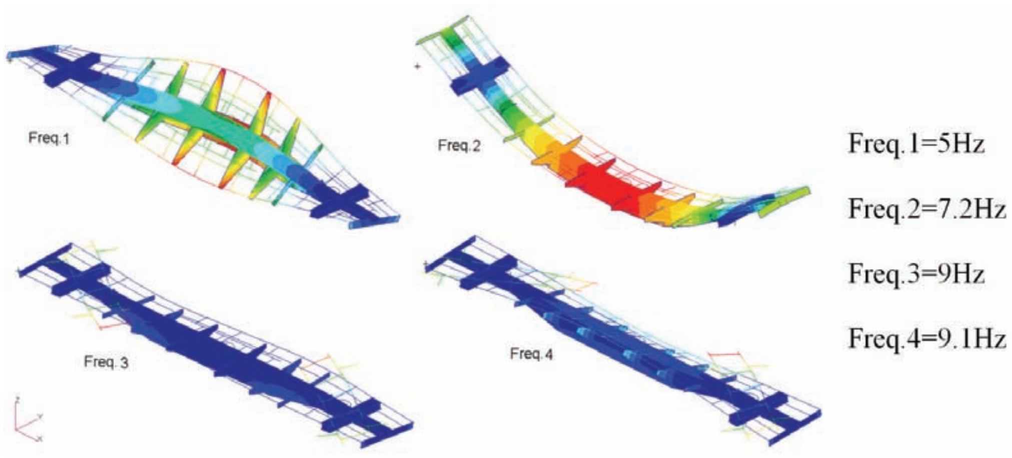

Fig. 8. Eigenmodes from the FEM analysis of the railway wagon

Matrix of damping C of examined system is determined according to the dependency (3):

$$[C]=[C^1]+[C^2]+\frac{G}{W3}[K], \tag{3}$$

where:

C_1 - damping elements and direct input damping matrix selection,
C_2 - direct input matrix and transfer functions,
G - overall structural damping coefficient,
$W3$ - frequency of interest in radians per unit time for the conversion of overall structural damping into equivalent viscous damping,
K - global stiffness matrix.

In transient response analysis, the system was loaded with deadweight and with a changeable in time load of three containers loaded with fine coal of the allowed value identical as in static analysis. The loading model was selected so as to represent hypothetical simultaneous loading of the containers on the carriage frame. The loading time is 1s. The analysis was conducted at different values of the structural damping coefficient *G*. Value *W3* [3] was determined from the dependency (4):

$$W3=2\pi\cdot f\ , \tag{4}$$

where *f* is frequency of free vibrations obtained from modal analysis. Value *W3*=50rad/s evaluated with an excess on the basis of frequency f_2 corresponding to pure bending of the system.

Figure 9 presents graphs of the components of vertical displacements of the central knot in the middle of the carriage length in the function of time. H-M-H stresses developed in the most loaded elements of the carriage frame model were presented in Fig. 10.

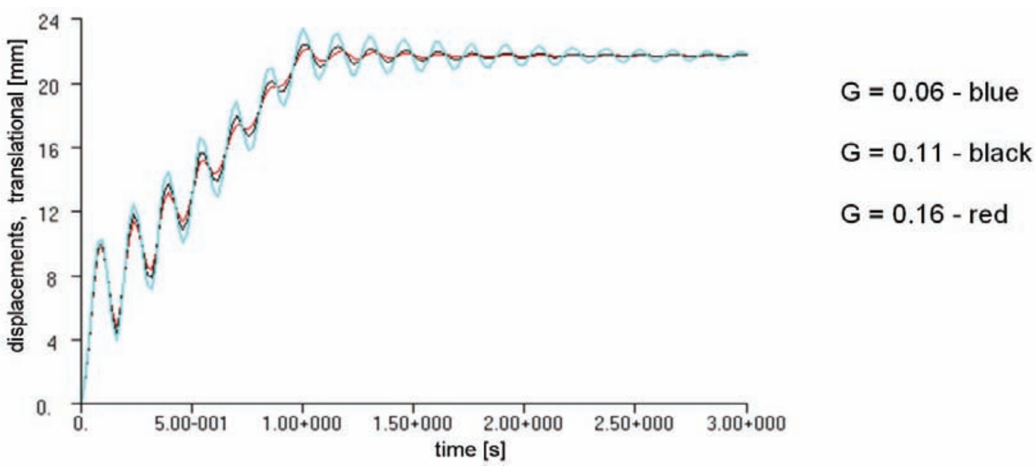

Fig. 9. Vertical displacements of the central knots in the middle of the carriage length

The greatest deflection of the carriage was obtained in the model with structural damping *G* = 0.06 in the time moment t=1 s, that means afterwards finishing the loading of containers on the platform. The maximum displacement of the carriage frame recorded on the vertical direction was 23.3 mm (in the static analysis the analogical value was 21.2 mm).

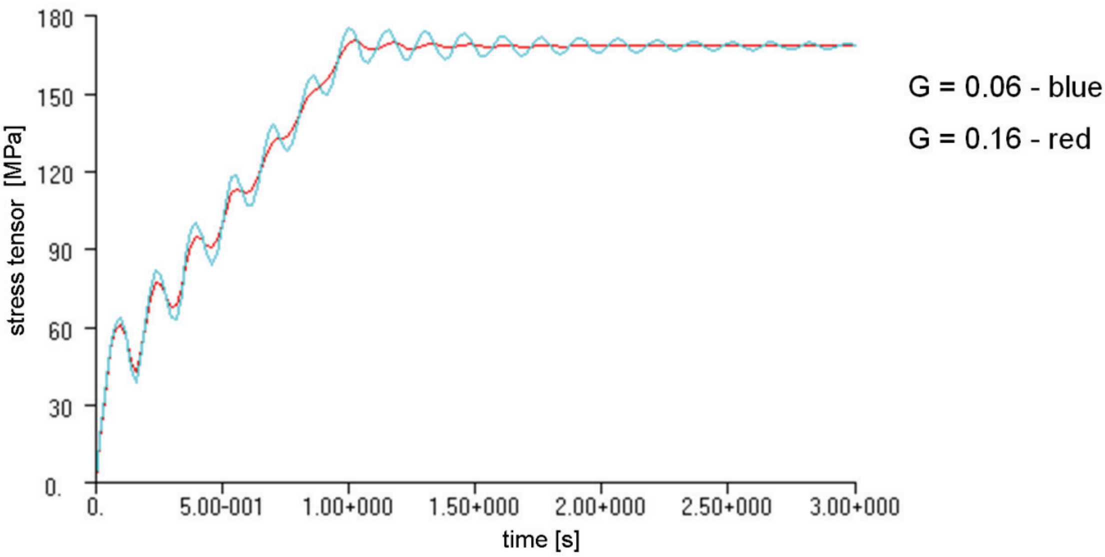

Fig. 10. Reduced strains of the most strenuous element of the model

It turned out that the most strenuous element during the simulation of loading the containers onto the frame was the element of the vertical reinforcing plate which is in the direct contact with the pivot support of the carriage. For the damping coefficient *G* = 0.06 in the time moment t = 1 s, the value of the maximum strains reached 175 MPa. H-M-H stresses from the analogical static analysis did not exceed the value of 168 MPa.

5. Conclusions

The paper presents the selected elements of the research methodology of the frame-carriage loading platforms system for specific transports. There was discussed the structure of numerical FE models of the system and selected results of static analysis. The models were verified with the use of results of an experimental measurement of the decline of the real rail carriage platform. The obtained value of the frame deflection from the numerical simulation of the static test corresponds to an experimental value (the maximum difference equals to approximately 8%). The H-M-H stresses do not exceed the allowed stresses, which value Re/x equals to 175MPa. On the basis of the static analysis, it can be concluded that the developed model is correct and can be used to further strength research.

The carriage model verified on the basis of load tests was used, after suitable modifications (specification of the mechanical characteristic), for the investigation of the influence of the allowed loading on the structure strength. In order to obtain this, there was conducted the modal analysis of the free vibrations and dynamical analysis in the scope of “transient response”. On the basis of the graphs of displacements and maps of stresses there was examined the influence of the damping parameter on the deflection of the carriage and the maximum stresses. It was verified that at the structural damping coefficient typical for such structures *G* = 0.06 in the moment t=1s, that is after full loading of the system with the simultaneous allowed load of three loading containers, the maximum displacement on the vertical direction of the frame and the greatest value of the H-M-H stresses in the carriage do not exceed the allowed values.

Acknowledgements

Paper financed by National Centre for Research and Development –PBR R10 0023 06/2009.

References

[1] Maziarz, W., *System ACTS. System przeładunku poziomego*, *www.logistyka.net.pl/index.php?option=com_content&task=view&id=1533&Itemid=40*, 2001.
[2] Reference Manual, *MSC.PATRAN*, Version r2, MSC.Software, 2007.
[3] Reference Manual, *MSC.NASTRAN*, Version r2, MSC.Software, 2007.
[4] Niezgodziński, M. i T., *Wytrzymałość materiałów*, PWN, Warszawa 2002.
[5] Dacko, M. i inni, *Metoda elementów skończonych w mechanice konstrukcji*, Arkady, Warszawa 1994.
[6] Kwaśniowski, S., Nowakowski, T., *Transport intermodalny w sieciach logistycznych*, Oficyna Wydawnicza Politechniki Wrocławskiej, Wrocław 2008.
[7] Domagała, M., Hawryluk, M., Lisowski, E., *Optymalizacja konstrukcji platformy kolejowej do przewozu ciężkich ładunków*, Transport Przemysłowy, Nr 2(34), s. 40-43, 1994.
[8] *www.kockumsindustrier.se/products/freight/Intermodal/Megaswing.htm*
[9] Chłus, K., Krasoń, W., *Influence of bearing model in part over carriage on chassis stiffness of railroad platform*, XXX Seminarium Kół Naukowych Mechaników WAT, Warszawa 2011.
[10] Rusiński, E., Czmochowski, J., Smolnicki, T., *Zaawansowano metoda elementów skończonych w konstrukcjach nośnych*, Oficyna Wydawnicza Politechniki Wrocławskiej, 2000.
[11] Ndambi, J. M., Giannopoulos, G., Vantomme, J., *Dynamic characterisation of an IPC pedestrian bridge*, Engineering Structures 27, 229–237, www.sciencedirect.com/science/article/pii/S0141029604003347, 2005.