Fastenings Parameters Determination for Highly Deformative Cargo, Taking Into Account Its Durability During Transportation In Cars And Trains

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***Abstract*—There are considered piece cargo transportation in the railway boxcar and car. Variable loadings influence the piece cargo during its transportation. These loadings depend on non-uniformity of the car's movement. There are installed internal connections parameters with which acceleration and displacement of a cargo relative to the car will not exceed admitted values. The mathematical model of a cargo and package is developed as systems with several freedom degrees on its foundation.**

Besides it is necessary to take into account the transported structures strength during fastening computations. Our studies displayed that additional stresses must be taken into account for developing of transporting fasteners for highly deformative cargoes, which can lead to elastic-plastic deformations of the transported device. There are viewing cooperative solution ways for the cargo's secure fastening and its strength maintaining.

***Index Terms*— fastening, elastic-plastic deformations, system dynamics**

I. Introduction

During transportation there are cases when it is impossible to realize cargo immobility about the car. Besides it is known, that introduction of additional connections between a rigid cargo and a car allows lowering the forces acting on cargo essentially. But it leads to cargo displacement in the car also. When the cargo mass is comparable to car mass, load displacements act to dynamics changing of the car properties. Besides, cargo displacement in the car leads to dynamic loads increase. There is a research necessity of unsteady movements of cars loaded by displaced cargo.

The damage boundary, acceleration damage boundary and displacement damage boundary theory for parameter's determining corresponding to the packaging system's working conditions without damage are developed in papers [1–8]. For the first time a damage boundary notion was proposed [1] in order to indicate the structure's sensitivity to the impact or its fragility. Since the criterion for the damage beginning moment determining is the load maximum acceleration, it is named the acceleration damage boundary [2-5]. All parameters must be taken into account. In this case, the values range under which the goods in the package safe transportation ensured is significantly less. In comparison with the case if only one of them is used. It is also indicated that the damage boundary determining method can be improved by introducing a criterion for the maximum allowable packaging material deformation. This method based on the packaged cargo acceleration. Small package stiffness could protect the structure from high acceleration.

In the paper [6] proposed a computational method for determining the damage boundary curves with help of the finite element method.

There was much attention paid to the research of the material's properties for packaging production [7]. It was notice that the cardboard package is most generally used and has a small cost.

Variable loadings influence the piece cargo during its transportation. These loadings depend on non-uniformity of the railway car's movement. Maximum loads occur during the boxcar's impact. Cargo placement and tie-downing in boxcars must be executed by taking into account the train's safety, shunting and fastening operations, full use of the boxcar capacity, safety of transported cargoes and cars [8–9].

The largest number of piece goods is transported in paperboard cases. Sometimes case are damaged during the boxcar's impact. The reasons can be not enough strength of the paperboard or inflexibility of the tie downing.

It is set a problem of definition of fastening parameters influence and properties of package system dynamics «cargo – package – car».

II. Modeling of the System «Car – Package – Cargo» Dynamics

Transportation safety is ensured by changing fastening elasticity in majority device's schemes designed for fastening rigid loads [10].

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The damage boundary, acceleration damage boundary and displacement damage boundary theory are intended for parameters definition of the packaging systems working conditions without damages. A damage boundary definition method is based on packaging load acceleration decrease. It is recommended to add to this method a displacement damage boundary criterion of the packaging material. Packaging with a small stiffness coefficient can protect a load from high acceleration, but thus it can be destroyed. Internal connections parameters with which acceleration and displacement of a cargo relative to the car will not exceed admitted values are installed.

We designed the generalized scheme (fig. 1) including transported cargo 1, package 2 and a car body 3. It was made for the account of different materials features properties which the package and fastenings of a cargo was made. There were elastic-viscous connections between a cargo and package, and also between package and car. Besides, there were forces of a dry friction between a car floor and package.

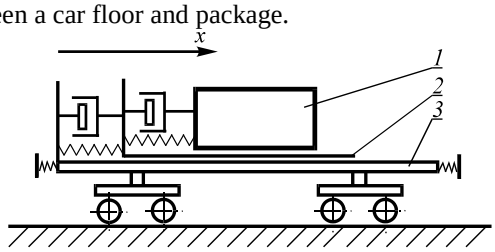

Figure 1. The scheme of the car with a cargo.

The mathematical model of a cargo and package was developed as systems with two freedom degrees on its foundation. The force of spring acted on a cargo. It was created by fastenings between cargo and package. Package interacted with car body through fastenings. Elastic forces depended on relative body's displacement nonlinearly. Not elastic resistance forces were proportional to velocity of this distance changing. There was obtained a system of second-order differential equations describing the cargo and packing movement. It was made a number of transformations and cargo acceleration expressions substitution. In that case the system differential equations take a form (1-2)

$$\ddot{x}_1 = \frac{k_1}{m_1}(x_2 - x_1)^{n_1} + \frac{\alpha_1}{m_1}(\dot{x}_2 - \dot{x}_1), \tag{1}$$

$$\ddot{x}_2 = -\frac{k_1}{m_2}(x_2 - x_1)^{n_1} - \frac{k_2}{m_2}x_2^{n_2} - \frac{\alpha_1}{m_2}(\dot{x}_2 - \dot{x}_1) - \\ -\frac{\alpha_2}{m_2}\dot{x}_2 - F_f, \tag{2}$$

where m_i – mass of the each system body; x_1 , x_2 – the generalized co-ordinate characterizing location in the car of load and package respectively; k_1, α_1, n_1 – stiffness coefficient, resistivity factor and exponent at displacement respectively for connection between package and load, k_2, α_2, n_2 – same parameters for connection between package and car body; F_f – friction force between package and car floor. The point over a variable quantity means a time differentiation.

We supposed that cargo displacement should not exceed x_{max}, load mass m, initial velocity was v_0, load stopped at end. Expression of approximate value stiffness on exponent n was (3)

$$k = \frac{m v_0^2}{2} \cdot \frac{n+1}{x_{\max}^{n+1}}. \tag{3.1}$$

This system solution was executed with MathCAD by a Runge-Kutta method on the fourth order accuracy. There had been observed various cases differing by internal forces parameters for x_{max} = 0,05 m, v_0 = 2,5 m/s.

At computations without not elastic resistance forces there was installed that deformation exponent increasing result to increasing of cargo value acceleration, that at $n_1 = n_2 = 4$ attained 30g. A stiffness coefficient changing practically did not influence on displacement of a cargo relative to a package and essentially influenced on displacement of a package relative to the car floor. On the basis of computations it had been received, that soft characteristic of spring with an exponent n_1 from 0,3 to 1 was most comprehensible from the point of view of minimization cargo acceleration. Thus the increasing exponent n_2 led to acceleration increase.

In addition taking into account not elastic resistance forces had displayed a distinction change of dependence on the maximum load acceleration from an exponent n_2. In this case the maximum was clearly looking through on a diagram at some intermediate value n_2. For example, at n_1 = 0,3 load acceleration increased at exponent increase n_2 to value 2, and then decreased (fig. 2).

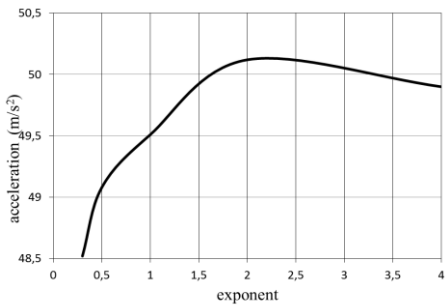

Figure 2. A cargo acceleration dependence on an exponent n_2 for the model with resistance forces.

Computation results had displayed a sharper load acceleration increment at dry friction coefficient increasing. The cargo acceleration least value (50,1 m/s^2) was attained at exponents $n_1 = 0{,}3$ and $n_2 = 2$ and coefficient of Coulomb friction between packaging and the car equal 0,2.

Thus the results analysis of computations on various initial values has allowed selecting the most rational parameters of fastening at which cargo displacement relative to the car does not exceed a legitimate value, and has minimum acceleration. The soft characteristic of elastic spring with exponents n_1 and n_2 from 0,3 to 1 is most comprehensible. Package and fastening with these

characteristics will allow securing cargo safety at transportation.

III. The Pipe Fastening Computation on Railway Car

A scheme design selection was an important stage that allowed evaluating the forces acting on the cargo, the platform and the fastening elements during device developing for cargo placement and fastening on a railway platform. The car and the fastening elements were influenced by cargo during car's impact and shunting operations, especially during unpick from sorting hills.

A platform with a cargo (fig. 3) presented a mechanical system with several freedom degrees. In the research, this complicated system was replaced by schemes with simple design and limited freedom degrees number. These schemes reflected the original scheme's basic properties and corresponded original goal.

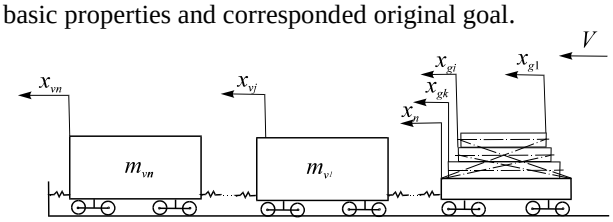

Figure 3. Scheme of the car impact with a group of motionless cars.

There was composed a differential equations system with using the d'Alembert method [11]. This equations system (3) was reflecting the motion of k cargo levels, moving platform and n cars for the presented scheme (fig. 3)

$$\left.\begin{aligned} &m_{gi}\ddot{x}_{gi} + T_{gi} + F_{gi} = 0; \\ &m_p\ddot{x}_p - T_p - F_{gk} + R_{v1} = 0; \\ &m_{vj}\ddot{x}_{vj} - R_{vj} + R_{vj+1} = 0; \end{aligned}\right\} \qquad (3.2)$$

where m_{gi}, m_p, m_{vj} – mass of the each cargo tier (i = 1, …, k), moving platform and cars (j = 1, …, n); $\ddot{x}_{gi}$, $\ddot{x}_p$, $\ddot{x}_{vj}$ – longitudinal acceleration of the each cargo level, moving platform and cars; T_{gi}, – longitudinal horizontal force projection in the elastic members of the cargo level fastening to the platform; T_p – total longitudinal horizontal force in the fastening's elastic elements of the cargo levels to the platform; F_{gi} – friction force between the cargo tiers and car floor; R_{vj} – force between cars.

The equation's system of (3) mattered when there was a transverse bandage and intermediate supports between the cargo levels in the device. In this case, the transverse bandage and intermediate supports were to be considered as the additional cargo levels. The cars initial impact velocity was taken into account in the computation.

The elasticity forces of the longitudinal fastening elements of the upper and lower cargo levels were determined (4)

$$T = c \cdot \Delta l \ \text{ at } \Delta l < 0; \quad T\text{=0, at } \Delta l\text{=0}. \qquad (4)$$

where c – rigidity coefficient of the longitudinal fastening elastic elements; Δl – longitudinal fastening elements elongation.

The forces between the cars with spring-friction absorbing devices were determined in direct ratio to the displacements which did not exceed the course of the absorbing apparatus by the equation from [12].

Numerical integration of the differential equations system (3) was carried out for different masses of a cargo at cars impact velocity of 5-9 km/h.

There was made a research of the effect cargo's dry friction forces on the device supports. And there was made a research of the fastening elements rigidity effect on the longitudinal displacement value of the cargo relative to the car. Also there was made a research of the fastening element's dynamic effects at different car impact velocity. At the same time, the rigidity coefficient's values were assumed to be the same and varied within 0,1–5 MN/m. The friction coefficient values between the cargo levels f_1 and the friction coefficient between the cargo and frame supports of the car f_2 were also assumed to be the same and varied within 0,2–0,6. Computations were carried out with the help of MathCAD Professional. They were presented in fig. 4.

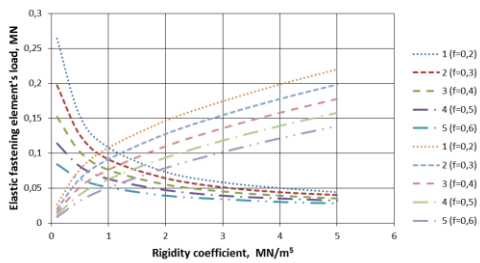

Figure 4. The cargo's lower level longitudinal displacement (downward curves) and the forces in fastening elastic elements (rising curves) at an impact velocity of the cars 5 km/h.

There were determined that the cargoes displacement and the forces in the elastic elements were practically equal during car's impact. Computations showed (fig. 4) that the biggest cargo displacement relative to the car and the force in the longitudinal fastening elements depend on the elastic fastening element's rigidity and the cargoes' friction coefficient on the supports. This dependence appeared at stiffness up to 1 MN/m in a high extent, and at stiffness above 1 MN/m - in a less extent. The friction coefficient was increased from 0,2 to 0,6 leads to the longitudinal displacement decrease of the cargo relative to the car. This led to a forces in the elastic fastening elements decrease in 2,4 times at a 0,5 MN/m rigidity and in 1,87 times - at a of 2 MN/m rigidity for a car impact velocity of 5 km/h. Rigidity reduction of the elastic elements (less than 0,5 MN/m) was accompanied by a significant increase in the cargo displacement relatively to the car during the car's impact. This could lead to the cargo output beyond the front of the car.

In addition, the computations of the cargo fastening devices for the railway car take into account the contact interaction features between structural parts [13]. The

stresses and deformations values in the fastenings details did not exceed the allowable values.

IV. Modeling Of The Piece Good's Case Paperboard Loading During A Boxcar's Impact

A. Finite Element Model

In this paper's part there was considered transportation of the paperboards with coils placed in several rows in width and height in the boxcar. Coils were cylindrical, boxes – rectangular. There were created computer models based on research work described in [14].

Places of the contact between parts had a complicated form. That's why the finite element method (FEM) based on ANSYS software was used in the stresses and deformations analysis of the engineering constructions.

The model was used for the pressure analysis of strapping acting the piece good's case paperboard. A case paperboard's part geometrical modelling was executed on the basis of initial data. This was the contact field of the strapping and the case paperboard (Fig. 5).

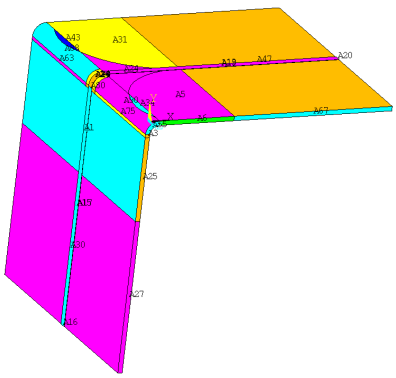

Figure 5. A case paperboard's quarter with strapping

The model analysis showed that the highest stresses occur in the contact field between the upper corners of the case paperboard and strapping.

There was made a computational scheme's development. On their basis there was made finite element deformation modeling of the upper corners of the piece good's paperboard mounted on a pallet.

Finite elements finer subdivision was executed in the places with contact parts of the case paperboard in order to determine the functional dependence of the contact pressure, the tangential contact stresses and slippage. Tetrahedral elements SOLID 187 and SOLID 185 were used to create the finite element mesh.

A packing material's and strapping's parameters were required for the calculations. Part of the mechanical properties was taken from the documentation of used materials. The second part was necessary to obtain by calculation.

To execute updated analysis of the packaging strength the definition of the other mechanical properties of packaging materials was required, because much of necessary for the calculations properties was not in normative documentation.

B. Experimental Determination Of Paperboard And Strapping Mechanical Properties

To execute calculations and refine the case paperboard's strength under the impact loads the experimental determination of the paperboard and strapping properties was done.

However, our test's results executing on the installation of INSTRON 5567 (fig. 6) showed that working normal tensile stress was 350 MPa and the breaking strength was 2572,5 N with the actual dimensions of the strapping cross-section (10,5×0,7 mm). Thus, the breaking strength's real value of the strapping was less than pointed in the documentation (2670 N). This could be one of the reasons for strapping break during a boxcar's impact during transportation. Also it was determined that the strapping actual value of the maximum elongation is 21 %. This value was set in documentation as 12 %. Thus, strapping had a lower stiffness than it's specified in documentation. This could led to a weakening of the strapping tension during a boxcar's impact and the subsequent package integrity's disruption.

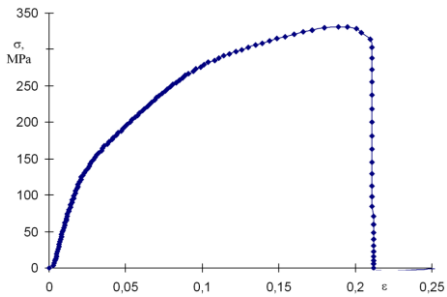

Figure 6. Diagram of the strapping *Tenax* 1718 stretching (cross-sectional dimensions of 0,7×10,5 mm).

Also experiments were performed to determine the case paperboard's mechanical properties of the brand P-35. Paperboard of this brand was used to make the packaging cases.

The tests were carried out for paperboard samples bending along and across the paperboard corrugations. Test results was shown in Fig. 7.

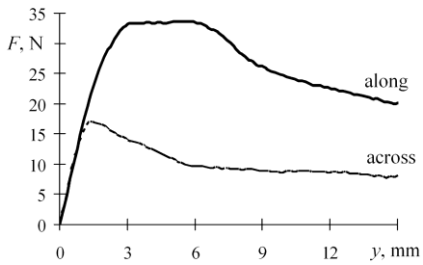

Figure 7. Deformation diagram of the sample paperboard bending along and across the paperboard corrugations.

Test result's analysis of the paperboard taken from the diagram (fig. 7) straight section was executing. In this area bending displacement values along and across the paperboard corrugations were the same. In this case elastic modulus value was obtained equal to E = 525,02 MPa.

The average values of the elasticity modulus were given in a modern handbooks and scientific literature [7]. They were adopted for the preliminary calculations. Elasticity modulus were specified in the along direction the paperboard – 3578,5 MPa, in the across direction of the paperboard's roll – 1367,8 MPa. As a result of our computations it was found that values for paperboard brand P-35 were closer to each other. This allowed using the material model with isotropic properties.

C. Finite Element Modeling Results

Computation results had shown the locations of the maximum stresses in the inflection place of the case paperboard. Fig. 8 showed this area with the stress scale. Same color corresponded to the same stress. Dangerous areas with damaged paperboard were allocated in the contact field of strapping with the inflection place of the case paperboard. Computations were carried out for different values of the boxcar's acceleration. They had shown that at acceleration value of 2,9g the case paperboard was damaged. The damage caused by the punching paperboard occurs according to National Standard 7376-89 for paperboard brand P-35 at a stress punching value of 1,6 MPa.

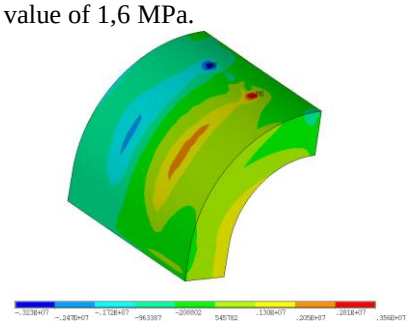

Figure 8. Tension τ_{yz} on the outside surface of the case paperboard.

It was clear from the figure that the paperboard will crush at the paperboard's top edge because of strapping during the impact. Paperboard would be destructed in this place.

Similar calculations were made for the case when adhesive tape was attached to the upper corners of the cardboard box under the binding strap. At the same time, stresses were reduced by 15-20%, which makes it possible to ensure the strength of the structure under impact loads.

V. Conclusions

Our studies displayed that additional stresses must be taken into account for developing of transporting fasteners for highly deformative cargoes. It is necessary to take into account the fasteners deformation and the cargo deformation at the same time.

The received results can be applied to development of new design of package and fastening securing cargo safe transportation.

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