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**V. G. Solonenko[1], N. M. Makhmetova[1], V. A. Nikolaev[2], M. Ya. Kvashnin[1], S. E. Bekzhanova[1], I. A. Bazanova[1], A. A. Malik[1]**

[1]Kazakh Academy of Transport and Communications named by M. Tynyshpaev, Almaty, Kazakhstan;
[2]Omsk State Transport University, Omsk, Russia.
E-mail: v.solonenko@mail.ru, makhmetova_n1958@mail.ru, NikolaevVA@omgups.ru,
kvashnin_mj55@mail.ru, s.bekzhanova@bk.ru, inna_bazanova@mail.ru, arman.kz_1677@mail.ru

# MODELING THE INFLUENCE OF THE ARTICLE DEVICE ON DYNAMIC INDICATORS OF FREIGHT WAGONS

**Abstract.** The article investigates the dynamic performance of a freight car with an articulation device - model SAC-1 RUS. As a criterion for assessing dynamic indicators, the acceleration of the car body and the dynamic deflections of the "spring-truck" system were selected, which characterize the ride and loss of stability of the car when it goes off the rails on various irregularities of the track.

When an articulated car moves along different path irregularities, the acceleration that occurs at various points along the length of the car is not the same. In compiling the system of differential equations of motion of the "spring - truck" system, the Lagrange equation of the second kind was used as the mathematical apparatus.

The body acceleration and the dynamic deflections of the springs were chosen as the criteria for assessing the running properties of the cars. The first characterizes the smoothness of the carriage, and the second characterizes the stability of the wheel, i.e. wheel derailment. It is known that the magnitude of the acceleration at different points along the length of the car is not the same, since the value of the vertical acceleration of bouncing is influenced by the vertical oscillations of the body's galloping. Therefore, when solving the system of differential equations, the body accelerations were determined at two points: above the center plates of the bogies and the center plate of the device for articulating freight cars or above the wheelsets, that is, in the places where acceleration sensors are installed during full-scale tests of cars.

Based on the analysis of the results obtained, it was revealed that articulated cars: exclude the possibility of relative vertical displacements of adjacent cars, with other equal dynamic parameters; lead to a noticeable deterioration in driving properties, accompanied by an increase in the value of vertical acceleration of the body. The values of accelerations and dynamic deflections of an articulated car are 6% higher than those of a four-axle freight car. The vertical acceleration of the front and rear of an articulated car is 30% greater than that of a typical car. Outside the critical speed for an articulated car, the value of vertical dynamic deflections significantly decreases in the case of an isolated irregularity with a length of 6 m or less.

**Key words:** articulated car, acceleration of the body, dynamic deflection, ride, articulation, unevenness of the path.

In 2019, a model articulated gondola car of increased payload capacity manufactured by Tikhvin Car-Building Plant JSC appeared on the CIS market. This gondola car model has a number of technical advantages, firstly, it improves transportation efficiency, and secondly, it provides a significant increase in the throughput of the railway network. Freight wagons should possess not only effective technical and economic characteristics, but also increased driving characteristics, stability and reliability. Therefore, it becomes necessary to conduct deep theoretical studies in order to determine the optimal dynamic performance of a freight car with an articulation device. Problems related to the study of the effective dynamic parameters of an articulated car taking into account the unevenness of the track are one of the urgent problems of transport technology and traffic safety in railway transport.

The aim of this work is to develop a mathematical model of a freight car with an articulation device that adequately describes the smoothness and stability of the car with various irregularities in the track, as well as to give an objective assessment of the dynamic performance of an articulated car based on the solution of differential equations of motion.

In the works of Russian scientists [1-3], the results of a study of the dynamic loading of a freight car on modernized bogies of increased carrying capacity are presented, where the maximum acceleration of the car body from the speed of movement, in an empty state, is determined.

A characteristic element of a modern wagon is a device for articulating freight wagons, which is supported by a biaxial trolley (see figure 1). Wagon sections connected by a device for articulating freight wagons of the SAC-1 RUS brand allow their mutual rotation in the horizontal and vertical plane. The device is designed to connect two successive sections of the carriage with support on one common trolley.

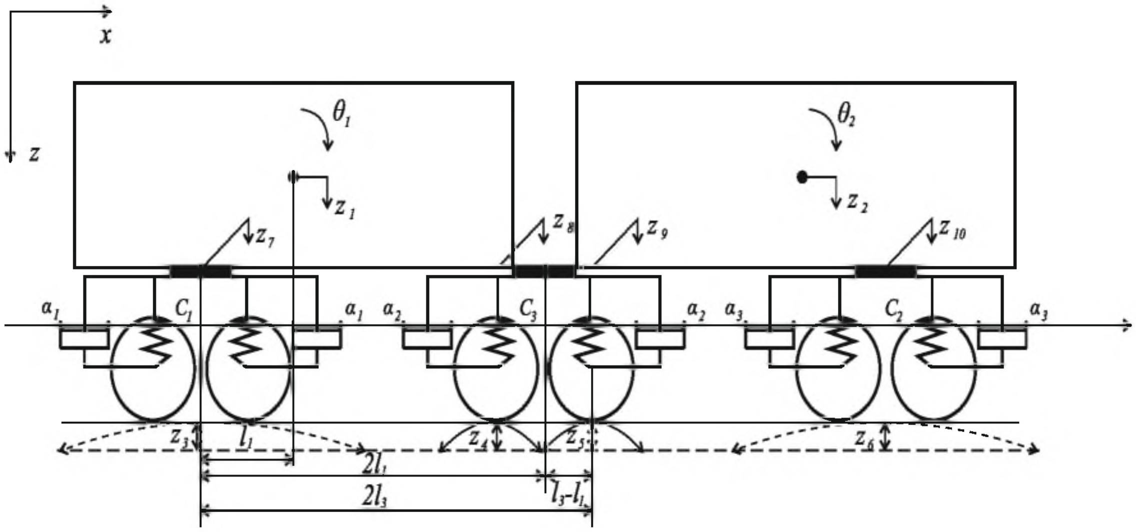


Figure 1 – Design diagram of an articulated wagon

The rotation of the sections of the car relative to the three coordinate axes is due to the presence of a spherical hinge in the structure. Consider the oscillations of an articulated car when moving along various irregularities of the path. Two types of bumps were considered, namely, periodic and isolated bumps at various lengths. Suppose that the car body is supported through the Fridays and the Friday of the articulation device on three trolleys with ordinary spring suspension. The rigidity of the carriage structure and the absorbing apparatus are taken into account through the rigidity of the articulation device, i.e. averaged stiffness is taken.

When studying plane vertical vibrations, an articulated car is considered as a special case of a generalized mechanical system, the mathematical model of which is defined as follows: an articulated car is a flat oscillatory system with degrees of freedom; suspension elements are characterized by independent parameters, i.e. mass and moment of inertia about the coordinate axes.

We assume that the wheelsets rotate relative to their own axes Oy and there is no translational movement relative to the side beams of the carts. The vertical movements of the two-part body are $z_1$ and $z_2$, respectively, and the galloping angles are - $\theta_1$ and $\theta_2$. The generalized coordinates are: $z_2$, $\theta_1$ and $\theta_2$.

The vertical displacement of the body $z_1$ is determined through the generalization of the coordinates using the expression [4]:

$$z_1 = z_2 - l_3(\theta_1 + \theta_2).$$

The positive directions of the displacements $z_1$, $z_2$ and the galloping angles $\theta_1$, $\theta_2$ are shown by arrows, the symbols $z_3$, $z_5$, $z_6$ are the irregularities of the path (see figure 1).

The movement of an articulated freight car along an absolutely rigid path is given by the second-order Lagrange equation

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_i}\right) - \frac{\partial T}{\partial q_i} + \frac{\partial \Pi}{\partial q_i} + \frac{\partial \Phi}{\partial \dot{q}_i} = Q_i, \quad (1)$$

where T - is the kinetic energy, П - is potential energy, Ф - is the dissipative function, $q_i$ - is the generalized coordinate, $\dot{q}_i$ - is the generalized speed, $Q_i$ – is the generalized force corresponding to the generalized coordinate $q_i$. The dissipative function, kinetic and potential energy are defined below by the relations

$$\Phi = \alpha_1\left[\dot{z}_2 - l_3\left(\dot{\Theta}_1 + \dot{\Theta}_2\right) - l_1\dot{\Theta}_1 - \dot{z_3}\right]^2 + \alpha_2\left(\dot{z}_2 - l_1\dot{\Theta}_2 - \dot{z_5}\right)^2 + \alpha_3\left(\dot{z}_2 - l_1\dot{\Theta}_2 - \dot{z_6}\right)^2, \quad (2)$$

$$T = \frac{1}{2}I_{oy}\dot{\theta}_1^2 + \frac{1}{2}I_{oy}\dot{\theta}_2^2 + \frac{1}{2}m\left[\dot{z}_2 - l_3\left(\dot{\theta}_1 + \dot{\theta}_2\right)\right]^2 + \frac{1}{2}m\dot{z}_2^2\,, \quad (3)$$

$$\Pi = c_1[z_2 - l_3(\theta_1 + \theta_2) - l_1\theta_1 - z_3]^2 + c_2(z_2 - l_1\theta_2 - z_5)^2 + c_3(z_2 - l_1\theta_2 - z_6)^2\,. \quad (4)$$

In expressions (2 - 4): $I_{oy}$- the central moment of inertia of the body; $c_1$, $c_2$, $c_3$ - stiffness of one spring set in kg/cm; $\alpha_1$, $\alpha_2$, $\alpha_3$– resistance coefficients of one hydraulic damper in kg·sec/cm.

To derive a system of differential equations of motion of an articulated car, we substitute relations (2-4) into equation (1), then after some transformations we obtain a system of differential equations in the form [5]:

$$\left.\begin{array}{c}
\left(I_{oy} + ml_3^2\right)\ddot{\theta}_1 + ml_3^2\ddot{\theta}_2 - ml_3\ddot{z}_2 + 2\alpha_1(l_1 + l_3)^2\theta_1 + 2\alpha_1 l_3(l_1 + l_3)\dot{\theta}_2 - 2\alpha_1(\dot{l_1} + l_3)\dot{z}_2 + \\
+2c_1(l_1 + l_3)^2\theta_1 + 2c_1 l_3(l_1 + l_3)\theta_2 - 2c_1(l_1 + l_2)z_2 = -2\alpha_1(l_1 + l_3)\dot{z}_3 - 2c_1(l_1 + l_3)z_3; \\
ml_3^2\ddot{\theta} + \left(I_{oy} + ml_3^2\right)\ddot{\theta}_2 - ml_3\ddot{z}_2 + 2\alpha_1 l_3(l_1 + l_3)\dot{\theta}_1 + (2\alpha_1 l_3^2 + 2\alpha_2 l_1^2 + 2\alpha_3 l_1^2)\dot{\theta}_2 + \\
+(-2\alpha_1 l_3 + 2\alpha_2 l_1 + 2\alpha_3 l_1)z_2^{\cdot} + 2c_1 l_3(l_1 + l_3)\theta_1 + (2c_1 l_3^2 + 2c_2 l_1^2 + 2c_3 l_1^2)\theta_2 + \\
+(2c_1 l_3 - 2c_2 l_1 + 2c_3 l_1)z_2 = -2\alpha_1 l_3\dot{z}_3 - 2\alpha_2 l_1\dot{z}_5 + 2\alpha_3 l_1\dot{z}_6 - 2c_1 l_3\dot{z}_3 - 2c_2 l_1\dot{z}_5 + 2c_3 l_1\dot{z}_6; \\
-ml_3\ddot{\theta}_1 - ml_3\ddot{\theta}_2 + 2m\ddot{z}_2 - 2\alpha_1(l_1 + l_3)\theta_1 + (-2\alpha_1 l_3 - 2\alpha_2 l_1 + 2\alpha_3 l_1)\ddot{\theta}_2 + \\
+(2\alpha_1 + 2\alpha_2 + 2\alpha_3)\ddot{z}_2 - 2c_1(l_1 + l_3)\theta_1 + (-2c_1 l_3 - 2c_2 l_1 + 2c_3 l_1)\theta_2 + \\
+(2c_1 + 2c_2 + 2c_3)z_2 = 2\alpha_1\dot{z}_3 + 2\alpha_2\dot{z}_5 + 2\alpha_3\dot{z}_5 + 2\alpha_3\dot{z}_6 + 2c_1\dot{z}_3 + 2c\,{}_2\dot{z}_5 + 2c_3\dot{z}_6;
\end{array}\right\} \quad (5)$$

Solving the system of differential equations (5) with respect to the highest derivatives, we obtain:

$$\left.\begin{array}{l}
\ddot{\theta}_1 = \frac{ml_1l_3^2 + 2I_{oy}l_1 + I_{oy}l_3}{2I_{oy}\left(I_{oy} + ml_3^2\right)}F_1 - \frac{l_3\left(ml_1l_3 + I_{oy}\right)}{2I_{oy}\left(I_{oy} + ml_3^2\right)}F_2 + \frac{l_3\left(ml_1l_3 - I_{oy}\right)}{2I_{oy}\left(I_{oy} + ml_3^2\right)}F_3, \\
\ddot{\theta}_2 = \frac{l_3\left(I_{oy} - ml_1l_3\right)}{2I_{oy}\left(I_{oy} + ml_3^2\right)}F_1 + \frac{I_{oy}(2l_1 - l_3) + ml_1l_3^2}{2I_{oy}\left(I_{oy} + ml_3^2\right)}F_2 - \frac{I_{oy}(2l_1 + l_3) + ml_1l_3^2}{2I_{oy}\left(I_{oy} + ml_3^2\right)}F_3, \\
\ddot{z}_2 = \frac{ml_1l_3 - I_{oy}}{2m\left(I_{oy} + ml_3^2\right)}F_1 + \frac{ml_1l_3 - 2ml_3^2 - I_{oy}}{2m\left(I_{oy} + ml_3^2\right)}F_2 - \frac{ml_1l_3 + 2ml_3^2 + I_{oy}}{2m\left(I_{oy} + ml_3^2\right)}F_3,
\end{array}\right\} \quad (6)$$

where:

$$F_1 = [z_2 - (l_1 + l_3)\theta_1 - l_3\theta_2 - z_2]2c_1 + \left[\dot{z}_2 - (l_1 + l_3)\dot{\theta}_1 - l_3\dot{\theta}_2 - \dot{z}_2\right]2\alpha_1;$$

$$F_2 = (z_2 - l_1\theta_2 - z_5)2c_2 + \left(\dot{z}_2 - l_1\dot{\theta}_2 - z_5\right)2\alpha_2;$$

$$F_3 = (z_2 + l_1\theta_2 - z_6)2c_3^{'} + \left(\dot{z}_2 + l_1\dot{\theta}_2 - z_6\right)2\alpha_3.$$

Based on a comparison of the structural elements of the articulated and typical carriage, the following geometric and mechanical characteristics were adopted: body length -19.540m, body height -3.2m, width -3.0m, weight -42t and the total static deflection at full load is 45mm. The parameters included in equations (5-6) take the following values: m = 39542 kg, $l_1$ = 5.21 m, $l_3$ = 6.38 m, $I_{oy}$ = 183· $10^4$ kg·m$^2$, $c_1$=$c_3$=394 Kg/cm, $c_2$=380 Kg/cm, $\alpha_1$=$\alpha_2$=26 Kg/cm, $\alpha_3$= 44 Kg/cm [6].

As criteria for assessing the running properties of cars, body accelerations and dynamic spring deflections were selected. The former characterizes the smooth running of the car, and the latter the stability of the wheel, i.e. wheel derailment [7]. It is known that the magnitude of the acceleration at different points along the length of the car is not the same, since the vertical vibrations of galloping of the body contribute to the value of the vertical acceleration of bouncing. Therefore, when solving the differential equations of motion, the accelerations of the body were determined at two points: above the Fridays of the trolleys and the Friday of the device for articulating freight wagons or above the wheelsets, that is, in places where acceleration sensors are installed during field tests of wagons. The depth of the roughness of the path, that is, the double amplitude of the sinusoid or the vertical distance between the lower and upper points of the roughness is denoted by $z_i$ (where i=3,4,5,6), which is 1 cm.

Figure 2 shows graphs of the dependence of the acceleration $\ddot{z}$ of the rear of the car on the speed of movement in the case of isolated bumps, respectively, with a length of 6, 12.5 and 25 m. Curves 1, 2 and 3 correspond to path irregularities of 6, 12.5 and 25 m in length. The acceleration value of an articulated wagon is 30% greater than that of a four-axle freight wagon.

Figure 3 shows the graphs of the dynamic deflections of the rear spring sets $z_\partial$ depending on the speed of movement in the case of an isolated roughness, respectively, with a length of 6, 12.5 and 25 m. Curves 1, 2 and 3 correspond to path irregularities of 6, 12.5 and 25 m in length. The magnitude of the dynamic deflections of an articulated wagon is 6% greater than that of a four-axle freight wagon.

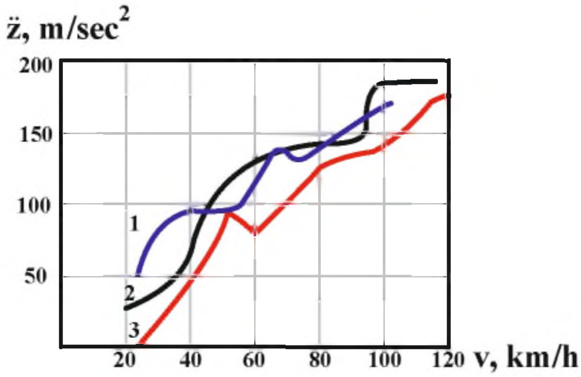


Figure 2 – Graphs of the dependence of the acceleration of the rear of the car on the speed in case isolated bumps. The curves correspond to: 1- the length of the roughness is 6 m; 2 - the length of the roughness is 12.5 m; 3- the length of the roughness is 25 m

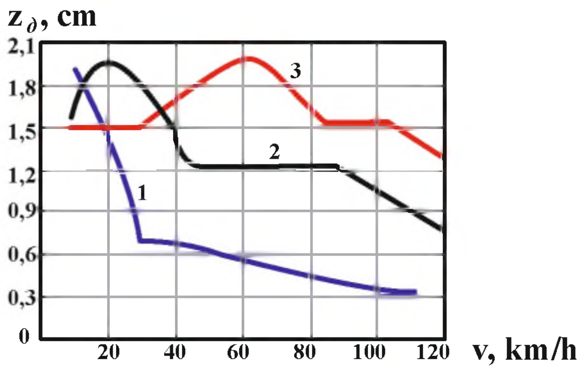


Figure 3 – Dynamic deflections of the rear spring sets depending on the speed of movement in the case of an isolated unevenness. The curves correspond to: 1- the length of the roughness is 6 m; 2 - the length of the roughness is 12.5 m; 3- the length of the roughness is 25 m

Figures 4-5 show graphs of the dependence of accelerations on speed with periodic irregularities of the path, respectively, for the front and rear parts of the car. Curves 1 and 2 correspond to irregularities of the path 12.5 and 25 m long. The values of the vertical accelerations of the front and rear of the articulated car are 30% greater than that of a typical car.

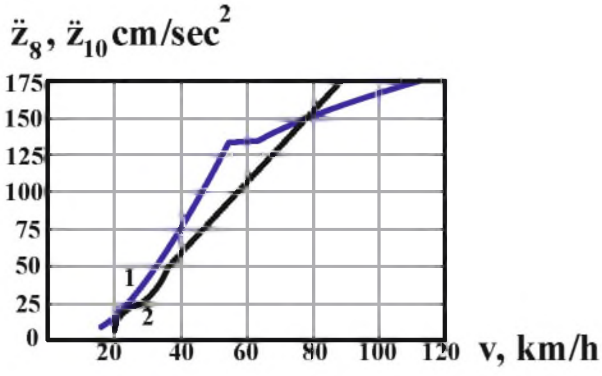


Figure 4 – Graph of the acceleration of the front parts of the car on the speed with periodic irregularities of the path. 1 - the length of the roughness is 12.5 m; 2- the length of the roughness is 25 m

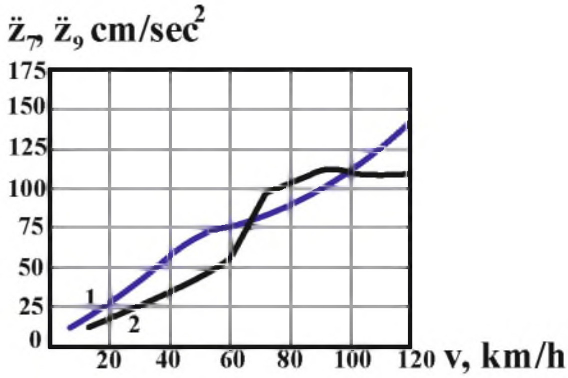


Figure 5 – Graph of the acceleration of the rear of the car from the speed. 1 - the length of the bumps is 12.5 m; 2- length of the roughness 25 m

In the case of isolated irregularities (lengths of 6, 12.5 and 25 m), the articulated car has a decrease in the value of vertical dynamic deflections with an increase in speed from 80 to 120 km / h. In the case of an isolated roughness of 6 m or less in length on an articulated car, the value of the vertical dynamic deflections significantly decreases beyond the critical speed.

In an articulated car, the vertical dynamic deflection reaches a maximum value of 2.0 cm in spring sets in the case of three irregularities, respectively, with a length of 6, 12.5 and 25 mm.

**Conclusions.** Based on the analysis of the results obtained, it was revealed that articulated wagons: exclude the possibility of relative vertical movements of adjacent wagons, ceteris paribus dynamic parameters; lead to a noticeable deterioration in driving properties, accompanied by an increase in the magnitude of the vertical acceleration of the body. The values of accelerations and dynamic deflections of an articulated wagon are 6% more than that of a four-axle freight wagon. It should be noted that the results obtained by the authors of [1-3] are in good agreement with the results of this work.

The vertical acceleration of the front and rear of an articulated car is 30% greater than that of a typical car. Outside the critical speed of an articulated car, the value of vertical dynamic deflections significantly decreases in the case of an isolated roughness of 6 m or less in length.

The analysis of the obtained results shows that the use of the articulation device of the SAC-1 RUS model worsens the dynamic characteristics of the articulated car, therefore, requires additional design solutions. Hence, as a consequence, it is necessary to conduct experimental studies in order to clarify the influence of dynamic loads on the track in accordance with the Technical Regulations for Traffic Safety in Railway Transport.

**В. Г. Солоненко[1], Н. М. Махметова[1], В. А. Николаев[2], М. Я. Квашнин[1], С. Е. Бекжанова[1], И. А. Базанова[1], А. А. Малик[1]**

[1]М. Тынышбаев атындағы Қазақ көлік және коммуникациялар академиясы, Алматы, Қазақстан;
[2]Омбы мемлекеттік жол қатынасы университеті, Омбы, Ресей Федерациясы

## ЖҮК ВАГОНЫНЫҢ ДИНАМИКАЛЫҚ КӨРСЕТКІШТЕРІНЕ МҮШЕЛЕУ ҚҰРЫЛҒЫСЫНЫҢ ӘСЕРІН МОДЕЛДЕУ

**Аннотация.** Мақалада моделі SAC-1 RUS мүшелеу құрылғысы бар жүк вагонының динамикалық көрсеткіштері зерттелген. Динамикалық көрсеткіштердің бағалау критерийі ретінде вагон шанашағының үдеу және «рессор-арба» жүйесінің динамикалық иілу сипаты алынған, олар қозғалыстың бірқалыптылығын және вагонның рельстен шығып кетпеу жағдайын сипаттайды.

Мүшеленген вагонның әртүрлі тегіссіздік бойымен қозғалғанда оның ұзындығы бойынша түрлі нүктелерде пайда болатын үдеу жағдайы бірдей болмайды. «Рессор - арба» жүйесінің дифференциалдық теңдеулер жүйесін құрастыруда математикалық аппарат ретінде екінші ретті Лагранж теңдеуі қолданылды. Вагон қозғалысының дифференциалдық теңдеулер жүйесінің шешімінен арба табанының үстіндегі және мүшелеу құрылғыларының үдеу сипаты, яғни далалық сынақ жүргізу кезінде үдеу датчиктері орнатылатын нүктелерде анықталды.

SAC-1 RUS маркалы мүшелеу қондырғысымен қосылған жүк вагондарының секциялары олардың көлденең және вертикаль жазықтықта өзара айналуына мүмкіндік береді. Құрылғы бір жалпы арбада тіреуіші бар екі тізбектей орналасқан вагон секцияларын қосуға арналған.

Вагон секцияларының үш координаталық оське қатысты бұрылуы құрылымда болатын сфералық шарнирге байланысты. Тегіссіздіктің екі түрі қарастырылды, атап айтқанда түрлі ұзындықтағы периодты және оқшауланған жол тегіссіздігі. Вагон құрылымының және жұту аппаратының қаттылығы мүшелеу құрылғысының қаттылығы арқылы ескеріледі, яғни орташа қаттылық алынады.

Доңғалақ жұптары өзінің Оу осіне қатысты айналады және арбалардың бүйірлік арқалығына қатысты ілгерлемелі орын ауыстырмайды. Екі бөліктен тұратын шанашақтың вертикаль орын ауыстыру жағдайы – $z_1$ және $z_2$, ал шоқырақтау бұрыштары – $\theta_1$ және $\theta_2$. Жалпыланған координаттар ретінде $z_2$, $\theta_1$ және $\theta_2$алынды.

Вагонның жүргізгіш қасиеттерін бағалау критерийлері ретінде шанашақтың үдеуі мен рессордың динамикалық иілуі алынды. Біріншісі вагон қозғалысының бірқалыптығын сипаттайды, ал екіншісі дөңгелек тұрақтылығын, яғни дөңгелектің рельстен шығуын сипаттайды. Вагон ұзындығы бойымен түрлі нүктелердегі үдеу шамасы бірдей емес екендігі белгілі, өйткені шанашақтың вертикаль шоқырақтау тербелісі вертикаль секіру үдеуінің сан мәніне әсер етеді. Сондықтан қозғалыс дифференциалдық теңдеуін шешкен кезде дененің үдеу үдерісі екі нүктеде анықталды: арба табандарының үстінде және жүк вагондарының мүшелеу құрылғысының табанында немесе жұп доңғалақтарының үстінде, яғни вагондарды далалық сынақтан өткізу барысындағы үдеу датчиктері орнатылған жерлерде.

Алынған нәтижелерді талдау негізінде мүшеленген вагондардың: брдей динамикалық параметрлі көршілес вагондардың салыстырмалы вертикаль орын ауыстырмайтындығы анықталды; шанақтың вертикаль үдеуінің сан мәнінің өсуі жүру сипатын айтарлықтай нашарлатады. Мүшеленген вагонның үдеуі мен динамикалық иілуінің сан мәні төрт осьті жүк вагонына қарағанда 6% артық. Мүшеленген вагонның алдыңғы және артқы бөліктерінің вертикаль үдеуінің сан мәні типтік вагонға қарағанда 30% үлкен. Ұзындығы 6 м немесе одан да аз оқшауланған тегіссіздікте критикалық жылдамдықтың шегінен асқанда, мүшеленген вагонның вертикаль динамикалық иілу үдерісі айтарлытай азаяды.

**Түйін сөздер:** мүшеленген вагон, шанақтың үдеуі, динамикалық иілу, қозғалыстың бірқалыптылығы, мүшелеу құрылғысы, жолдың тегіссіздігі.

**В. Г. Солоненко[1], Н. М. Махметова[1], В. А. Николаев[2], М. Я. Квашнин[1], С. Е. Бекжанова[1], И. А. Базанова[1], А. А. Малик[1]**

[1]Казахская академия транспорта и коммуникаций им. М. Тынышпаева, Алматы, Казахстан;
[2]Омский государственный университет путей сообщения, Омск, Россия

## МОДЕЛИРОВАНИЕ ВЛИЯНИЯ УСТРОЙСТВА СОЧЛЕНЕНИЯ НА ДИНАМИЧЕСКИЕ ПОКАЗАТЕЛИ ГРУЗОВЫХ ВАГОНОВ

**Аннотация.** В статье исследуются динамические показатели грузового вагона с устройством сочленения – модели SAC-1 RUS. В качестве критерия оценки динамических показателей выбраны ускорения кузова вагона и динамические прогибы системы «рессора – тележка», которые характеризуют плавность хода и устойчивость вагона от схода с рельсов при различных неровностях пути.

При движении сочлененного вагона по различным неровностям пути ускорение, возникающее в различных точках по длине вагона, не одинаково. При составлении системы дифференциальных уравнений движения системы «рессора – тележка» в качестве математического аппарата использовано уравнение Лагранжа 2-го рода. Из решения системы дифференциальных уравнений движения вагона определялись ускорения над пятниками тележки и устройством сочленения, т.е. в точках где, как правило, устанавливаются датчики ускорения при натурных испытаниях.

Секции вагона, соединенные устройством для сочленения грузовых вагонов марки SAC-1 RUS, допускают их взаимный поворот в горизонтальной и вертикальной плоскости. Устройство предназначено для соединения двух последовательно расположенных секций вагона с опиранием на одну общую тележку.

Поворот секций вагона относительно трех осей координат происходит за счет наличия в конструкции сферического шарнира. Рассматривались два типа неровности, а именно периодическая и изолированная неровность пути при различных длинах. Жесткость конструкции вагона и поглощающего аппарата учитываются через жесткость устройства сочленения, т.е. берется осредненная жесткость.

Колесные пары вращаются относительно собственных осей Оу и отсутствует поступательное перемещение относительно боковых балок тележек. Вертикальные перемещения кузова, состоящего из двух частей, соответственно, $z_1$ и $z_2$, а углы галопирования - $\theta_1$ и $\theta_2$. В качестве обобщенных координат приняты: $z_2$, $\theta_1$ и $\theta_2$.

В качестве критериев оценки ходовых свойств вагонов выбраны ускорения кузова и динамические прогибы рессор. Первые характеризуют плавность хода вагона, а вторые – устойчивость колеса, т.е. сход с рельсов колеса. Известно, что величина ускорения в различных точках по длине вагона неодинакова, поскольку на значение вертикального ускорения подпрыгивания вносит вклад вертикальные колебания галопирования кузова. Поэтому при решении дифференциальных уравнений движения ускорения кузова определялись в двух точках: над пятниками тележек и пятником устройства для сочленения грузовых вагонов или над колесными парами, то есть в местах, где устанавливаются датчики ускорения при натурных испытаниях вагонов.

На основе анализа полученных результатов выявлено, что сочлененные вагон: исключают возможность относительных перемещений по вертикали соседних вагонов, при прочих равных динамических параметрах; приводят к заметному ухудшению ходовых свойств, сопровождающихся ростом величины вертикальных ускорений кузова. Величины ускорений и динамических прогибов сочлененного вагона больше на 6%, чем четырехосного грузового вагона. Вертикальные ускорения передней и задней части сочлененного вагона больше на 30%, чем у типового вагона. За пределами критической скорости у сочлененного вагона величина вертикальных динамических прогибов существенно уменьшается в случае изолированной неровности длиной 6м и менее.

**Ключевые слова:** сочлененный вагон, ускорение кузова, динамический прогиб, плавность хода, устройство сочленения, неровность пути.

**Information about authors:**

Solonenko Vladimir Gelevich, Professor, Doctor of Technical Sciences, Kazakh Academy of Transport and Communications named by M. Tynyshpaev, Almaty, Kazakhstan; v.solonenko@mail.ru; https://orcid.org/0000-0001-6503-6598

Makhmetova Narzankul Musaevna, Professor, Doctor of Technical Sciences, Kazakh Academy of Transport and Communications named by M. Tynyshpaev, Almaty, Kazakhstan; makhmetova_n1958@mail.ru; https://orcid.org/0000-0001-7324-5832

Nikolaev Victor Alexandrovich, Professor, Doctor of Technical Sciences, Omsk State Transport University, Omsk, Russia; NikolaevVA@omgups.ru; https://orcid.org/0000-0002-0850-1796

Kvashnin Mikhail Yakovlevich, Associate Professor, Candidate of Technical Sciences, Kazakh Academy of Transport and Communications named by M. Tynyshpaev, Almaty, Kazakhstan; kvashnin_mj55@mail.ru; https://orcid.org/0000-0002-3969-9299

Bekzhanova Saule Ertaevna, Professor, Doctor of Technical Sciences, Kazakh Academy of Transport and Communications named by M. Tynyshpaev, Almaty, Kazakhstan; s.bekzhanova@bk.ru; https://orcid.org/0000-0001-6272-9567

Bazanova Inna Amandykovna, Associate Professor, Doctor of Technical Sciences, Kazakh Academy of Transport and Communications named by M. Tynyshpaev, Almaty, Kazakhstan; inna_bazanova@mail.ru; https://orcid.org/0000-0003-1899-0092

Malik Arman Aydinovich, Assistant-Teacher, Kazakh Academy of Transport and Communications named by M. Tynyshpaev, Almaty, Kazakhstan; arman.kz_1677@mail.ru; https://orcid.org/0000-0003-3651-1702

## REFERENCES

[1] Luks D.Yu. Improving the reliability of freight cars in operation // Railway transport, M. N 11, 2013. P. 52-56 (in Russ.).

[2] Nekhaev V.A. Investigation of the influence of the parameters of the nonlinear system "car-track" on the dynamics of the railway crew // Izvestia Transsib, Omsk. 2017. N 3 (31). P. 6-78. (in Russ.).

[3] Petrov V.V. Mathematical model for assessing the impact of deviations of the design parameters of the carriage bogie from the nominal values on its dynamic properties // Izvestia Transsiba, Omsk. N 2 (38), 2019. P. 55-65 (in Russ.).

[4] Musayev Janat, Vladimir Solonenko, Narzankul Mahmetova, Mikhail Kvashnin, Azamat Alpeisov, Algazy Zhauyt. Some aspects of the experimental assessment of dynamic behavior of the railway track // Journal of theoretical and applied mechanics. Vol. 55, Issue 2. P. 421-432. Warsaw 2017 (in Eng.).

[5] Solonenko V.G., Makhmetova N.M., Musaev J.S., Bekzhanova S.E., Kvashnin M.Ya. The method of limiting speed when passing turnouts of railway vehicles with bogies of zk-1 // News of the National academy of sciences of the Republic of Kazakhstan. Series of geology and technical sciences. 2019. Vol. 1, N 433. P. 151-162 (in Eng.). https:doi.org/10.32014/2019.2518-170X.19

[6] Solonenko V.G., Makhmetova N.M., Musayev J.S., Bekzhanova S.E., Kvashnin M.Ya. Stresses in elements of metal railway bridges under the action of the crew // News of the National Academy of Sciences of the Republic of Kazakhstan. Series of geology and technical sciences. Vol. 2, N 434. 2019. P. 159-165 (in Eng.). https://doi.org/10.32014/2019.2518-170X.50

[7] Solonenko V.G., Makhmetova N.M., Nikolaev V.A., Kvashnin M.Ya., Bekzhanova S.E., Bondar I.S., Mirzabaev S.A. Analysis of the stressed-deformed state of travel pipes with the use of hardware and software complex // News of the National Academy of Sciences of the Republic of Kazakhstan. Series of geology and technical sciences. Vol. 1, N 439. 2020. P. 181-188 (in Eng.). https://doi.org/10.32014/2020.2518-170X.22