Research of friction indices influence on the freight car dynamics

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S u m m a r y . The work is devoted to study of friction indices influence in the "body – bogie" system in freight cars (the change of the friction coefficient in the side bearers during operation) on their basic indices – the coefficients of the horizontal and vertical dynamics, vehicle body acceleration, frame strength, and derailment stability coefficient. The study was conducted by mathematical modeling of the freight car dynamic loading using the software package «DYNRAIL». The focus was mainly concentrated on the influence of the friction force change between the body and bogies. Theoretical studies of dynamic loading of the freight open cars were conducted in both the empty and loaded conditions with the bogies model 18–100 during motion in the tangent and curved railway sections with different radii and the established motion speeds. In this case, the basic dynamic parameters of the freight car were calculated. The theoretical research results in determination of freight car dynamic indices taking into consideration the friction coefficients in the system "body – bogie" allow for an adequate assessment of the friction coefficient between the freight car bearers parameters, on the railway traffic safety factors (coefficients of dynamics, vehicle body acceleration, frame strength and derailment stability coefficient).

K e y w o r d s . Freight cars, bearers, dynamic parameters.

INTRODUCTION

The railway transport in developed countries plays an important role in social and economic life of the country and carries out the large containment of transportation activities [3, 4, 7].

This motivates the transport industry to move towards innovation changes and increase its significance as an important transit subsystem on the way of renovation of both the infrastructure and the strategy of all transportation process components including the interaction with other transport modes [15, 18].

First of all the basic areas of the railway industry activity are the following:

– development of the high-speed train traffic,

– improvement of the road safety,

– development of new rolling stock and modernization of the existing fleet.

PROBLEM DEFINITION

According to the previous publications [6, 11], the researches of freight car dynamics is a complex theoretical problem. Its purpose is the definition of permissible and safe speeds in terms of interaction between wheels and rails.

Due to the urgency of this subject one should cope with the task to study the effects of various factors and characteristics of technical conditions of the freight cars running gears (which are unavoidable to arise during operation) on their basic dynamic indices. Among these factors the system "body – bogie" plays an important role. This paper focuses on the impact of the friction force changes between the body and bogies [24, 25].

Theoretical studies were conducted by the mathematical modeling of dynamic loading of the open car in both the empty and loaded conditions with the bogies TsNII-Kh3 (model 18 – 100) during motion in the tangent and curved railway sections with different radii and the established motion speeds. In this case, the basic dynamic parameters of the freight car were calculated [2, 9].

Mathematical modeling of the freight car dynamic loading was carried out using the software package «Dynamics of Rail Vehicles» («DYNRAIL») [15, 16, 17] developed in Dniepropetrovsk National University of Railway Transport named after Academician V. Lazaryan.

THEORETICAL BACKGROUND

The first studies on sliding friction were conducted by Kulon and were repeated by Moren [1, 26]. The founder of the theory of friction with lubrication is a scientist M. Petrov. Further the theory was developed in the writings of M. Zhukovsky and other researchers.

As we know, there are two cases of sliding friction:

1) friction at rest and, in particular, the friction at the start of motion,

2) friction in motion.

When the system is in equilibrium state, the pressure of horizontal surface on the element has a resultant *N*, normal to the surface, which is equal and opposite to the weight P of the body with the weight (Fig. 1, a).

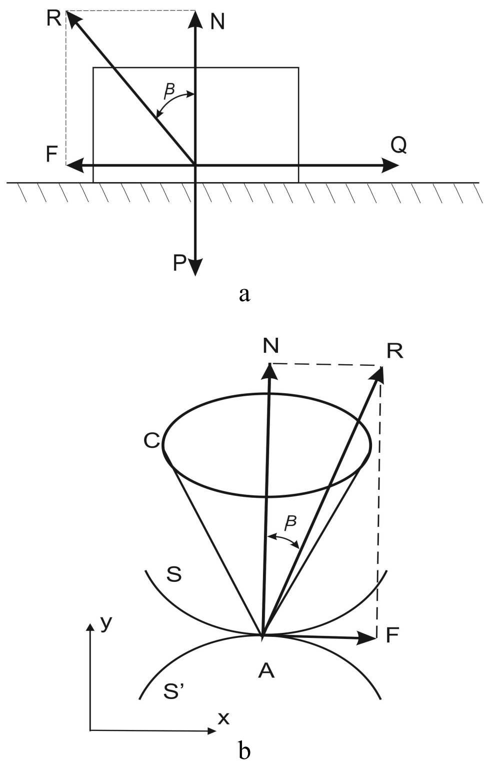


Fig. 1. Schematic illustration of the forces between bodies including friction: a – at rest, b – with one contact point – friction cone

The reaction of surface *R* on the body is equal and opposite to the resultant of weight *P* and the applied horizontal force *Q*. This reaction is decomposed into two parts: the normal *N*, equal and directly opposite to the force *P* and the tangential *F*, which is equal and opposite to the force *Q*. Tangential component is the force of friction. For the angle *β* between the reaction *R* and the normal *N* we have:

$$tg\beta = \frac{F}{N} = \frac{Q}{P}. \qquad (1)$$

If one gradually increases the *Q*, then there is a moment when the force reaches the value F_p, at which body is put into motion. The corresponding numerical value F_p of the force *F* is called the friction at the start of motion. The corresponding value φ of the angle β for which:

$$tg\varphi = \frac{F_p}{P}, \quad (2)$$

is the angle of friction.

Sliding starts from the moment when the resultant of forces *P* and *Q*, applied to the body, makes an angle with normal that exceeds φ.

Kulon measured the values F_p and φ during the experiment, on the results of which he deduced three laws [1, 26]:

1. Friction at the start of motion does not depend on the area of surfaces that are in contact.

2. It depends on the nature of these surfaces.

3. It is proportional to the normal component of the reaction, or the normal component of the pressure.

Constant correlation of the friction force F_p at the start of the motion to the normal reaction *N*, or to the normal pressure *P*, is the friction coefficient *f*:

$$f = \frac{F_p}{N} = \frac{F_p}{P}. \quad (3)$$

The angle of friction φ is determined by the formula:

$$tg\varphi = f. \quad (4)$$

In practice, equilibrium of bodies with friction at one contact point is the prevalent case. In this case it is considered the body *S* (Fig. 1, b) laid on another body *S′* with which it has a contact on the very small part of the surface [1, 26]. It is assumed that the latter reduced to one point A. The reaction *R* of the body *S′* on the body *S* is the normal reaction *N* and the tangent reaction *F*, the direction of which is unknown and the maximum is equal to $f \cdot N$. The angle β between *R* and *N* will be less than the friction angle φ. In order to bring the body *S* to equilibrium, a balance between the forces applied directly to the body *S* and the reaction *R* is necessary, or the forces applied to the body should have a resultant equal and directly opposite to the force *R*, that is:

a) passing through the point *A*,

b) directed in the way to press the body *S* to the body *S′*,

c) making an angle with the normal *AN*, which is less than the angle of friction.

These necessary demands are sufficient and if they are met, it is possible to assume that the resultant of the applied forces is moved directly to the point *A* and is decomposed into two forces: the normal force *P* and the tangent force *Q*. Under the influence of these forces the sliding is absent, as the angle of the resultant with the normal is less than φ, therefore (Fig. 2, b):

$$\frac{Q}{P} < f, \; Q < f \cdot P, \quad (5)$$

and the tangential component is less than the friction at the start of the motion. If the cone of revolution with the axis *AN*, which makes an angle φ with *AN* is considered, then to put it to equilibrium it is necessary and sufficient the forces had the resultant, the direction of which passes through the points *A* and *C*, lying in the middle of the cone.

From the previous considerations one can conclude that any applied to the body force, which passes through the point *A* and makes an angle with the normal, less than φ, i.e. the force lying in the middle of the cone *C* is balanced by the reaction of the body, since this force can be decomposed as we described above.

To calculate the angle of friction let us set up a static equations, from which we obtained:

$$R_1 = R \cdot \cos\beta = N, \quad (6)$$

$$R_2 = R \cdot \sin\beta = F_{sr} \le F_{sr}^{\lim} \le N \cdot f = \cos\beta, \quad (7)$$

where: R_1, R_2 are the components R.

After performing the arithmetic operations we can see that:

$$R \cdot \sin\beta \le R \cdot f \cdot \cos\beta\,, \tag{8}$$

$$tg\beta \le \beta\,. \tag{9}$$

Friction angle – is the force angle with *N*, the tangent of which is equal to the coefficient of friction:

$$tg\phi_{sr} = f\,,\ \beta \le \phi_{sr}\,. \tag{10}$$

Coefficient of friction f – is a dimensionless quantity, it is determined empirically and depends on the material of the contacting bodies and surfaces (character of treatment, temperature, humidity, etc.).

The coefficient of friction f_0 for some materials:

– wood friction 0,4 – 0,7,

– metal friction 0,15 – 0,25,

– steel on ice friction – 0,027.

In the case of motion, it is assumed that it is moving the solid body, limited by some surface and contacting with another body in the point. If there is a friction, the reaction of one body on the second is decomposed into two forces: the normal *N*, which is called a normal reaction, and tangent *F*, which is the force of friction and is subject to the following three laws:

1. The friction force is directed to the side opposite to the relative velocity of the material pointing relation to the body surface.

2. It does not depend on the velocity magnitude.

3. It is proportional to the normal reaction: $F = f \cdot N$, coefficient f is the friction coefficient at the start of the motion.

According to Hertz's experiments, these laws can be applied mainly in the case of direct friction (i.e., when the friction surfaces are dry). They should be changed if the surfaces have lubricants. In this case the ratio F/N depends on the speed and the force N [26]. During engineering calculations one usually comes from the number of empirically established laws that with the sufficient accuracy reflect the basic features of the friction phenomenon.

Dynamic coefficient of the sliding friction f is also the dimensionless quantity and is determined empirically. The coefficient value depends not only on the material and condition of the surfaces, but also, to some extent, on the speed of moving bodies. In most cases, when the velocity increases, the coefficient f initially decreases and then keeps almost a constant value [19].

MAIN PART

Establishment of admissible car speeds in tangent and curved track sections is a challenging engineering task that requires a differentiated approach and takes into account the technical condition of the track superstructure (TS) and the running gears of the rolling stock [8, 12]. Permissible speeds were determined at the results of comparison of the obtained dynamic parameters with the permissible values according to the “Standards” [10].

Among all the friction pairs in determining the dynamic loading of the freight cars the friction in the system "body – bogie" is one of the dominants. It is the study of the system in connection with the technical condition of the freight cars running gears and determination of their basic dynamic parameters this research is dedicated to [20, 21, 22, 23].

The bearing connection of the body and bogies is the most important subsystem of the freight car. Dynamic and other technical and economic characteristics of the car depend on the correct choice of structural schemes and parameters of this subsystem. The car body oscillates during the motion and makes angular rotations relative to the vertical, longitudinal and transverse axes. The main bearing connection of the body and bogie is center plate – center bowl, in which is realized the friction torque, preventing rotation of the bogie around the vertical axis, as well as frictional forces, preventing movement along the longitudinal axis of the body and along the transverse axis [13].

The main functional purpose of the body bearers and the bolster is to prevent excessive swaging of the body center bowl of the bogie and to reduce the wabbling. In this moment the friction in the bearing connection center plate – center bowl – bearers should not exceed certain values in order to avoid the excessive impact on the track, wheel sets and axle box truck [27, 28].

RESULTS OF RESEARCHES

In the study of the friction influence on the dynamic loading of the car several consitions considered:

– the normal condition, at which the damping factor is taken as 1,

– the condition of the low friction, which arises in the bogie construction when the gib is higher as compared to the normal condition, in that case the coefficient φ is taken as 0,2 or 0,5,

– overdamped condition of the system, at which the coefficient φ is taken as 1,5,

– full absence of the friction in the system, at which the coefficient φ is taken as 0.

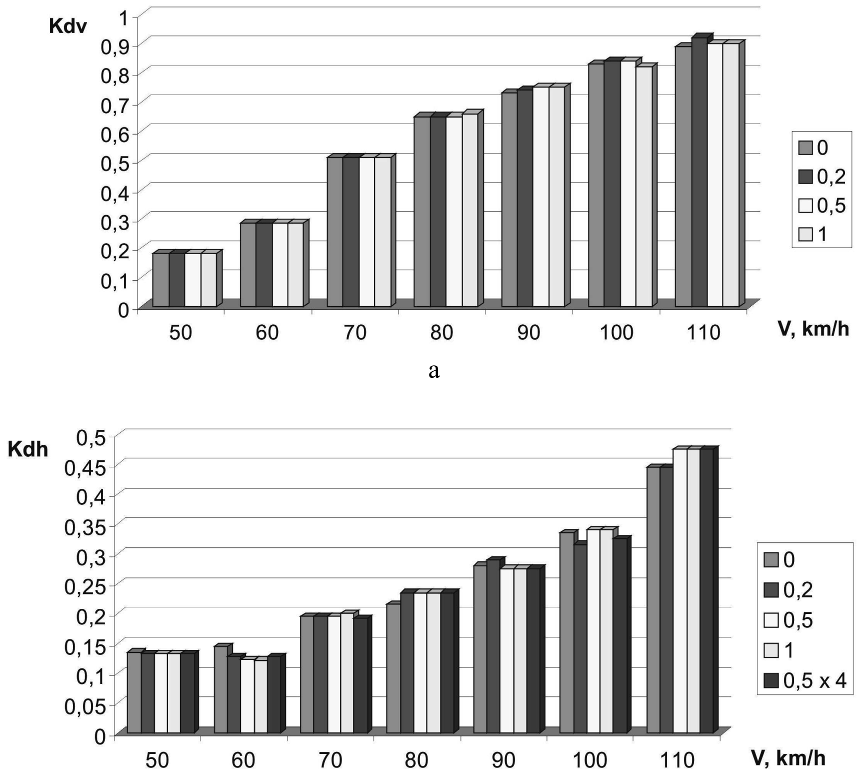
As a result of the calculations the dependency diagrams of the basic dynamic parameters were constructed (Fig. 2):

– coefficients of the vertical and horizontal dynamics,

– frame strength,

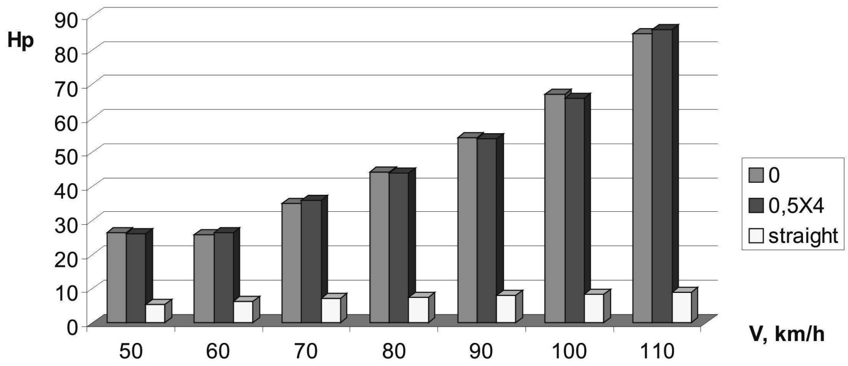
– coefficient of resistance,

– horizontal and vertical acceleration of the four–axle freight open car body, taking into account the speed of motion from the friction coefficient of in the system "body – bogie".

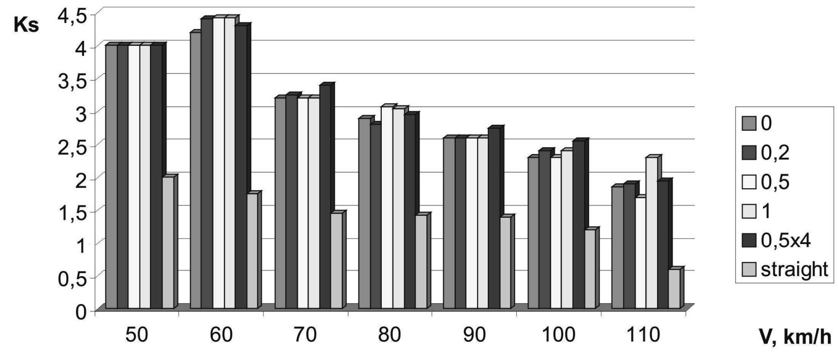


a

b



c



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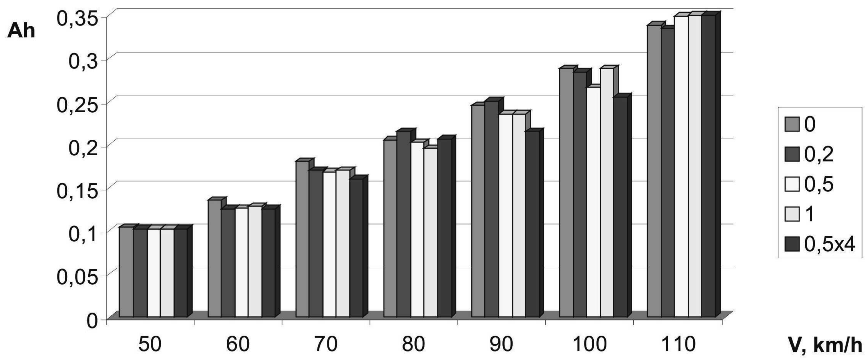
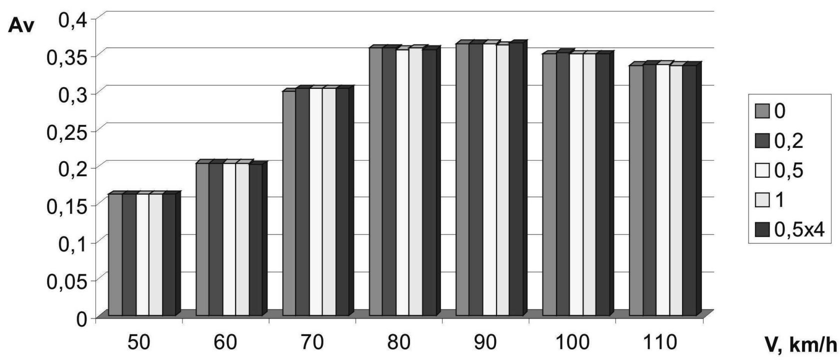


e



f

Fig. 2. Change diagram of dynamic performance from the motion speed:

a – the coefficient of vertical dynamics, b – the coefficient of horizontal dynamics, c – frame strength, d – stability coefficient, e – horizontal acceleration of the body, f – vertical acceleration of the body

From the resulted diagrams one can see that the basic dynamic performance of the four–axle freight open car is not significantly depends on the friction coefficient and at the motion speed up to 100 km/h within the allowable values, corresponding to the "Standards" [10].

The theoretical calculations allow concluding that the friction in the bearing connection center plate–center bowl–bearer of the freight open car in empty and loaded conditions with the bogies TsNII–Kh3 (model 18 – 100) doesn't have a significant impact on the road safety performance. It is the radii of the curved track sections and the outer rails height, etc influence the road safety performance.

CONCLUSIONS

1. As a result of researches the dependencies of the basic dynamic coefficients of the four–axle open car from the friction coefficient in the system "body – bogie", taking into account the motion speed were obtained.

2. Thus, the obtained results of calculations allow evaluating the impact of the technical condition of the car running gears on the road safety performance.

REFEReNCEs

1. **Appel P., 1960.:** Static. Dynamics of a point. Gos. publishing physical and mathematical literature. Moscow, 515. (in Russian).
2. **Belodedov V., Nosko P., Fil P., Stavitskiy V., 2007.:** Parameter optimization using coefficient of variation of intervals for one-seed sowing apparatus with horizontal disk during maize seeding. TEKA Commission of Motorization and Power Industry in Agriculture, V. VII, 31-37.
3. **Dailidka S., Myamlin S., Lingaitis L., Neduzhaja L., Jastremskas V., 2011.:** Renewal of locomotive stock of Lithuanian Railways. Proc. Donetsk Railway Transport Institute, issue 28, 174-179 (in Russian).
4. **Dailidka S., Myamlin S., Lingaitis L, Neduzhaja L., Jastremskas V., 2012.:** Innovative solutions in creating a mainline locomotive for railways in Lithuania. Bulletin of the East Ukrainian National University named after V. Dahl. Issue 3, 52-58 (in Russian).
5. **Danovich V., Korotenko M., Myamlin S., Neduzhaja L., 1999.:** Mathematical model of spatial oscillations of electric locomotive with the modernised scheme of body and bogies connection: collection of scientific papers. Transport. Increase of operating efficiency of electric transport equipment. Interuniversity collect. of sc. papers, DGTURT, Dniepropetrovsk, 182-189 (in Russian).
6. **Danovich V., Malysheva A., 1998.:** Mathematical model of spatial oscillations five cars connection moving along a straight track section: collection of scientific papers. Transport. Stress loading and durability of a rolling stock: collect. of sc. papers, DGTURT, Dniepropetrovsk: "Science and education", 62-69 (in Russian).
7. **Danovich V., Myamlin S., Neduzhaja L., 2000.:** Overview of solutions undercarriage design of certain types of locomotives. Dniepropetrovsk, ITM. Issue 2, 111-119 (in Russian).
8. **Danovich V., Rybkin V.,. Myamlin S., Reydemeister A., Tryakin A., Halipova N., 2003.:** Determination of permissible speeds of freight cars movement along railway tracks with 1520 mm gauge. Bulletin DGTURT. Issue 2, p. 77-86 (in Russian).
9. **Golubenko A., Malohatko A., Klyuev S., Klyuev A., 2011.:** The application review on the rolling stock of devices for turn of wheel pairs in the horizontal plane. TEKA Kom. Mot. i Energ. Roln: Ol pan, 11a, 5-11.
10. **Instruction on examination, service, overhaul and make up of wheelsets. 2006.:** CV–CL–0062. Kyiv, Ukrzaliznytsa, 108 (in Ukrainian).
11. **Lazaryan V., 1964.:** Dynamics of cars. Moscow, Transport, 256. (in Russian).
12. **Litvin V., Myamlin S., Malysheva A., Neduzhaja L., 1994.:** Dynamic parameters of some cars types. Mechanics of transport: train weight, speed, safety of movement. Interuniversity collect. of sc. Papers. Dniepropetrovsk, DIIT, 95-104 (in Russian).
13. **Lukhanin M., Myamlin S., Neduzha L., Shvets A., 2012.:** Dynamics of freight cars with the transverse displacement of trolleys. Proc. Donetsk Railway Transport Institute. Edition 29, 234-241 (in Ukrainian).
14. **Mukhanov V., Ten A., Myamlin S., Neduzhaja L., 2010.:** Innovative developments in the field of freight car building. Proc. Donetsk Railway Transport Institute. Edition 22, 76-82 (in Russian).
15. **Myamlin S., 2001.:** Connection of dynamic parameters of laden gondola car with

acceleration of axle boxes frame: collection of scientific papers. Transport: collect. of sc. papers, DNURT, Dniepropetrovsk. Issue 7, 86-89 (in Russian).

16. **Myamlin S., 2002.:** Simulation of railway vehicles dynamics. Dniepropetrovsk, «New ideology», 240. (in Russian).
17. **Myamlin S., 2003.:** Author's rights registration certificate on product №7305. Computer program «Dynamics of Rail Vehicles» («DYNRAIL»), registered 20.03.2003 (in Russian).
18. **Myamlin S., Dailidka S., Neduzha L., 2012.:** Mathematical Modeling of a Cargo Locomotive. Proceedings of 16th International Conference "Transport Means 2012". Kaunas, 310-312.
19. **Myamlin S., Neduzha L., Shvets A., 2009.:** Preparation of initial data for modeling the dynamic parameters of freight cars. Transport handling machinery. Issue 4, 152-160 (in Russian).
20. **Myamlin S., Neduzha L., Shvets A., 2012.:** Author's rights registration certificate on product №42263. Product research and practical "Algorithm to calculate of computer programs "The Programmatic complex for determination of moments of inertia of car bodies", registered 15.02.2012 (in Ukrainian).
21. **Myamlin S., Neduzha L., Ten A., Shvets A., 2010.:** Spatial Vibration of Cargo Cars in Computer Modelling with the Ac-count of Their Inertia Properties. Proceedings of 15th International Conference. Mechanika. Kaunas. 325-328.
22. **Myamlin S., Neduzha L., Ten A., Shvets A., 2012.:** Author's rights registration certificate on product №42224. Computer program "The Programmatic complex for determination of moments of inertia of car bodies", registered 13.02.2012 (in Ukrainian).
23. **Myamlin S., Neduzha L., Ten A., Shvets A., 2013.:** Determination of dynamic performance of freight cars taking into account technicalcondition of side bearers. Science and transport progress bulletin of Dniepropetrovsk national university of railway transport named after academician V. Lazaryan. Scientific journal, 1 (43) 2013, 162-170.
24. **Myamlin S., Neduzhaja L., Ten A., 2010.:** Theoretical research of gondola car dynamics. Proc. Donetsk Railway Transport Institute. Edition 24, 143-151 (in Russian).
25. **Myamlin S., Neduzhaja L., Ten A., Shvets A., 2011.:** Definition specifics of inertia moments of freight cars bodies. Proc. Donetsk Railway Transport Institute. Edition 25, 137-144 (in Russian).
26. **Targ S., 1986.:** Short course of theoretical mechanics. Textbook for high schools. Moscow, "Higher school", 418. (in Russian).
27. **Verigo M., Kogan A., 1986.:** Track and rolling stock interaction. Moscow, Transport, 560. (in Russian).
28. **Vershinsky S., Danilov V. Chelnokov I., 1972.:** Car dynamics. Moscow, Transport, 304. (in Russian).

ИССЛЕДОВАНИЕ ВЛИЯНИЯ ПОКАЗАТЕЛЕЙ ТРЕНИЯ НА ДИНАМИКУ ГРУЗОВОГО ВАГОНА

Сергей Мямлин, Лариса Недужая,
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Аннотация. Работа посвящена исследованию влияния показателей трения в системе «кузов – тележка» грузовых вагонов (изменение значения коэффициента трения в скользунах во время эксплуатации) на их основные показатели – коэффициенты горизонтальной и вертикальной динамики, вертикальные и горизонтальные ускорения кузова, рамную силу, коэффициент устойчивости от схода с рельсов. Исследование проводилось методом математического моделирования динамической нагруженности грузового вагона с использованием программного комплекса «DYNRAIL». Основное внимание уделялось в основном влиянию изменения силы трения между кузовом и тележками. Теоретические исследования динамической нагруженности грузового полувагона проводились в порожнем и груженном состоянии с тележками ЦНИИ-Х3 (модель 18–100) при движении в прямых и кривых участках железной дороги различных радиусов с установленными скоростями движения. В результате исследований, с целью определения динамических показателей грузовых вагонов с учетом влияния показателей трения, получены зависимости основных динамических показателей от изменения значения коэффициента трения в скользунах с учетом скорости движения на прямых и кривых малого и среднего радиуса участках железной дороги. Результаты теоретических исследований определения динамических показателей грузовых вагонов с учетом показателей трения в системе «кузов – тележка» позволяют объективно оценить влияние коэффициента трения между скользунами грузовых вагонов на показатели безопасности движения по железной дороге (коэффициенты горизонтальной и вертикальной динамики, вертикальные и горизонтальные ускорения кузова, рамную силу, коэффициент устойчивости от схода с рельсов).

Ключевые слова: грузовые вагоны, скользуны, динамические показатели.