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РУХОМИЙ СКЛАД І ТЯГА ПОЇЗДІВ

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ABOUT THE EVALUATION OF THE LONGITUDINAL FORCES LEVEL EFFECTING THE TRACK DISPLACEMENT AT TRANSIENT MODES OF TRAIN MOVEMENT

Purpose. Study the transient modes effect of movement on the track displacement for the freight train safety control is supposed in this paper. For this it is necessary to investigate the longitudinal dynamics of a train on the track displacement. Simultaneously to assess the longitudinal forces level of a track and rolling stock interaction. **Methodology**. The level of the longitudinal forces, effecting the track displacement, was evaluated using mathematical modeling of longitudinal vibrations of the trains at transient modes of motion caused by braking. It was considered that each train vehicle consists of a body (solid) and the wheel sets, connected with the body by friction bearings (inelastic link). It was believed that during the movement of each train vehicle the vertical plane of its symmetry coincident with the vertical plane of symmetry of the assembled rails and sleepers. At simulation it was also supposed that in the process of translational motion of the vehicle body wheels make pure rolling along the rail without slipping on it. **Findings**. In the results of calculations the values of the longitudinal forces at different types of braking were obtained (it is regenerative braking and pneumatic one) under quasi-static and shock transients. For this various initial state of clearances in the inter-car connections up to beginning of transient was considered. The level of dynamic additives to longitudinal forces of interaction between wheel and rail that are substantially depending on vehicle accelerations was assessed. **Originality**. The transient regimes effect of trains movement caused by braking on the level of the longitudinal forces of track and rolling stock interaction was investigated. The longitudinal load of freight trains with regenerative and pneumatic braking was researched. The effect of the initial state of the train and different modes of braking on a dynamic additive to the longitudinal forces of the interaction between the track and rolling stock, which may effect the displacement of assembled rails and sleepers, was estimated. **Practical value**. The obtained results can be used to select rational modes of braking of freight trains, especially on lengthy down grade, from the positions prevent possible track displacement.

Keywords: mathematical modeling; transient modes of train movement; pneumatic braking; recuperative braking; interaction forces between the track and rolling stock

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Introduction

As the experience of freight trains operation shows, track displacement occurs when traction-braking running is applied in order to keep given speed of mode, especially on an excessive gradient and downhill length accordingly [2, 3, 6−7, 11−15].

Purpose

Freight trains safety control requires studying the effect of transient mode of their movement on track displacement.

Methodology

Processes of longitudinal forces occurrence of interaction between the track and rolling stock, caused by transient modes of trains movement, were studied by mathematical modeling of longitudinal vibrations of the train using known methods of numerical integration of nonlinear differential equations describing its motion [1, 4, 5, 9, 10].

As a simplified model of the train a chain of bodies (vehicles), interconnected by links (inter-car links) was considered. At this it was assumed that each train vehicle consists of a body (solid) and the wheel sets, connected with the body by friction bearings (inelastic link). The elastic properties of the track and wheel sets were not taking into account. It was thought that during the movement of each train vehicle the vertical plane of its symmetry coincident with the vertical plane of symmetry of the assembled rails and sleepers.

At simulation it was also supposed that in the process of translational motion of the vehicle body wheels make pure rolling along the rail without slipping on it. Such wheel motion was considered as compound, consisting of translational motion with rate V_C and acceleration a_C of center of body masses (Fig. 1) and rotary motion about the axis of the wheel set with an angular velocity ω and angular acceleration ε. Then during pure rolling $\omega = \frac{V_C}{r}$ and $\varepsilon = \frac{a_C}{r}$, where r – wheel radius (Fig. 2).

It was supposed that longitudinal force Q acts on each vehicle of the train (Fig. 1), which includes a component of the vehicle gravity on the slope of the track, the efforts in the links between vehicles (in inter car links), resistance force of translational motion, for example, from the wind load.

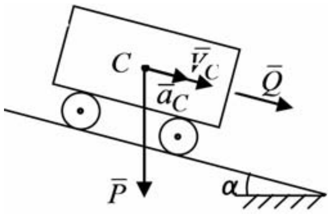

Fig. 1. The computational model of the train vehicle

At this resistance forces moment M_{res}^{ws}, arising in bearings, acts on each wheel set of the vehicle (Fig. 2), and braking moment M_{brak}^{ws} can act in the result of regenerative and pneumatic (locomotives) braking of the vehicle.

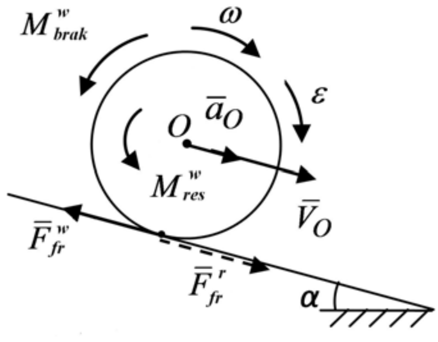

Fig. 2. Force load wheeled of vehicle wheelset

It can be shown that the dynamic equation which describes the motion of the train vehicle in these cases has the form:

$$(m_v + N \cdot \frac{I_O^{ws}}{r^2}) \cdot a_C = Q - N \cdot \frac{M_{brak}^{ws}}{r} - N \cdot \frac{M_{res}^{ws}}{r},$$

where m_v – body mass of the vehicle, N – wheel set number of the vehicle, I_O^{ws} – inertia axial moment of the wheel set.

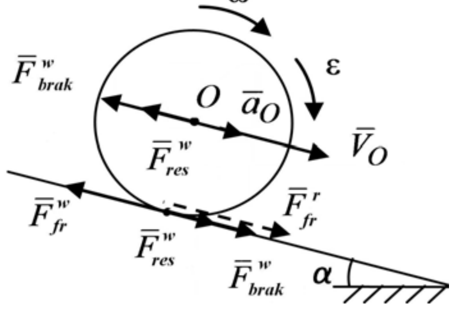

Fig. 3. Force diagram, acting in the center of wheel masses and at the point of wheel and rail contact

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It should be taken into consideration that braking moment M_{brak}^{w} and moment of resistance force M_{res}^{w}, applied to the wheel of the vehicle (Fig. 2), one can change with corresponding moment of forces pair, one of which is attached to the wheel axle, and the other – to the contact point of a wheel and a rail (Fig. 3).

Then each of these moments can be expressed through the moment of the relevant force couples ($F_{brak}^{w}, F_{res}^{w}$): $M_{brak}^{w} = F_{brak}^{w} \cdot r$, $M_{res}^{w} = F_{res}^{w} \cdot r$).

As a result pointed moments for wheel sets correspond to the formulas

$$M_{brak}^{ws} = 2F_{brak}^{w} \cdot r = F_{brak}^{ws} \cdot r ,$$

$$M_{res}^{ws} = 2F_{res}^{w} \cdot r = F_{res}^{ws} \cdot r$$

and $\dfrac{M_{brak}^{ws}}{r} = F_{brak}^{ws}$, but $\dfrac{M_{res}^{ws}}{r} = F_{res}^{ws}$.

With recent expressions dynamic equation of the vehicle is the following

$$(m_v + N \cdot \frac{I_O^{ws}}{r^2}) \cdot a_C = Q - N \cdot F_{brak}^{ws} - N \cdot F_{res}^{ws}$$

or

$$(m_v + N \cdot \frac{I_O^{ws}}{r^2}) \cdot a_C = Q - F_{\text{brak}}^{v} - F_{\text{res}}^{v} ,$$

where $F_{\text{brak}}^{v} = N \cdot F_{\text{brak}}^{ws}$ – it is braking force that acts on the vehicle, and $F_{res}^{v} = N \cdot F_{res}^{ws}$ – it is resistance force to motion from friction in the bearings of the vehicle.

Then acceleration of masses center of the vehicle may be expressed as

$$a_C = \frac{Q - F_{brak}^{v} - F_{\text{res}}^{v}}{m_v + N \cdot \dfrac{I_O^{ws}}{r^2}} .$$

Interaction forces between a wheel and a rail in cases in question are the friction forces arising in the contact point of a wheel and a rail (Fig. 3); at this, if $\overline{F}_{fr}^{w}$ – it is friction force, acting on a wheel from the rail side, then $\overline{F}_{fr}^{r}$ – it is friction force, acting on a rail from the wheel side; because the action is a reaction, then $\overline{F}_{fr}^{r} = -\overline{F}_{fr}^{w}$.

In order to find out what determines the frictional force $\overline{F}_{fr}^{ws}$ that acts on wheel set, it is necessary to make a dynamic equation of rotational motion of wheel set about its axis:

$$I_O^{ws} \cdot \varepsilon = \sum M_O ;$$

$$I_O^{ws} \cdot \frac{a_C}{r} = -M_{brak}^{ws} - M_{res}^{ws} - F_{fr}^{ws} \cdot r ,$$

where $\overline{F}_{fr}^{ws} \cdot r$ – it is net moment relatively to the axis of wheel set of the friction forces applied to wheel set from the rails side.

Then the expressions for the determination the total frictional force acting on the wheel set and a vehicle have the form:

$$F_{fr}^{ws} = \frac{M_{brak}^{ws}}{r} + \frac{M_{res}^{ws}}{r} + I_O^{ws} \frac{a_C}{r^2} ,$$

$$F_{fr}^{v} = N \cdot \frac{M_{brak}^{ws}}{r} + N \cdot \frac{M_{res}^{ws}}{r} + N \cdot I_O^{ws} \frac{a_C}{r^2}$$

or

$$F_{fr}^{v} = F_{brak}^{v} + F_{res}^{v} + N \cdot I_O^{ws} \frac{a_C}{r^2} = F_{brak}^{v} + F_{res}^{v} + d ,$$

where d – it is dynamic additive to forces F_{brak}^{v}, F_{res}^{v}, which depends on the acceleration of the vehicle.

Acceleration values of vehicles can be significant at transient modes of train movement. That is why it seems to be interesting to investigate the processes of longitudinal forces rise of interaction between a track and rolling stock at transient modes of train movement and primarily which are caused by their braking.

Findings

Regenerative (electric braking locomotives) and pneumatic braking of the train with a speed of 40 km/h on horizontal sections of the track and slopes were simulated. In some cases, the train before braking was pre-compacted, in others it was extended.

It was assumed that the train consists of 50 four-homogeneous gondola cars, weight 80 tons and four locomotives, type VL-11. Joint of three

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locomotives was in the front of a train and one in the rear end of the train.

It was also believed that cars are equipped with an air distributor No. 483 and composition brake shoe while inter-car links with elastic and friction draft gear SH-1-TM.

Levels of longitudinal forces F_{fr}^{w}, acting on rails from the side of the train vehicle, and values of dynamic additives *d* were determined.

Below as an example the oscillographes chart of the longitudinal forces (Fig. 4−5) and accelerations (Fig. 6−7) are presented at regenerative braking (all locomotives realize 24 position of electric brake) in prior extended and pre-compacted trains.

As one should expect, the highest level of longitudinal forces and accelerations occur when regenerative braking of prior extended trains in the rear end sections of the train.

The dependences of the dynamic additives from motion time for the 1st, 4th, 26th and 52nd vehicles during regenerative braking in prior extended and pre-compacted trains correspondingly are shown in Fig. 8−9. The total dynamic additive curve (red line) and the braking force for the entire train are shown in Fig. 10−11.

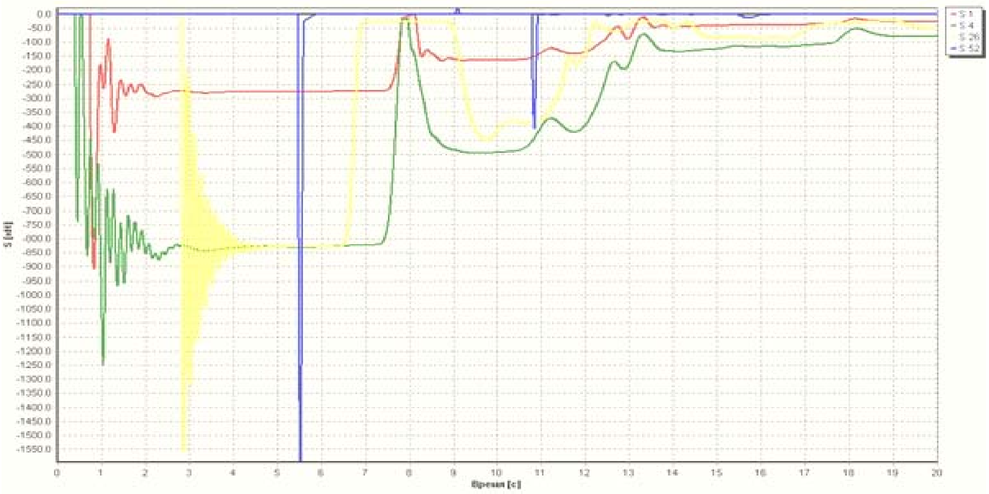

Fig. 4. Oscillograms of longitudinal forces after the first locomotive, 4th, 26th and 52nd vehicles at regenerative braking in prior extended train

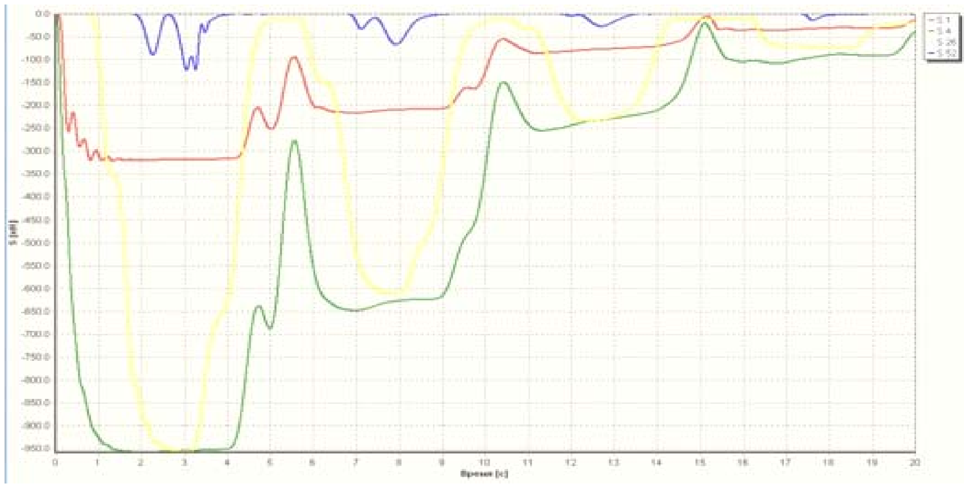

Fig. 5. Oscillograms of longitudinal forces after the first locomotive, 4th, 26th and 52nd vehicles at regenerative braking in pre-compacted train

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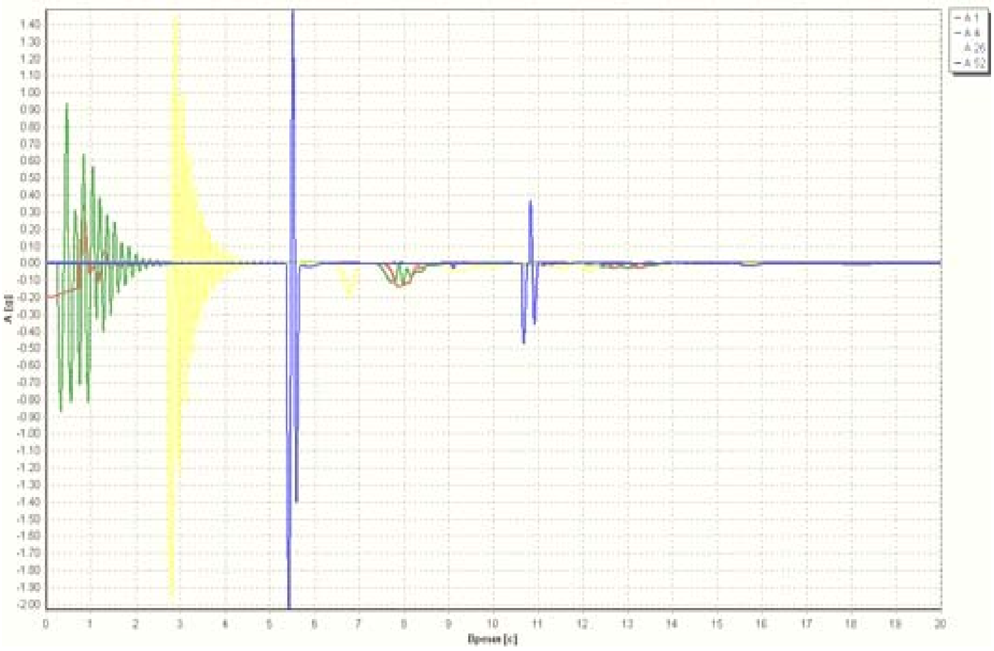

Fig. 6. Oscillograms of longitudinal accelerations
of the first locomotive, 4th, 26th and 52nd vehicles
at regenerative braking in prior extended train

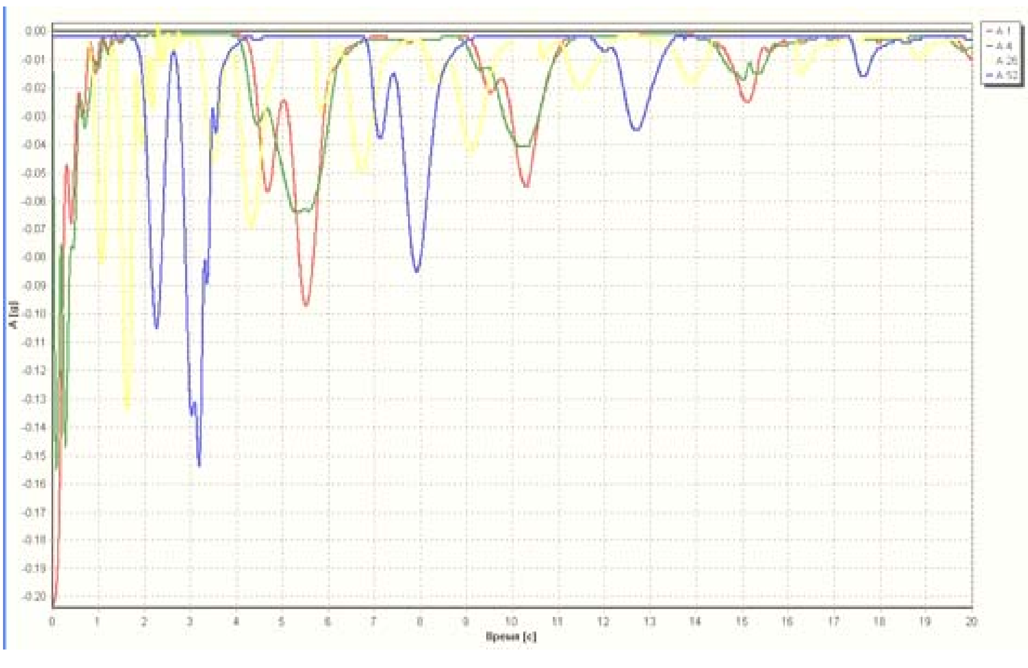

Fig. 7. Oscillograms of longitudinal acceleratons
of the first locomotive, 4th, 26th and 52nd vehicles
at regenerative braking in pre-compacted train

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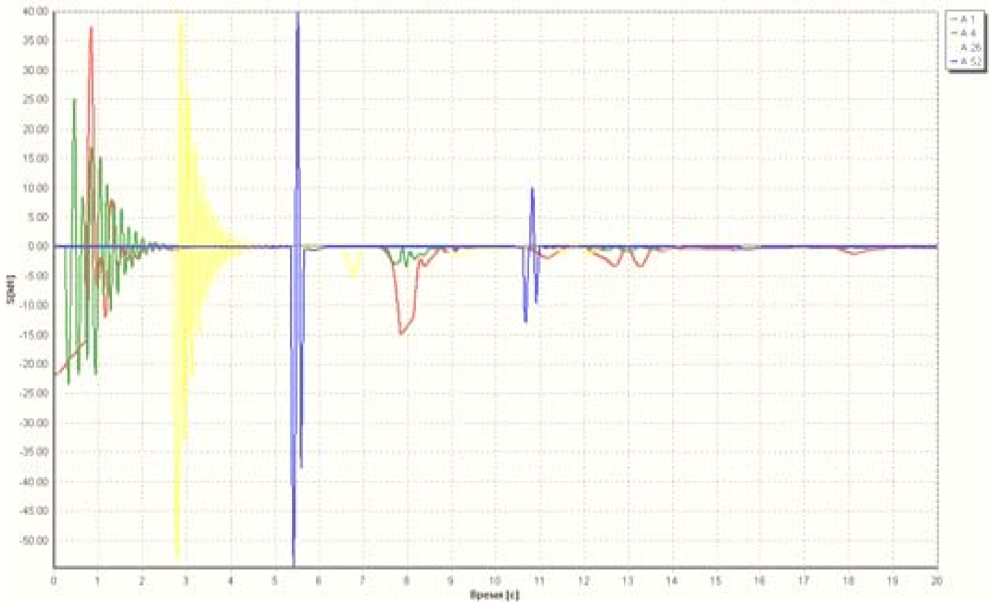

Fig. 8. Dependences of dynamic additives from motion
time for the 1st, 4th, 26th and 52th vehicles
at regenerative braking in prior extended train

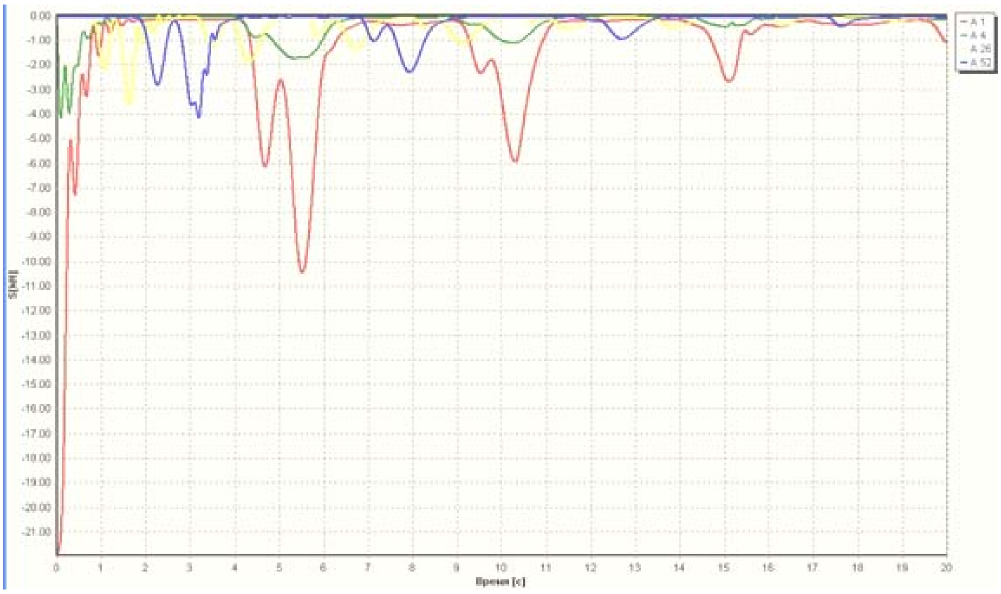

Fig. 9. Dependences of dynamic additives from motion time
for the 1st, 4th, 26th and 52th vehicles
at regenerative braking in pre-compacted train

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Fig. 10. Value change of the total dynamic additive (upper line) and the total braking force for the entire train, depending on the motion time at regenerative braking movement in prior extended train

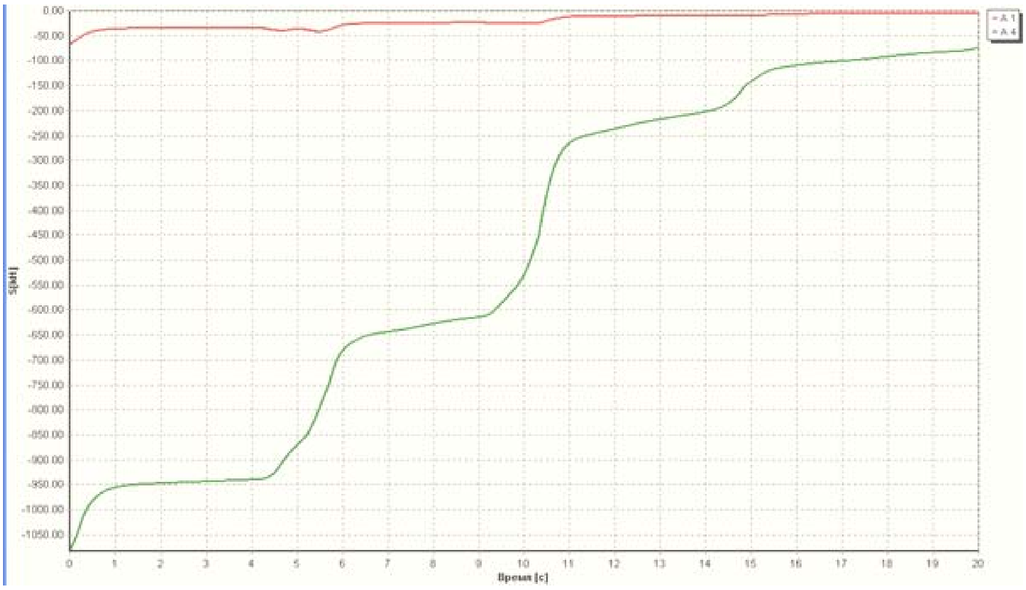

Fig. 11. Value change of the total dynamic additive (upper line) and the total braking force for the entire train, depending on the motion time at regenerative braking in pre-compacted train

At braking in the prior extended train the highest level of compressive longitudinal forces of shock behavior for examined sections of a train is about 1500 kN (Fig. 4) and the quasistatic ones - 950 kN (Fig. 5).

The highest level of longitudinal accelerations is 20 m/s^2 at shock transients (Fig. 6) and 2 m/s^2 (Fig. 7) – in quasistatic ones.

The maximum level of total value additives for the train takes the value of 55 kN at shock processes (Fig. 8) and 21 kN – at quasistatic ones (Fig. 9).

From the graphs shown in Fig. 8−11, one can conclude that the maximum value of dynamic additive is registered in that section of the train where

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the greatest value of the longitudinal acceleration occurs. Therefore, at regenerative braking the greatest value of dynamic addition in a prior extended train 2.5 times more of that value which occurs than for pre-compacted train. At braking of the prior extended train the greatest value of dynamic additive occurs in the rear end section, and at braking of the pre-compacted train occurs in front of the train.

The total values of the dynamic additives and braking forces (Fig. 10−11) in the train do not depend on the initial state of the gaps in the intercar links.

Similar dependences during pneumatic braking by the Ist stage with discharging of brake of 0.5 atm are presented in Fig. 12−19.

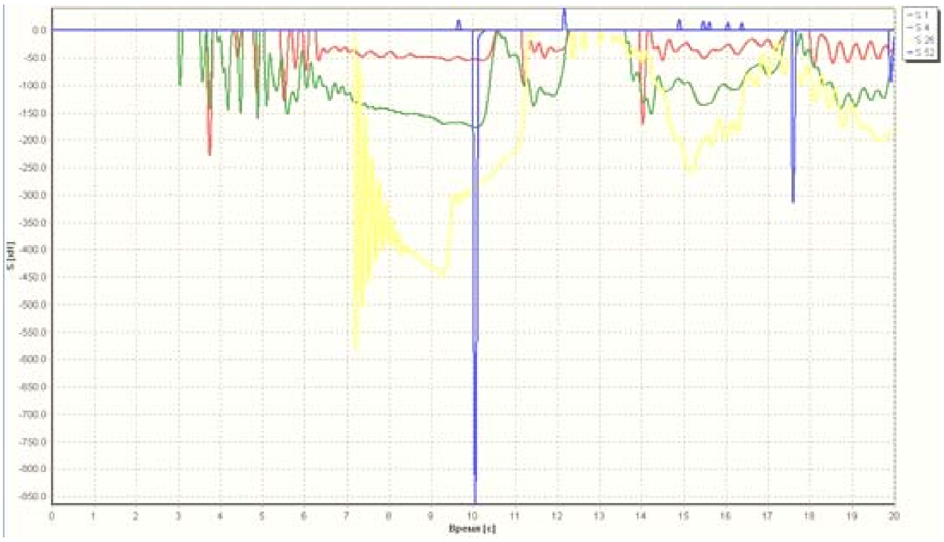

Fig. 12. Oscillograms of longitudinal forces after the first locomotive, 4th, 26th and 52nd vehicles at pneumatic braking by the ^{1st} stage in prior extended train

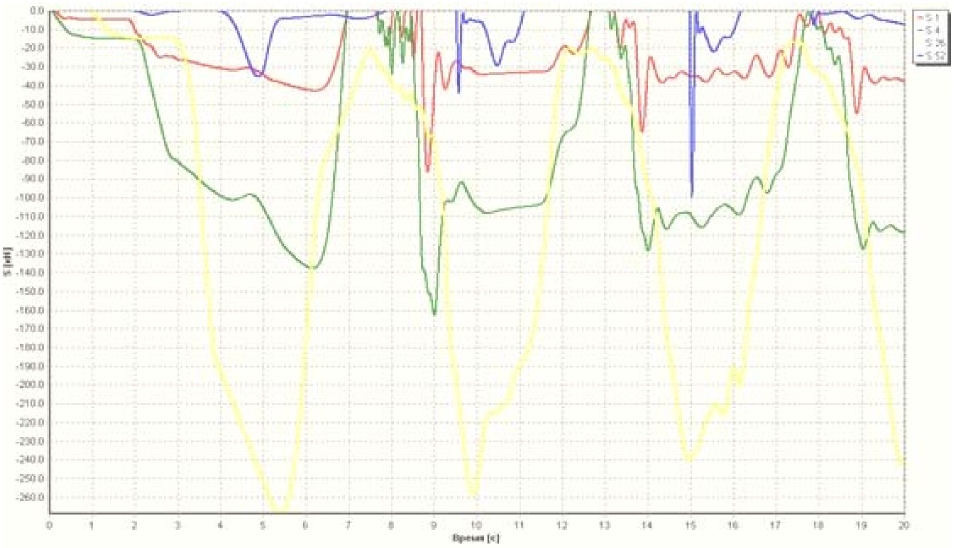

Fig. 13. Oscillograms of longitudinal forces after the first locomotive, 4th, 26th and 52nd vehicles at pneumatic braking by the ^{1st} stage of pre-compacted train

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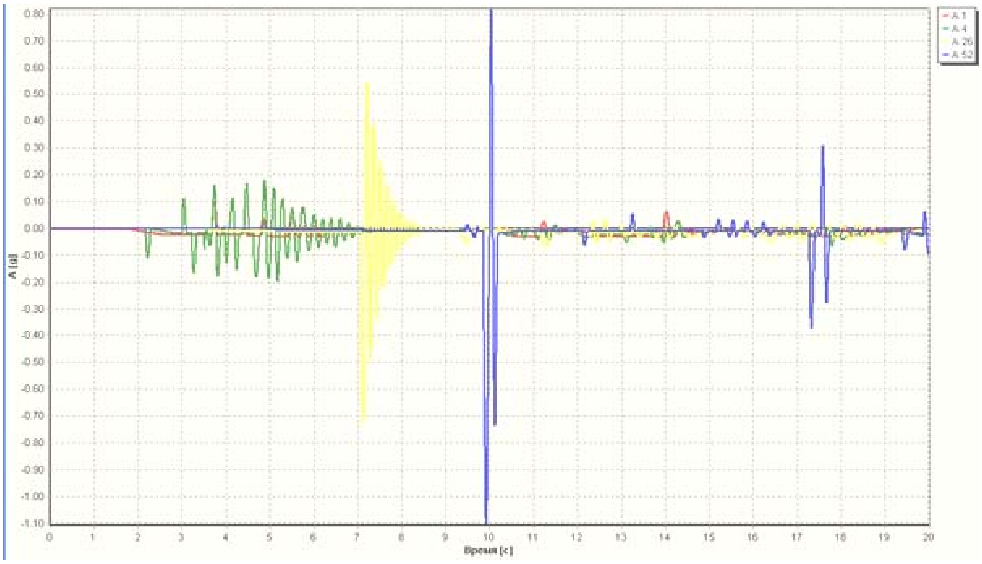

Fig. 14. Oscillograms of longitudinal accelerations of the first locomotive, 4th, 26th and 52nd vehicles at pneumatic braking by the 1st stage in prior extended train

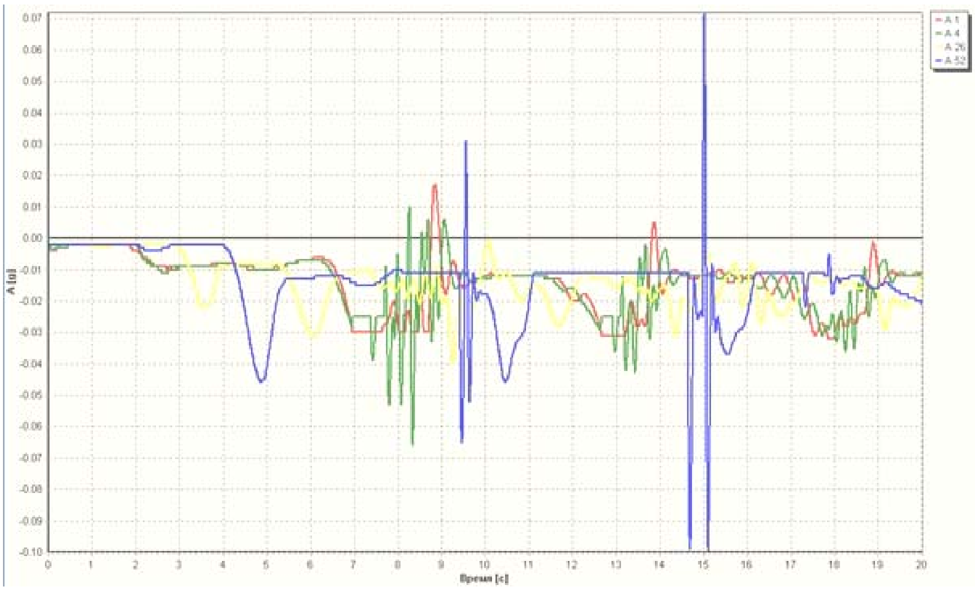

Fig. 15. Oscillograms of longitudinal accelerations of the first locomotive, 4th, 26th and 52nd vehicles at pneumatic braking by the 1st stage in pre-compacted train

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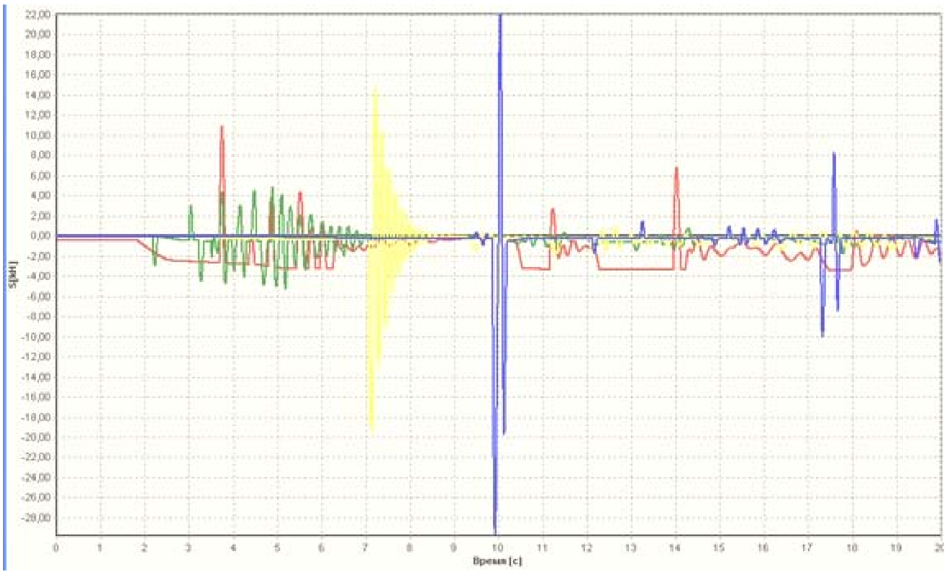

Fig. 16. Dependences of dynamic additives from motion time for the 1st, 4th, 26th and 52nd vehicles at pneumatic braking by the 1st stage in prior extended train

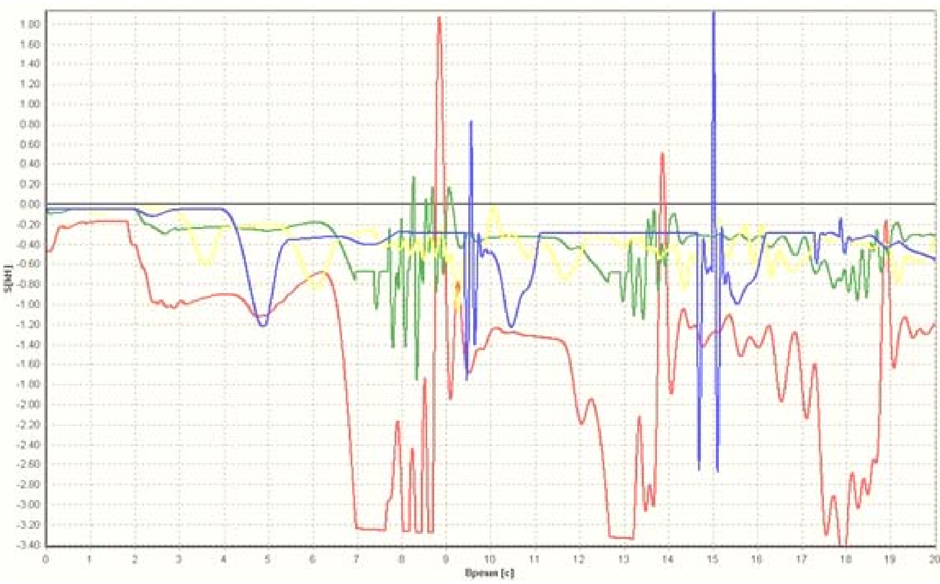

Fig. 17. Dependences of dynamic additives from motion time for the 1st, 4th, 26th and 52nd vehicles at pneumatic braking by the 1st stage in pre-compacted train

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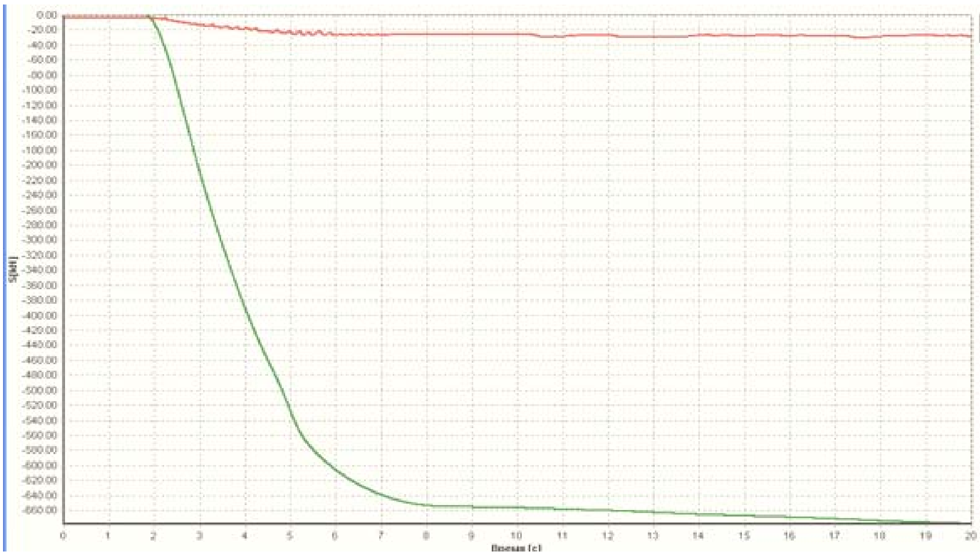

Fig. 18. Value change of the total dynamic additive (upper line)
and the total braking force for the entire train, depending on the motion time
at pneumatic braking by the 1st stage in prior extended train

As can be seen from the graphs shown in Fig. 12−17, oscillograms behavior of longitudinal forces and accelerations essentially depends on the initial state of gaps in the intercar links. At braking of the prior extended train the greatest value of dynamic additive occurs in the rear end section, as the greatest acceleration arises there. At braking of the pre-compacted train the greatest value of dynamic additive occurs in the front of the train, as in this case due to lack of shock loads, acceleration of a locomotive substantially exceeds longitudinal acceleration of other vehicles.

Comparison of the results presented in Fig. 8−9 and 16−17, showed that the greatest value of the dynamic additive of the regenerative braking is almost 2 times higher than similar value, obtained during braking by the 1st stage of the prior extended train and almost 6 times higher at regenerative braking of the pre-compacted trains.

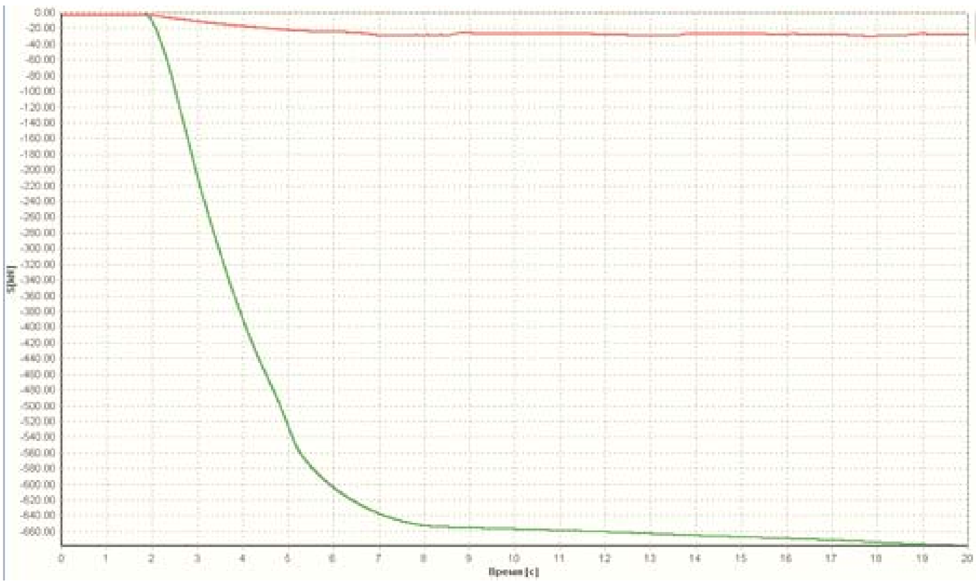

Fig. 19. Value change of the total dynamic additive (upper line) and the total braking force
for the entire train, depending on the motion time at
pneumatic braking by the 1st stage in the pre-compacted train

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The highest total value of dynamic additive at braking by the 1th stage occurs in the rear end of the train regardless from the initial state of train set.

When comparing the results in Fig.10−11 and Fig. 18−19 it is clear that the greater value of total dynamic additive to longitudinal forces of interaction between a track and rolling stock for the entire train occurs during regenerative braking, and 2 times higher than similar value arising at pneumatic braking. It is evidence that the regenerative braking is more dangerous for track displacement.

It should be also noted that regardless of the braking type (regenerative or pneumatic) and initial state of gaps in intercar links, dynamic additive value was much less than arising braking forces. That is why the level of longitudinal forces arising in intercar links at the considered modes of movement has little effect on the track displacement.

Originality and practical value

The longitudinal loading of freight trains with regenerative braking and pneumatic one was investigated. The impact of initial state of the train and the different modes of braking on the dynamic additive to the longitudinal forces of interaction between a track and rolling stock was estimated. It may affect the assembled rails and sleepers. Obtained results can be used to select the rationale braking modes of freight trains, especially downhill length, from a position to prevent possible track displacement.

Conclusions

Obtained results show that the dynamic additive to longitudinal forces in the wheel and rail interaction depends on the occurring accelerations.

The total value of the dynamic additive was greater at the regenerative braking and does not depend on the initial state of a train set. The level of total dynamic additive in the train was much less than the level of resulting braking forces. Therefore, the level of longitudinal forces in intercar links has little effect on the track displacement.

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ПРО ОЦІНКУ РІВНЯ ПОЗДОВЖНІХ СИЛ, ЩО ВПЛИВАЮТЬ НА УГОН КОЛІЇ, ПРИ ПЕРЕХІДНИХ РЕЖИМАХ РУХУ ПОЇЗДА

Мета. В статті передбачається вивчити вплив перехідних режимів руху на угон колії для забезпечення безпеки руху вантажних поїздів. Для цього необхідно дослідити вплив поздовжньої динаміки поїзда на угон колії, оцінивши при цьому рівень продольних сил взаємодії колії та рухомого складу. **Методика.** Рівень поздовжніх сил, що впливають на угон шляху, оцінювався за допомогою математичного моделювання поздовжніх коливань поїзда при перехідних режимах руху, викликаних різними видами гальмування. При цьому передбачалось, що кожен екіпаж поїзда складається з кузова (тверде тіло) та колісних пар, сполучених із кузовом підшипниками ковзання (зв'язок непружний). Вважалося, що в процесі руху кожного екіпажу поїзда вертикальна площина його симетрії збігалася з вертикальною площиною симетрії рельсошпальної решітки. При моделюванні вважалось також, що в процесі поступального руху кузова екіпажу колеса роблять чисте кочення по рейці без прослизання щодо нього. **Результати.** У результаті розрахунків були отримані значення поздовжніх сил при різних видах гальмування (рекуперативних і пневматичних) у квазістатичних та ударних перехідних процесах. Для цього розглядався різний початковий стан зазорів у міжвагонних з'єднаннях до початку перехідного процесу. Оцінений також рівень динамічних добавок до поздовжніх сил взаємодії колеса та рейки, істотно залежних від прискорень екіпажів. **Наукова новизна**. Досліджено вплив перехідних режимів руху поїздів, викликаних гальмуванням, на рівень поздовжніх сил взаємодії колії та рухомого складу. Досліджена поздовжня навантаженість вантажних поїздів при рекуперативних та пневматичних гальмуваннях. Оцінено вплив початкового стану поїзда та різних режимів гальмувань на динамічну добавку до поздовжніх сил взаємодії колії й рухомого складу, яка може впливати на угон рельсошпальної решітки. **Практична значимість.** Отримані результати можуть бути використані для вибору раціональних режимів гальмування вантажних поїздів, особливо на затяжних спусках, із позицій запобігання можливого угону шляху.

Ключові слова: математичне моделювання; перехідні режими руху поїзда; гальмування пневматичне; гальмування рекуперативне; сили взаємодії колії та рухомого складу

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ОБ ОЦЕНКЕ УРОВНЯ ПРОДОЛЬНЫХ СИЛ, ВЛИЯЮЩИХ НА УГОН ПУТИ, ПРИ ПЕРЕХОДНЫХ РЕЖИМАХ ДВИЖЕНИЯ ПОЕЗДА

Цель. В статье предполагается изучить влияние переходных режимов движения на угон пути для обеспечения безопасности движения грузовых поездов. Для этого необходимо исследовать влияние продольной динамики поезда на угон пути, оценив при этом уровень продольных сил взаимодействия пути и подвижного состава. **Методика**. Уровень продольных сил, влияющих на угон пути, оценивался с помощью математического моделирования продольных колебаний поезда при переходных режимах движения, вызванных различными режимами торможения. При этом полагалось, что каждый экипаж поезда состоит из кузова (твердое тело) и колесных пар, соединенных с кузовом подшипниками скольжения (связь неупругая). Считалось, что в процессе движения каждого экипажа поезда вертикальная плоскость его симметрии совпадала с вертикальной плоскостью симметрии рельсошпальной решетки. При моделировании полагалось также, что в процессе поступательного движения кузова экипажа колеса совершают чистое качение по рельсу без проскальзывания относительно него. **Результаты**. В результате расчетов были получены значения продольных сил при различных видах торможения (рекуперативных и пневматических) в квазистатических и ударных переходных процессах. Для этого рассматривалось различное начальное состояние зазоров в межвагонных соединениях к началу переходного процесса. Оценен также уровень динамических добавок к продольным силам взаимодействия колеса и рельса, существенно зависящих от ускорений экипажей. **Научная новизна**. Исследовано влияние переходных режимов движения поездов, вызванных торможением, на уровень продольных сил взаимодействия пути и подвижного состава. Исследована продольная нагруженность грузовых поездов при рекуперативных и пневматических торможениях. Оценено влияние начального состояния поезда и различных режимов торможений на динамическую добавку к продольным силам взаимодействия пути и подвижного состава, которая может влиять на угон рельсошпальной решетки. **Практическая значимость.** Полученные результаты могут быть использованы для выбора рациональных режимов торможения грузовых поездов, особенно на затяжных спусках, с позиций предотвращения возможного угона пути.

Ключевые слова: математическое моделирование; переходные режимы движения поезда; торможение пневматическое; торможение рекуперативное; силы взаимодействия пути и подвижного состава

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