t – время;

Δt – шаг интегрирования.

В основу модели пневматической тормозной системы грузового поезда положены графические зависимости величины давления в тормозных цилиндрах (ТЦ) грузовых вагонов при торможении и отпуске (рис. 1) [5].

В разработанной модели учтены неодновременность срабатывания воздухораспределителей из-за конечной скорости распространения тормозной волны вдоль поезда, падение давления в тормозной магистрали по длине поезда из-за неизбежных утечек, неоднородность состава по типу тормозов, а также специфика работы основных тормозных приборов.

В результате использования математических моделей был произведён имитационный расчёт движения однородного грузового поезда массой 8600 т в режиме регулировочного торможения по спуску крутизной –10‰. Торможение осуществлялось со скорости движения v=50 км/ч первой ступенью разрядки тормозной магистрали на 0,7 кгс/см^2.

Получены следующие графики зависимостей (рис. 2): *а* – скорость движения поезда $v(t)$; *б* – давление в тормозном цилиндре первого экипажа $P_{ТЦ1}(t)$; *в, г, д* – силы, действующие в серединах первой $F_I(t)$, второй $F_{II}(t)$ и последней $F_{III}(t)$ третей поезда; *е* – распределение наибольших продольных сил вдоль поезда F_{max}. Хорошо видно при этом, что максимальные продольно-динамические силы не превышают допустимого по условиям безопасности уровня, составляющего для режима служебного торможения 1 МН.

Таким образом, разработанные математические модели движения поезда могут с успехом применяться для решения целого ряда задач, связанных с изучением торможения длинносоставных грузовых поездов, позволяя при этом детально исследовать их динамику.

ЛИТЕРАТУРА

1. Блохин Е. П., Манашкин Л. А. Динамика поезда (нестационарные продольные колебания). – М.: Транспорт, 1982. – 222 с.
2. Лисицын А. Л., Мугинштейн Л. А. Нестационарные режимы тяги: Тяговое обеспечение перевозочного процесса. – М.: Интекст, 1996. – 158 с.
3. Расчеты и испытания тяжеловесных поездов / Блохин Е. П., Манашкин Л. А., Стамблер Е. Л. и др.; под ред. Е. П. Блохина. – М.: Транспорт, 1986. – 262 с.
4. Hauptmann, D. *et al.* Eisenbahntechnische Rundschau, 1998, № 5, pp. 306–309.
5. Никифоров Б. Д., Головкин В. И., Кутыев Ю. Г. Автоматизация управления торможением поездов. – М.: Транспорт, 1985. – 263 с. ●

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SIMULATION OF REGULATING BRAKING MODE OF LONG TRAIN

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ABSTRACT

From the theory of train longitudinal dynamics it is known that with an increase in length and weight of a train, longitudinal forces grow. In addition, in the course of train movement in areas of rolling grade of the track to perturbations from kinks of the profile, perturbations associated with motion control can be imposed. Adverse overlay can lead to the emergence of longitudinal forces, hazardous under the terms of strength and stability of cars against derailments due to squeezing or pulling [1, 2]. The most unfavorable in this sense is the mode of regulating braking by air brakes, because then rate of change of brake forces remains unregulated. Therefore, when assessing the greatest forces in the train, and even more so in a long train, it is necessary to first take into consideration given control mode – pneumatic braking.

The authors use mathematical and engineering methods, mathematical simulation. In line with the study of longitudinal vibrations of a train a discrete multibody model of trains as a system of rigid bodies is considered. With its help regulating braking mode by air brakes is calculated, which allows to ensure that maximum longitudinal forces in the train do not exceed the level allowed under the terms of traffic safety.

Keywords: railway, freight train, longitudinal dynamics theory, mode, mathematical modeling.

Background. *In the simulation of the train movement for a thorough study of longitudinal oscillations a discrete multibody model is used, representing a train in the form of solids, interconnected essentially by nonlinear deformable elements. The motion of each vehicle (car or locomotive) is described by a following system of differential equations [3]:*

$$\begin{cases} \dot{v}_i = \dfrac{F_i - F_{i+1} + F_{i\Sigma}}{m_i},\ i=\overline{1,n};\ F_{n+1}=0\,; \\ \dot{q}_i = v_{i-1} - v_i,\ i=\overline{2,n}\,; \\ \dot{x}_1 = v_1\,; \\ x_i = x_{i-1} - \dfrac{l_i + l_{i-1}}{2} - q_i,\ i=\overline{2,n}\,, \end{cases} \qquad (1)$$

where n is a number of vehicles in a train;

q_i is a deformation of the i-th coupling cable;

$\dot{q}_i$ is deformation rate;

m_i is weight of the i-th vehicle;

v_i is speed of the center of mass of the i-th vehicle;

F_i is reaction in the i-th coupling cable;

$F_{i\Sigma}$ is total external force, acting on the i-th vehicle;

x_i is coordinate (along the trajectory) of the center of mass of the i-th vehicle;

l_i is length of the i-th vehicle.

Total external force $F_{i\Sigma}$, acting on the i-th vehicle is resultant from F_{eli} traction, braking force (electric B_{eli} and pneumatic B_{pni}), forces of main W_{mi} and additional W_{ai} resistance to train motion:

$$F_{i\Sigma} = F_{\mathrm{el}i} - B_{\mathrm{el}i} - B_{\mathrm{pn}i} - W_{\mathrm{m}i} - W_{\mathrm{a}i}. \qquad (2)$$

It is evident that the presence in the equation (2) of terms, taking into account the thrust force F_{eli} and force of electric braking B_{eli} will be meaningful only if a considered vehicle is a locomotive, which can be located not only at the head of the train, but also also in diffrently within the train. Otherwise, these terms are zero. Resistance forces W_{mi} and W_{ai} are calculated separately for each vehicle in accordance with the procedure described in the rules for traction calculation of train operations.

Virtually the entire railway rolling stock of Russia is equipped with spring friction center coupler draft gear. In this case, the forces acting in the automatic couplers, are calculated as follows:

$$F_i = F_{fi}(q_{fi}) \cdot sign\ q_i\,,$$

$$q_{fi} = \begin{cases} |q_i|,\ \ if\ q_i < 0 \\ 0,\ \ if\ 0 \le q_i \le \delta_{0i} \\ q_i - \delta_{0i},\ \ if\ q_i > \delta_{0i}\,, \end{cases} \qquad (3)$$

where

$$F_{fi} = \begin{cases} \min\{F_{li}, F_{ui}\}\,, \\ \quad if\ (q_{fi} < \Delta_i) \wedge (q_{fi}(t) \ge q_{fi}(t-\Delta t))\,; \\ \max\{F_{ui}, F_{ei}\}\,, \\ \quad if\ (q_{fi} < \Delta_i) \wedge (q_{fi}(t) < q_{fi}(t-\Delta t))\,; \\ F_{ei},\ \ if\ q_{fi} \ge \Delta_i\,; \end{cases} \qquad (4)$$

$$\begin{cases} F_{ei} = \tilde{S}_i + k_{ui}(q_{fi}(t) - q_{fi}(t-\Delta t)) + \beta_i \dot{q}_{fi} \cdot sign\ q_{fi}\,; \\ \tilde{S}_i = \left(F_{li}(t-\Delta t) \vee F_{ui}(t-\Delta t)\right), \\ \quad if\ F_{fi}(t-\Delta t) = \left(F_{li}(t-\Delta t) \vee F_{ui}(t-\Delta t)\right)\,, \\ otherwise \\ \tilde{S}_i = F_{fi}(t-\Delta t) - \beta \dot{q}_{fi}(t-\Delta t) \cdot sign\ q_{fi}(t-\Delta t)\,; \\ F_{li} = F_{0i} + k_{li} q_{fi}\,; \qquad F_{ui} = (1-\eta_i) k_{li} q_{fi}\,. \end{cases} \qquad (5)$$

In these expressions:

q_i is relative movement of the center of mass of the (i–1) -th and i- th vehicles;

q_{fi} is amount of compression of center coupler draft gear and deformations of body;

δ_{0i} is a gap in the i-th coupling;

Δ_i is a value of q_{fi}, at which center coupler draft gears close;

F_i is a force, deforming coupling;

F_{li} and F_{ui} are values of F_i on the branches of loading and unloading of power characteristics of center coupler draft gears;

F_{ei} is a value of force F_i during transition from loading to unloading, and vice versa;

F_{0i} is force of initial tightening of center coupler draft gears of the i-th coupling;

k_{li} and k_{ui} are stiffness coefficients in the calculation of forces F_{li} and F_{ei};

η_i is coefficient of coupling's energy dissipation at work of center coupler draft gears;

β_i is coefficient of viscous resistance to deformation of vehicle construction;

t is time;

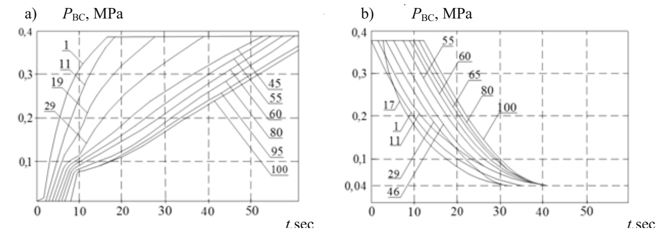
Δt is integration step.

Objective. *The objective of the authors is to investigate regulating braking mode and to present a discrete model multibody model.*

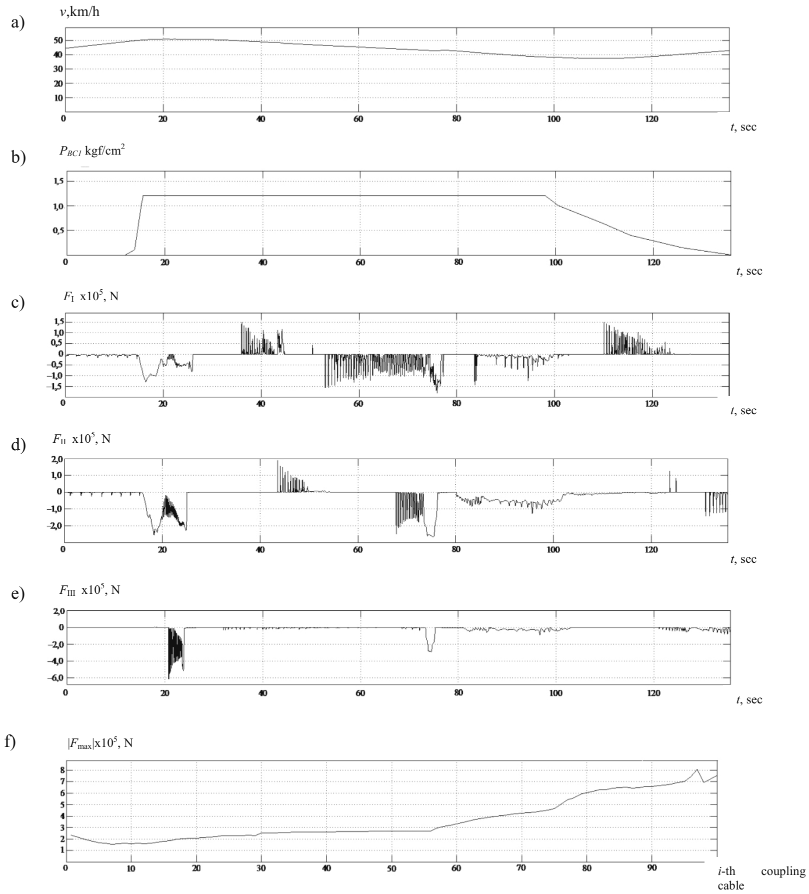
Methods. *The authors use mathematical and engineering methods, mathematical simulation.*

Results. *The model air brake system of a freight train is based on graphical dependences of pressure in brake cylinders (BC) of freight cars during braking and release (Pic. 1) [5].*

The proposed model takes into account not simultaneously triggering of diffusers due to the finite speed of propagation of brake waves along the train, pressure drop in the brake pipe along all the length of the train due to unavoidable leaks, the heterogeneity on the type of brakes, as well as the specifics of the work of main brake devices.



Pic. 1. Graphs of filling (a) and emptying (b) of brake cylinders of cars of a freight train.



Pic. 2. Simulation results of a homogeneous freight train movement.

As a result of use of mathematical models of simulation movement calculation was made related to a freight train weighing 8600 tonnes in the regulating braking mode on downhill slope $-10\ ^0/_{00}$. Braking was carried out from a speed v = 50 km / h by the first stage of brake pipe discharge of 0,7 kgf/cm².

Conclusion. *The following dependency graphs are obtained (Pic. 2): a – train speed $v(t)$; b – pressure in the brake cylinder of the first vehicle $P_{BC1}(t)$; c, d, e – forces acting in the middle of the first $F_I(t)$, the second $F_{II}(t)$ and the last $F_{III}(t)$ thirds of the train; e -distribution of the largest longitudinal forces along the train F_{max}. It is clearly seen that in this case the maximum longitudinal dynamic forces do not exceed the permissible level for safety, which is 1 MN for service brake mode.*

Thus, developed mathematical models of train movement can be successfully used to solve a number of problems related to the study of braking of long freight trains, while allowing the detailed study of their dynamics.

REFERENCES

1. Blokhin, E.P., Manashkin, L. A. Train dynamics (unsteady longitudinal oscillations) [*Dinamika poezda (nestatsionarnye prodol'nye kolebaniia)*]. Moscow, Transport publ., 1982, 222 p.

2. Lisitsyn, A.L., Muginshtein, L.A. Unsteady traction modes: Traction supply of a transportation process [*Nestatsionarnye rezhimy tiagi: Tiagovoe obespechenie perevozochnogo protsessa*]. Moscow, Intekst publ., 1996, 158 p.

3. Calculations and tests of heavy trains [*Raschety i ispytaniia tiazhelovesnykh poezdov*]. Blokhin, E.P., Manashkin, L.A., Stambler, E.L. *et al.* Ed. by Blokhin, E. P. Moscow, Transport publ. 1986, 262 p.

4. Hauptmann, D. *et al.* Eisenbahntechnische Rundschau, 1998. Iss 5, pp. 306–309.

5. Nikiforov, B.D., Golovkin, V.I., Kutyev, Yu.G. Automation of train braking control [*Avtomatizatsiia upravleniia tormozheniem poezdov*]. Moscow, Transport publ., 1985, 263 p.

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