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DETERMINING THE KINEMATIC PARAMETERS OF RAILCAR MOTION IN HUMP YARD RETARDER POSITIONS

ODREĐIVANJE KINEMATSKIH PARAMETARA KRETANJA VAGONA PRI POLOŽAJIMA RETARDERA U RANŽIRNOJ STANICI

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- hump yard
- railcar braking distance
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- kinematic parameters

Abstract

Hump yards perform the engineering function of train shunting and departure, moving and accumulating cars in the yard, etc. Hump yard designs, in order to ensure safe railway operations, should take into account the kinetic dynamics of rolling stock and individual railcars. The theorem of change in the kinetic energy of a point object is used to obtain formulas for determining the braking distance of a railcar in the braking area of a hump yard. This research also shows why using the formula for the free fall rate of a body accounting for the inertia of rotating parts, to determine a railcar's velocity in the braking zone, is incorrect. Results obtained show accuracy in the range from 3.7 to 1.52 %, which is within an acceptable window of accuracy for engineering calculations.

Ključne reči
- ranžirna stanica
- put kočenja vagona
- kinetička energija
- količina kretanja
- kinematski parametri

Izvod

U ranžirnim stanicama se obavljaju inženjerske funkcije manevrisanja šinskim vozilima, njihov polazak, kretanje i nagomilavanje vagona u stanici, itd. Radi bezbednosti u železničkom saobraćaju, pri projektovanju ranžirnih stanica treba razmatrati dinamiku kretanja kompozicija i pojedinačnih šinskih vozila. Teorema promene kinetičke energije materijalne tačke se koristi za izvođenje obrazaca za određivanje kočionog puta vagona u okviru zone kočenja u ranžirnoj stanici. U radu je takođe pokazano zašto je pogrešno primeniti formulu za brzinu slobodnog pada tela, uzimajuću u obzir inerciju rotirajućih delova, za određivanje brzine vagona u zoni kočenja. Dobijeni rezultati pokuzuju tačnost u oblasti 3,7 do 1,52 %, što je unutar prihvatljivih granica tačnosti za inženjersku primenu.

INTRODUCTION

Railway vehicles are transport systems designed for safe and smooth movement on individual tracks. They differ significantly from automobiles and other transport systems in that they operate on tracks consisting of straight and curved sections, and use propulsion and brake system without active steering means, such as wheel steering systems or rudders. At the same time, railway vehicles must satisfy conflicting demands: straight-line stability on straight-line tracks (running stability) and turning on curved tracks. In addition, it is important for them to maintain a high ride quality and be immune to vibrations when passing through irregular sections of track or switches. In other words, it is very important that railway vehicles have good running stability and turning characteristics, while ensuring good ride quality, /1/.

The railway industry is essentially conservative. Only in recent years have modern scientific methods of analysis been applied to the problem of rail vehicle dynamics. Its complexity is extreme, but the need for increased velocities and greater carrying capacity, both of which create new problems in terms of wear and stability, forces railway operators and equipment suppliers to solve these problems in a more systematic and fundamental way /2/, /3/, /4/.

Chenxu Lu and Jin Shi /5/ study the dynamic behaviour of a vehicle and track over a long and inclined section of a high-velocity railway under braking conditions. A model of dynamic interaction of the vehicle and the track was built on the basis of two longitudinal models of this interaction. In the model, the vehicle is considered a multi-rigid system with 21 degrees of freedom, consisting of an automobile body, two railroad trucks and four wheels; using the finite element method, the rail track is modelled as an Euler beam; the method of 'circular path' to reduce the degree of freedom of the model for modelling a long journey; two models of the longitudinal interaction of a wheel and a rail are considered: the Polach creep theory (suitable for modelling high creep as a result of strong braking), /6/, and the longitudinal theory of hard contact. The dynamic characteristics of substructures during vehicle braking, calculated using models based on the Polach creep model and longitudinal contact models, show a small difference, but the Polach creep model can fully take into account wheel movement and greater wheel-rail creep during braking and can accurately analyse damage to the wheel-rail interface.

Analysis of dynamic interaction between the wheel and rail under various conditions shows that a large braking torque will cause some or all wheels to slide, damaging the wheel-rail contact. This will increase the braking distance and time, as well as extend the sliding time of the locked wheels, increasing the risk of damage to the wheel from rail contact. Braking torque should be kept below a reasonable value so the braking distance and braking time can be as short as possible without causing the wheel to slip along the track. According to the calculations performed in this study, a reasonable braking torque under dry conditions and wet conditions should be 7 and 4 kNm, respectively.

A number of works have investigated the dynamics of a braking railway car in retarder positions /7, 8/. An important task is regulating velocities in railcar braking areas (BA) and in retarder positions (RP). High capacity hump yards are equipped with various kinds of car retarders, which are the primary means for regulating the velocity of rolling stock /9/. Meanwhile, two types of braking are required - interval and sight (targeted). Interval braking provides the necessary intervals between cars for their safe passage through turnouts and braking devices in the classification bowl. Sight (targeted) braking allows the velocity of the car to be adjusted depending on the distance that it must go in the hump yard. Kozachenko, Bobrovskyi, Grevtsov, and Berezovyi /10/ describe a variety of factors (propulsion properties of railcars, their gravity, range, curves and straightaways in the study path along the profile of the yard, weather conditions, as well as the human factor) that affect the difficulty of railcar braking in hump yards. It also describes the purpose and importance of applying each braking position (first, second, third - RP1, RP2, RP3).

The transport task of determining the travel time and braking distance of a railcar in a brake position area was considered intractable /11, 12/. In articles /13, 14/, as well as in the existing methodology of structural and engineering humping calculations, this problem was solved using the concept of 'brake position power.' Note that the authors of works /13, 14/, when performing humping calculations in both high-velocity and braking areas, do not use formulas to determine railcar (negative) acceleration.

The engineering problem of determining braking time and distance of a railcar in retarder position sections is poorly studied.

The objectives of this study are building mathematical models of railcar movement based on classical provisions of theoretical mechanics and developing formulas to determine braking distance; using the modelling outcomes to confirm the correctness and applicability of mathematical models for railcar BA in second and third brake positions.

METHODS

This study offers four solutions to the engineering problem of determining the kinematic characteristics of a railcar based on the:

- basic law of dynamics for imperfect connections (D' Alembert's principle);
- theorem on the motion of the inertia center of a system of point objects;
- theorem on change in kinetic energy of a point object in final form;
- theorem on the change in momentum of a point.

We write the theorem on change in the kinetic energy of a point object on segment AB /12/, where the car can move, factoring in initial velocity v_0 as applies to the problem under consideration, in the form:

$$\frac{G}{2g}(v_B^2 - v_A^2) = A_{Fx} \, , \tag{1}$$

where: v_A is the velocity at point A (beginning of motion); v_B is velocity at point B (end of motion); G is gravity; g is gravitational acceleration of 9.8 m/s^2.

$$A_{Fx} = A_{Gx} + A_{Ff} \, , \tag{2}$$

where: A_{Gx} is a projection of the force of gravity Gx on the x axis with the movement of x_{AB} between points A and B,

$$A_{Gx} = G_x x_{AB} = G \sin\psi x_{AB} \, , \tag{3}$$

where: A_{Ff} is the work of friction force F_f (or generally any resistance forces F_r) with the movement of x_{AB} between points A and B, /12/.

$$A_{Ff} = -F_f x_{AB} = -k_f G \cos\psi x_{AB} \, , \tag{4}$$

where: $k_f = 0.25$ is the coefficient of friction of the wheels of a railway car on rail threads; ψ is the angle of inclination of section AB of the hump profile.

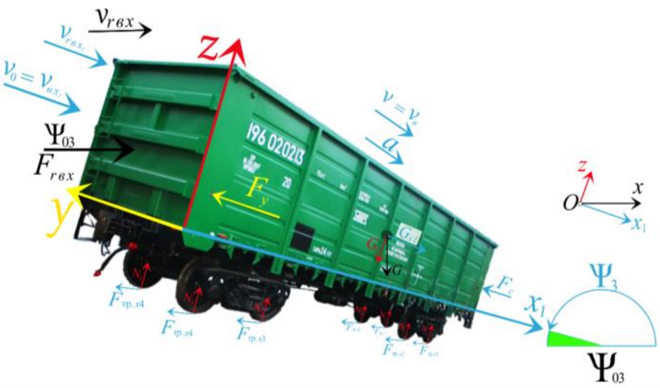

Figure 1. Mathematical model parameters of the railcar movement.

Figure 1 is indicated by: 0 - origin of mobile coordinate system $0x_1yz$, rigidly connected to the car; $0x$ - horizontal axis; ψ_{03} - angle of removal (descent) section 1TP of the hill; v_{rvx} - relative air speed; $v_0 = v_{vx1}$ - initial speed of car; v_w - speed of car; $F_S = F_{SR}$ - resistance force of the medium; N and $F_{TR.x}$ - normal and tangent components of coupling reaction (rail threads). $N = N_1 + N_2 + N_3 + N_4$ and $F_{TR.x} = F_{TR.x1} + F_{TR.x2} + F_{TR.x3} + F_{TR.x4}$, as parallel forces.

The increment ΔE of the system's kinetic energy is equal to the sum of the corresponding work of active forces A_{Gx} and constraint reactions A_{Ff},

$$\Delta E = A_{Gx} + A_{Ff} \, . \tag{5}$$

Substituting Eqs.(3) and (4) into Eq.(2), taking into account Eq.(1), after simplifications we can obtain a formula for determining the velocity of a car in a BA in a retarder position:

$$v_{Bi}^2 - v_{Ai}^2 = 2g(\sin\psi_i - k_f \cos\psi_i)x_{ABi} ,$$

where: i refers to the numbers of path profile sections (i = 1, ...9); v_{Ai} is the velocity at point A_i (the beginning of movement at the i-th section of the profile); v_{Bi} is the velocity at point B_i (the end of movement at the i-th section of the profile); ψ_i is the angle of inclination of section $AiBi$ of the hump profile; or when $x_{ABi} = l_i$, the braking distance on the i-th section of the profile is

$$v_{Bi}^2 = v_{Ai}^2 + 2g(\sin\psi_i - k_f \cos\psi_i)l_i . \tag{6}$$

Hence, when $v_{Bi} = 0$:

$$0 = v_{Ai}^2 + 2g(\sin\psi_i - k_f \cos\psi_i)l_i . \tag{7}$$

From Eq.(7) we obtain braking distance l_i,

$$l_i = \frac{v_{Ai}^2}{2g(k_f \cos\psi_i - \sin\psi_i)} . \tag{8}$$

If we consider that for small angles (less than 5°) in relation to the profile along the entire length of the hump yard path $\sin\psi_i \approx \psi_i = i_i$ and $\cos\psi_i \approx 1$, then Eqs.(6) and (8) will look like:

$$v_{Bi}^2 = v_{Ai}^2 + 2g(i_i - k_f)l_i , \tag{9}$$

$$l_i = \frac{v_{Ai}^2}{2g(k_f - i_i)} . \tag{10}$$

As we see, the braking distance l_i is directly proportional to the square of initial velocity v_{Ai} and inversely proportional to friction coefficient k_f and the slope of path i_i.

The absolute value of car (negative) acceleration ($|a_i|$) with equally slowed-down motion in the BA is found using the formula, /12/:

$$|a_i| = \frac{|\Delta F_{fi}|}{M_{r0}} 10^3 , \tag{11}$$

where: $|\Delta F_{fi}|$ is the resulting force under the influence of which the car's wheel pairs are forced to slide along the rolling surfaces of the rail threads and the brake buffers of the car retarder in BA in RP sections, /12/, defined as

$$|\Delta F_{fi}| = F_{xi} + |F_{ci}| , \tag{12}$$

where: F_{xi} is the force that moves the car into the BA in RP sections, taking into account the influence of a small magnitude tailwind force; F_{ci} in general refers to all kinds of resistance (resistance to dry sliding friction of the contact surfaces of the wheelset rims and the brake buffers of the car retarder, primary (or running) resistance, resistance from air and wind, snow and hoarfrost resistance) under the influence of which the car can be braked until completely stopped by the car retarder; M_{r0} is the resulting mass of the wagon with its load and non-rotating parts (i.e. the wagon body and railroad trucks) when the wheelset, forced by 'compressed' brake buffers in the car retarder in BA in RP sections, slides cleanly like the dry friction pair 'steel on steel.'

If we know the (negative) acceleration value $|a_i|$ from Eq.(12) with equally slowed-down car movement, then we can determine the velocity until the car stops using the velocity formula:

$$v_{fi}^2 = v_{Ai}^2 + 2|a_i|l_i . \tag{13}$$

Using the velocity formula

$$t_i = \frac{v_{Ai} - v_{fi}}{|a_i|} , \tag{14}$$

we can find the braking time t_i until the car stops, $t_i < t$, where t is the current time in seconds,

$$t_i = \frac{v_{Ai} - v_{fi}}{|a_i|} . \tag{15}$$

Thus, applying the theorem on change in kinetic energy of a point object in its final form in railcar braking areas in RP sections, using Eqs.(8) or (10) has made it possible to determine the railcar's distance in braking section l_i.

To calculate l_i , the following options are considered:

a) the direct entry of the first wheelset l_{Ai} and/or the wheel pairs of the front truck l_{st} into the retarder position section;
b) the entrance of the car to the site based on the car base's length,

which are necessary to establish the initial velocity of the car's entry into the braking area v_{Ai} = 3.57 m/s.

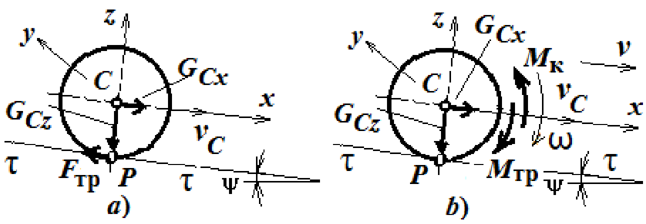

Figure 2. Schemes for determining the braking section for options: a) direct entry of first wheelset; b) entry based on car base length.

The car's braking distance can be determined using:

$$l_i = v_{Ai}t_i + \frac{1}{2}g(\sin\psi_i - k_f \cos\psi_i)t_i^2 . \tag{16}$$

The car's (negative) acceleration during equally slowed-down movement in the braking area $|a_i|$, unlike Eq.(12), can be defined as /12/:

$$|a_i| = a_f (i_{fi} - |w_i|) , \tag{17}$$

where: $a_f = const$ is the linear (negative) acceleration of the car during equally slowed-down movement in braking areas in RP sections, /12/, defined as:

$$a_f = \frac{G}{M_{r0}} 10^3 , \tag{18}$$

where: i_{fi} is a dimensionless quantity describing the slope of the hump profile in RP sections when taking into account the projected influence of tailwind force F_{wx}, defined as

$$i_{fi} = i_{fxi} + k_{wx} , \tag{19}$$

where: k_{wx} is a dimensionless quantity that takes into account the projected influence of small magnitude tailwind force F_{wx} on the x axis, contributing to the accelerated movement of the car in fractions of G; $k_{wx} = 0$ in the absence of wind; $|w_i|$ is the specific resistance to movement in braking areas in RP sections, /12/.

Now, Eq.(16) in accordance with Eq.(17), can be written as:

$$l_i = v_{Ai} t_i + \frac{1}{2} a_f (i_{fi} - |w_i|) t_i^2 , \tag{20}$$

and taking into account Eqs.(17) and (20), it can be represented as:

$$l_i = v_{Ai} t_i + \frac{1}{2} a_f |a_i| t_i^2 . \tag{21}$$

Railcar stop time t_i is defined as:

$$t_i = \frac{v_{Ai}}{g(k_f \cos\psi_i - \sin\psi_i)} . \tag{22}$$

After comparing Eq.(16), obtained according to the theorem on the the motion of the inertia centre of a system of point objects, and Eq.(8), derived using the theorem on change in kinetic energy of a point object in its final form, with the formula of the distance taken from elementary physics Eq.(21), they are clearly different in form.

The relative error in calculating the car braking distance using Eqs.(21) and (16) is 1.52 %, and with Eqs.(21) and (20) it is 9.2 %, which is within a reasonable window of accuracy for engineering calculations.

The car braking distance calculated by Eq.(21) is $l_i = 13.35 \approx 13.4$ m, and with Eq.(9) $l_i = 12.86 \approx 12.9$ m. The relative calculation error is 3.7 %, which confirms the correctness of Eq.(8) outcomes.

RESULTS

To analyse the results, graphic representations of braking distance versus braking time are constructed based on Eqs.(21), (16) and (20) with t varying from 1.0 to 2.0 with a step of $\Delta t = 0.1$ s (Fig. 3).

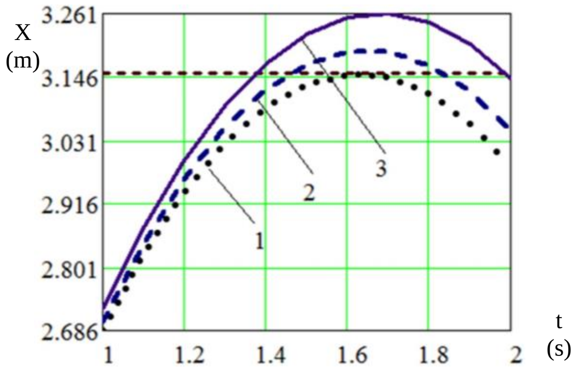

Figure 3. Dependencies of braking distance on braking time, according to: 1- Eq.(21); 2- Eq.(16); 3-Eq.(20).

In Fig. 3, graphs show increasing quadratic function until the car stops. Maximum values for braking distance l_i = 3.152, 3.195 and 3.262 m correspond to braking times t_i = 1.625, 1.648 and 1.682 s.

Graphical dependence of braking distance on velocity, constructed using Eq.(8) with a variation of v_{Ai} from 0 to 5 m/s with the step $\Delta v_{Ai} = 0.25$ m/s, is shown in Fig. 4.

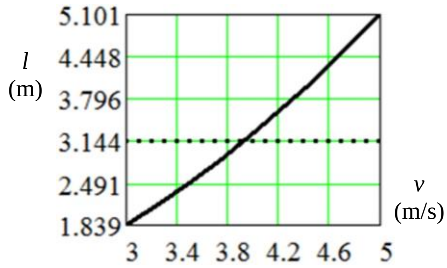

Figure 4. Dependence of braking distance on velocity.

Figure 4 shows how at $v_{Ai} = 0$, the braking distance is $l_i = 0$. This confirms the argument about the importance of a railcar entering the braking area at retarder positions with an initial velocity of $v_A > 0$, otherwise the car stops completely before the car retarder is engaged.

When $v_A > 0$, the kinetic energy E_0 of the car with mass M and initial velocity v_A will be completely expended to overcome the work A_r and drag force F_r which occur when the car retarder is engaged.

In turn, the work of various resistance forces A_r accumulated on the rim of the car's wheelsets, on rail threads and on the brake buffers of the car retarder, will dissipate into the environment in the form of heat. When the car is completely stopped, i.e. $v_B = 0$, the following condition will be met: $E_0 + (-A_r) = 0$.

DISCUSSION

In /11/ it is noted that the existing methodology for humping calculations in hump yards /10, 13, 14/, is mainly aimed at determining the height of the hump yard from its top to the calculated point. Such kinematic parameters of railcar movement as (negative) acceleration and movement time in the braking area are not taken into account. The car's braking distance is also not calculated.

Calculating a railcar's braking distance using Eq.(8), based on an elementary physics velocity formula Eq.(10) made it possible to observe that using the same initial velocity value gives the same results.

In summarizing the results of our calculations of railcar braking time and distance using Eqs.(8), (10) and Eqs. (16), (20), (21)-(23), we can note that using the same initial velocity values has yielded acceptable results for engineering calculations. This confirms the correctness of our mathematical models as applied to a railcar BA in all retarder position sections (RP1 and RP3).

Mathematical models for the movement of a railcar (chain) along the entire length of an RP section of a hump yard under the influence of a small magnitude tailwind made it possible to develop a new methodology for calculating railcar dynamics in this yard section, which allows the kinematic parameters of the car (rate and velocity) for a given geometric parameter (length) to be determined for the section of yard under consideration.

Calculations to determine the kinematic parameters of a car according to the new methodology made it possible to determine, for a known travel distance along the entire length of the RP section (*per pass*), the time required for uniform acceleration of a car in this section of the hump.

CONCLUSION

The task of a railway station is to sort railcars from incoming trains and build outgoing trains by appropriately grouping specific classifications of cars. The classification process involves the physical movement of trains, cars and engines between receiving tracks, hump yards and departure tracks. When designing a hump yard profile and the corresponding retarder system, the hump should be high enough to provide railcars with enough kinetic energy to easily overcome rolling resistance and track resistance and roll an estimated distance beyond the touch point.

When the car reaches the beginning of the track (the touch point), it may need to roll back a distance of 30 to 1000 m, depending on the number of cars already standing on the track it is switching to. Due to the large difference in distances and the behavior of cars during rolling, the velocity of each car should be regulated so that no serious collisions occur. At the same time, the cars must have enough energy to connect with other cars that are already waiting on classification routes. The results of this study can be used to adequately address problems in calculating and designing hump yards.

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