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## **SIMULATION OF DYNAMIC PROCESSES OF INTERACTION OF CAR AND RAILWAY TRACK DURING TRAIN PASSAGE OF CURVED SECTIONS OF THE TRACK**

**Summary.** In this article, the considered principles for the development of calculation schemes and the subsequent formation of vibration equations are a special case of classical simulation of motion in space of solid body systems connected in space by kinematic connections. This approach is useful with limited computational capabilities and an assumption of the relatively small body movements inherent in railway crew bodies, and reduces the task of motion research to analysis of fluctuations. Various design schemes of freight car truck, mathematical modeling of systems dynamics, "crew-track" safety of freight car movement and withdrawal under different technical conditions of running parts and track are considered in this work.

### **1. INTRODUCTION**

Simulation of dynamic processes of interaction of rolling stock and railway track became possible with the help of modern computer equipment evaluation of efforts acting on elements of a track when considering the operation of rolling stock-track system in dynamics. To achieve this, it is necessary to build special models of rolling stock and railway track.

Most research works have examined the dynamics of rolling stock and the path for which deformations for computed forces were evaluated by finite-element analysis. During the research, the parameters of the track and rolling stock, corresponding to the target function, were found. As reported in the reports, it was noted that the results of the calculations of the efforts on the track elements and rolling stock were well in line with their actual observed values. In connection with the integration of Kazakhstan railways into the European and Asian railway network, it is necessary to carry out calculations by modern ISO 9000/9001 certified methods, understood and accepted in the countries of the common market [1, 2]. Despite the great potential of modern personal computers, calculations of the interaction between the track and rolling stock still have to be carried out according to the stages given in Table 1. It is possible that soon (with a significant increase in computer power) it will be possible to solve these tasks for the whole system at once.

### **2. ANALYSIS OF FREIGHT CAR PARK**

The main task in solving the problem of safety of rolling stock movement is to find effective methods and means, which make it possible to describe with a high probability the dynamic processes that arise during rolling stock movement along the railway track taking into account the real states of the car-track system. These methods and means should provide not only qualitative but also quantitative information

about these processes and, at the same time, be adequate in terms of real analogues of the movement of natural cars on concrete sections of the way. The obtained results should be in the form of graphical oscillograms showing the change of each parameter during the car movement time. Initial technical data for the study shall describe in detail the values of mechanical and physical properties of the rolling stock-track system [2, 3].

At present, with powerful computing computer systems, the general problem of non-linear car fluctuations in motion, both in straight and curved sections of the real path, can be solved.

The Company's freight fleet of cars is 65,521 units. The main share of the car park is gondola cars: 49.8%. The remaining types of cars in the car park are distributed as follows: covered - 15.2%, platforms - 5.1%, tanks - 9.9% and others - 20.0%. The accepted axial load of freight wagons is not more than 230 kN.

Since the 1960s, the United States, Canada, Australia and other countries have produced four-axle cars with a capacity of 90 tons (axial loads of about 294 kN) and operate a car fleet with loads of up to 340 kN on trains weighing 12 - 20 thousand tons. Foreign manufacturers widely use aluminum alloys to make bodies of freight cars, which allows to significantly reduce the weight of car containers to 17 - 23 tons with a load capacity of 117 - 120 tons. For comparison, 1520 mm gauge freight cars have a relatively low load capacity (60-70 tons), require additional costs related to loading, unloading and mounting of goods, have small inter-repair runs and a low level of specialization; the axial load is 230 kN and the weight of containers is 240 kN [3].

### **3. EVALUATION OF DYNAMIC FORCES, ACTING ON THE RAILS FROM ROLLING STOCK, USING THE ADAMS/RAIL SOFTWARE SYSTEM**

Deformations of the railway track under load and imperfections of the track, its retreat within the limits of maintenance standards together with imperfections of the rolling stock, uneven wear and defects of the wheels of the cars - all this causes fluctuations of the rolling stock during its movement along the track. The oscillations contribute dynamic additives to the static load. The value of these additives can be determined by calculation. Let us take a look at the simulation steps in Table 1 in more detail [3, 5].

The ADAMS/Rail software system provides for the introduction of the main characteristics of the rolling stock, affecting its smooth movement and oscillation (characteristics of masses, suspension springs and dampers taking into account their location relative to the car body, wheel profile) and railway track section characteristics (transverse rail profiles, rail head irregularities along the track, longitudinal rail slopes, slope, outer rail elevation in the curve), spiral lengths, curve radii and circular curve lengths.

Enter the track description file as input (longitudinal track profile - longitudinal and vertical coordinate, line plan, spatial geometry of right and left rail head - wheel and rail contact lines - single-point or two-point contact, file describing irregularities on rail head rolling surfaces), the car wheel cross-profile description file, files describing the characteristics of all subsystems and elements of the car [6].

The presence of differences in the vertical coordinates of rail threads within the curve, bearing in normative documents the name "elevation of the outer rail" (measured usually in millimeters) and described in ADAMS/Rail parameters of block [CANT\_ANGLE\_PATH] of the railway description file (\*.trk), is specified by the angle of inclination of the segment connecting the right and left rail thread relative to the horizontal, measured in radians.

Mutual orientation of wheels and rails, and adopted coordinate systems for wheels and rails are shown in Fig. 3. The left and right wheels of the car are considered separately with their coordinate systems. As an example, Fig. 2 shows a diagram of the attachment of the springs and dampers of the freight car cart.

The calculations take into account the peculiarities of the rail (its geometric deviations in plan and profile from the design position, which can be modeled by sinusoids, trapezoidal and step functions, or by introducing a table of coordinates of points of the rail axis, by introducing characteristics of spectral

densities of the deviations). Displacement of wheel and rail contact surfaces by value "u" (Fig. 8) is taken into account, which is determined by calculation in accordance with the values of geometric deviations of rail threads and dynamics of wheels movement.

Table 1

Simulation Stages

No.	The applied program complexes	The applied models	Purpose of a stage
1	ADAMS/Rail, COSMOS/M	Car model on the track	Evaluation of dynamic forces acting on the rails from rolling stock. Calculation of deformations and stresses of railway track.
2	COSMOS/M	Track model, with loads from rolling stock	Load distribution from rolling stock by track design. Determination of the values of the forces of interaction of the elements of the track.
3	COSMOS/M	Fastening knot model	Strenuously deformed condition of the intermediate rail attachment unit and determination of loads on reinforced concrete tie.
4	COSMOS/M	Model of reinforced concrete sleeper in ballast prism	Calculation of the stress-deformed state of reinforced concrete tie, determination of optimal parameters of tie, determination of the distribution of tie pressure on ballast, distribution of stresses on layers of ballast prism and in layers of non-rolled materials and in plates - insulation.
5	COSMOS/M	Model of laminated earth web on elastic base	Calculation of strenuously deformed state of earth bed, extraction of the dense core of fill and assessment of stresses in the zones of contact between the core and the sloping part of the fill, assessment of the impact of fill cavities or fill base on its precipitation

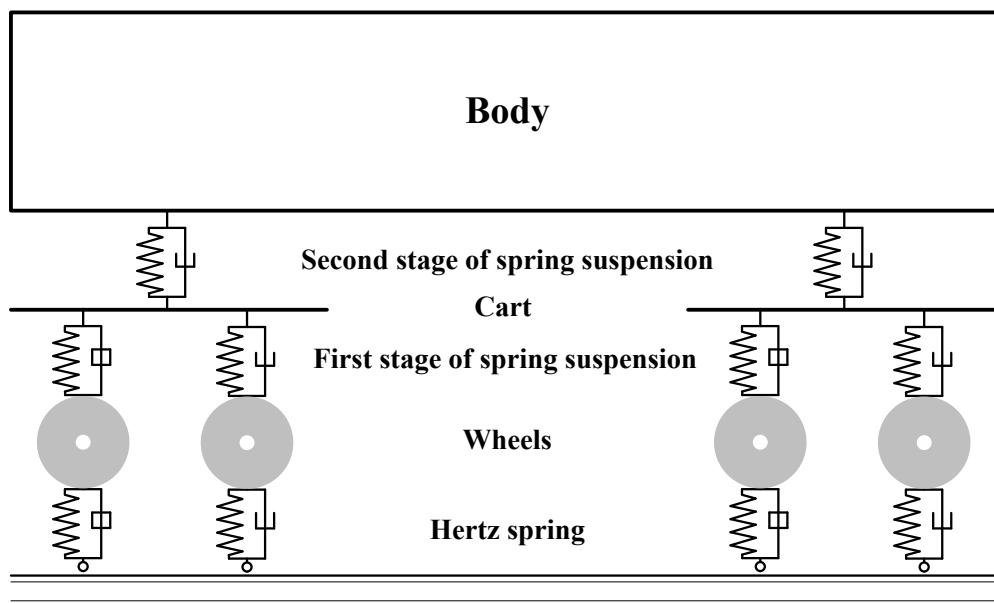


Fig. 1. Design diagram of the car

Fig. 5-7 and Table 2 show coordinate systems and defined parameters, wheel and rail contact characteristics [7, 8].



Fig. 2. Diagram of attachment of springs and dampers of a freight car cart

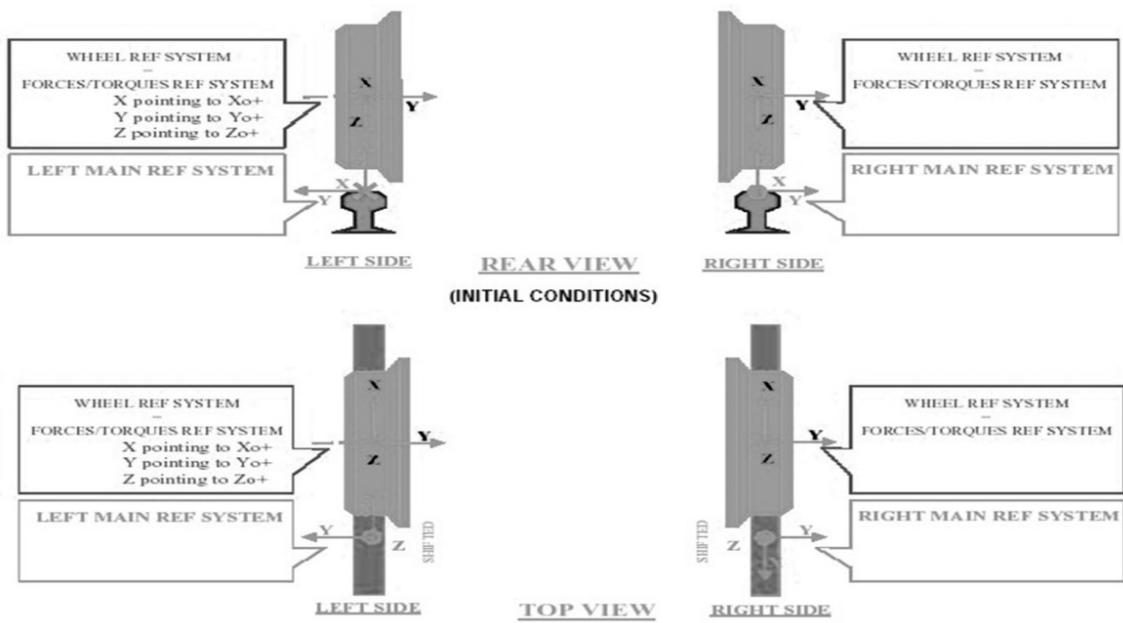


Fig. 3. Orientation of wheel-rail system elements along the left and right rail threads

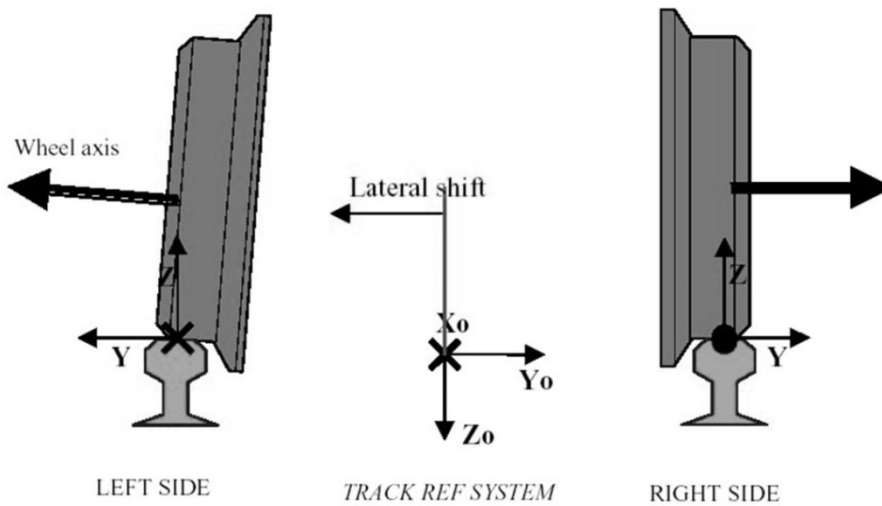


Fig. 4. Coordinate system orientation

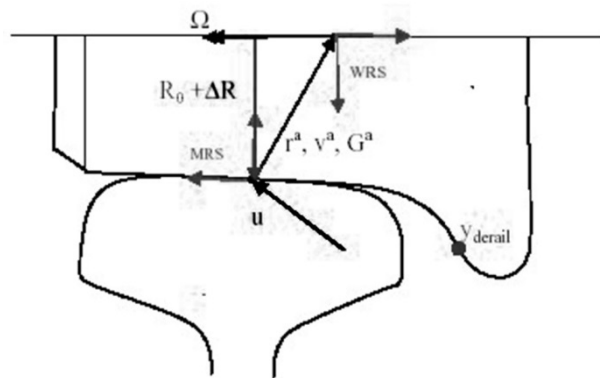


Fig. 5. Orientation of local coordinate systems at the wheel–rail contact point

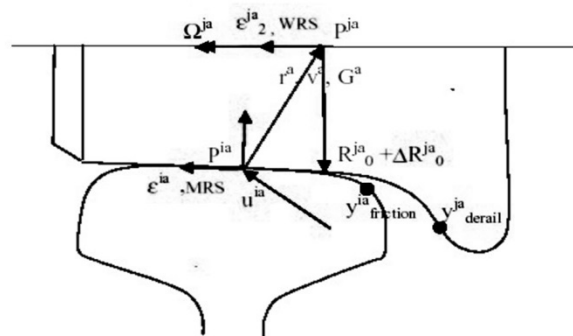


Fig. 6. Wheel and rail coordinate systems, defined parameters

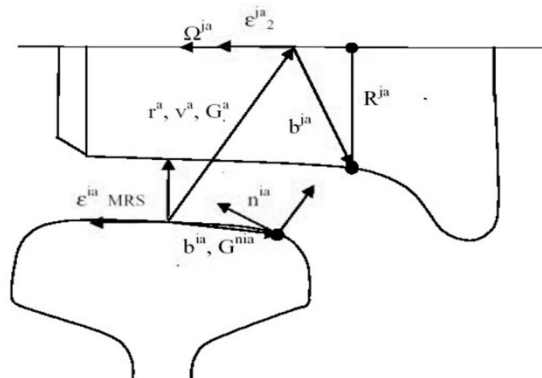


Fig. 7. Wheel and rail contact parameters

In the first approximation, the task can be solved for the measured plan and profile of the path obtained by passing the load device with the specified load on the axle [9]. According to Professor G.M. Shahunyantz, the error from the representation of an absolutely rigid path in the forces of interaction between the rolling stock and the way does not exceed 5% because the rigidity of the track as a whole is more than an order of magnitude higher than the rigidity of the suspension of the rolling stock; however, this approach does not allow determination of the redistribution of dynamic loads and deformations between individual elements of the track.

All quantities between "discrete points" for which data are input in the source data files can be interpolated by various methods (linearly, quadratic ally, etc., splines and various other functions).

The dynamics of rolling stock when passing curves using the ADAMS/Rail complex can be studied by non-stationary processes of passing curves by the car. Until recently, these crucial issues were not resolved, and the recommended amount of elevation of the outer rail in the curve was determined for steady-state carriage movement. In fact, the established movement of the car is not a rule, but an

exception. With short spirals and short straight inserts between the reverse curves, the car's oscillations do not have time to come to a calm state. Deviations in the geometry of the track (vertical and lateral irregularities) cause additional oscillations of the wagon [10, 11].

Table 2

## Parameters describing wheel–rail contact

$e^{ia}$	Coordinate system for the contact element on the rail. This is the same system as that for the transverse rail profile (MRS)
$p^{ia}$	Beginning of coordinates of MRS
$e^{ja_2}$	Wheel orientation vector. Rotation of wheel profile relative to this axis forms the wheel body. Direction of rotation as with wheel profile. Determined in the relative coordinate system of WRS wheel
$p^{ja}$	The initial orientation of the vector $e^{ja_2}$ in the WRS coordinate system
$n^{ia}$	Coordinate system for determining the position of the contact location on the rail
$w(e^{ja_2})$	Function, describing the wheel cross profile, value $R^{ja_0}$ added
$r(e^{ja_2})$	Function, describing the cross profile of the rail
$y^{ia}_{friction}$	Range of change of friction characteristics along the rail
$y^{ja}_{derail}$	Limit position of wheel contact point on the rail corresponding to wheel derailment. If the contact point is lifted on the wheel ridge, a derailment message appears
$R^{ja_0}$	Distance from the wheel profile coordinate system to the wheel axis of rotation (not always to the wheel axis)
$\Omega^{ja}$	Wheel angular speed
$r^a$	Vector between $p^{ia}$ and $p^{ja}$
$v^a$	Vector of the relative speed of wheel movement on the rail
$G^a$	Rail Contact Coordinate System and Rail Coordinate System Transformation Matrix
$\Delta R^{ja_0}$	Eccentricity of a nozzle of a wheel
$u^{ia}$	Amount of deviation of the rail in plan (displacement only)
$b^{ja}$	Contact point vector on wheel
$b^{ia}$	Rail contact point vector
$G^{mia}$	Transformation matrix between the rail contact coordinate system and the rail coordinate system
$\mu$	Friction coefficient
$w$	Coefficient of rigid sliding
$\lambda$	Equivalent extremity
$\varepsilon$	Contact Point Angular Parameter
$\sigma$	Parameter of an angle of rotation
$e_0$	Half-wheel diameter, nominal distance to the wheel center line
$r_0$	Rated rolling radius of the wheel
$\delta_0$	Nominal contact angle in the MRS system
$u$	Cross shift of a wheel
$\varphi$	Integrated variable
$\alpha$	Angle of wheel run on rail
$\delta_u$	Undeformed distance between the rail profile and the contact line on the wheel
$w_{el}$	Elastic deformation of contacting wheel and rail

As the results of the measurements show, plane-parallel movement of the car body is not observed even when the car moves along straight sections. Due to deviations on the way from the standards of content and due to irregularities on wheels, the car constantly oscillates; the directions (forms) of oscillations depend on the forcing frequencies and directions of loads. As can be seen from the data in

Fig. 9, for example, the side roll is sharply increased and then sharply weakened on even sections of the track.

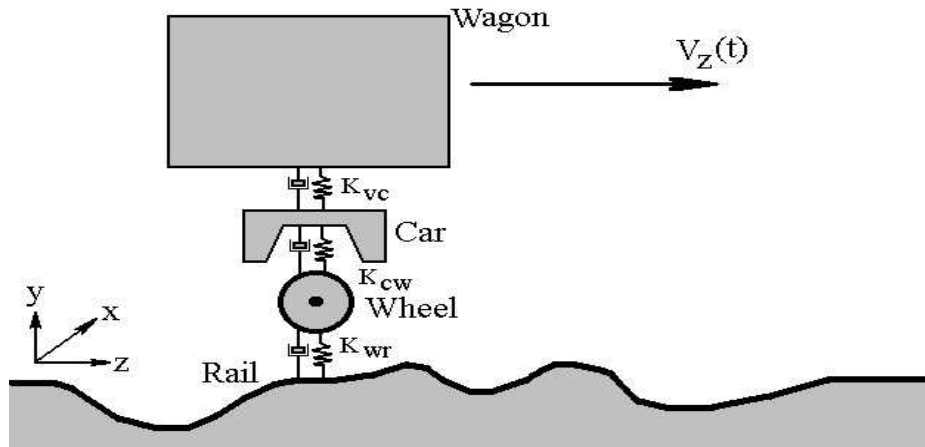


Fig. 8. Example of model for the determination of vertical dynamics of rolling stock in the case of an absolutely rigid (non-deformable) track

As an example, Figs. 10, 11 and 12 show some of the vertical and lateral force calculations carried out in the ADAMS/Rail software system for the movement of a freight car along a 1500 m radius curve and adjacent straight sections (outer rail elevation – 40mm, speed - 10-70 m/s).

The simulation allows estimation of the unloading of the car wheels during its movement in the curve, which is impossible for traditional calculations of the way to determine, and such estimates are necessary to determine the level of traffic safety.

By introducing the actual characteristics of the geometry of the path, which always has some kind of deviation from the content standards, it is possible to estimate the technical requirements for the rolling stock and the content standards of the track [12, 13].

Fig. 9 shows the graphs of vertical and transverse forces change in the wheel–rail contact for the left rail (thrust thread) and the right rail (internal in the curve rail). Changes in the relative maximum, average and minimum values of forces in wheel–rail contact for a passenger car are shown.

As the calculations have shown, the dynamics of the car when fitting into the curves depends significantly on the geometry of the rail threads and the speeds of movement (Figs. 9, 11-14). As can be seen from the analysis of Figs. 11 and 13, the maximum loads on the external rail thread increase nonlinearly with an increase in speed. When speeds of 65-70 m/s are reached even on a curve with a strictly maintained geometry and complete absence of irregularities on the surface of the rails, there is a complete de-loading of individual wheels (Figs. 12 and 14), which indicates a high probability of the wheelset falling or even crashing. This calculation illustrates the procedure [14], and its results after analysis taking into account other factors will allow to assign the maximum permissible speed of movement on this curve not faster than 50 m/s or to check the efficiency of elevation increase of the outer rail of the curve.

It can be seen that when the speed of movement increases to 70 m/s, the vertical forces acting on the thrust thread of the curve increase by 2.12 times compared to the static ones, and the de-loading of the wheel moving on the inner rail at the fluctuations of the car can reach 100%. Starting from speeds greater than 40 m/s, there is a large increase in the lateral force applied to the abutment thread. It is obvious that for this car design, the wheel stability on the rail at a speed of 70 m/s is not ensured [15, 16]. To ensure the stability of the car, it is necessary to change the track parameters or characteristics of the car suspension.

In order to assess the change in the force effect of the car on the way when fitting into circular and reverse curves (S-shaped curve with a radius of 300 m, corresponding to the movement on the side path along the arrow translation of the 1/11 mark), simulation of the passenger car movement is performed (Figs. 10, 15).

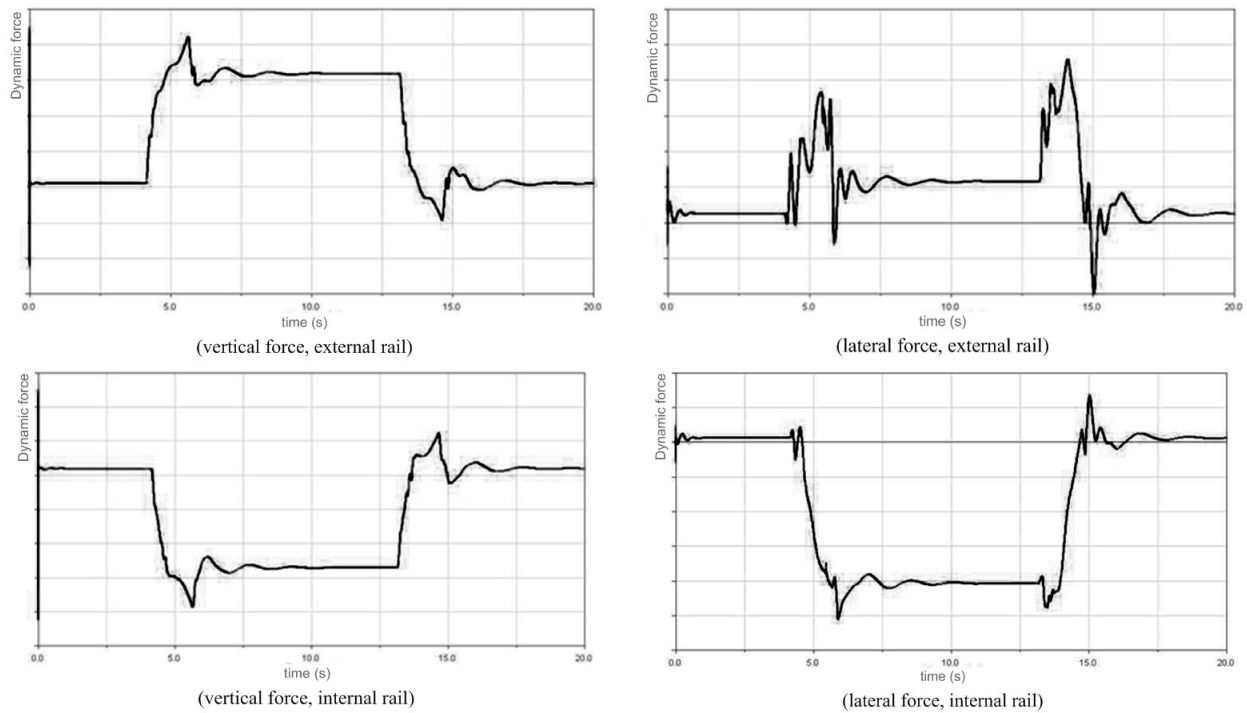


Fig. 9. Dynamic forces acting on rail threads in curve  $R = 1500$  m at  $v = 30$  m/sec

As the analysis of the calculation results for short S-shaped curves with a radius of 300 m with short (less than 25 m) straight inserts between curves showed, no steady motion was observed on the curve. As the speed of movement increases, the maximum and minimum loads on the rail threads change nonlinearly (Fig. 15).

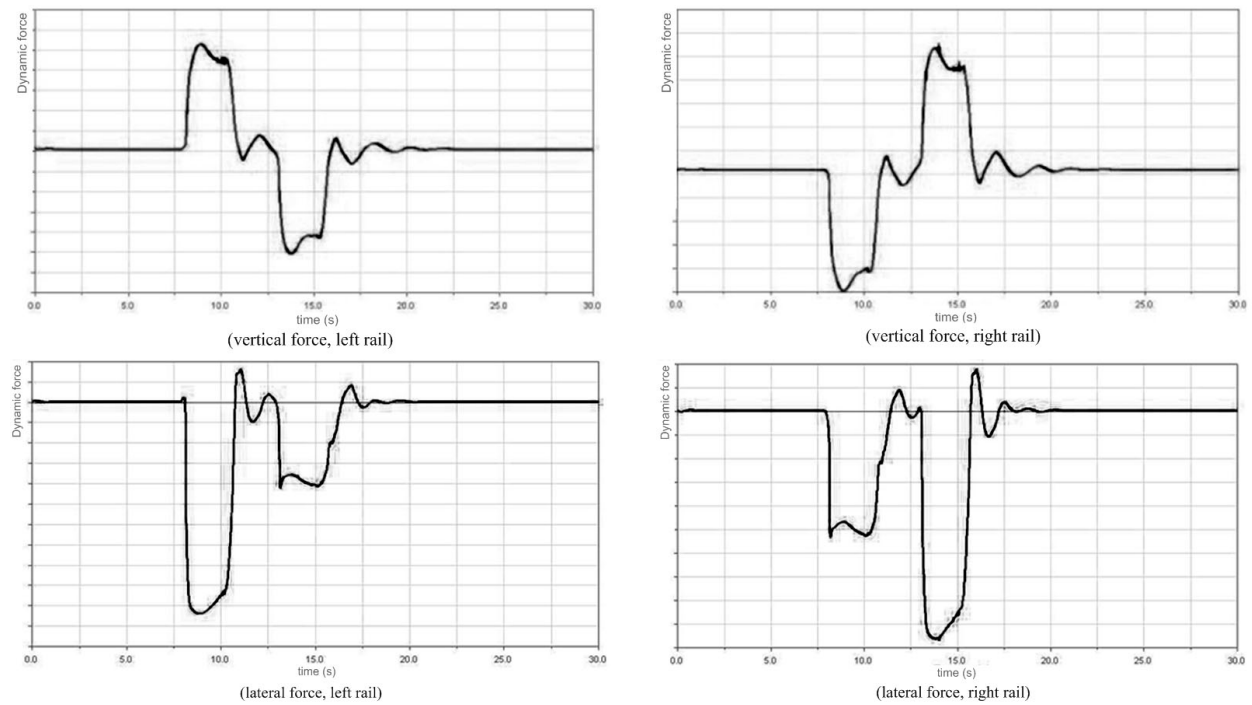


Fig. 10. Dynamic forces acting on rail threads in an S-shaped curve of 300 m radius from the first axis of the first bogie (in the course of movement) at a speed of 10 m/s



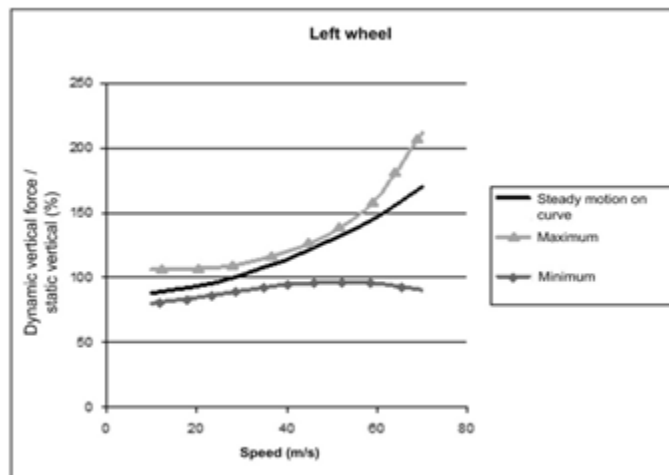


Fig. 11. Dependence on the speed of vertical dynamic forces acting on the external rail thread in the curve of radius 1500 m at an elevation of 40 mm

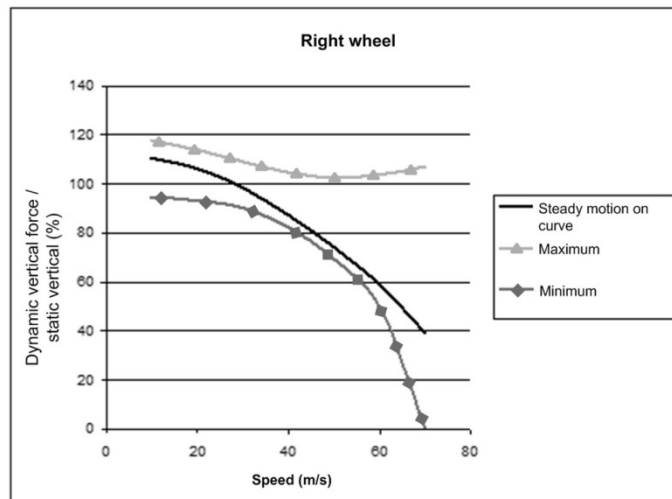


Fig. 12. Dependence on the speed of vertical dynamic forces acting on the internal rail thread in the curve of radius 1500 m at an elevation of 40 mm

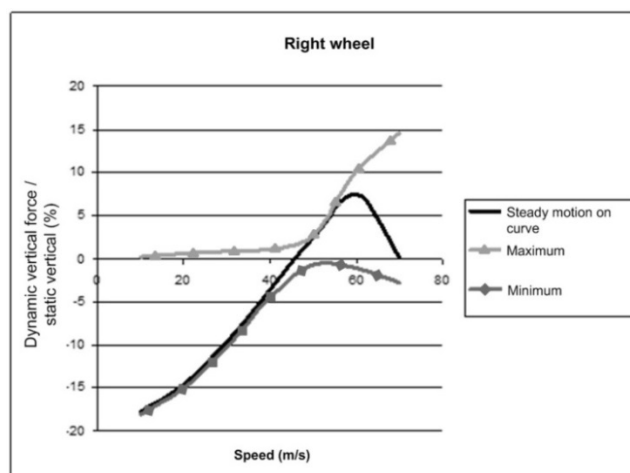


Fig. 13. Dependence on the speed of transverse dynamic forces acting on the external rail thread in the curve of radius 1500 m at an elevation of 40 mm

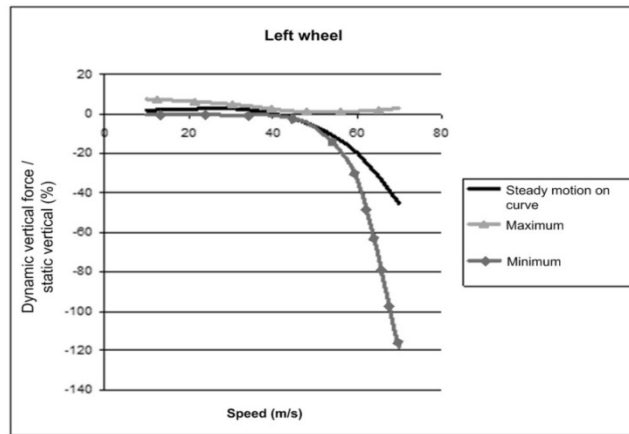


Fig. 14. Dependence on the speed of transverse dynamic forces acting on the internal rail thread in the curve of radius 1500 m at an elevation of 40 mm

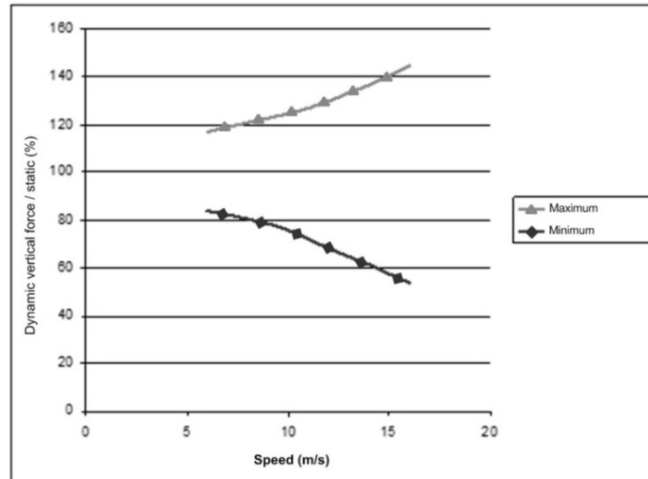


Fig. 15. Dependence on the speed of vertical dynamic forces acting on rail thread in an S-shaped curve of 300 m radius

Figs. 10 and 15 show the results of calculations of vertical and lateral forces at movement of the specified car from the straight line to the curve with a radius of 300 m with a spiral length of 3 m and zero elevation of the external rail (simulation of movement of the car with a straight line on the shift to the side track). If we compare the change of these forces along the switch path without joints with the change of forces at the entrance to the circular curve with the elevation branch of the outer rail along the spiral, we see a significant difference. When the car fits into the switch (where there is no elevation of the outer rail), the vertical and lateral pressures of the wheel on the rail are much higher and the car oscillations do not decrease.

Determination of the dynamic characteristic of the car on the curved section of the track is carried out with mismatch of the outer rail elevation retraction and curvature retraction.

In the practice of the current content of circular curves, it is common to require the end of the outer rail elevation retract to coincide with the beginning of the circular curve.

The magnitude of the elevation of the outer rail in the circular curve is determined by the known formula:

$$h = 12.5 \cdot V_{\text{spec}} \cdot \frac{2}{R}, \quad (1)$$

where  $V_{\text{spec}}$  is the reduced train speed in km/h and  $R$ — is the curve radius, m.

The established ideas about the equilibrium of centrifugal and centripetal forces from the elevation of the outer rail arose from the consideration of the "static picture of the balance of forces;" when the passenger car follows a long circular curve, there is a dynamic equilibrium. In a freight car that is not normally transported at a speed of more than 80 km/h, the effect of spring suspension begins to be affected usually at speeds of more than 60 km/h, and at lower speeds, the freight car moves as an uncompressed mass and the condition accepted by the CP774 for it is fair. The passenger car does not have time to come to balance as quickly as the cargo car and, for it, the condition does not reflect the essence of the processes of vibration attenuation at the entrance and exit from the curve. On lines where passenger traffic prevails, these circumstances need to be taken into account when planning curve straightening.

For trains with less and more speed than the given one, the elevation of the outer rail turns out to be excessive or insufficient than that determined by the formula. Some heavy-duty cranes and other heavy equipment in accordance with the standards of JSC "NC" KTZ "cannot be transported along railway tracks if the elevation of the outer rail exceeds 80 mm. Lowering the elevation of the outer rail in the curve only for a single pass of heavy equipment is very not favourably; thus, an attempt is made to avoid excessive overestimation of the elevation of the outer rail in the distance of the track.

According to the method of elevation of the outer rail, there are no general opinions in the curved sections of the railway tracks. Some experts believe in the benefits of overestimating the estimated elevation by about 20-30%.

It is obvious that further research is required on the assignment of the elevation of the outer rail of the curve, but with a more general approach with consideration not of "statics," but of the dynamics of the processes taking place.

As shown in the previous section, the oscillations of the passenger car in short curves of small radius do not have time to fade and there must be different criteria for assigning elevation than the condition of the CP774.

#### 4. CONCLUSIONS

Analysis of the results of numerical calculations of the change of the pressure of the wheels of the passenger car on the outer and inner rails of the circular curve showed that the change of the speed of movement of the car in the curve has a significantly greater impact (up to 40-50% at a speed of up to 40 m/s) on the change of vertical loads of the wheels on the rails than the mismatch of the points of the end of the elevation retraction and curvature retraction.

At the circular curve, there is not decreasing of the car oscillations at the points of the end of elevation retraction coinciding with the end of the spiral. Usually, at least 2-5c is required to decrease the car's oscillations and form a dynamic equilibrium.

Usually, when calculating the interaction between the track and the rolling stock, the hypothesis of continuous rolling of the wheel on the rail is considered, which is not always fair. Modeling in the ADAMS/Rail software complex does not require this hypothesis and allows to determine cases when continuous rolling is broken, which happens not only in joints and in the presence of sliders on wheels but also in certain combinations of irregularities on the way.

According to analysis of results of investigation of vertical oscillations of rolling stock-track system, when moving along short waves of periodic irregularities with significant amplitudes of dynamic equilibrium of interaction does not occur (in this case), local tear-off movement of wheels and sharp increase of loads (3-6 times more than static).

These examples show a significantly greater possibility of calculation by numerical methods to correctly reflect the process of interaction of the track and rolling stock and to assess traffic safety using the ADAMS/Rail software complex and the possibility of scientific justification of technical requirements of mechanical parameters of cars and track and their content standards. Traditional calculations do not offer such possibilities.

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