



Study and Monitoring of Driving Forces that Arise at the Contact Point Between the Wheel Border and the Rail during the Motion of the Rail Vehicle on a Curved Path of the Railway Line

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Abstract

One of the most acute problems faced today by Railway Transport Companies is the rapid wear of locomotive wheel borders in the freight transport. Hence, the purpose of this article is to find ways to reduce the forces acting on the wheels of railway vehicles, generally during their movement in small radius turns. For means of simplification has been assumed that the train is moving at uniform speed during a curved path. Analyzing the scheme of action of forces on the pair of wheels, we see that one of the ways to reduce this consumption phenomenon is to reduce the parasitic moment of frictional forces. With reference to the practice of roller-contact studies, we modeled mathematically the redistribution of frictional forces at the point of contact of the wheel – rail pair and realized two computer models for two variants of wheel profiles: conical and curved.

Keywords: Modeling, dynamics of railway vehicles, wheel and rail profile, wear reduction

1. Introduction

The movement of a railway vehicle in the curved sections of the line is associated with the emergence of moments, which impede the rotation of the carriage during the movement and lead to an increase in the driving force on the curb of the wheel. The larger this driving force, the greater the consumption of the curb.

Therefore, if we want to reduce the consumption of the curb (which is the main reason that during the re-profiling of the rolling ring we are forced to remove a lot of material in it), and to improve traffic safety, we must reduce this driving force during the movement of the vehicle in the curved parts.

The motion stability in the straight sections of the railway line, and the normative forces during the movement in curved sections, are related to the dynamic properties of the vehicle.

To enable the study of this phenomenon, it is necessary to study the contemporary literature on this problem.

The suspension system serves us precisely to achieve the stability of the movement. This system performs this function through the moment of rotation of its elastic elements, and the moment of frictional forces that oppose the rotation of the carriage in relation to the suspension module in the horizontal plane. Also, these frictional forces that arise

during the movement in sharp turns, increase the driving forces acting on the rim of the wheel, bringing an additional consumption of it.

The movement of a railway vehicle in curved parts of the line has been studied by numerous researchers in this field. In order to make this problem as simple as possible, simplifications have been made by researchers, for example the rolling rings have been taken as cylindrical rings, the coefficient of friction is constant, etc., which has brought a reduction of the problem to a system of transcendent equations, the solution of which is numerical [1, 2].

Therefore, to describe the movement in the first place, the dynamic task must be solved, which is related to the movement in straight and curved sections of the road, turning the problem into the construction of a linear model, using extensively linearization. For this we must build a model of stable motion in curved parts, taking into account the constructive features of objects that interact during motion, as described in references [3, 4].

During the movement of the vehicle in the curve, in the contact of the wheel-rail pair, dynamic forces act, which cause us to drag the wheel on the rail in the transverse direction (Fig. 1).

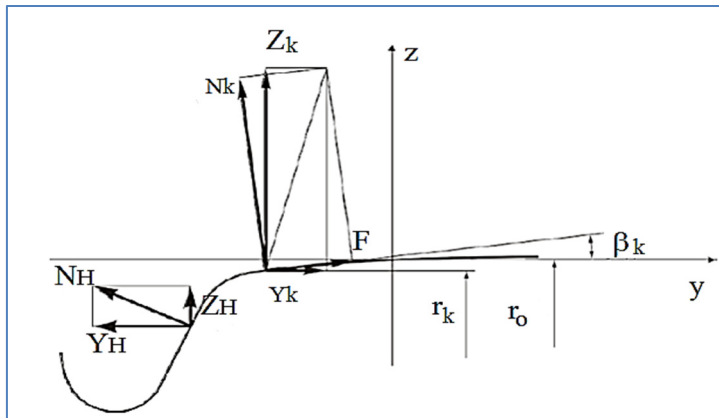


Fig. 1. Forces acting at the point of contact of the railway wheel system, during the movement of the vehicle in curved parts, or in lines with transverse irregularities.

Referring to the authors dealing with this problem, we can say that the wheel of the first pair of wheels that travels on the outer rail of the curve, has contact with the rail both on the treading part and on its side with the curb. Thus, the interaction of the wheel-rail pair with two points of contact is realized: namely one on the rolling surface and one on the curb.

Since the pair of wheels is positioned on the carriage in the longitudinal and transverse directions, then the frictional forces F (Fig. 1) of all the wheels of the vehicle form an additional moment, and as a result we have the increase of the driving force Y_H .

Therefore, if we seek to reduce the driving force Y_H we must reduce the moment of the frictional forces, and transfer the work of the lateral forces to the treading part of the wheel contact surface.

The model of wheel-rail interaction is a model which can be described in terms of a nonlinear theory of wheel axle drag according to equation (1) [5].

$$u_{yijk} = \psi_{ij} - \frac{1}{v} (\dot{\eta}_{ij} - \dot{\eta}_{Rijk}) \quad (1)$$

where ;

i, j, k - indices, which determine the number of the carriage, the pair of wheels and the wheel ($k = 1$ - outer wheel, $k = 2$ inner wheels);

$\eta_{-}(i, j)$ - the speed of transverse displacement of the wheel pair;

ψ_{ij} - Angle of deviation of the pair of wheels from the radial position

$\eta_{-}Rijk$ - the speed of transverse displacement of the railway track which is below the wheel with index k .

As seen from the above equation, the transverse slip in curves depends mainly on the angle of rotation of the wheel pair, while moving in curved parts.

Also the movement of the wheel pair will be accompanied by the longitudinal slip of the wheel, which refers to the non-uniformity of the road and wheels, equation (2).

$$u_{xijk} = \Delta r_{ijk} * r_0^{-1} + (-1)^k * b_s (\rho_{ij} - \dot{\psi}_{ij} * V^{-1}) \quad (2)$$

where ;

Δr_{ijk} - change of radius of the circle of rotation during transverse displacement, of the pair of wheels

r_0 - average radius of rotation of the wheel;

ρ_{ij} - Corresponding curvature of the road under the pair of wheels;

$\dot{\psi}_{ij}$ - the speed of inclination of the wheel pair;

b_s - half the distance between the wheel roll circles.

Consequently, the multiple magnitude of the velocity of the wheel on the rails in relation to it can be written by equation (3).

$$u_{ijk} = \sqrt{u_{xijk}^2 + \left(\frac{u_{yijk}}{\cos \beta_{ijk}}\right)^2} \quad (3)$$

β_{ijk} - is the Tangent Angle at the point of contact, with k-index of the wheel and the rail for the pair of wheels with index ij.

As we said above, we have made many simplifications for the realization of the task given as we here have neglected the impact on the speed of the slide and the direction of its vector, the change of the radius of rotation and the angle of formation of the wheel profile.

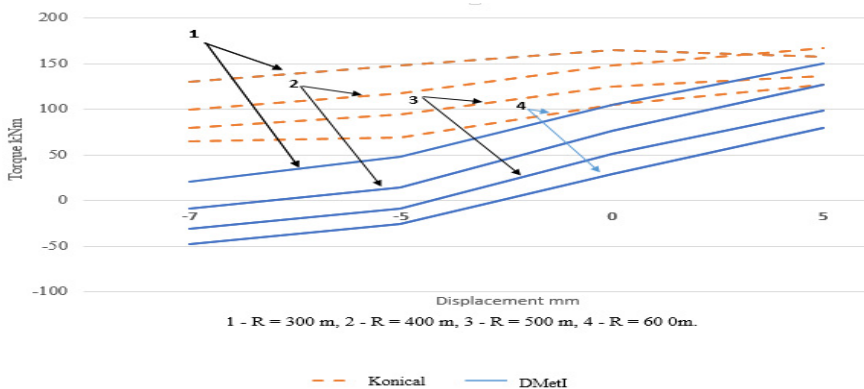
2. Materials and Methods

Let us now determine the values of the moment of friction at the contact of the wheel with the rail, during the movement of the vehicle in a turn with a radius of less than 500-650m, a turn which can also be called "sharp".

The movement of the load in a "sharp" circular turn can be called stable when the static component of the lateral forces of wheel interaction exceeds by a considerable value the magnitude of the dynamic forces.

Such a thing happens when we take the derivatives in relation to time equal to zero in the above expressions. In order to make the calculations, we used the method described in the references [6, 7]. As an object study we have taken an electric locomotive whose carriage is with two axes.

When moving through a given turn referring to the direction of movement, the first pair of wheels roll on the rails, while the second may be in the gap. The moment of frictional forces in contact depends on the radius of rotation and the positioning of the vehicle on the rails, i.e. from the position of the second pair of wheels (Fig. 2).



The calculation is made for two variants of the wheel profile: Standard conical (Fig. 3) and curved type DMetI (Fig. 4). In the profile description we have used the methodology described in works [8, 9].

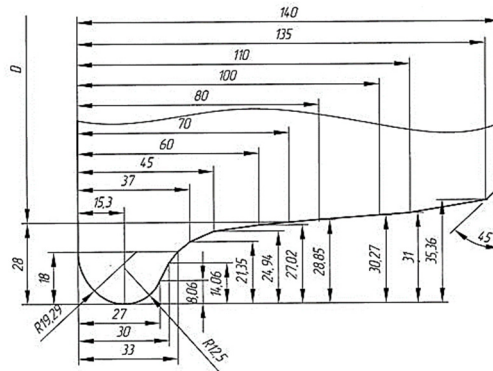


Fig. 3. Standard conical profile

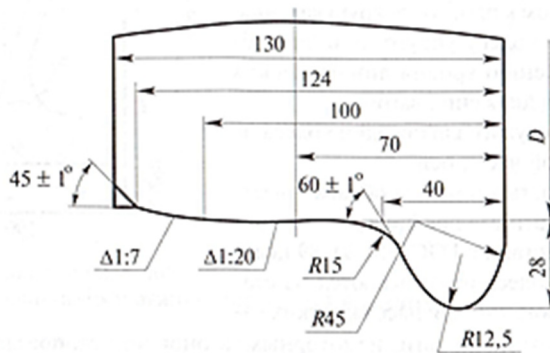


Fig. 4. Curved type DMeTI profile

For a conical profile, this moment of frictional forces significantly exceeds the moment of return acting from the chassis to the carriage. The moment of frictional forces of the wheels on the rails for a conical profile is determined mainly by the transverse components of the frictional forces and depends slightly on the position of the vehicle, above the railway gauge.

3. Results and Discussions

Numerous studies have shown that to improve the passage of the tool in turns, it is good to use a profile which has a curved shape. In this profile, small carriage displacements in the railway lead to a change in moment magnitude (Fig. 5). For the curved wheel profile the moment of frictional forces has a smaller magnitude, due to the increase of the difference between the wheel roll circles and the reduction of the wheel drag in the longitudinal direction. Below are the results of the simulation of a dynamic system in stable motion on a curve with a radius of 350 to 2000 m, in a line without geometric irregularities, for the two types of profile listed above. (Fig. 6-7).

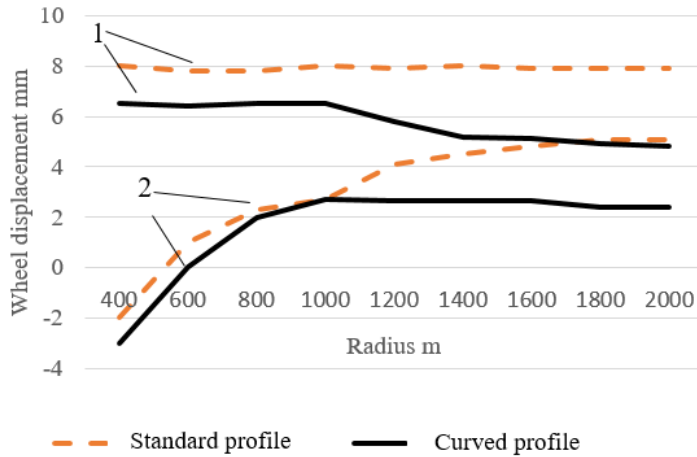


Fig. 5. Displacement of the pair of wheels on the railway gauge. 1, the front pair of wheels 2, the rear pair of wheels. Radius in meters.

From the results we see that the conical profile maintains contact at two points, even in smooth turns (Fig. 3, 4). While the curved profile of the wheel allows the non-contact movement of the wheel curb, ie it performs the roll with a point of contact of the outer wheel on the turn with a radius of more than 1000 m.

The movement in sharp turns passes in contact with the curb, which makes the vehicle position. For this, the value of the steering force decreases while the friction forces perform work mainly on the contact surface of the wheel. With increasing radius of rotation, the frictional forces of the wheel on the rail in the longitudinal direction, contribute significantly to the reduction of torque. (Fig. 5).

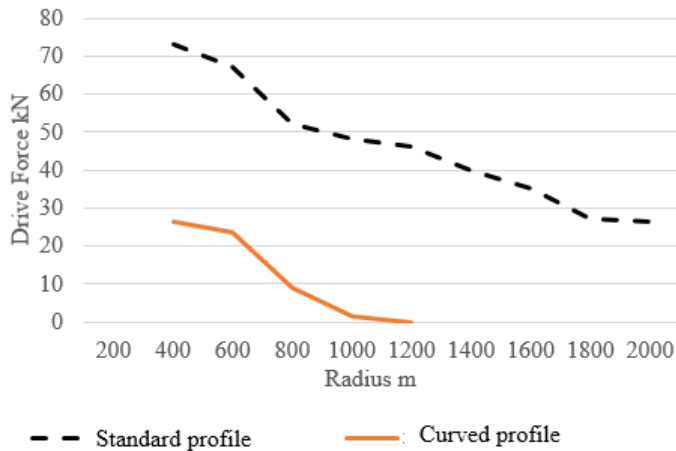


Fig. 6. Driving forces