THEORETICAL APPROACHES OF LINEAR AND NON-LINEAR ACTIVE CONTROL OF RAILWAY VEHICLES. STUDY OF THE FRICTION AND ADHESION OF THE WHEELS OF RAILWAY VEHICLES WITH THE HELP OF DYNAMIC MODELING.

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ABSTRACT: The paper describes the development of the form of auto-oscillations in case of rotation of the traction unit which is affected by a combination of random conditions and the development of rotation, the effects of shock vibrations in operation, the presence of subharmonic resonances and external disturbances at frequencies close to auto-oscillations. Due to the fact that the self-oscillations of a pair of wheels are the most typical form of self-oscillation for known drive structures, in order to reduce the self-oscillations to a safe level, it is proposed to use energy dissipation on impacts in the transmission gear.

KEYWORDS: wheels, shocks, self-oscillations, resonances.

1. Introduction

1.1 Friction and tightening of railway vehicle wheels

Adhesion or adhesion coefficient is given by the ratio of tangential longitudinal force (i.e. braking or tracking) to the normal force at wheel rail contact. The tangential force that a braked or trailed wheel can exert on a rail is limited by the friction coefficient available between the surfaces coming into contact for a given normal load [1]. In contact with clean steel, the friction coefficient is known to be higher than the adhesion requirements for normal traction and braking operations of existing rolling stock. However, contamination such as leaves, grease and water can easily occur at the wheel-rail contact reducing friction leading to low grip problems [2]. When there is a low grip, train delays can be the clearest consequence for commuters traveling by rail. However, many other negative effects can occur, such as damage to the wheels and rails, overtaking of the station platform and even collisions. Therefore, not only the reliability, but also the also the safety and costs of rail transport can be compromised.

1.2. Friction

Friction is defined as the force of resistance tangent to the common boundary between two bodies when, under the action of an external force, one body moves or tends to move relative to the surface of the other body [1].

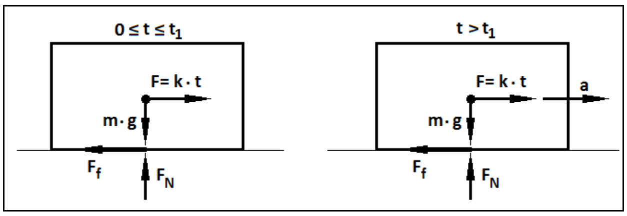

Fig. 1. The forces applied on a on a solid body rectangular on a horizontal plane

The friction is normally the friction coefficient f, which is defined as the ratio between the frictional force F_f to the normal force F_N in contact between the two bodies, as shown in relation (1):

$$f = \frac{F_f}{F_N} \tag{1}$$

Figure 1 shows a solid rectangular body of mass m leaning on a horizontal plane. If a force F is applied, parallel to the plane and increasing in time, to the center of mass, as shown in Figure 1, where k is an arbitrarily positive constant and t time, there will be a time t_1 when the body begins to slide on the plane. The friction that opposes the start of the movement is called static friction.

The coefficient of static friction fs is given in relation (2), where g is the acceleration due to gravity. From that moment, the body slides with an acceleration a, and the force that opposes the sliding motion of the body is called kinetic friction (or dynamic friction). The instantaneous kinetic coefficient of friction f_k at a t_2 is given by the relation (3), where a_2 is the acceleration of the respective body at the t_2 moment.

$$f_s = f(t = t_1) = \frac{k \cdot t_1}{m \cdot g} \tag{2}$$

$$f_k = f(t = t_2) = \frac{k \cdot t_2 - m \cdot a_2}{m \cdot g} \tag{3}$$

In most tribological pairs, static friction is greater than kinetic friction, with the difference being dependent on materials and contact conditions (relation 1). In the case of steel used for rail wheels and rails, laboratory investigations have shown that the coefficient of static friction can be up to almost twice the coefficient of kinetic friction (relation 3).

2. Modeling of the railway vehicle wheels

Modeling means many things, so it is necessary to start by defining what this means regarding research. The aim of this paper is to investigate advanced braking strategies. Modeling and simulation are part of the analysis and design process in most engineering projects and are essential in all but the simplest system. The difference between modeling and simulation is that a model is a simplified representation of a system, and simulation is an adapted model for simulation on a computer, i.e. mathematical or logical relationships and operational rules embedded in the computer program, which are known together as a simulation model on a computer or simple simulation model [1].

The simulation is similar to laboratory experiments performed by scientists to gain insight into existing theories or to develop and validate new theories. Studying the behavior of the system through these indirect methods (i.e. by modeling and simulation) becomes a necessity in many situations where no other alternative is possible (e.g. observation, analysis, experimentation, non-destructive testing etc.) or the available alternatives are not effective or they are too expensive.

This paper presents a model that illustrates the interaction due to braking between the wheel of a railway vehicle and the rail. It also shows the derivation of the motion equations for a single set of wheels of the railway vehicle. These well-established equations were used a few years ago to solve the problems of railway dynamics.

The paper also describes the methods that are currently being developed to evaluate the braking control systems of railway vehicles. These systems are needed for high performance control strategies [2]. The effects of shaping the rail and wheel are demonstrated when the brake shoes are applied to the wheel. Both linear and non-linear situations are studied, as well as the effects they have on the vehicle dynamics.

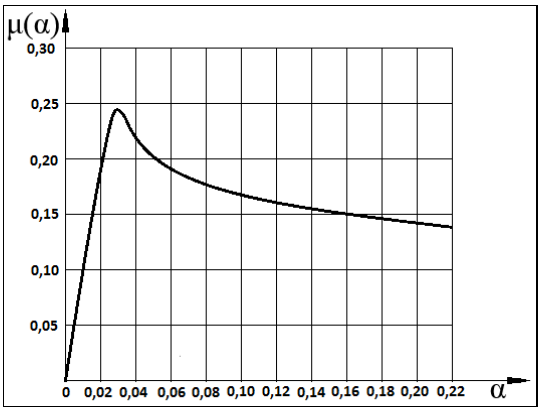

Fig. 2. Coefficient μ to a α in dry conditions

In the first phase, brake shoes were used that acted on the edge of the wheel. At present, the brake discs are mounted either on the wheel itself or on the mounted axle, and in some cases in both places. Clog brakes on the wheels of the railway vehicle are the type of brakes that remain the most widely used worldwide, although technology has changed and improved a lot over time.

A fundamental aspect is that the maximum towing effort (or braking) that can be transmitted is the product of the predominant adhesion factor and the vertical reaction force. The maximum axle load depends on the maximum speed and unprotected mass and therefore the maximum towing or braking effort that could be transmitted. For wheel-rail contact, surface conditions, both in terms of smoothness and cleanliness, significantly affect adhesion. The adhesion factor also varies depending on the speed of sliding between the wheel and the rail [3]. The distribution at a low friction condition is accentuated, as shown in Figure 1, which shows the coefficient of friction compared to the slip coefficient.

This fact, coupled with the high inherent inertia in all rail wheels, means that the most sliding systems focus on detecting slip as quickly as possible and taking corrective action.

Mathematical models are developed for the design and evaluation of advanced braking control strategies for high performance railway vehicles [4]. The modeling includes both linear dynamics and non-linear braking forces generated at the wheel-rail contact. The equations of motion for a single wheel and the braking effect of a single wheel are discussed in the next section.

The results of a progression of increasing complexity models are then presented. As the understanding of the dynamic behavior of a wheel is fundamental to the study of the dynamics of a railway vehicle, it is first considered an isolated wheel moving along the rail [5]. These equations are extended to the case of a rigid frame supported on two wheels, representing a single bogie. This model is further extended to represent a single vehicle and finally to a multi-vehicle train.

2.1 Interaction between wheel and rail

The wheels of the railway vehicle are in the form of two wheels rigidly mounted on a common axle. In a superficial view, the behavior of a wheel of a railway vehicle is determined by purely geometric effects. Pure rolling motion is altered by the action of tangential forces at the wheel-rail contact point. These forces induce sudden slip or slight slip. This is important in the behavior of wheel-rail contact [6].

The mechanism of the braking system works according to the following sequence. The brake shoes are pushed on a disc mounted either on the axle or on the wheel. The force generated by the support reacts through a suspended brake caliper connected to the bogie frame. This opposes the rotation of the wheel and creates an elastic deformation of the contact space between the wheel and the rail. As a result, a longitudinal braking force is developed, the longitudinal braking force. The theory of this mechanism is based on sliding or factional difference between the peripheral speed of the wheel and the speed of the train. The relationship between the coefficient of friction μ and the slip α, depends on the materials, but for steel wheels and rails, it is usually as shown in Figure 10, which also indicates the adhesion limit [7].

The geometric features of interest refer to the behavior of the point of contact on each wheel where the slight slip is created. A formal definition of slip based on the variables shown in Figure 2 is given in relation 4:

$$\alpha = \frac{2 \cdot (RW \cdot \omega - V)}{(RW \cdot \omega + V)} \quad (4)$$

The longitudinal slip is defined in terms of the speed with which the rolling stock passes through the rail-wheel contact area and is expressed as the difference between the components in the longitudinal direction of these rigid body speeds divided by the wheel forward speed. The slip is produced by a tangential force and a moment of the normal axis. The tangential force is usually also solved by its longitudinal and lateral components, although this work only refers to sliding and force in the longitudinal direction. Instead, the slip can be considered as specified and the force and moment calculated from it. In the theoretical work, this is the general case.

2.2 The movement equation of the wheel

The dynamic model of the wheel should be as simple as possible. For simulation and analysis purposes it should also contain all the important parameters of the particular properties that are being investigated on the dynamic system of the whole vehicle [8].

Figure 3 shows the forces acting on a single wheel under the braking action.

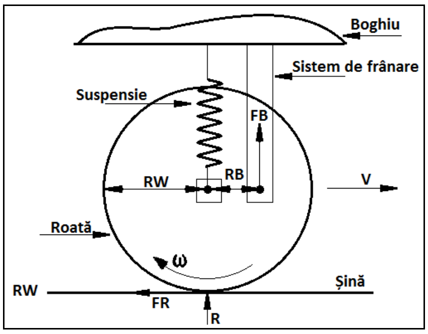

Fig. 3. Vehicle dynamic system parameters

For the motion of a mounted axle with small speed differences in relation to the path on which it travels, the slip α is given by the relation:

$$\alpha = \frac{(RW \cdot \omega - V)}{V} \quad (5)$$

in which RW is the radius of the wheel, ω is the angular velocity and V is the forward speed of the wheel. For braking, result negative values of an angular velocity value ω.

When the speed difference is a significant proportion of the wheel or rail speed, it is more appropriate to use the following equation, in which the difference is divided by the average speed (relation (5)). The relationship (6) was used because, although there is some increase in computational complexity, the simulation was expected to reach high levels of sliding in certain circumstances and therefore a more precise equation must be used.

The dynamic equation of the wheel is therefore as follows:

$$IW \cdot \alpha = FR \cdot RW - EB \cdot RB \quad (6)$$

in which IW is the moment of inertia of the mounted axle and where:

$$FR = \mu(\alpha) \cdot R \tag{7}$$

The coefficient of friction $\mu(\alpha)$ is a non-linear relation, usually presented (figure 10) for dry rails, where the same characteristic applies to the negative values of the slip encountered during braking [9]. The equation can be substantially linear for a small braking force (for which the grip is approximately proportional to the slip) but is obviously very nonlinear as the grip limit approaches.

The equation of the linear longitudinal braking force FR is:

$$FR = \frac{R(RW \cdot \omega - V) \cdot C_c}{V} \tag{8}$$

where R is the reaction force and C_c is the slope of the graph for the small value α .

The longitudinal braking force FR also depends on the reaction force R . However, this force will change as result of the redistribution of wheel loads during braking, an effect which also depends on the dynamic properties of the vehicle's suspension [10]. This effect creates the interaction between the braking systems of the different axles mounted in a train [11] and which must be properly understood and modeled to allow the determination of the maximum potential of high-performance braking systems.

2.3 Breaking of one wheel

There are several parameters that are used to model the braking behavior of a single wheel. These are the wheelbase RW = 0.5 m, the distance from the center of the wheel to the brake arm, RB = 0.25 m and a quarter of the mass of the train, R = 7853.5 kg for 77042.8 N. For all preliminary studies a train starting speed V = 20 ms-1, although at a later stage the results were also evaluated at higher and lower speeds. For a single dynamic mounted axle [12] it is simply represented by:

$$MV = -FR \tag{9}$$

A brake application was simulated to illustrate the effect of brakes on the speed, slip and angular velocity of a single wheel running on a railway.

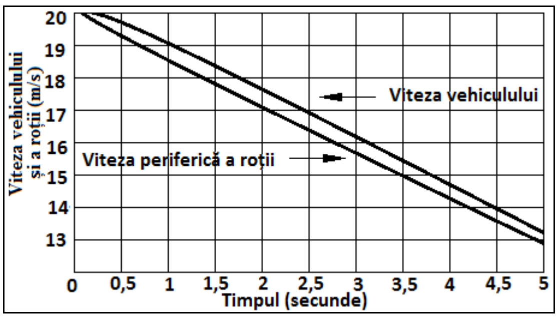

Fig. 4. Linear wheel speed and peripheral wheel speed

In this case, a linear adhesion-slip feature was used (only the FB limitation, so that the adhesion limit is not exceeded). The simulation results show that as the brakes are applied, the vehicle speed and the wheel speed start to decrease, the speed of the fall depends on the braking force [13]. The simulation was stopped when the speed of the mounted axle [14] reached zero. Figure 4 shows the linear wheel speed and the peripheral wheel speed, while figure 5 shows the longitudinal braking force.

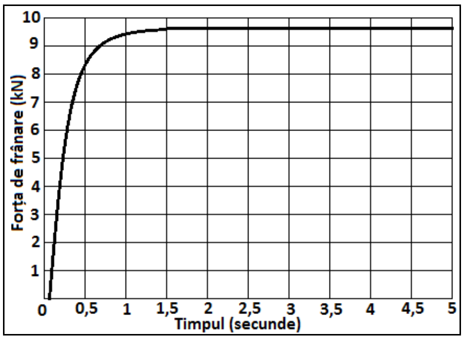

Fig. 5. Longitudinal braking force

The results of a steadily increasing force FB, which creates a longitudinal braking that exceeds the adhesion limit, are shown in Figure 5. The chart above shows the speed of the train and the peripheral speed of the wheel. The difference is small due to the low level of slip that exists for the steel at the contact of the steel. The graph below shows the longitudinal braking force, FR, which is generated at the wheel-rail contact point. At this stage, the braking force FB = 60 kN and the longitudinal braking FR are as shown in Figure 6, but all other parameters are the same as before.

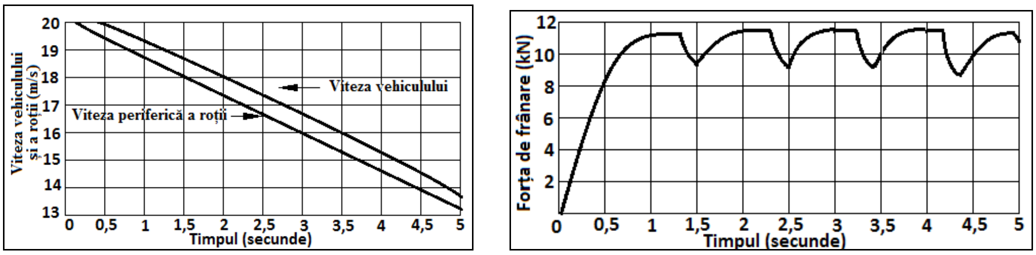

Fig. 6. Braking of a single mounted axle with easy wheel slip control

The developing simulation includes a simple control law representative of current practice, which stops the braking effort if the slip exceeds a value of 0.03 and reapplies it when the wheel stops slipping.

2.4 Wheel sliding control

Wheeled vehicles are usually braked on their wheels and, although there are used different types of auxiliary braking systems, friction brakes are normally used as the most popular means of decelerating or stopping the vehicle [15].

The frictional force is developed between the wheel and the rail when the wheel is braked. The braking force is the available output force that will be used to decelerate or stop the train. Figure 7 shows the diagram of the braking scheme, including a first-order filter to represent the dynamics of the braking system. The input is a slip signal and if the slip value is below the maximum allowable value, then it will allow braking to be applied [16].

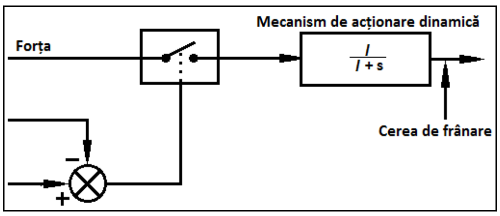

Fig. 7. The diagram of a single-wheel controlled braking

If the input signal has matched or exceeded the maximum allowed value, the brake will stop [17]. This force could have any profile, such as a step or a ramp, and a brake frame was used in these studies [18]. The time constant of the braking dynamics is typical of a pneumatic braking system [19].

The adhesion between wheel and rail is the limiting factor in rail braking performance [20].

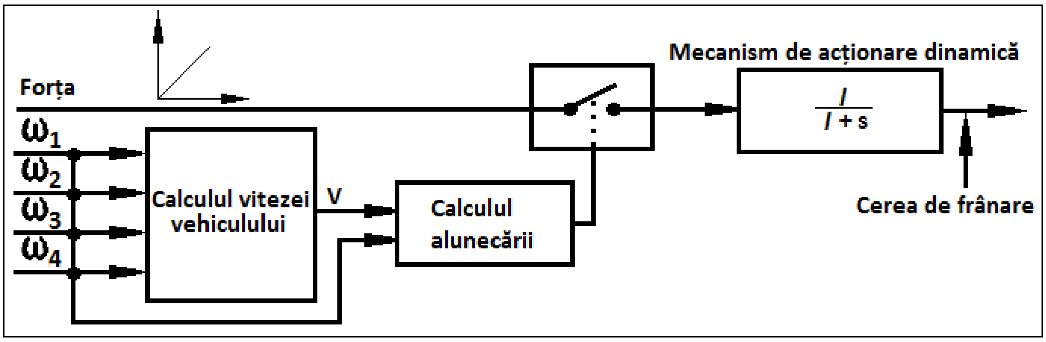

Fig. 8. The control design diagram of the vehicle device on a wheel

If the slip limit is exceeded once the brakes have been applied, then they are stopped immediately [21]. When the wheel recovers from slipping, the brakes are reapplied and this cycle is repeated. This control method [22] was implemented for the results in Figure 9.

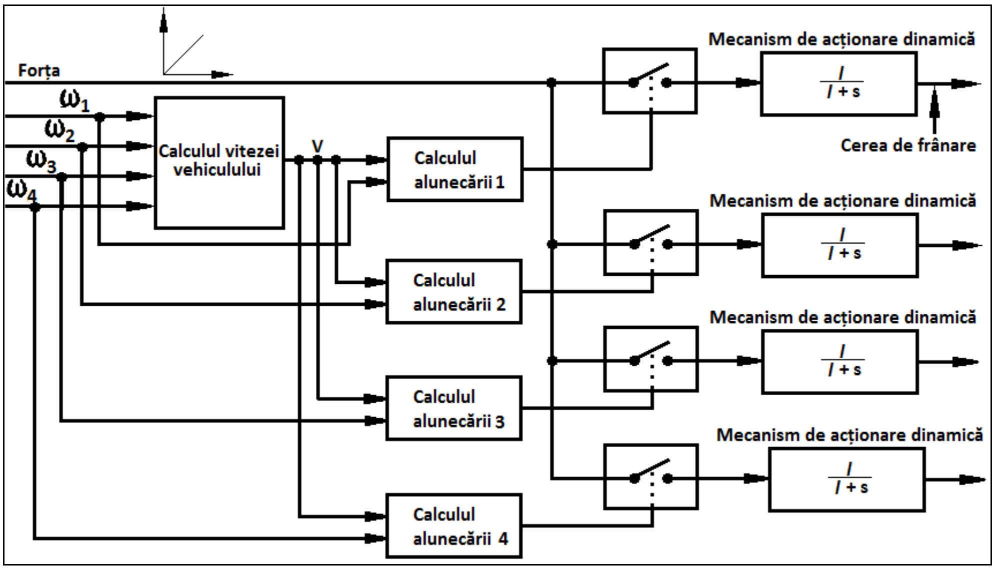

Fig. 9. The control design chart for four-wheel drive vehicles

The same unique sliding control system used on the mounted axle has been used in the bogie system, in which each mounted axle has an independent control system [23].

3. Conclusions

One can obviously speak of an advantage of the roller or ball follower (Module B), compared to the classic sole follower (Module C). Therefore, high speeds and higher efficiency can be obtained with the help of module B. By designing and adapting such a distribution mechanism on 2100 HP Diesel Electric locomotives, the vibrations from the distribution shaft (s) are largely eliminated, a better efficiency is obtained and the theoretical operation chart of the Diesel Engine is close to the ideal one.

The development of this or that form of self-oscillation in the case of traction unit rotation is affected by a combination of random conditions from the beginning and the development of rotation, the effects of shock vibrations in driving, the presence of subharmonic resonances and external disturbances at frequencies close to auto-oscillations frequencies.

Since the self-oscillations of a pair of wheels are the most typical form of self-oscillation for known drive structures, to reduce the self-oscillations to a safe level, it is proposed to use energy dissipation on impacts in the transmission gear.

The coefficient of adhesion is the ratio between the longitudinal tangential braking force and the traction force and the normal force in the wheel-rail contact plane. The tangential force that a braked wheel can exert on the rail is limited by the coefficient of friction available between the wheel and the rail at a given normal force. The coefficient of friction in clean steel contacts is usually higher than is required for normal braking and traction operations of rolling stock. However, leaf, grease and water pollution can easily occur in wheel-rail contact and reduce the level of friction due to problems of adhesion lacking. In recent years, problems have arisen in our country due to the presence of moisture and leaves and other pollution on the rails. Due to a lack of adhesion on the rails, voyages delay and many other negative effects can occur, such as damage to the wheels and the rail, ignoring signals and even collisions. Therefore, not only the reliability, but also the safety of rail transport is jeopardized due to adhesion problems.

The studies focused on the theoretical aspects, as well as on the aspects of numerical modeling and simulations. The technical approaches have been implemented by knowing the dynamic behavior of railway vehicles, specifying the actuators of active suspensions and their implementations. The technical aspects were primarily addressed by knowing the dynamics of the vehicles, their behavior, the analysis of driving stability and the related standardized indicators. Knowledge of previous experiences, research into new technologies and simulations have led to the synthesis of functional specifications and the choice of electromechanical technology to define the transverse and vertical actuations of active suspensions. The introduction of uncertainties in the nominal scheme leads to a change in equations and representation schemes, algebraic schemes. This new formulation can be used to analyze the stability of the assembly, once the control defined for the nominal system or to be used for the synthesis of a new controller, for a modified representation diagram. Synthesis (or DK iteration) is a method recommended especially in these situations.

The element that ensures the guidance of the railway vehicles on the two rails of the track is the mounted axle. The force acting on the wheel-rail contact area creates a moment of overturning of the axle and, thus, an additional load transfer depending on the wheel radius.

The effect of the interaction between the braking action of the individually mounted axle was analyzed. The evaluations included the effect of simple control laws to prevent the wheels from slipping during braking.

The next step in the research program is to use these dynamic models to design brake regulators that will very effectively control the braking effort to maximize the use of available adhesion. It is expected that each vehicle will have a single loop control for each set of wheels, the design of which will take into account the interactive nature of the four brake control loops. The study will also take into account the change in adhesion level, for which connections between controllers on different vehicles are expected to become important.

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