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## SIMULATION MODEL OF WHEEL SLIP AND SLIDE PROTECTION SYSTEMS FOR RAIL VEHICLES

### MODEL SYMULACYJNY UKŁADU WYKRYWANIA I LIKWIDACJI POŚLIZGU PRZY ROZRUCHU I HAMOWANIU DLA POJAZDU SZYNOWEGO

#### Abstract

A simplified model of a rail vehicle - electric locomotive, with particular emphasis on wheel-rail cooperation model, is presented. The principles of detecting the slip and algorithms that detect and eliminate the slip, both during acceleration and braking using dump valves are presented. Two anti-slip system simulation models, implemented in Matlab Simulink are described taking into account each subsystem model. The following models of hardware and software subsystems are described: the rotational speed sensors, the dumping valves and the subsystem determining the reference speed. The study program, simulation results, analysis and conclusions are presented.

*Keywords: electric traction, traction rail vehicle, braking, wheel slip, wheel slide, simulation, model*

#### Streszczenie

Przedstawiono uproszczony model pojazdu szynowego – lokomotywy elektrycznej, ze szczególnym uwzględnieniem modelu współpracy koła z szyną. Zaprezentowano zasady wykrywania poślizgu oraz opisano algorytmy wykrywania i likwidacji poślizgu, zarówno przy rozruchu, jak i przy hamowaniu z udziałem zaworów upustowych. Opisano modele symulacyjne obu układów przeciwoślizgowych zaimplementowane w środowisku Matlab Simulink® z uwzględnieniem modeli poszczególnych podsystemów. Nakreślono modele podsystemów sprzętowych: model czujników obrotowo-impulsowych i zaworów upustowych oraz modele podsystemów programowych: podsystemu wyznaczania prędkości referencyjnej. Opisano program prób i badań obu układów. Przedstawiono wyniki badań symulacyjnych oraz analizę eksperymentów symulacyjnych wraz z wnioskami.

*Słowa kluczowe: trakcja elektryczna, pojazd trakcyjny, hamowanie, poślizg, symulacja, model*

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## 1. Introduction

In the paper, a simulation model of wheel slip and slide protection systems for rail vehicles is presented. The control principles have been described for both systems. Software simulation models of both protection systems implemented in Matlab Simulink® have been described, taking into consideration models of particular sub-systems, such as speed sensors, dump valves and the reference speed calculation subsystem. A simplified model of a rail vehicle, which is an electric locomotive, has been presented. Test programs for both systems have been described, and the results have been presented.

## 2. Simulation models

### 2.1. Simulation model of a rail vehicle

The dynamics of the vehicle  $i$ -th axle set rotary motion is given with the following equation set [1, 10]:

$$J \frac{d\omega_i}{dt} = M_i - F_{ai} r \quad (1)$$

$$F_{ai} = \psi_i(s_i, p) Q_i \quad (2)$$

$$s_i = \frac{v_i - v_T}{v_T} \quad (3)$$

where:

- $J$  – moment of inertia of rotating elements associated with an axle set reduced to the axle set axis,
- $\omega_i$  – angular velocity of an  $i$ -th vehicle axle set,
- $M_i$  – driving or braking torque of an  $i$ -th axle set,
- $F_{ai}$  – adhesion force of an  $i$ -th axle set,
- $r$  – wheel radius,
- $\psi_i$  – instantaneous value of wheel-rail adhesion coefficient for wheels of an  $i$ -th axle set,
- $s_i$  – relative slide of  $i$ -th axle wheels against the rails,
- $p$  – vector of parameters of  $\psi$  vs.  $s$  characteristics,
- $Q_i$  – instantaneous load of an  $i$ -th axle set,
- $v_i$  – circumferential velocity of  $i$ -th axle set wheels,
- $v_T$  – translational vehicle velocity.

The dynamics of a translational vehicle motion is described with the following equation:

$$m \frac{dv_i}{dt} + \sum_{i=1}^n \frac{J}{r} \frac{d\omega_i}{dt} = \sum_{i=1}^n F_{ai} - F_r \quad (4)$$

where:

- $m$  – vehicle mass,
- $n$  – number of driven or braked axles,
- $F_r$  – movement resistance force.

The most sophisticated part of the model is the characteristics of the instantaneous coefficient of adhesion  $\psi$  versus the relative slide  $s$ . According to (2), the adhesion coefficient is a non-linear function of wheel-rail relative slide, and a set of factors, hereby denoted as  $p$ , which comprise the following [3–5, 13]:

- kinematic state of a wheel,
- state of the rail,
- physical and chemical characteristics of the wheel-rail system,
- ambient conditions,
- vehicle velocity,
- circumferential deceleration of the wheels,
- slide energy  $E$  developed in the wheel and a rail contact point.

The  $\psi$  vs.  $s$  characteristics are described with a non-linear empiric equation set. The shape and parameters of the curve depend on the abovementioned factors in a way, which is difficult or impossible to establish. Generalised characteristics of the instantaneous coefficient of adhesion  $\psi$  versus the relative slide  $s$  are shown in Fig. 1.

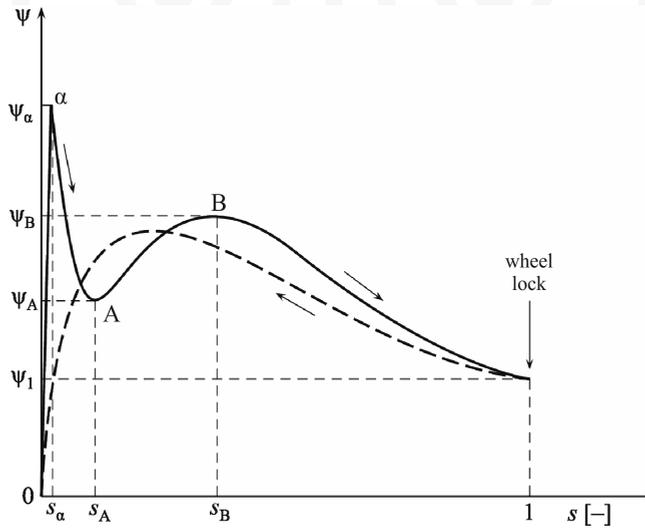


Fig. 1. Generalised characteristics of the instantaneous coefficient of adhesion  $\psi$  versus the relative slide  $s$  [4, 13]

## 2.2. Simulation model of the pneumatic brake

The locomotive torque  $M$  demanded by the driver during traction is completely released by the traction system. During braking, however, the braking torque is released partially by the traction system (electro-dynamic braking) and partially by a pneumatic system. This process is called blending and can be controlled by a pneumatic panel controller, which sends

an ED brake demand to the traction system, receives feedback signals, calculates the brake cylinder pressures necessary to release the remaining brake force from the pneumatic brake system, and controls the pneumatic panel in order to obtain the demanded pressures. The diagram of the simulation model of the pneumatic board implemented in Matlab Simulink® is shown in Fig. 2. The pressures generated by the pneumatic panel are then fed to the dump valves (see point 2.2.2.), and the resulting pressures are fed to the pneumatic brake.

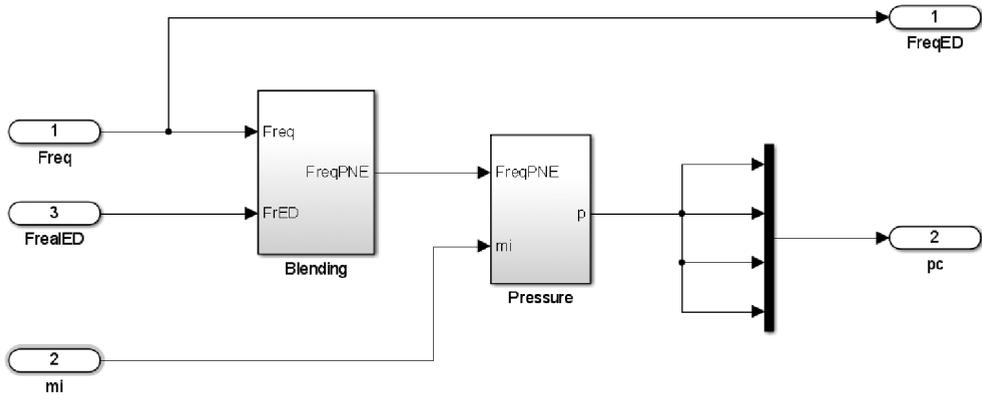


Fig. 2. Diagram of the simulation model of the pneumatic board implemented in Matlab Simulink®

The pneumatic brake model consists of the following subsystems:

- model of the lever system,
- model of the friction elements.

The model of the lever system describes the relation between the pressure in the brake cylinder  $p_c$  and the pressure of the friction linings of the brake blocks onto the wheel treads  $N_k$ , which is exerted by the piston of the brake cylinder. This relation is highly non-linear and is characterised by hysteresis, resulting from friction in the brake cylinder and in the joints of the clamp mechanism. It is shown in Fig. 3.

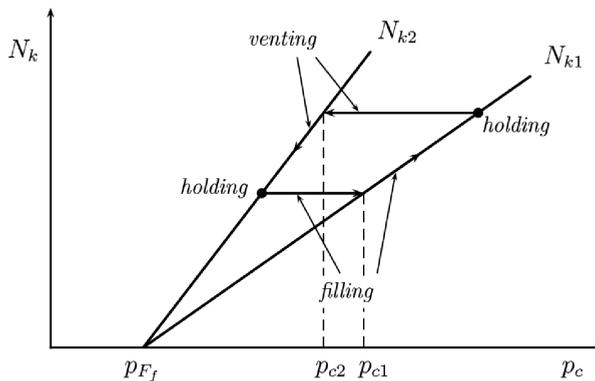


Fig. 3. Relation between the air pressure in brake cylinders  $p_c$  and the brake blocks pressure  $N_k$

The model of the friction elements describes the relation between the pressure exerted by the brake blocks and the braking torque  $M_b$ . For a block brake, this relation is given with equation (5), where  $r$  is the wheel radius and  $\mu$  is the friction coefficient of the brake block friction lining, which depends in a strongly non-linear way on the wheel speed and pressure of the friction linings.

$$M_b = N_k \cdot \mu \cdot r \quad (5)$$

In order to obtain a braking torque, this pneumatic torque should be added to the braking torque obtained from the traction motors.

### 2.3. Simulation model of the electro-dynamic traction and brake system

Voltage inverters supplying the induction motors are connected to the traction overhead line through the input low pass filters. The traction inverter is a power electronic device, the static and dynamic properties of which depend mainly on the type of controllers and power electronic elements (GTO thyristors, IGBT transistors), which are electric switches (valves). The mathematical model of an electric valve, which has been adopted for this project, is a lossless switch between two modes: lossless conducting mode and blocking mode. Possible states of the valves in the three inverter branches are represented by variables  $K_A$ ,  $K_B$ ,  $K_C$ . Each of these variables can adopt value 1 or 0. The output inverter voltage is a value resulting from  $K_A$ ,  $K_B$ ,  $K_C$  and a value of the voltage at capacitor  $C_f$  described with (6):

$$\begin{bmatrix} U_{AN}(t) \\ U_{BN}(t) \\ U_{CN}(t) \end{bmatrix} = \begin{bmatrix} K_A \\ K_B \\ K_C \end{bmatrix} \begin{bmatrix} u_f(t) \end{bmatrix} \quad (6)$$

A precisely described position of a stator voltage space vector corresponds to each of the configuration of valves. Voltage inverters applied in traction rail vehicles should make controlling the frequency and amplitude of the fundamental harmonic frequency possible. Besides, it is necessary to minimise distortions of the motor currents in order to limit parasitic torques negatively influencing the vehicle operation and to limit distortion of the current in the traction overhead line, influencing the trackside devices.

The brake circuit of a locomotive consists of a chopper (PH), resistor  $R_h$ , capacitor  $C_f$  and an inverter. The frequency of PH operation is given with (7) [3]:

$$f_{pH} = \frac{I_{FH0} (U_{fH0} - I_{FH0} R_h)}{U_{fH0} H C_f} \quad (7)$$

where:

$$H = u_{fH_{\max}} - u_{fH_{\min}}, I_{FH0} - \text{average brake current.}$$

The dynamics of the changes of the instantaneous voltage value on the capacitor is described with (8). Signal  $K_{HP}$ , which controls the operation of the brake converter valve, is generated in the brake controller and its logical value 1 or 0 depends on the value of voltage on capacitor  $C_f$ .

$$\frac{d}{dt}u_{fH}(t) = \frac{i_{FH}(t)}{C_f} \bar{K}_H + \left( \frac{u_{fH}(t)}{C_f R_h} - \frac{i_{FH}(t)}{C_f} \right) K_H \quad (8)$$

Induction motor is supplied with a distorted voltage from an inverter or a sinusoidal generator of changeable amplitude and frequency. Mathematical model of the motor can be given as non-stationary state equation. The elements of the state-transition matrix depend on the instantaneous mechanical angular velocity of the rotor  $\Omega_m$ . The equations are given with (9) [8]:

$$\dot{X}_e = A_e(\Omega_m) X_e + B_e U_e \quad (9)$$

where:

$A_e$  – state-transition matrix,  
 $X_e = [\varphi_{s\alpha} \ \varphi_{s\beta} \ i_{s\alpha} \ i_{s\beta}]$  – state-transition vector,  
 $U_e = [u_{s\alpha} \ u_{s\beta}]^T$  – control vector

$$B_e = \begin{bmatrix} 1 & 0 \\ 0 & 1 \\ \frac{1}{\sigma_1 L_s} & 0 \\ 0 & \frac{1}{\sigma_1 L_s} \end{bmatrix}$$

Electromagnetic torque of an induction motor is given with (10):

$$M_e = \frac{3}{2} p_b (\varphi_{s\alpha} i_{s\beta} - \varphi_{s\beta} i_{s\alpha}) \quad (10)$$

where:

$p_b$  – liczba par biegunów.

## 2.4. Simulation model of the wheel slip and slide protection system

### 2.4.1. Simulation model of wheel speed sensors

A simplified simulation model of sensors calculates the discrete value of circumferential wheel speed on the basis of a continuous value of this speed. The calculation period is adopted as 10 ms.

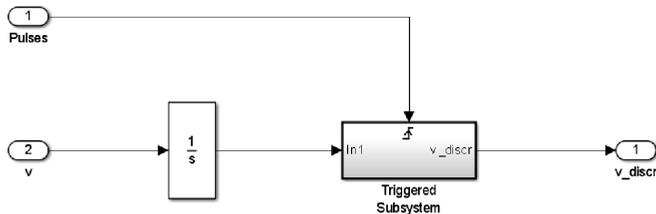


Fig. 4. Diagram of the simulation model of wheel speed sensor implemented in Matlab Simulink®

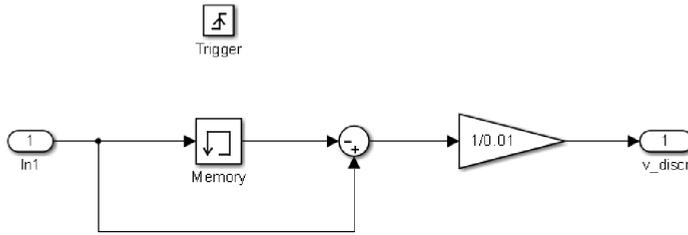


Fig. 5. Diagram of the triggered system of the model of wheel speed sensor implemented in Matlab Simulink®

#### 2.4.2. Simulation model of dump valves

Dump valves are actuators of the Wheel Slide Protection System. They make filling the brake cylinders with compressed air possible, venting them or maintaining the pressure value, and the same increasing, decreasing or maintaining the braking force, making controlling the slide possible. The output pressure value of a dump valve during filling and venting is given with (11) and (12). Subsystem “Dump valve” implemented in Matlab Simulink® is shown in Fig. 6.

$$p_c = p_{ch} + (p_{cin} - p_{ch}) \left( 1 - e^{-\frac{t-t_{ch}}{T_F}} \right) \quad (11)$$

$$p_c = p_{ch} \cdot e^{-\frac{t-t_{ch}}{T_V}} \quad (12)$$

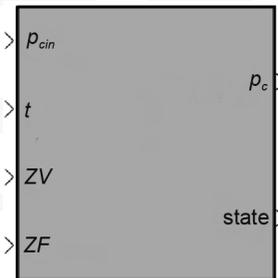


Fig. 6. Subsystem “Dump valve” implemented in Matlab Simulink®

where:

- $p_c$  – brake cylinder pressure,
- $p_{ch}$  – cylinder pressure at the moment of change of the control state,
- $p_{cin}$  – input pressure of a dump valve,
- $t$  – time,
- $t_{ch}$  – time of change of the control state,
- $T_F$  – brake cylinder time constant at filling,
- $T_V$  – brake cylinder time constant at venting,
- $ZV, ZF$  – control signals of respectively venting and filling valves.



$$\widehat{F}_\mu = -h_1 \widehat{F}_\mu + \frac{h_1 M_n}{R_k} - \frac{h_1 J}{R_k} \dot{\Omega}_k \quad (13)$$

where:

$h_1$  – system parameter.

In Fig. 8, two graphs of the adhesion force value have been shown: the real value, obtained from simulation (1), and an estimated value (2). The input values are: angular wheel velocity, traction torque (also estimated on the basis of a motor current and motor parameters). The parameters are: driven wheel diameter and dynamic constant of the system  $h_1$ .

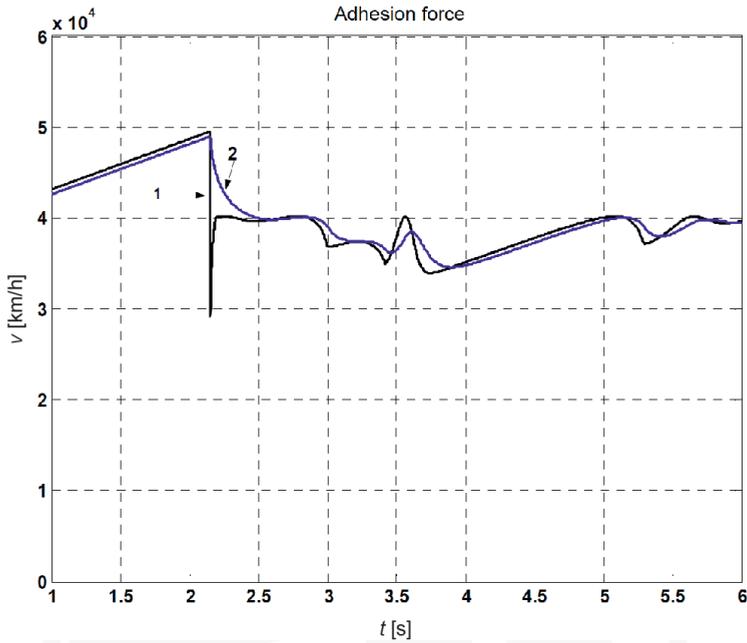


Fig. 8. Graph of adhesion force: 1) real value, 2) estimated value

As it can be concluded from the simulation results shown in Fig. 6, the estimate is settling more slowly than the real value. The velocity and accuracy of mapping of the real value depends on the parameter  $h_1$ . A compromise should be chosen between excessive influence of noise, e.g. measurement noise, on the instantaneous value of the estimate, and the slow settlement of the estimate after change of the wheel-rail adhesion coefficient.

## 2.5. Simulation model of the Wheel Slide Protection system

A braked rail vehicle as a controlled object, as concerns the WSP systems, features the following characteristics [1]:

- several signals crucial for WSP control cannot be measured directly (adhesion coefficient, adhesion force and usually cylinders pressure),

- the controlled object is highly non-linear and non-stationary,
- the model of the controlled object is stochastic,
- both feedback and control signals are discrete.

In Fig. 9, a practical control diagram of the WSP control loop, being a modification of a well-known diagram of classical control, is shown [2]. Inputs of a WSP controller may be wheels slide estimate (because reference speed is an estimate of the real vehicle speed), and first as well as second order differences of wheels speeds. For the abovementioned reasons, classical controllers are seldom used for WSP systems. In the project, presented in this paper, a Fuzzy Controlled has been used. It was a Fuzzy PI controller, the inputs of which are wheels slide estimate and first order difference of wheel speeds [2].

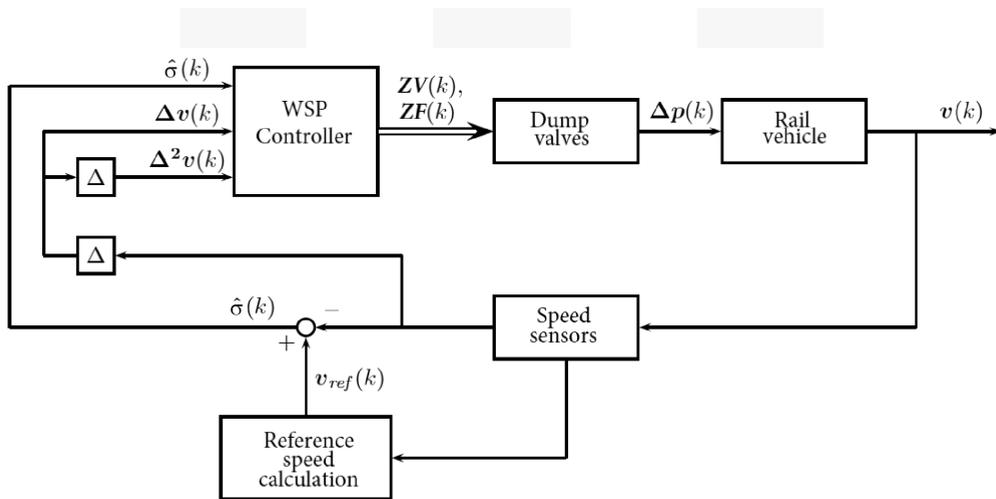


Fig. 9. Practical block diagram of WSP control system [2]

### 3. Real time simulation results

#### 3.1. Simulations of the Wheel Slip Protection system

For the simulation, it has been assumed that the traction torque increases in a linear way. Initial wheel-rail adhesion condition is good (dry rail). At simulation time  $t_4 = 2.15$  [s], the driving wheel enters the area of wet rail. The decision of decreasing the driving torque is taken on the basis of adhesion force observer. When the adhesion force value reaches the maximum, the requested torque is reduced by 10%, and subsequently it is increased linearly again. The simulation results are shown in Fig. 10.

On the basis of the simulation results, it can be concluded that the proposed method of the anti-slip device operation is efficient, and not only does it prevent the development of macro-slip, but it also helps maintaining the adhesion force in the maximum range for the current adhesion conditions.

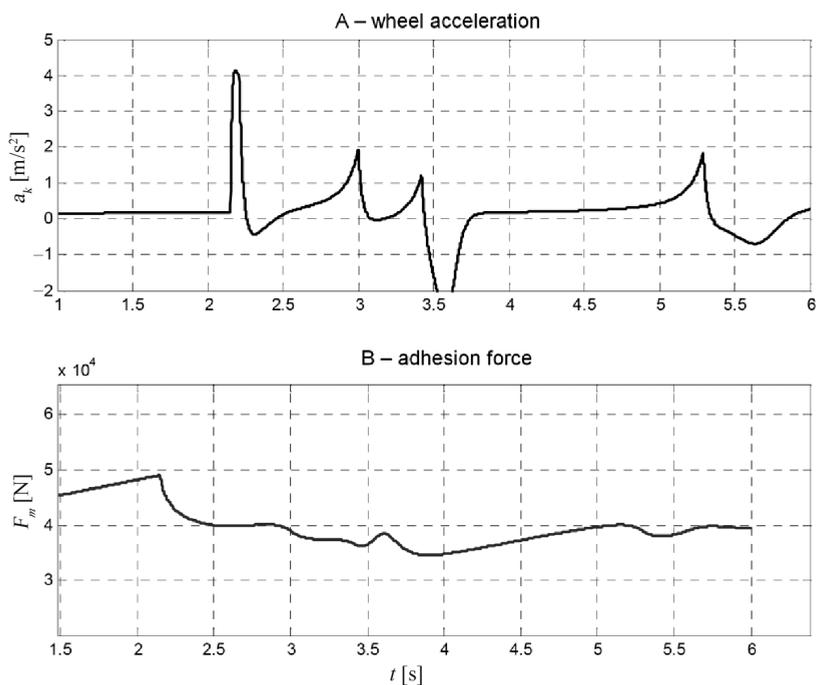


Fig. 10. Simulation results: A) acceleration, B) adhesion force

### 3.2. Simulations of the Wheel Slide Protection system

In order to test a Wheel Slide Protection system against normative requirements, it should undergo a series of tests defined in standard EN 15595 [12] for a given vehicle. This standard provides a specification of the requirements for both structure and function of the WSP devices. It also provides a specification of tests, which verify meeting by a WSP device the normative requirements concerning functionality of the WSP control system. The test program consists of several tests. The exact specification of tests depends on the vehicle type: passenger coaches, wagons, locomotives and train-sets or high speed trains. The initial vehicle speed and brake position, as well as additional conditions, are specified for each test.

For example, for an electric locomotive with an electro-dynamic brake, there are sixteen various tests. They comprise braking until standstill (the so-called slide tests) at lowered adhesion (sprayed rail) from 120 km/h and 160 km/h at various brake intensity, braking from 120 km/h at rails covered with soap in a given distance, braking of a vehicle dragged with velocity of 100 km/h on sprayed rails, braking from low velocity (50 km/h) on sprayed rails until standstill, and braking from 100 km/h at very low adhesion until standstill. All tests are performed with the ED brake on and off. In the abovementioned standard, the assessment criteria for all of the tests are defined. In Fig. 11, an exemplary simulation result is shown (braking from 120 km/h until standstill, at R+E brake mode, at lowered adhesion (sprayed rail). For this type of test, the maximum permissible brake distance is 700 m, while during simulations, 519 m were obtained, so the test result is positive.

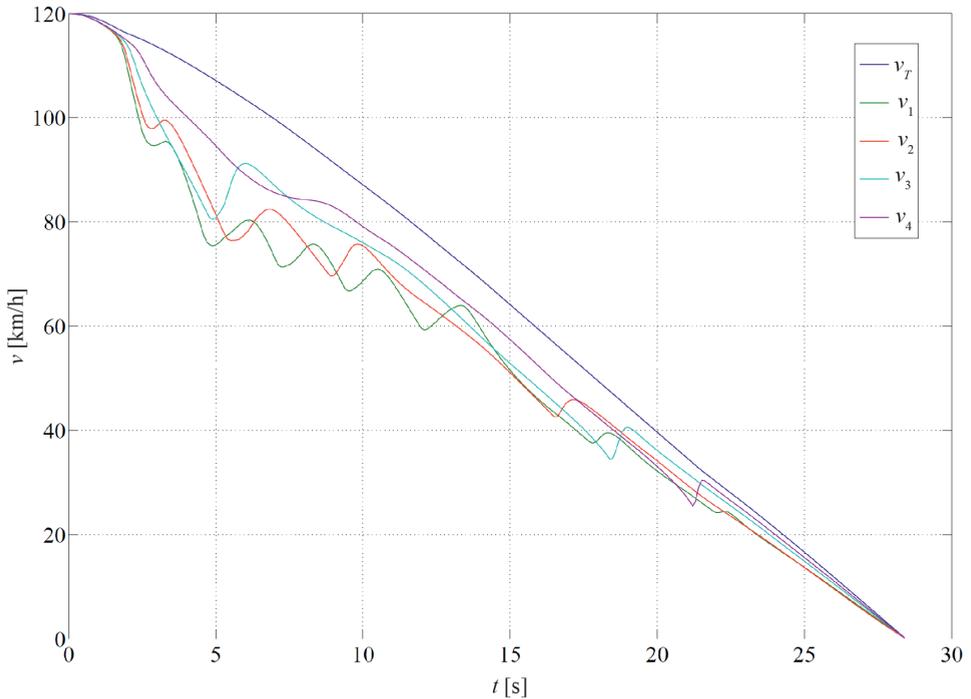


Fig. 11. Test results of simulated braking of an electric locomotive with ED brake from

#### 4. Conclusions

The developed model of the traction vehicle with asynchronous drive and models of anti-slip and anti-slide devices make performing simulation tests of various types of vehicles at a varying adhesion possible. The results of the performed simulations make determining the settings for the controllers of anti-slip and anti-slide devices possible.

Applying optimised anti-slip and anti-slide devices provides for an optimal exploitation of the available adhesion, which makes increasing both the driving and braking torque of the traction vehicle possible.

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