Matlab Simulink® Model of a Braked Rail Vehicle and Its Applications

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1. Introduction

When a braking force applied to the axle sets of a rail vehicle exceeds a critical value, which depends on the wheel-rail adherence, the wheels start to slide. If no corrective action is taken, in a very short time the wheels can be locked. Locking of the wheels or their excessive slide can result in increased braking distance and damage to the wheel rims (flat spots, also called "flats"). Wheel flats are sources of vibration and noise, they lower riding quality of a vehicle as well as passenger comfort, but first and foremost increasing of the braking distance directly impairs safety of the train staff and passengers and also of people nearby. In order to prevent excessive wheel slide and wheel lock, rail vehicles are equipped with Wheel Slide Protection (WSP) systems.

From the point of view of a WSP controller, the rail vehicle is a strongly non-linear and non-stationary plant. From the other hand there exist intuitive expert knowledge concerning the way the slide should be controlled. For these two reasons fuzzy logic controllers (FLC) are widely used in WSP systems (Caldara et al. (1996), Barna (2009)).

One of the basic problems concerning FLCs is lack of formal methods of design. Designing the rule bases of the fuzzy controllers is usually performed using a trial-and-error method, which in turn requires performing numerous tests of the controlled plant. When a controlled plant is a rail vehicle, possibilities of performing such tests are very limited due to high costs of tests and danger of damaging the wheels. A commonly used solution to this problem is employing a simulator of a braked rail vehicle, which can be used for preliminary design, optimization and tuning of the controllers. Tests on a real object are performed at the last stage of the design process in order to verify the designed controller.

The purpose of this chapter is to present the Matlab Simulink® model of a braked rail vehicle and its many applications, which comprise both designing and testing of the WSP systems. In the next two introductory sections the slide phenomenon as well as the structure and principle of operation of WSP systems are described in order to provide the readers who are not familiar with the WSP devices with some useful information which would facilitate reading of this chapter.

2. Slide during braking of a rail vehicle

During braking of a rail vehicle, a braking torque generated by a rail vehicle braking system is applied to the axle sets. When a value of this torque exceeds a boundary value, which depends on the instantaneous value of wheel-rail adherence, than the circumferential speed of the wheels starts to decrease (which is called sliding). If no corrective action is taken, in a very short time the wheels can be locked.

Locking of the wheels of a rail vehicle, as well as an excessive wheel slide have several negative consequences, out of which two are critical. First and foremost, due to decreasing of the adhesion coefficient, the braking force remains constant at a small value, which makes impossible effective braking of a vehicle and results in significant increasing of the braking distance. As an example, the braking distance of a 150A passenger car braked at 120 km/h amounts, according to both field and simulation tests approximately 480 m, while at braking with all the wheels locked at the beginning of braking it can be increased even up to 800 m. Increasing of the braking distance impairs the safety of passengers, train staff and people in proximity. Secondly, when locked wheels slide along the rails, flat spots (called "flats") can be produced on wheel treads. The depth of a wheel flat can amount up to several millimeters (Pawełczyk (2008)), especially in case of a prolonged slide. In Fig. 1 and Fig. 2 photographs of wheel flats of a passenger car wheels are shown.

Fig. 1. Flat spot on a wheel tread 1

Fig. 2. Flat spot on a wheel tread 2

Increase of the temperature of the sliding wheel and rapid cooling of the outer wheel layer, caused by regaining speed after the wheel slide has ceased, may result in forming of martensite around the flat spot. Martensite is a form of steel, firm, but fragile, which is subjected to cracks and crumbling. It develops first of all at the surface, and then propagating far into the wheel material. It results in developing the loss of the material of the wheel treads (Jergéus (1998); Kwaśnikowski & Firlik (2006)).

Damage to the wheel treads may be caused not only by wheel lock, but also by excessive difference (above 30 km/h) between vehicle velocity and circumferential velocity of a wheel.

Vibrations also lower the comfort of the passengers and the trains staff, negatively influencing their mood and ability to work after the journey, and also their health. In Fig. 3 simulation characteristics are shown for both a vehicle with round wheels and a vehicle with one flat spot on one wheel, at speed 140 km/h. The influence of a flat spot is seen the most clearly for frequency of app. 13,5 Hz. Lowering of comfort, defined as increasing of the effective value of vertical acceleration in the middle of the body amounts in this particular case to app. 30 %, and it depends on the vehicle velocity (Ofierzyński (2008)).

The concern about wheel slide has increased, as rail vehicles began running at higher speeds, because not only braking torque which is necessary to brake a vehicle needs to be bigger, which increases the probability of the wheel slide, but also consequences of such event are more serious when it occurs at high vehicle velocities.

To prevent the described above situations, rail vehicles are equipped with Wheel Slide Protection (WSP) devices, which detect slide of the vehicle wheels and adequately control the braking torque, not only preventing wheels from being locked, but also increasing the adhesion coefficient value, thus making the braking distance as short as possible.

Because a safety critical system is a system the failure of which can result in severe consequences, e.g. death or injury of people, or significant damages to property or environment, thus it is evident that a WSP device is a safety critical system (Barna & Kaluba (2009)). Standard CEN (2009) and leaflet UIC (2005) contain specification of requirements for both structure and functions of WSP devices.

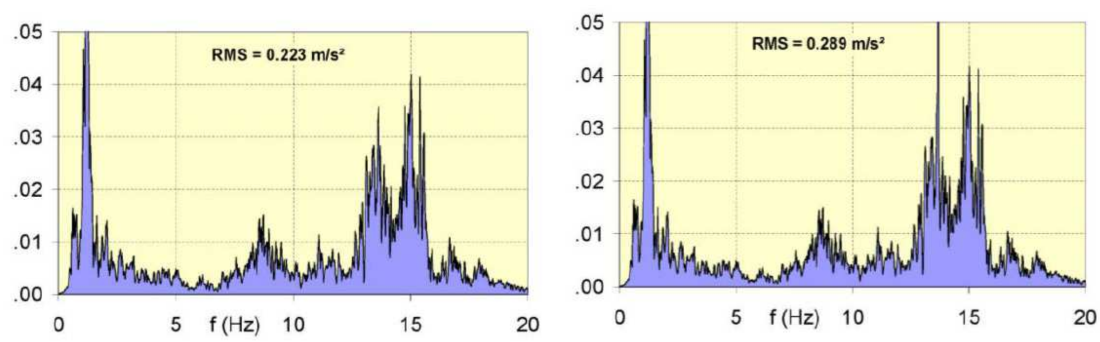

Fig. 3. Swing and frequency characteristics of vertical acceleration in the middle of the passenger car body, respectively without and with a wheel flat (Ofierzyński (2008))

3. Wheel Slide Protection systems

A WSP system consists of the following elements: wheel speed sensors, controller and dump valves. The block diagram of a WSP system for a two-axle bogie with block brake is shown in Fig. 4 (Barna (2009), UIC (2005), CEN (2009)).

A WSP controller on the basis on signals from speed sensors of all wheel-sets, performs calculation of circumferential wheel velocities and determines circumferential wheel

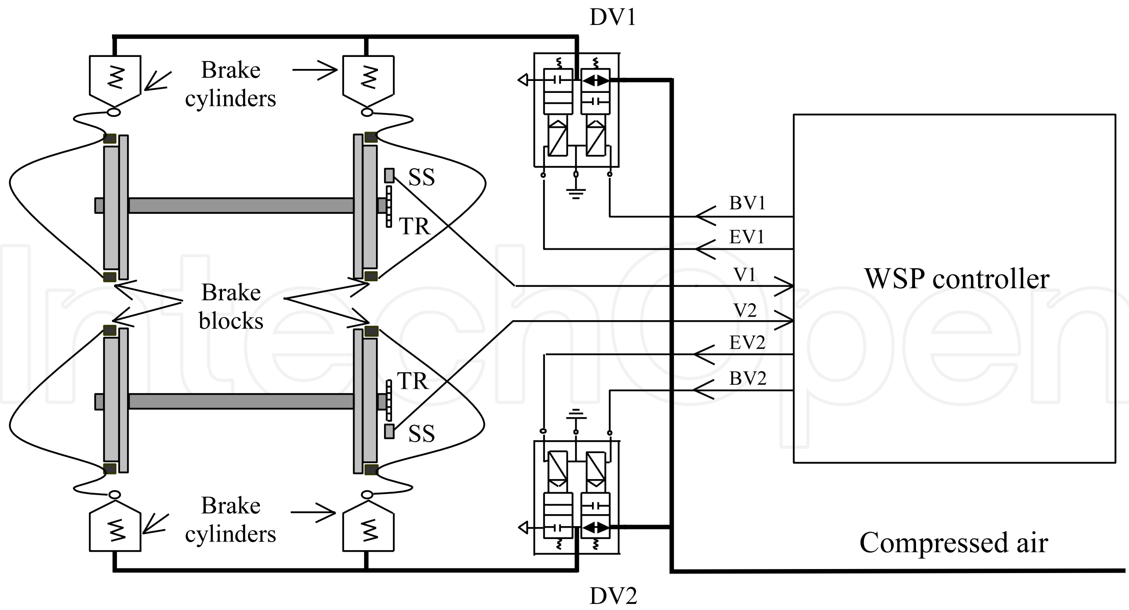

Fig. 4. Block diagram of a WSP system: DV1, DV2 - dump valves, V1, V2 - speed measurement signals, EV1, EV2, BV1, BV2 - dump valve control signals

accelerations and wheel slides, performs estimation of vehicle velocity (called the reference velocity), and then, on the basis of the obtained values appropriately controls the dump valves, in order to adjust the braking torque to the instantaneous adhesion conditions (Barna (2010a)). When a slide is detected, the braking torque should be adequately decreased by proper control of the dump valves; after recovery of adhesion the torque should be increased in order to provide effective braking of the vehicle.

Speed sensors make possible measuring the angular speeds of the wheels – they generate square-wave signals from which circumferential wheel speeds can be calculated, and which are main input signals for the WSP controller.

The WSP actuators are dump valves, mounted as close as possible to the brake cylinders. The valves can adopt one of the three states, which makes possible increasing, decreasing or maintaining the pressure value in the brake cylinders:

- filling of the cylinders (F)
- venting of the cylinders (V)
- cutting-off the air supply without venting (H).

Both filling and venting of the dump valves can be performed in one of the two following ways (Boiteux (1999)):

- *continuous*: applying either F or V state
- *graduated*: increasing or decreasing of the pressure are realized by applying repeated sequences of respectively F and H or V and H states.

In Barna (2009) seven levels of pressure change have been proposed, designated P1, P2, P3, U1, U2, U3, H. The levels are obtained by cyclical applying sequences of the three mentioned above states of the valves **P3**: P, **P2**: P H, **P1**: P **H**: H, **U1**: U H H, **U2**: U H, **U3**: U. Each of the states is applied for a defined period of time, e.g. 100 ms.

4. Characteristics of the WSP controllers

Wheel Slide Protection (WSP) devices control braking torque, not only preventing wheels from being locked, but also increasing the adhesion coefficient value, making the braking distance as short as possible. Thus the main task of WSP systems is "*to make the best use of available adhesion for all intended-operating conditions by a controlled reduction and restoration of the brake force to prevent axle sets from locking and uncontrolled sliding due to low adhesion*" (CEN (2009)).

The main difficulties concerning the control algorithm resulting from the characteristics of a braked rail vehicle as a controlled plant are listed below, the first two out of which are the most crucial:

- instantaneous value of the coefficient of adhesion between wheel and rail ψ cannot be measured
- translation vehicle speed v_T is usually not measured due to costs
- controlled plant is highly non-linear
  - characteristics $\psi = f(s)$ is non-linear (see Fig. 13)
  - characteristics $M_b = f(p_c)$ is characterized by hysteresis
  - dump valves behavior is non-linear
- controlled plant is non-stationary — parameters of characteristics $\psi = f(s)$ change in a way which is possible to comprehend in stochastic manner only
- direct control of a braking torque is not possible; actuators control the pressure values in the brake cylinders
- controlling of the brake cylinder pressures is performed indirectly and discretely
- instantaneous values of the cylinder pressures are not known.

The task of a WSP controller is to achieve the highest possible at the current adhesion conditions adhesion forces (forces transferred to the wheels of the axle sets), which is equivalent with achieving the shortest possible braking distance. This goal is achieved, as mentioned in Section 3 by proper control of the dump valves of each axle set.

In Fig. 5 a trajectory $\psi = f(s)$ for an optimally working controller has been shown, superimposed on a generalized adhesion versus slide curve. A WSP device equipped with such a controller, called in Boiteux (1999) a Second Generation WSP, realizes control in the neighbourhood of the point B of an adhesion curve and maintains a value of a relative slide s close to its optimum value s_B (see subsection 5.2.4).

A control system such as described above is called an *extremum regulation system*. The task of the extremum regulation system is maintaining a controlled signal as close as possible to the extremum value, which changes depending on disturbance signals (Kaczorek (1977)). In the case of a braked rail vehicle and a WSP controller this extremum value should be the instantaneous value of adhesion coefficient between wheel and rail ψ.

In Fig. 6 an overall block diagram of a WSP control system is shown, which is a base for practically realized control algorithms of WSP controllers.

A braked rail vehicle as a control plant is strongly non-linear and non-stationary, therefore the WSP controllers are usually designed using Fuzzy Logic and Expert

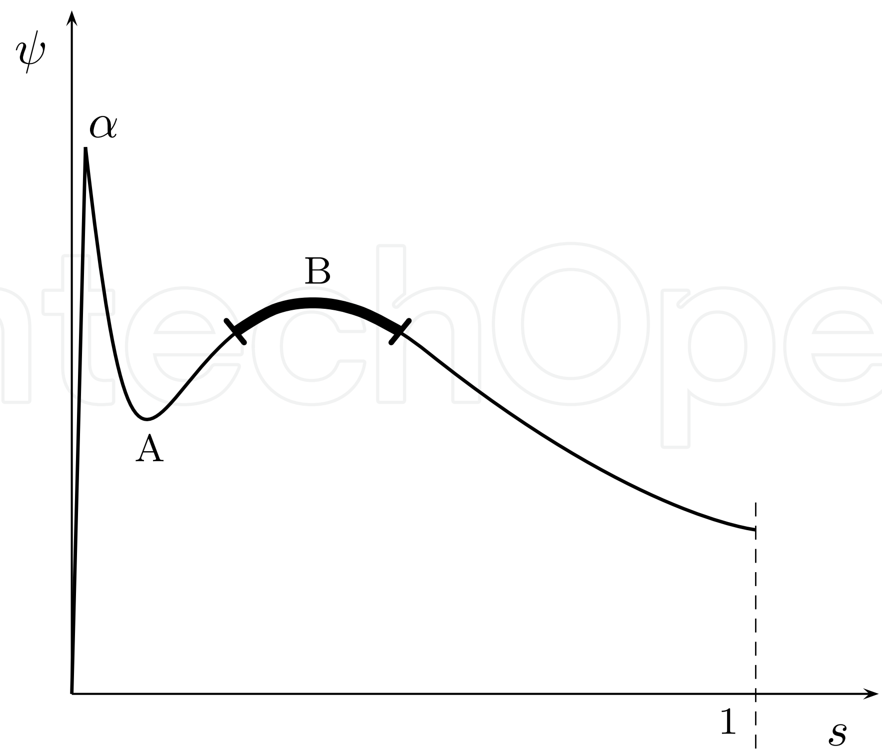

Fig. 5. The principle of operation of an optimal Wheel Slide Protection (Boiteux (1999))

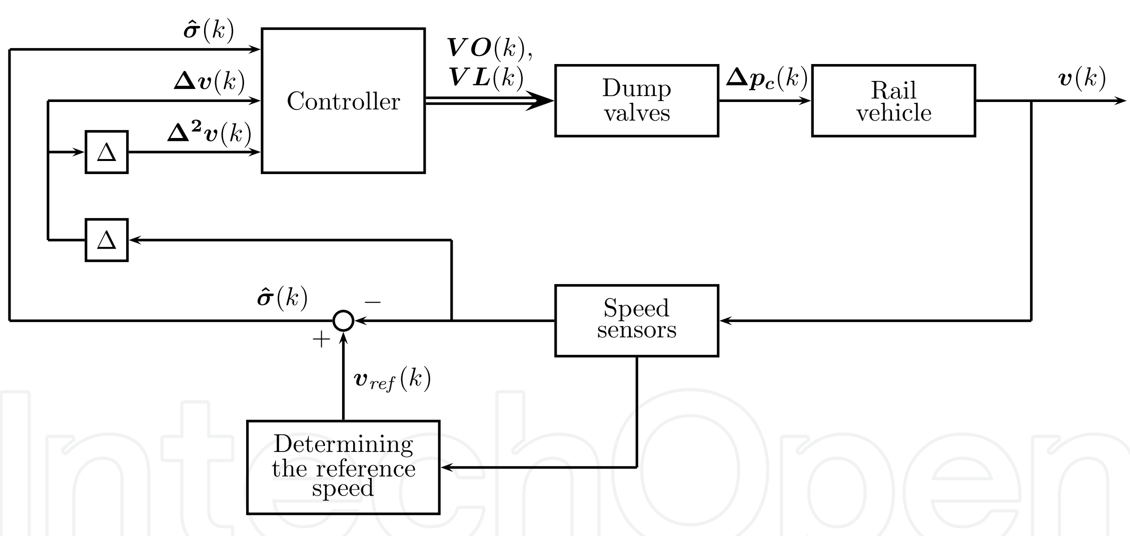

Fig. 6. Practical block diagram of WSP control system (Barna (2009))

Knowledge techniques (Caldara et al. (1996); Cheok & Shiomi (2000); Mauer (1995); Sanz & Pérez-Rodríguez (1997); Will & Żak (2000)).

The WSP control system using Fuzzy Logic controllers implemented in Matlab Simulink® is shown in Fig. 7. The control system is realized as Triggered Subsystem Simulink® block, triggered every 100 ms, which corresponds to a real controller program cycle time. The controller blocks for every axle are implemented with Fuzzy Logic Controller Simulink® blocks. The inputs for each FLC block are absolute axle slide σ [km/h] and axle acceleration a, which are calculated on the basis of the measured wheel circumferential velocities and vehicle reference velocity v_{ref}, as shown in Fig. 6. The measured wheel circumferential velocities are obtained by feeding the calculated continuous wheel velocities (see section 5.2.3) through a model of speed sensors. Fuzzy Logic Controllers are of PI type. The Simulink® subsystem determining the reference velocity v_{ref} is shown in Fig. 8. The output of each FLC is an integer in the range from -3 to 3, designating one of seven levels of pressure change (see section 3), which is then transformed in a post-processing block into signals controlling a dump valve.

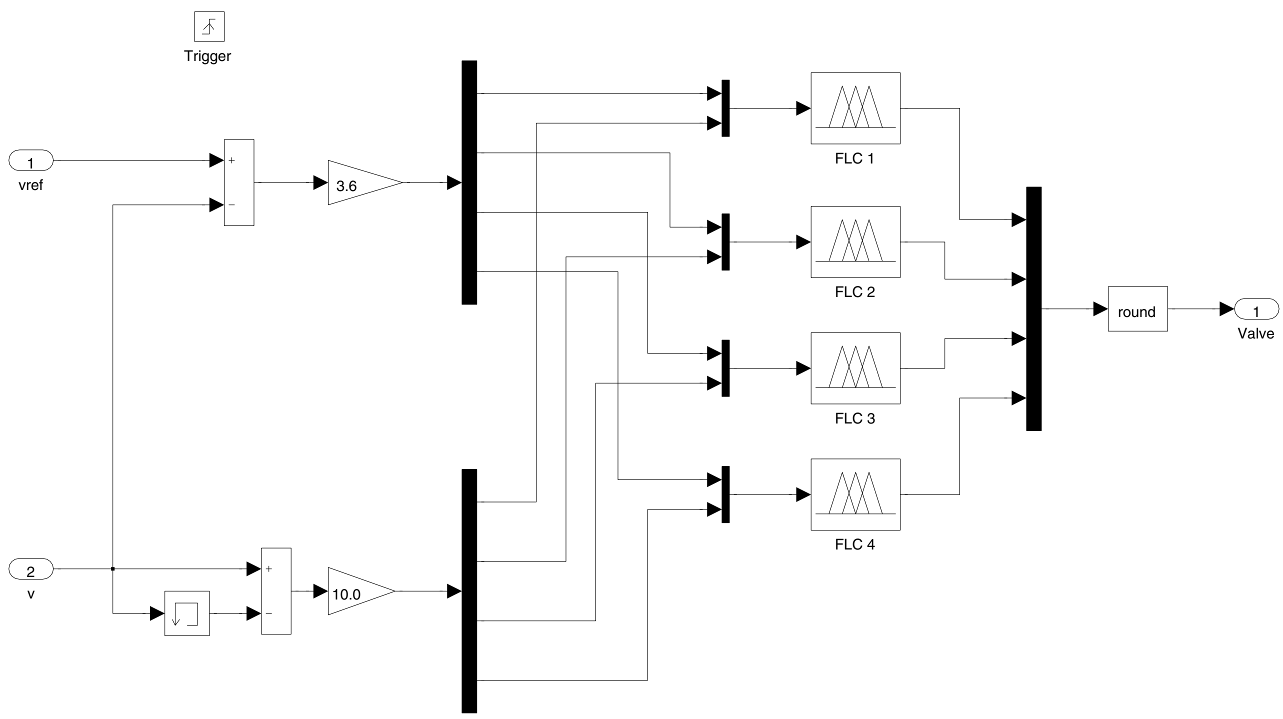

Fig. 7. Fuzzy Logic WSP control system implemented in Matlab Simulink®

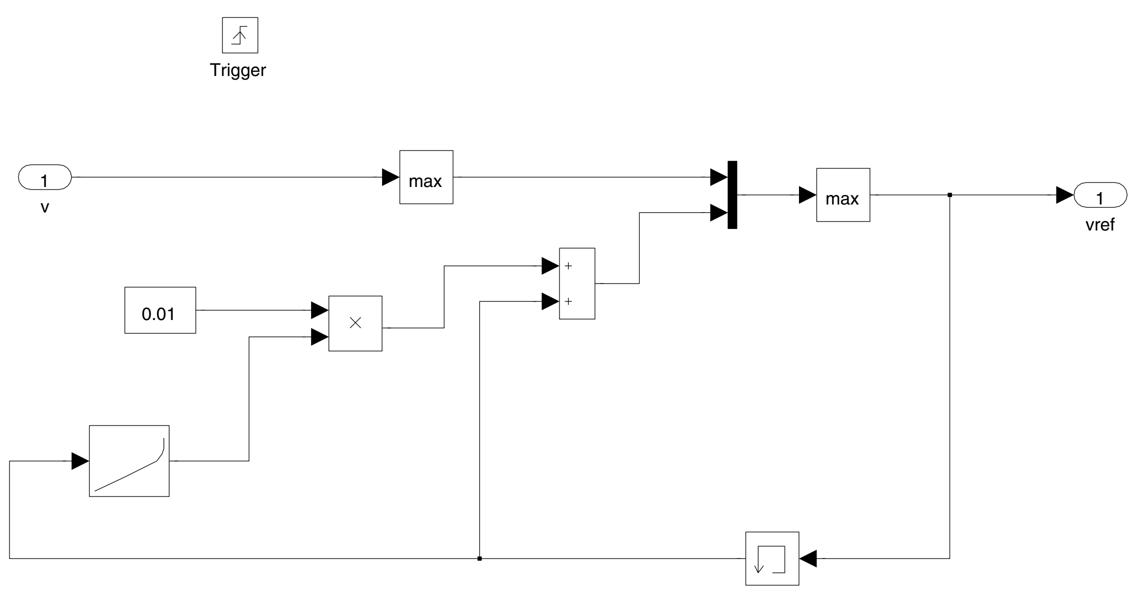

Fig. 8. Matlab Simulink® subsystem determining the reference velocity v_{ref}

5. Simulator of a braked rail vehicle

5.1 Introduction

Designing and testing of the FLC WSP controllers demands performing numerous experiments. Performing tests using a real rail vehicle is not feasible, because of high costs and possibility of damaging the object, therefore for designing and testing WSP controllers a test stand is indispensable. One of the most important elements of the test stand is a simulator of a braked rail vehicle.

The mathematical model of a braked rail vehicle, simulation model and various versions of a simulation stand are presented in this subsection.

5.2 Mathematical model of a braked rail vehicle

5.2.1 Introduction

The basis for a rail vehicle simulator is a mathematical model of a braked rail vehicle, taking into consideration the basic phenomena occurring during sliding and omitting the phenomena that are of no or slight significance.

The model has the following parameters:

- parameters describing properties of the vehicle (e.g. mass, number of axles, inertia of the axle sets)
- vehicle velocity in the moment of commencement of braking
- intensiveness of braking defined as the maximum pressure in the brake cylinders.

The inputs to the model are:

- state of dump valves generated by the WSP controller
- state of the rail (adhesion coefficient).

The purpose of the model is performing the simulations of braking of a rail vehicle at reduced adhesion. Therefore all the basic phenomena influencing the wheel speeds during braking must be modeled as faithfully as feasible. A special attention must be put to all the main non-linear subsystems of the system. The most important example is the wheel-rail adhesion coefficient.

Some simplifying assumptions have been adopted, which simplify the model, while do not significantly alter its functionality. One of these assumptions is considering the vehicle as a rigid body with the agglomerated mass.

The model consists of the following subsystems:

- model of a braking system
- model of a rotational motion of a wheel-set
- model of the adhesion curve
- model of a translation vehicle motion.

The subsystems of the model are described in the subsequent subsections.

5.2.2 Model of a braking system

The model of the braking system describes the relation between pressure in the brake cylinders of the axle set and its braking torque.

This model can be divided into the following three subsystems:

- model of the pneumatic system and the dump valves
- model of the lever system
- model of the friction elements.

5.2.2.1 Model of the pneumatic system and the dump valves

The pressure at the input of a dump valve can be described with the following equation (Kaluba (1999)):

$$p_{c_{in}} = p_{c_{max}} \left(1 - e^{-0.75t}\right). \tag{1}$$

The dump valves are a part of delivery of the WSP system. However, from the modeling point of view, because they are integrated into the pneumatic system of the vehicle, they are considered as part of this system.

The dump valve has been modeled as the inertia element of adjustable time constant independently for filling T_F and venting T_V.

The pressure in the brake cylinders supplied by the valve are given with one of the following equations:

for filling

$$p_c = p_0 + (p_{c_{in}} - p_0)\left(1 - e^{-\frac{t-t_z}{T_F}}\right), \tag{2}$$

for venting

$$p_c = p_0 e^{-\frac{t-t_0}{T_V}}, \tag{3}$$

and for holding

$$p_c = p_0. \tag{4}$$

5.2.2.2 Model of the lever system

The relation between the pressure of the friction linings of the brake blocks onto the wheel treads of the brake disks, which is exerted by the piston of the brake cylinder and the pressure in the brake cylinder is highly non-linear and is characterized by hysteresis, resulting from friction in the brake cylinder and in the joints of the clamp mechanism (Knorr (2002); Tao & Kokotovic (1996)). This relation is shown in Fig. (9). The two rays limiting the area are given with:

$$N_{k1} = (p_c A_k - F_f)\eta_s \eta_r i_p, \tag{5}$$

$$N_{k2} = (p_c A_k - F_f)(2 - \eta_s \eta_r) i_p. \tag{6}$$

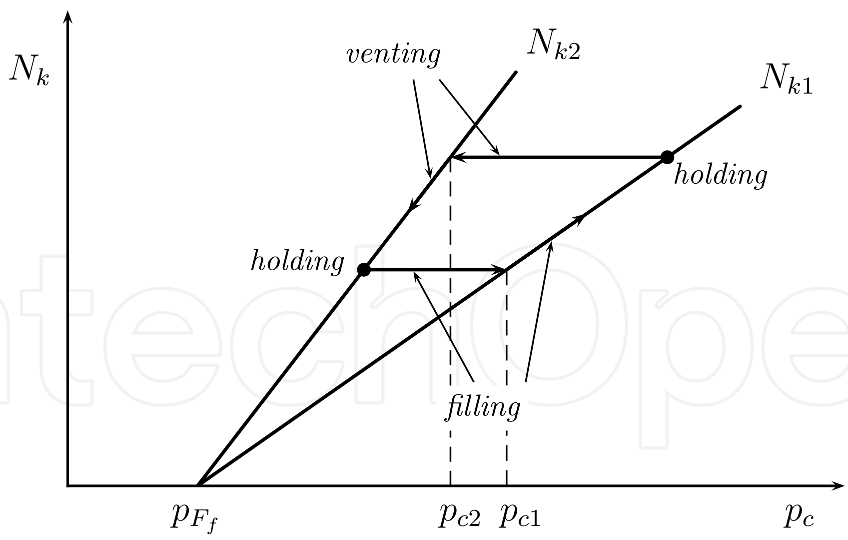

Fig. 9. Relation between the air pressure in brake cylinders and the brake blocks pressure

5.2.2.3 Model of the friction elements

Because some vehicles are equipped with disk brakes and some with block brakes, models of both types of brakes have been prepared.

For both types of the brakes the braking torque is given with:

$$M_b = N_k \mu r_b, \tag{7}$$

where r_h is the braking radius.

For the block brake, the braking radius r_h is equal to the wheel radius r, for the disk brake it is equal to the radius of the brake disk r_d.

The friction coefficient between the friction linings of the brake blocks and the wheel tread or the brake disk μ depends in a non-linear way on many factors, first of all on wheel velocity and pressure of the disk block.

A simplified value of the friction coefficient μ for a block brake is given with the empirical equation. In this model the equation given in Sachs (1973) has been adopted:

$$\mu = 0,16 \sqrt[3]{\frac{40,25}{v}} \sqrt[3]{\frac{2,461}{p_x}}, \tag{8}$$

while:

$$p_x = \frac{N_k}{A_x}. \tag{9}$$

In Fig. 10 a surface of the friction coefficient μ vs. circumferential speed of the vehicle wheel and the unitary pressure of the friction linings for a block brake is shown.

For the disk brake the friction coefficient μ has been determined on the basis of the results presented in Kaluba (1999). In Fig. 11 a surface of the friction coefficient μ vs. circumferential speed of the vehicle wheel and the sum of unitary pressure of the friction linings for a disk brake is shown.

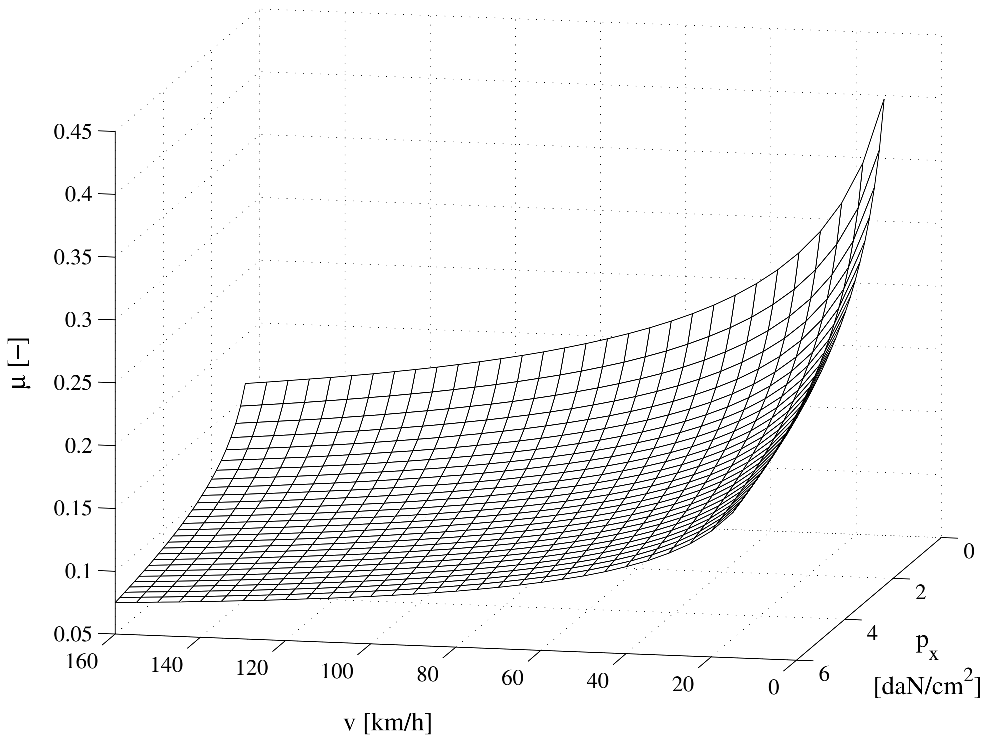

Fig. 10. A surface of the friction coefficient μ vs. circumferential speed of the vehicle wheel and the unitary pressure of the friction linings (Sachs (1973))

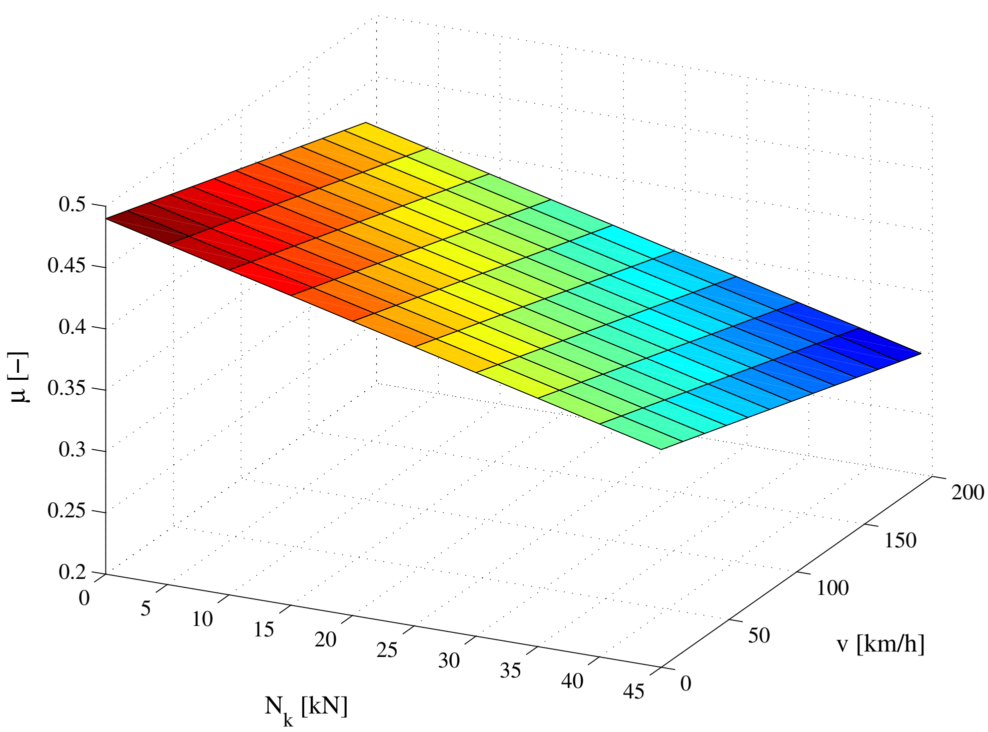

Fig. 11. A surface of the friction coefficient μ vs. circumferential speed of the vehicle wheel and the sum of unitary pressure of the friction linings for a disk brake (Kaluba (1999))

5.2.3 Model of a rotational motion of a wheel-set

A simplified diagram of forces acting upon a braked axle set is shown in Fig. 12.

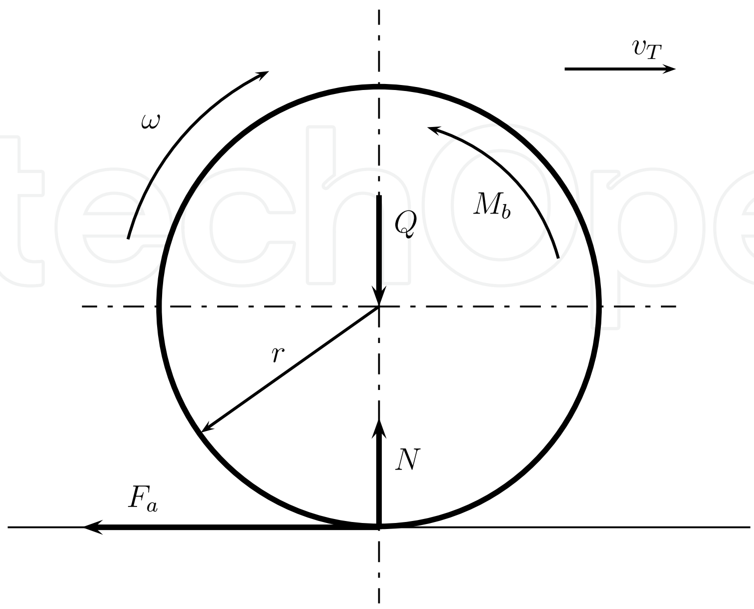

Fig. 12. A simplified diagram of forces acting upon a braked axle set

The dynamics of the axle set rotary motion is given with the following equation set:

$$J\frac{d\omega_i}{dt} = F_{ai}r - M_{bi}, \tag{10}$$

$$F_{ai} = \psi_i(s_i, \mathbf{p})Q_i, \tag{11}$$

$$s_i = \frac{v_T - v}{v_T}, \tag{12}$$

where (11) defines the adhesion force, and (12) defines the relative slide. The model of value of ψ is described in the following subsection.

5.2.4 Model of the adhesion curve

Generalized characteristics of instantaneous adhesion coefficient ψ versus relative slide s is shown in Fig. 13. Value of adhesion coefficient at point B (ψ_B) is called *maximum exploitable adhesion* and corresponding slide (s_B) is called *optimal slide* (Boiteux (1987)).

The parameters of the generalized curve are the following values:

- relative slide at point α s_α
- available adhesion coefficient ψ_α
- relative slide at point A s_A
- adhesion coefficient value at point A ψ_A
- optimum relative slide s_B
- maximum exploitable wheel-rail adhesion coefficient ψ_B
- adhesion coefficient value for wheel lock ψ_l.

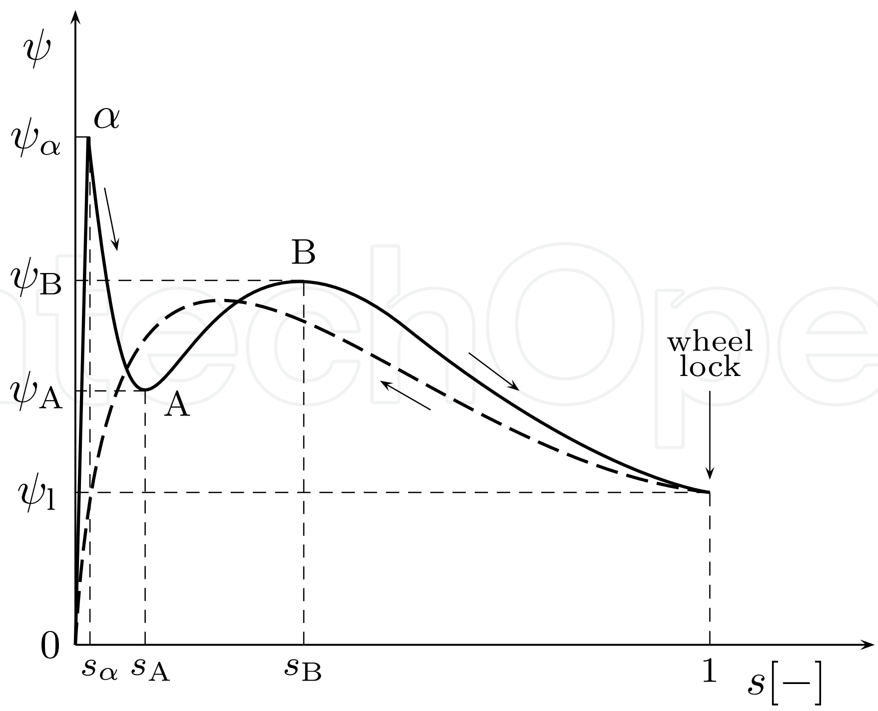

Fig. 13. Generalized characteristics of instantaneous coefficient of adhesion ψ versus relative slide s (ORE (1985))

The shape and parameters of the curve depend on many factors, most of which are difficult or impossible to establish. The most important factors are listed below (Boiteux (1987; 1990; 1998); ORE (1985; 1990)):

- state of the rail
- vehicle velocity v_T
- absolute wheel slide σ
- relative wheel slide s
- circumferential deceleration of the wheels d
- slide energy E developed in the wheel and a rail contact point.

The mathematical model of adhesion coefficient, based on Boiteux (1987; 1990; 1998); ORE (1985; 1990), is described in detail in Barna (2009).

As an example, according to Boiteux (1987), maximum exploitable wheel-rail adhesion coefficient ψ_B and optimum relative slide s_B can be estimated with the following equations:

$$\psi_B = k\sqrt{\frac{Q\,\psi_\alpha\,E_o}{2\,d_m}}, \tag{13}$$

and

$$s_B = \frac{1}{v_T}\sqrt{\frac{2\,E_o\,d_m}{Q\,\psi_\alpha}}. \tag{14}$$

One of the most important behavior of the adhesion curve, which has been taken into consideration is, that during the braking at low adhesion with efficiently operating WSP system, the value of the maximum exploitable wheel-rail adhesion coefficient ψ_B increases, so it can double or even triple within 10 s.

Simulated characteristics of adhesion coefficient ψ vs. time for a braking from the initial velocity of 120 km/h, showing increase of the maximum exploitable wheel-rail adhesion coefficient in the course of braking is presented in Fig. 14

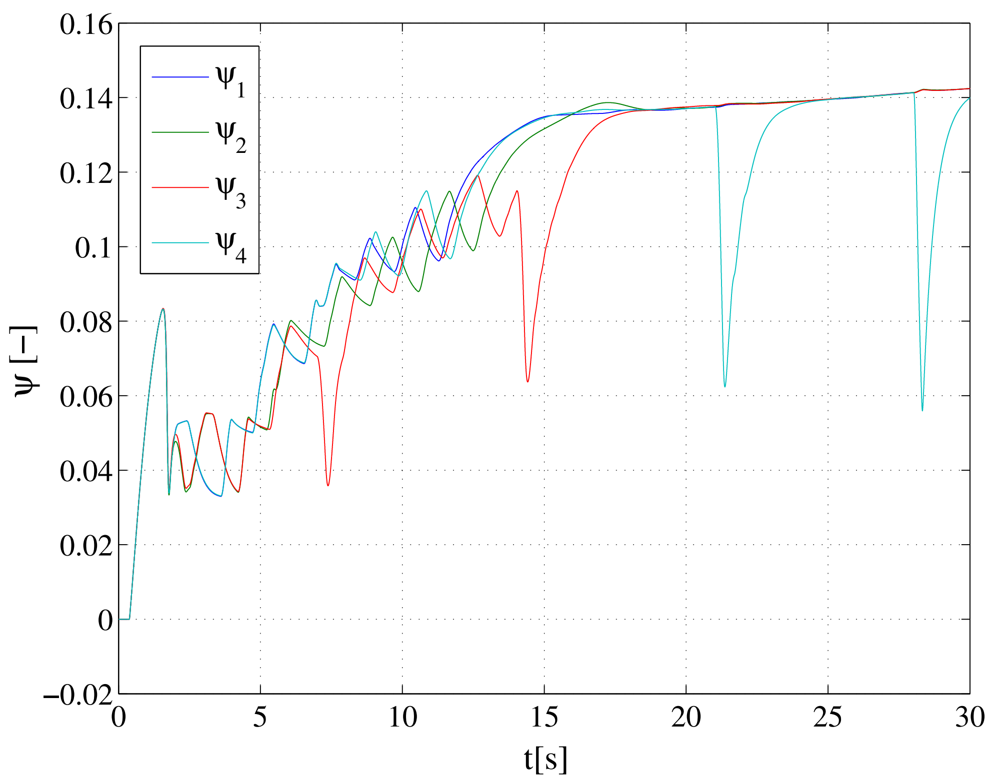

Fig. 14. Simulated characteristics of adhesion coefficient ψ vs. time, where ψ_i – adhesion coefficient for i-th axle set (Barna (2009))

5.2.5 Model of a translation vehicle motion

The model of a translation vehicle motion is described with:

$$m\frac{dv_T}{dt} + \sum_{i=1}^{n} \frac{J_i}{r}\frac{d\omega_i}{dt} = -F_{a\Sigma}, \tag{15}$$

where i is the number of axle sets of the vehicle.

5.3 Simulation model and a simulation stand

5.3.1 Requirements

In UIC (2005) and CEN (2009) there are contained requirements concerning the simulator of a braked rail vehicle. According to these normative documents the simulator should make possible simulating various conditions and scenarios concerning the wheel-rail adhesion as well as vehicle and track parameters:

- adhesion coefficient ψ between 0,02 (extremely poor adhesion) and 0,15 (dry rail)

- adhesion variation as occurs in real life
- sudden changes of adhesion as the ones occurring when a wheel encounters a spilled oil or dead leaves
- dynamic variation of wheel-rail adhesion coefficient vs. vehicle velocity and the slide controlled by WSP system
- maximum rail slope of 50‰
- drag braking test (when a vehicle is hauled at constant speed).

Simulator should make possible modeling the vehicles of the following parameters:

- vehicle velocity up to 240 km/h
- changes of parameters such as wheel diameter and inertia of axle sets, mass of the vehicle and particular bogies as well as location of the centroid, loading with passenger or cargo as well as braking force characteristics
- various brake positions with braking rate values λ being within the range 25% do 200% (for the braking systems independent of adhesion)
- braking systems which, apart from the friction brake are additionally equipped with the brake independent of adhesion, e.g. track brake

It is recommended, that for the traction vehicles it should be possible to simulate blending (electrodynamic brake cooperating with friction brake).

5.3.2 Simulation model

The simulation model of the braked rail vehicle integrated with the WSP system has been implemented in Matlab Simulink®. A simplified block diagram of the model is shown in Fig. 15. Validation of the model is basically performed by comparing the braking distances obtained from simulations performed with the experimental results (CEN (2009); UIC (2005)).

5.3.3 Simulation stands

In the Figures 16, 17 i 18 block diagrams showing various possibilities of realizing the laboratory test stand have been shown.

In Fig. 16 a stand described in UIC (2005) and CEN (2009) is shown, in which the computer simulation is minimized. This stand comprises a hardware model of a brake system with dump valves installed and pressure sensors mounted in brake cylinders. These pressure values are read by the analog inputs of the computer vehicle simulator, which calculates the adhesion forces and resulting wheel velocities. Fast analog output or counter output board of the computer vehicle simulator outputs pulse train simulating impulses from the WSP speed sensors.

A real WSP controlling device inputs the simulated speed signals from the computer vehicle simulator and outputs signals controlling dump valves in the hardware model of a brake system. The advantage of this stand is fidelity of simulating the pneumatic part of the vehicle braking system, which makes possible obtaining accurate values of brake cylinder pressures during WSP operation. The disadvantage is relatively high cost of the stand.

In Fig. 17 another version of a test stand is shown in which, comparing to the previous stand, also the pneumatic system is computer simulated. A reliable model of a pneumatic system

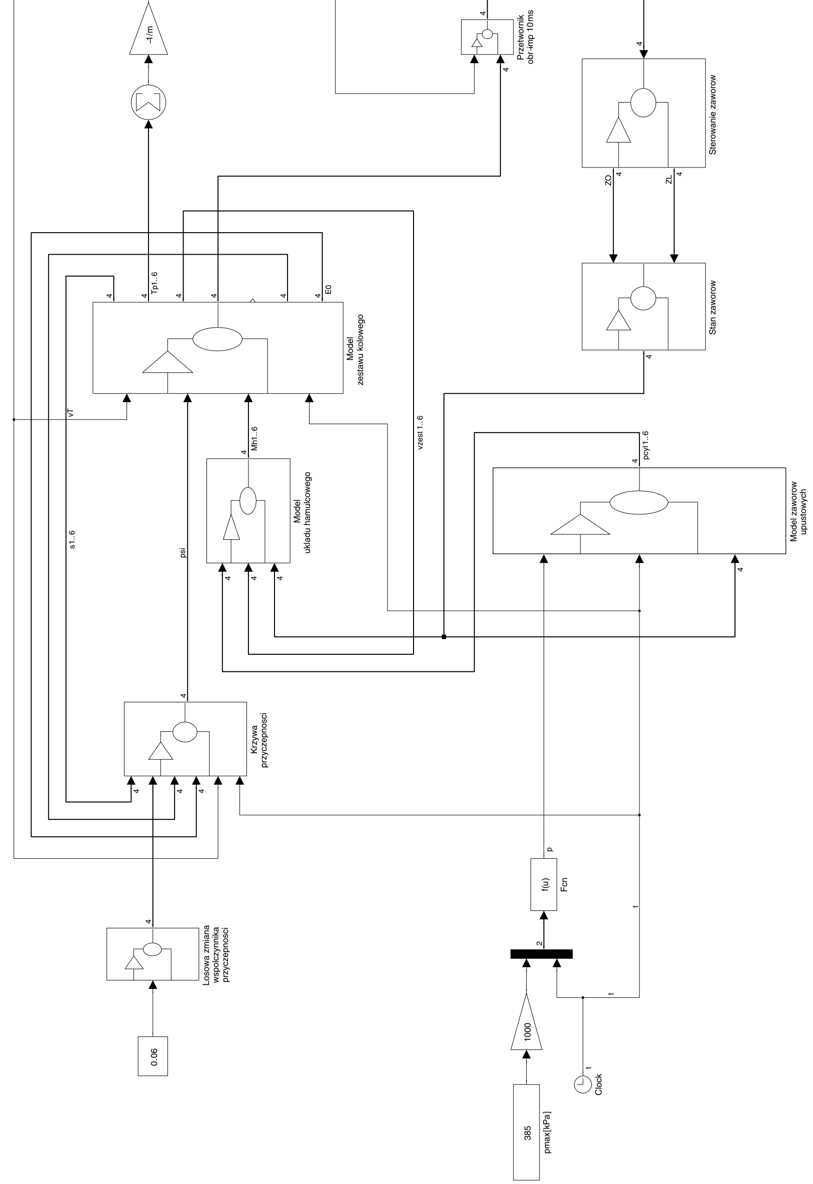

Fig. 15. A simplified block diagram of the braked rail vehicle integrated with the WSP system implemented in Matlab Simulink®

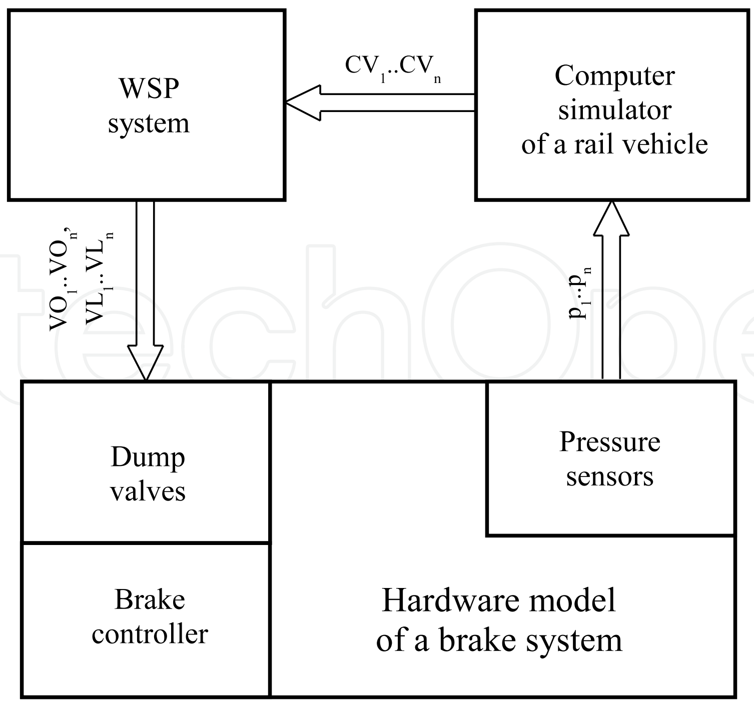

Fig. 16. Laboratory test stand using a hardware model of a brake system

and of the dump valves is additionally needed. The advantage of the stand is much lower cost. It is also much cheaper in operation, because it does not require supplying with the compressed air. However it cannot be used for homologation tests of WSP systems according to UIC (2005) and CEN (2009).

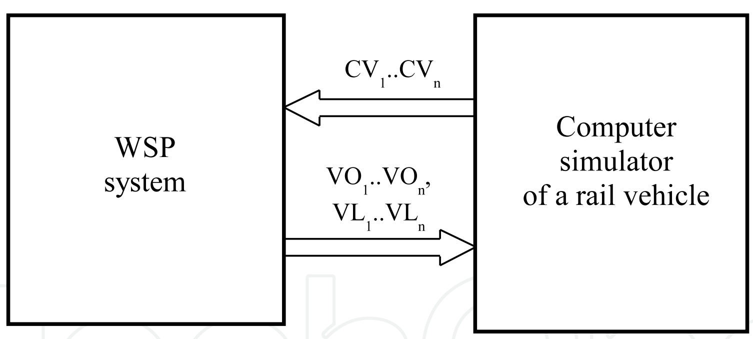

Fig. 17. Laboratory test stand with computer simulation of the pneumatic system

In Fig. 18 a third test stand is shown, in which a WSP controller is also computer simulated. Such stand can be realized if a control algorithm of a WSP system is known. The biggest difference comparing to the previous stand is, that the fast counters of the WSP system are also simulated. The stand is realized using only a computer with a Matlab Simulink® program, thus the advantage is a very low cost of the test stand. The stand is a good choice at the stage of development of WSP control algorithm, because the WSP developer can perform numerous tests of the WSP systems, examining various variants of the control algorithm. However, this stand does not allow for the testing of the ready WSP control device. One of the described above test stands should be used for such purpose. It is this variant of the test stand that is used for simulations described in this chapter.

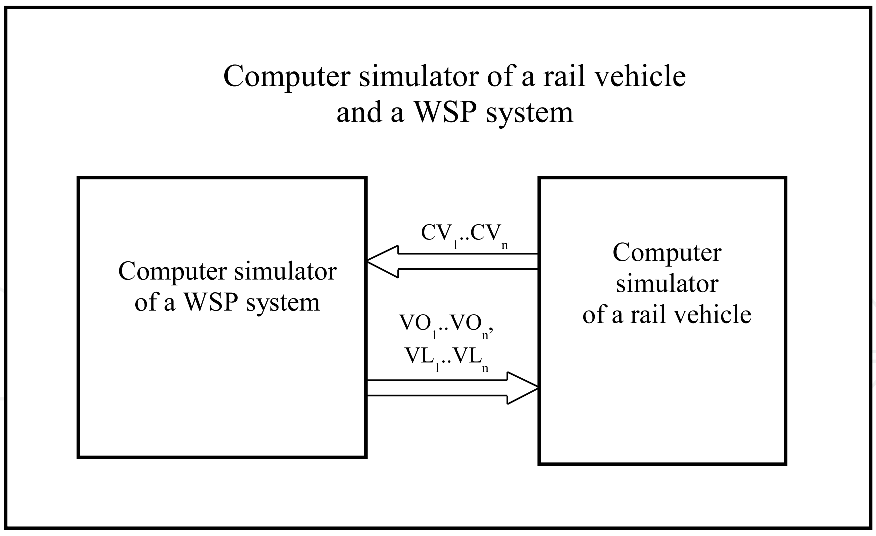

Fig. 18. Laboratory test stand with computer simulation of all the systems

6. Application and simulation results

6.1 Introduction

The simulator of a braked rail vehicle can be used for manifold purposes, which fall into two categories. The first one is developing WSP control algorithms, and it may comprise:

- acquiring the numerical data concerning the critical values used in the control algorithms
- establishing the influence of the changes of the control plant parameters
- establishing the influence of the changes of the controller parameters
- acquiring the reference data for design of the FLC WSP controllers
- tests of the WSP controllers.

The second category is preliminary testing of the WSP control algorithms or the designed WSP control devices or systems against the requirements given in CEN (2009) and UIC (2005).

In the remaining part of this section the three chosen examples of using the simulation stand for some of the mentioned above purposes are shown.

6.2 Acquiring reference data for designing the WSP control algorithms

By performing simulated brakings at both good and poor adhesion conditions, it is possible to acquire data, which can be used for designing both structure and parameters of WSP controllers. Several examples are given below.

In Fig. 19 the characteristic of vehicle acceleration for simulated braking from 120 km/h at maximum brake cylinder pressure $p_{c_{\max}}$ = 385 kPa at good adhesion is shown. Such data, acquired for various brake positions and initial test speed are especially valuable for developing algorithm of calculating the reference speed.

In Fig. 20 the results of simulated braking at poor adhesion conditions (initial value of $\psi_B \approx 0,06$) are presented. The initial test velocity was 160 km/h and the brake position was R ($p_{c_{max}}$ = 385 kPa). In the figure vehicle velocity, circumferential wheel velocity and circumferential

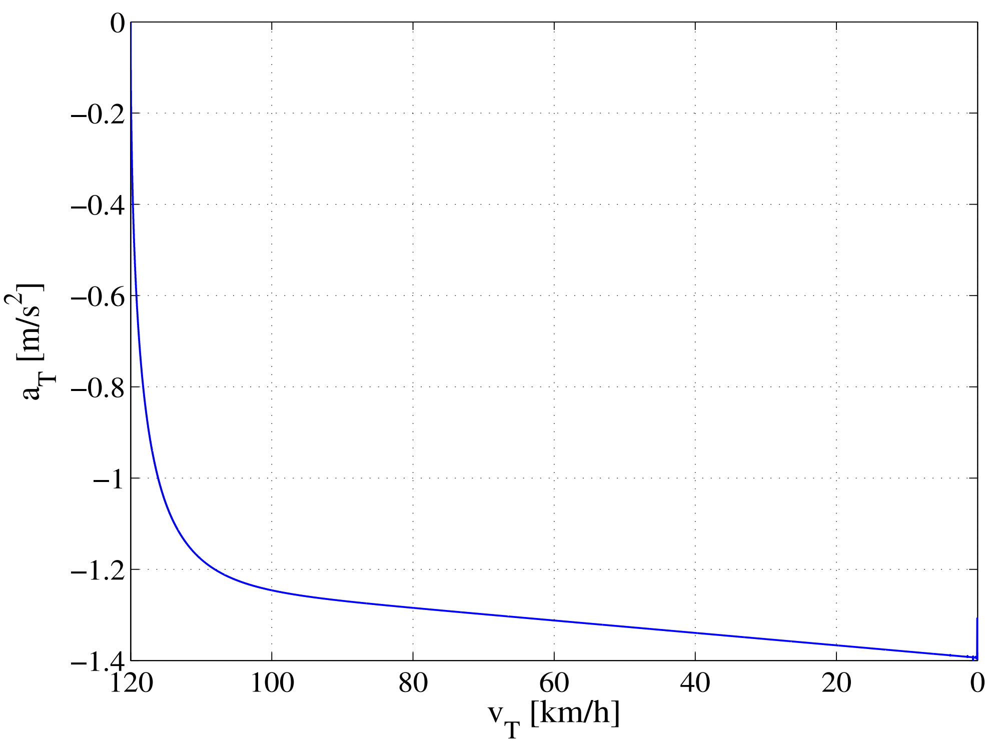

Fig. 19. Characteristic of vehicle acceleration a_T vs. vehicle velocity v_T for simulated braking from 120 km/h at maximum brake cylinder pressure $p_{c_{max}}$ = 385 kPa at good adhesion (Barna (2009))

wheel accelerations are shown. The numerical data can be used for setting and adjusting the parameters of the WSP control algorithms.

With a braked rail vehicle simulator it is also possible to perform simulations of braking at poor adhesion conditions aimed at acquiring data concerning the behavior of circumferential wheel speeds and accelerations, which can facilitate the process of developing the WSP control algorithms. In Fig. 21 the characteristics of vehicle velocity and circumferential wheel speed and acceleration during the first cycle of slide detection and suppression.

6.3 Designing the rule bases of FLC WSP

The rule bases of fuzzy controllers can be designed using expert knowledge and numerical simulation results, provided that such data is available and that the controller which has been used for obtaining the data has controlled the slide in an effective way. In Cheok & Shiomi (2000) a method of designing the fuzzy controller rule base has been proposed, which uses both expert knowledge and chosen numerical results obtained during operation of a PID controller. Basing on the numerical results some rules have been determined, and the lacking rules have been defined on the basis of expert knowledge concerning the slide control. In a simulator of a braked vehicle it is possible to access the signals, which are nor available during normal operation of the systems, i.e. wheel-rail adhesion coefficient and translation velocity of the vehicle. An advantage of having a simulator can be taken to create so called reference controllers which can be used for generating data for FLC WSP rule base design.

A block diagram of a WSP control system with a reference controller is shown in Fig. 22.

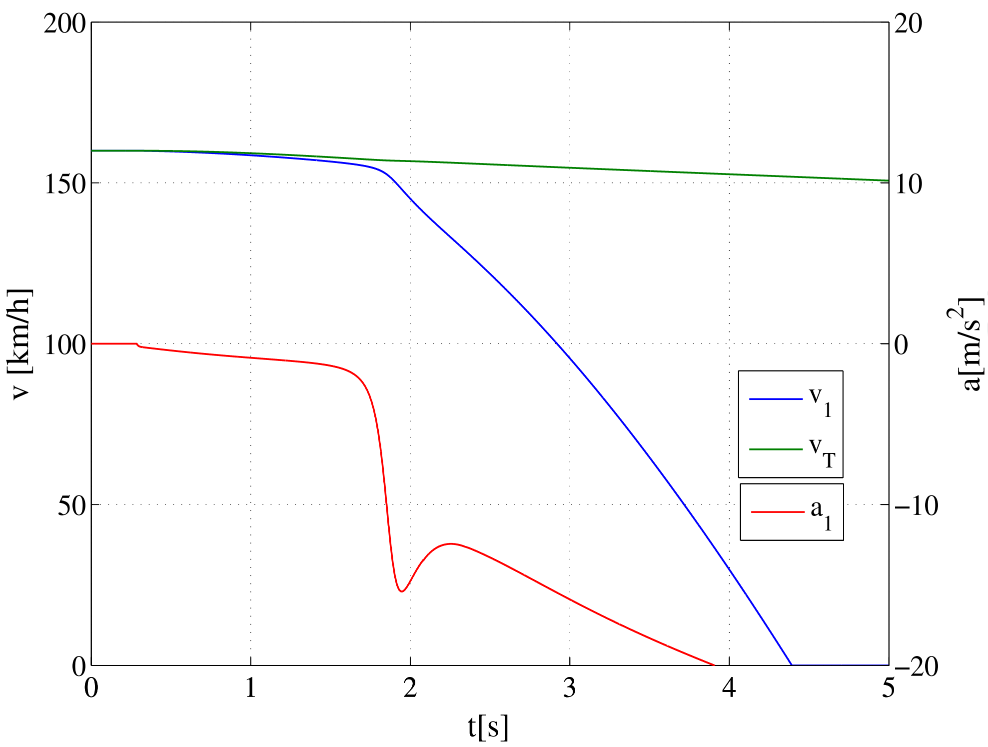

Fig. 20. Results of simulated braking at poor adhesion conditions (initial test velocity 160 km/h, initial value of $\psi_B \approx 0,06$, brake position R ($p_{c_{max}}$ = 385 kPa)) (Barna (2009))

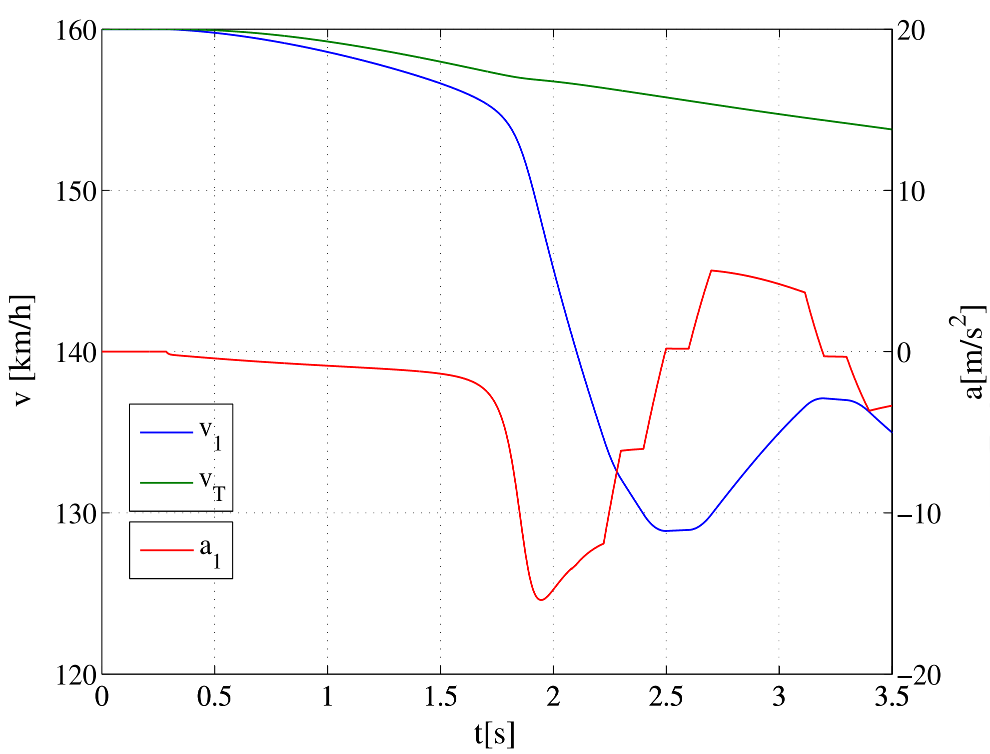

Fig. 21. Characteristics of vehicle velocity and circumferential wheel speed and acceleration for the first cycle of slide detection and suppression (Barna (2009))

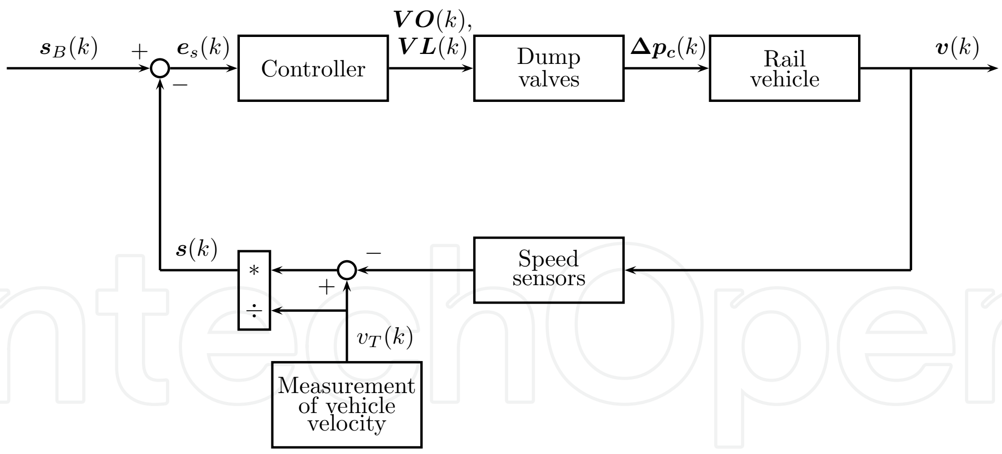

Fig. 22. Block diagram of a WSP control system with a reference controller (Barna (2009))

In Barna (2010b) a method of designing the rule bases of WSP fuzzy controllers for rail vehicles is presented. In this method results of simulation of two reference controllers are used, both of which use signals which are nor available during normal operation of the systems, i.e. optimal relative slide s_B and translation velocity of the vehicle v_T. Two types of reference controllers have been designed in order to obtain data for designing a real-life FLC WSP . One of them is a Fuzzy Logic controller, and the other is an expert knowledge based controller.

A block diagram of a Fuzzy Logic reference WSP control system implemented in Matlab Simulink® is shown in Fig. 23. It's structure is similar to a regular Fuzzy Logic WSP controller, but the reference value is the optimal relative slide s_B. A standard Mac-Vicar Whelan rule base has been adopted (Yager & Filev (1995)).

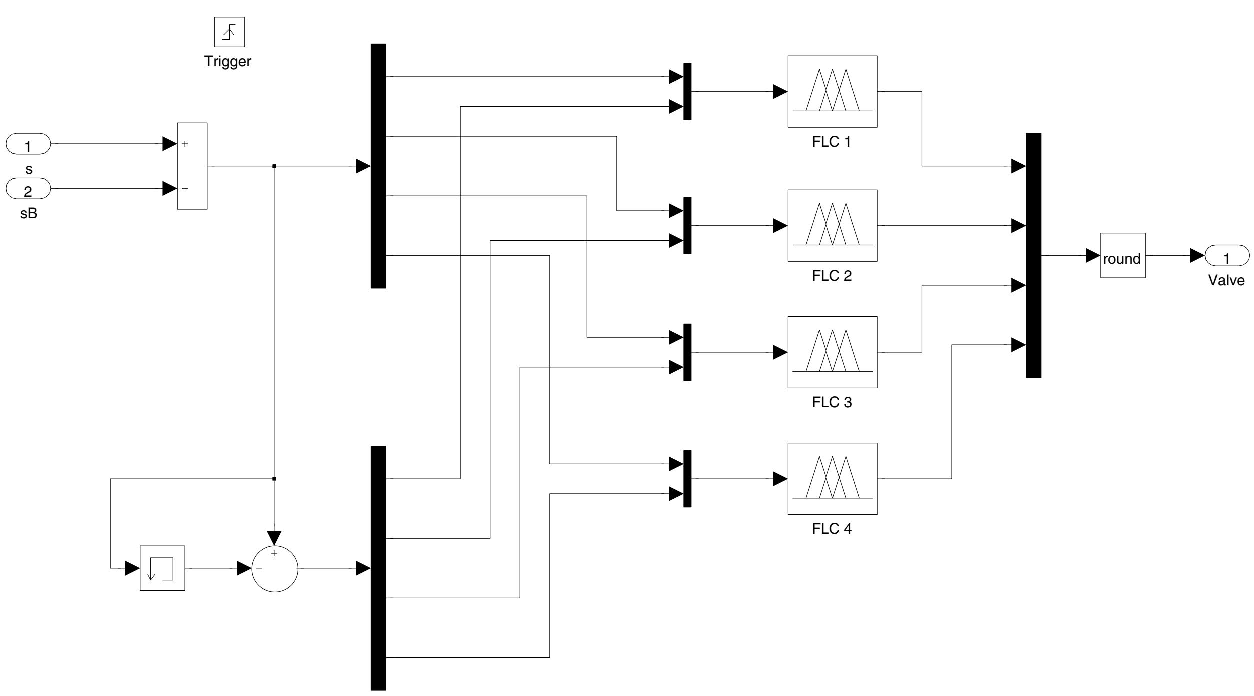

Fig. 23. Fuzzy Logic reference WSP control system implemented in Matlab Simulink®

A knowledge based algorithm is based on a MSG1 WSP controller made by Knorr-Bremse and presented in Boiteux (1999).

The input controller values for each axle set are:

- absolute wheel slide σ
- circumferential wheel acceleration a.

The output controller values are the same as in the Fuzzy Logic controllers described earlier. Three reference functions are defined, which divide the slide space into four ranges versus the reference velocity v_{ref}: $\Delta\sigma_1(v_{ref})$, $\Delta\sigma_2(v_{ref})$ and $\Delta\sigma_3(v_{ref})$. These functions are shown in Fig. 24.

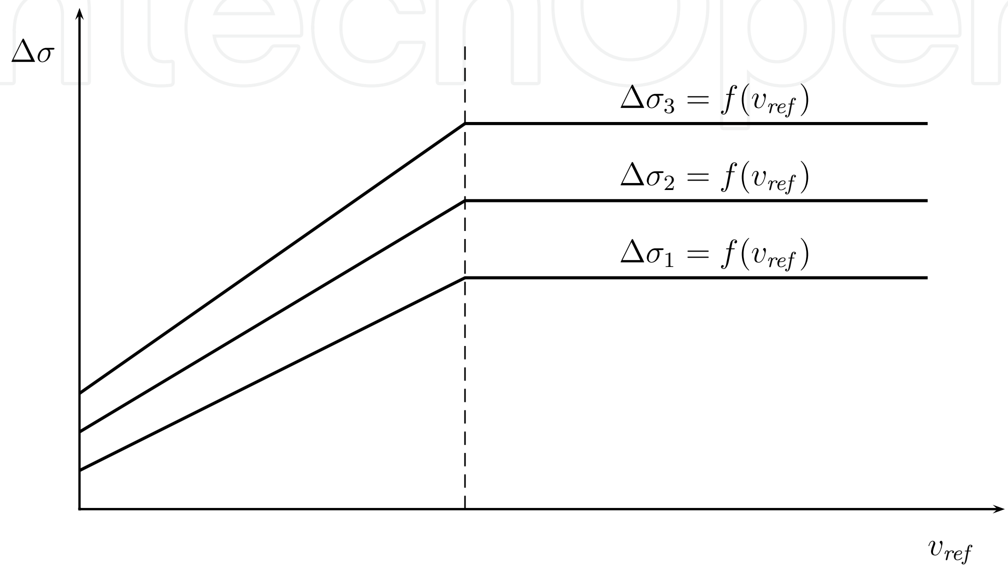

Fig. 24. Reference values of absolute slide used in control algorithm of MSG1 WSP system (Boiteux (1999))

Four values of circumferential wheel accelerations are also defined: a_1, a_2, a_3 oraz a_4. These values are demonstratively pictured in Fig. 25.

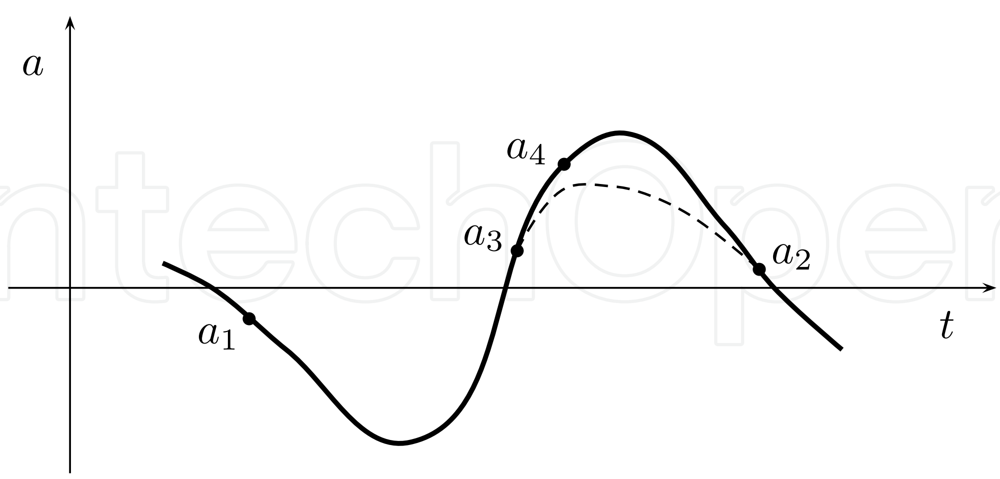

Fig. 25. Values of circumferential wheel accelerations used in control algorithm of MSG1 WSP system pictured demonstratively (Boiteux (1999))

The decision table of MSG1 WSP system control algorithm is presented in Table 1.

A block diagram of a knowledge based WSP control system implemented in Matlab Simulink® is shown in Fig. 26.

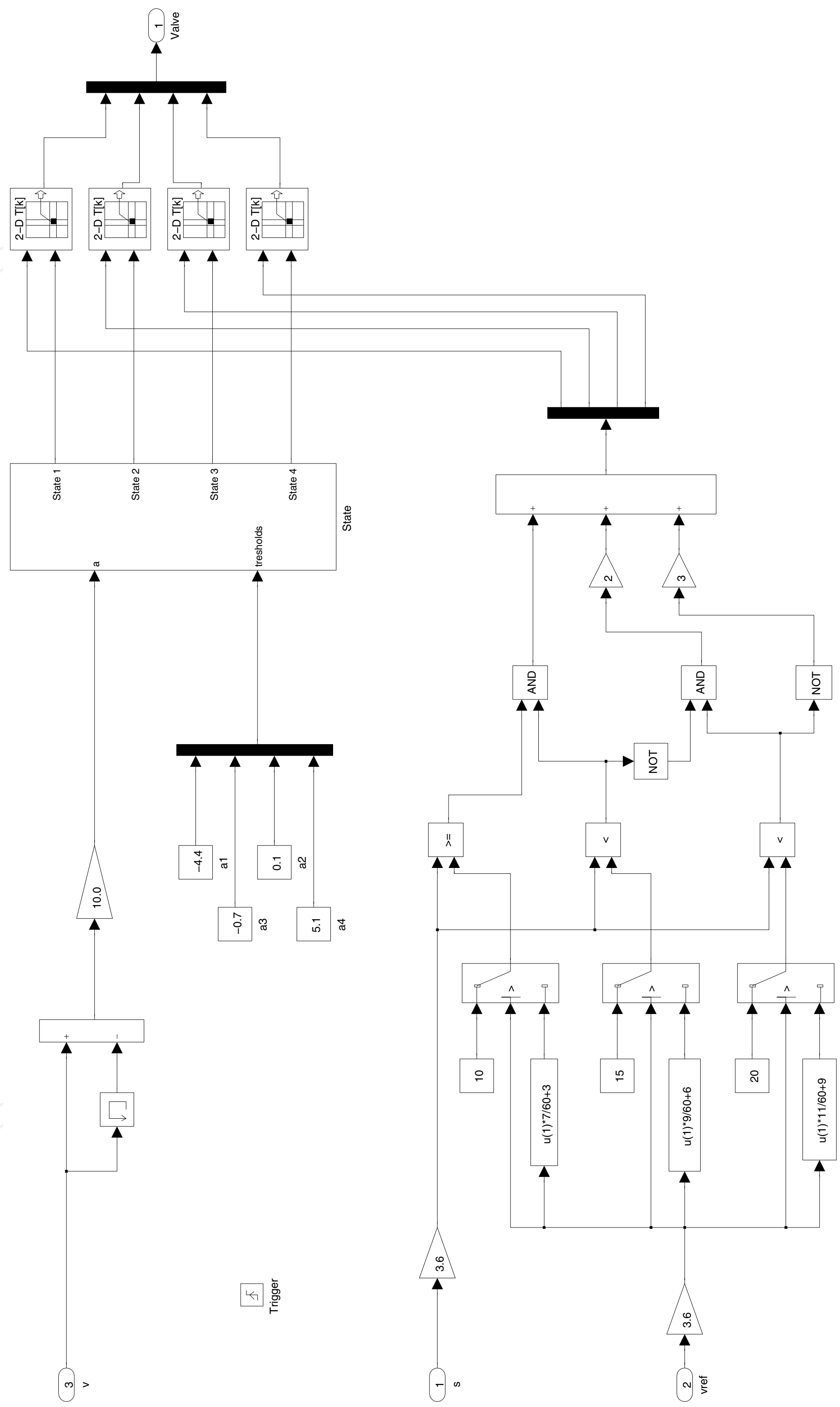

Fig. 26. Block diagram of a knowledge based WSP controller implemented in Matlab Simulink®

$a \rightarrow$					
σ	$\boldsymbol{a_1 \rightarrow a_3}$	$\boldsymbol{a_3 \rightarrow a_4}$	$\boldsymbol{a_4 \rightarrow a_4}$	$\boldsymbol{a_4 \rightarrow a_2}$	$\boldsymbol{a_2 \rightarrow a_1}$
$\downarrow$ $\boldsymbol{\sigma < \Delta\sigma_1}$	H	H	P3	H	P2
$\boldsymbol{\Delta\sigma_1 < \sigma < \Delta\sigma_2}$	U1	H	P2	H	P1
$\boldsymbol{\Delta\sigma_2 < \sigma < \Delta\sigma_3}$	U2	H	P1	H	H
$\boldsymbol{\Delta\sigma_3 < \sigma}$	U3	U3	U3	U3	U3

Table 1. The decision table of MSG1 WSP system control algorithm (Boiteux (1999))

Simulations have been performed with a simulation model of a braked rail vehicle and controllers implemented in Matlab Simulink®. In Fig. 27 exemplary results of simulated braking from 120 km/h with a FLC reference controller at decreased adhesion have been shown.

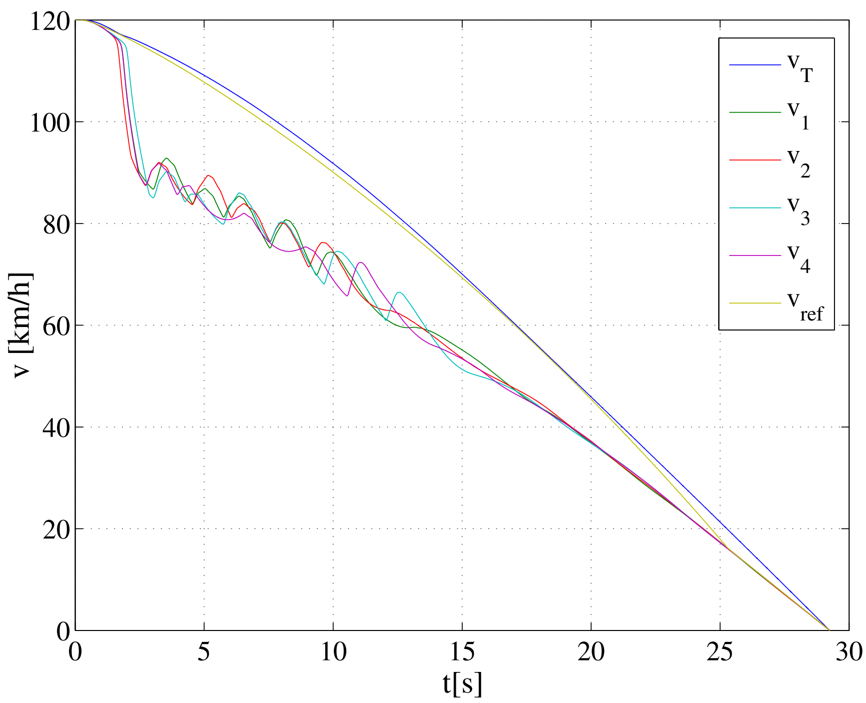

Fig. 27. Characteristics of velocities $\boldsymbol{v}$, v_T and v_{ref} for simulated braking from 120 km/h at maximum brake cylinder pressure $p_{c_{max}}$ = 385 kPa at decreased adhesion for a FLC reference controller (Barna (2009))

After running a series of tests contained to UIC 541-05 leaflet and EN 15595, simulation data have been used to develop a preliminary rule base using a dedicated C program.

The controller has been tested against the UIC 541-05 requirements with positive results. An exemplary simulated braking is shown in Fig. 28.

6.4 Testing the WSP systems

When a WSP system has been designed, simulated tests can be performed in order to check whether the WSP meets the requirements of CEN (2009) and UIC (2005). The test program consists of several tests, divided into three groups: slip tests, drag braking test and a test at low adhesion. The exact specification of tests depends on a vehicle type, which can be one of

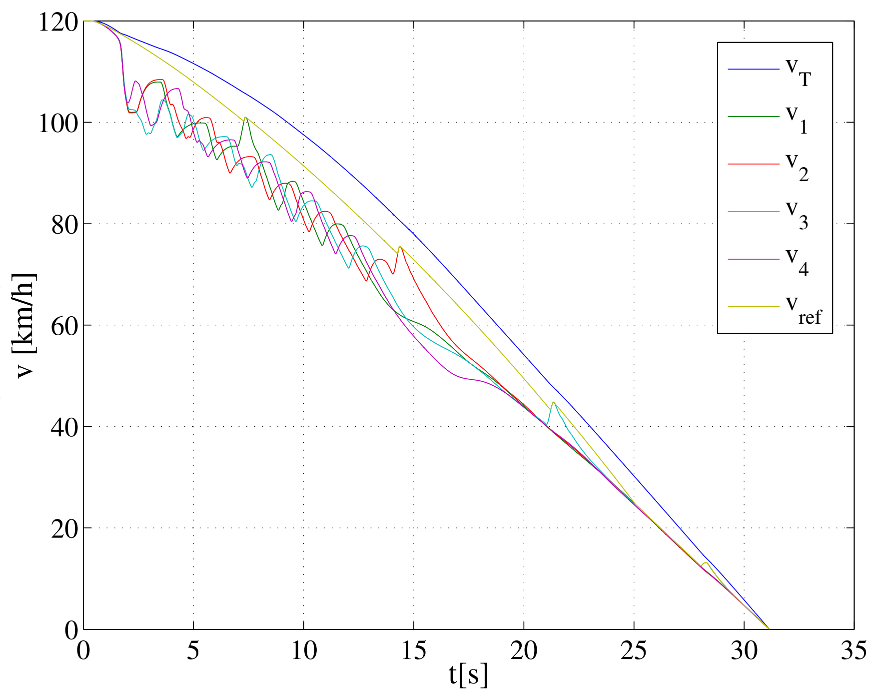

Fig. 28. Characteristics of velocities $\boldsymbol{v}$, v_T and v_{ref} for simulated braking from 120 km/h at maximum brake cylinder pressure ($p_{c_{\max}}$ = 385 kPa) at decreased adhesion for a WSP controller (Barna (2009))

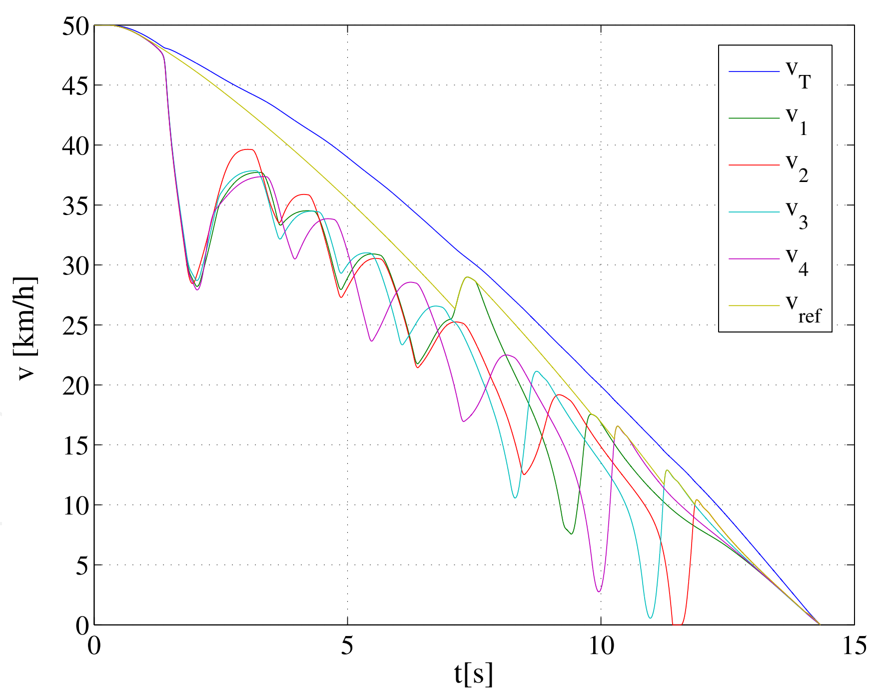

Fig. 29. Characteristics of velocities $\boldsymbol{v}$, v_T and v_{ref} for simulated braking from 50 km/h at maximum brake cylinder pressure ($p_{c_{\max}}$ = 385 kPa) at decreased adhesion for a WSP controller (Barna (2009))

the following: a passenger coach, a wagon, a locomotive, a train-set or a high speed train. The initial vehicle speed and brake position, as well as additional conditions, are specified for each test. In order to check whether a WSP meets the requirements of the normative documents, appropriate simulated tests should be performed, and the results assessed.

Testing WSP systems against normative references is described in Barna (2009) and Barna (2010a). In Figures 29, 30 and 31 exemplary test results are shown: in Fig. 29 the results of tests from 50 km/h, and in Fig. 30 the results of tests for rail covered with soap. In Fig. 31 characteristic of pressure in a brake cylinder for simulated braking from 120 km/h is shown.

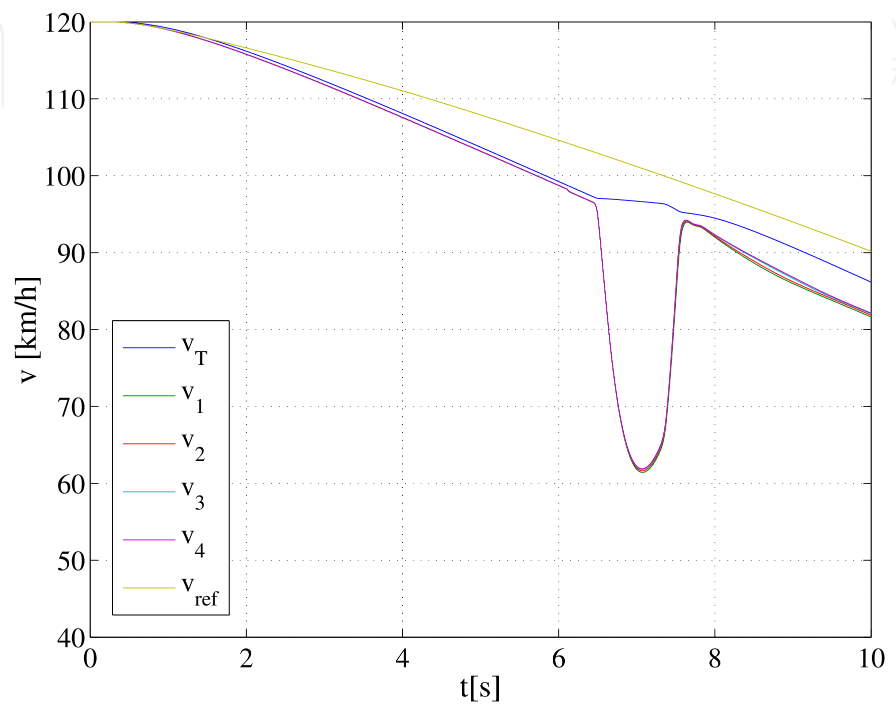

Fig. 30. Characteristics of velocities $\boldsymbol{v}$, v_T and v_{ref} for simulated braking from 120 km/h at maximum brake cylinder pressure ($p_{c_{\max}}$ = 385 kPa) for rail covered with soap for a WSP controller (Barna (2009))

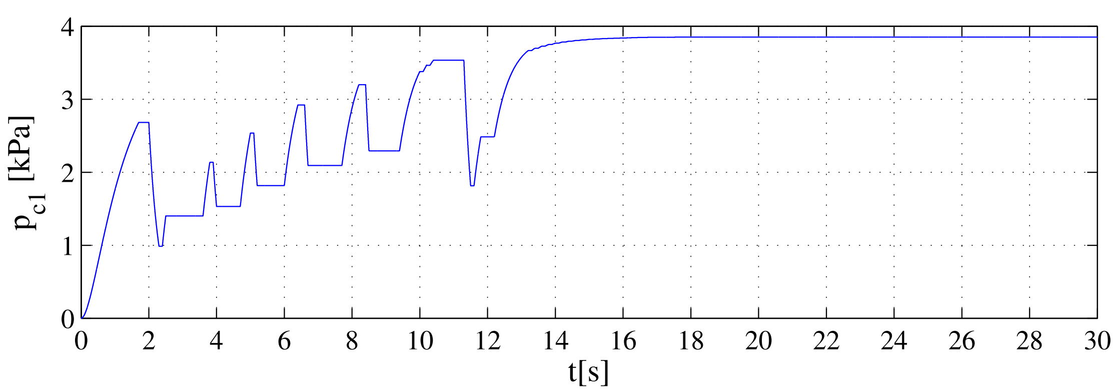

Fig. 31. Characteristics of pressure p_c for simulated braking from 120 km/h at maximum brake cylinder pressure ($p_{c_{\max}}$ = 385 kPa) at decreased adhesion for a WSP controller (Barna (2009))

7. Conclusions

In order to design an efficient WSP controller, a simulator of a braked rail vehicle is indispensable. The mathematical model of the wheel-rail adhesion must comprise regeneration of adhesion by controlled slide.

The simulator can be used for manifold purposes, including developing and testing of WSP controllers.

One of possible applications is using the simulation results of the reference controllers for designing the rule bases for WSP FLC controllers. From the analysis of performance of the WSP fuzzy controller, the rule base of which has been designed in this way, it can be concluded, that this method makes possible designing efficient controllers.

The test program realized by the simulator must comply with the requirements of CEN (2009) and UIC (2005), thus making possible testing of the controllers in the whole possible range of slide and acceleration values.

The futer research concerning the simulator of the braked rail vehicle would comprise:

- further development of the mathematical model of particular sub-models, especially the model of the adhesion coefficient and model of the pneumatic system
- developing a software/hardware simulator with xPCTarget toolbox using a computer fitted with I/O boards.

8. Acknowledgements

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9. List of symbols

a [m/s^2] – circumferential wheel acceleration
a_T [m/s^2] – vehicle acceleration
A_k [m^2] – area of piston of the brake cylinder
A_x [m^2] – area of friction surface of the brake disk
d_m [m/s^2] – average value of circumferential wheel deceleration from the beginning of braking and the moment when the slide value reaches 0.4
e [$-$] – control error
E [J] – instantaneous value of axle set slide energy, from the start of the braking until the instant moment t
E_o [J] – value of axle set slide energy from the start of braking until reaching of the working point (s, ψ) point B
F_f [N] – force of the return spring of the brake cylinder
F_a [N] – adhesion force for an axle set
$F_{a\Sigma}$ [N] – sum of adhesion forces for all axle sets of a vehicle
i_p [$-$] – final ratio of clamp mechanisms of an axle set
J [kg m^2] – moment of inertia of rotating elements associated with an axle set, reduced to the axle set axis

L_h [m] – total braking distance
M_b [Nm] – braking torque of an axle set
m [kg] – vehicle mass
N [N] – vertical rail reaction force for an axle set
N_k [N] – pressure exerted by brake discs, reduced to the braking radius
n [−] – number of axle sets of a vehicle
p_c [Pa] – brake cylinder pressure
$\mathbf{p}$ [−] – vector of parameters of ψ vs. s characteristics
$p_{c_{max}}$ [Pa] – maximum pressure in a brake cylinder for a given brake position
$p_{c_{in}}$ [Pa] – instantaneous air pressure at inlet of a dump valve
p_{c_0} [Pa] – outlet dump valve air pressure value at the moment of change of the dump valve state
p_x [daN/cm^2] – unitary pressure of the friction linings of a brake block
Q [N] – instantaneous load of the axle set
r [m] – wheel radius
r_b [m] – brake radius
r_d [m] – brake disk radius
s [−] – relative slide of wheels against the rails
s_B [−] – optimum relative slide
s_α [−] – relative slide for point α
t [s] – time
T_h [s] – total time of braking until standstill
T_V [s] – time constant of dump valve venting
T_F [s] – time constant of dump valve filling
v [km/h] – circumferential velocity of axle set wheels
v_{ref} [km/h] – vehicle reference velocity
v_T [km/h] – vehicle velocity
γ [−] – coefficient of rotational mass
η_r [−] – coefficient of increasing the efficiency of the lever mechanism of the brake clamp mechanism in motion
η_s [−] – static efficiency of the lever mechanism of the brake clamp mechanism
λ [−] – braking rate
μ [−] – friction coefficient of the brake block friction lining
ψ [−] – instantaneous value of wheel-rail adhesion coefficient
ψ_B [−] – maximum exploitable wheel-rail adhesion coefficient
ψ_l [−] – adhesion coefficient for locked wheels
ψ_α [−] – available adhesion coefficient
σ [km/h] – absolute slide of wheels against the rails
$\hat{\sigma}$ [km/h] – estimated absolute slide of wheels against the rails
ω [rad/s] – angular velocity of a vehicle axle set

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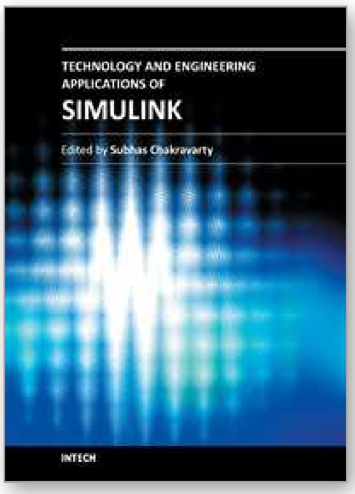

Technology and Engineering Applications of Simulink

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Building on MATLAB (the language of technical computing), Simulink provides a platform for engineers to plan, model, design, simulate, test and implement complex electromechanical, dynamic control, signal processing and communication systems. Simulink-Matlab combination is very useful for developing algorithms, GUI assisted creation of block diagrams and realisation of interactive simulation based designs. The eleven chapters of the book demonstrate the power and capabilities of Simulink to solve engineering problems with varied degree of complexity in the virtual environment.

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