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MODELLING AND CONTROL OF HYSTERESIS-CHARACTERIZED BRAKE PROCESSES

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ABSTRACT - The hysteresis in mechanical subsystems is a well-known effect. It causes a delay of system reaction and reduces the control accuracy. For brake system, the static hysteresis is usually considered in loading-unloading cycle. Usually for simulation studies, hysteresis is expressed as a time delay or as linear functions. The paper introduces investigations of hysteresis-characterized processes in the brake system and relevant dynamic model of this effect for further application to ABS and ESC control algorithms.

The paper is organized as follows: the first part analyzes the literature sources and previous research works on the experimental data of hysteresis-characterized processes under different operational conditions. The hysteresis process is analyzed for the cases: (i) "input pressure vs. brake piston displacement", and (ii) "input pressure vs. realized brake moment in the friction pair". The experimental results demonstrated that the hysteresis taking place in the brake mechanism has a nonlinear asymmetric form.

The second part of the article introduces a mathematical model based on the Bouc-Wen method and its computer realization in Matlab to describe an asymmetrical hysteresis. The method is verified through the experimental results. The analysis of hysteresis influence on vehicle dynamics has been provided for the case of the ABS straight-line braking by considering two variants of the ABS controller: PID control and sliding mode control.

The nonlinear behaviour of hysteresis requires a more complex control law to achieve precise tracking of reference signals. This issue is marked out as future work.

INTRODUCTION

The mechanical hysteresis is a well-known effect attributed to many vehicle systems like engine, brakes, shock absorbers, tyres and other. In the case of a mechanical system under control, the hysteresis causes a delay of the system reaction and reduces the control accuracy by iterative processing. The nonlinear and often asymmetric form of the hysteresis complicates the compensation of these undesirable effects.

A reaction delay caused by the hysteresis is especially critical for the brakes as components of anti-lock braking systems (ABS) or electronic stability control (ESC) due to the adverse influence on safety and stability of vehicle motion. Any investigation of this problem is a complex task because a conventional hydraulic brake system has many hysteresis elements such as vacuum booster, pipelines, brake cylinders, friction pair pad-disk, and others.

The literature analysis shows different methods and approaches to describe the hysteresis of the brake system and its components. Gerdes and Hedrick [1] developed a mathematical model of brake system dynamics using reduced-order models to reproduce static hysteresis in vacuum booster. Augsburg and others investigated relevant tribological processes related to the brake pad behaviour [2] and the brake pedal feel characteristics [3]. Tretsiak and other [4] carried out experimental investigations of hydraulic brake drive and defined the influence of single components (vacuum booster, disk brake mechanism, brake master cylinder, and pipelines) on the hysteresis losses. It should be noted that hysteresis characteristics of vacuum booster and master cylinder are well known [1, 4] and, due to their location in the brake drive, have no influence on control systems like ABS.

The research presented in this paper addresses the hysteresis in wheel brakes from viewpoint of the development of relevant dynamic model for applications to vehicle control systems like ABS. The starting point for the dynamic model is the choice of proper representation of the hysteresis. It can be made through the constant time delay [5, 6], look up tables [7], or linear characteristic (backlash). In addition, a wide range of hysteresis models such as Preisach, Duhem, Bouc-Wen and others are known. For example, Hoseinnezhad and others [8] proposed to use the Maxwell-slip model and implemented real-time calibration technique for the clamp-force model in electromechanical brakes.

This paper introduces a simplified mathematical model of the brake hysteresis especially developed on the basis of the Bouc-Wen method for applications to ABS and ESC algorithms. The subsequent parts of the work introduce the model description, experimental works, and analysis how the hysteresis effect influences controlled brake dynamics.

EXPERIMENTAL INVESTIGATIONS OF HYSTERESIS EFFECT

As applied to the wheel brake operation within a control circuit of ABS or ESC, the brake hysteresis must be considered as a cumulative effect from superposition of several hysteresis-related processes in brake hydraulics, actuators, friction pair, and so on. These processes each need specific test technique to obtain relevant experimental characteristics. However, such thorough elaboration is not necessarily required on the stage of development design of brake controllers. *In this context, the performed experiments were not addressing a separate analysis of pure dynamic hysteresis of the friction pair "brake pad – brake disc".* Within the scope of the presented research, the objectives of experimental works were:

- *Qualitative assessment* of hysteresis loops for the purpose of their subsequent replication during the model development;
- *Comparative analysis* of hysteresis loops with different input/output parameters of the brake system.

The experimental testing was performed for the hydraulic brake system with the wheel brake having a fixed caliper. To obtain dynamic hysteresis characteristics, a series of tests was carried out using controlled law of input pressure on the brake dynamometer of Ilmenau University of Technology. The test rig scheme and general view of the brake assembly are shown in Fig. 1.

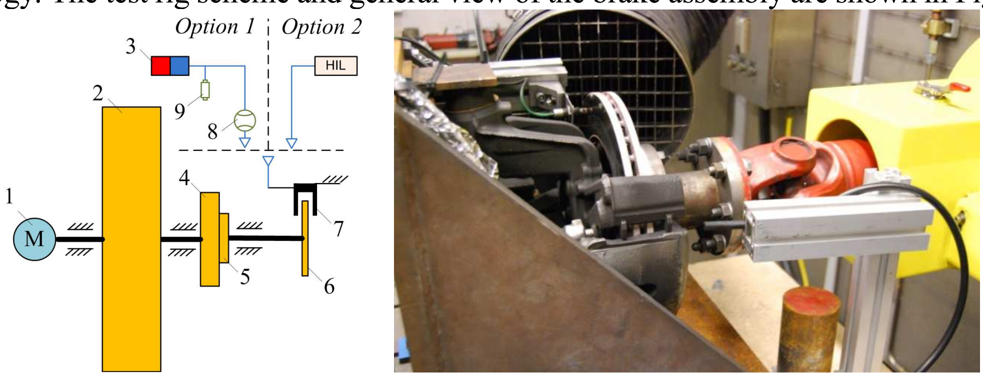

Figure 1: Test rig scheme and brake assembly:
1 – motor, 2 – inertia disk assembly, 3 – air-hydraulic actuating mechanism, 4 – inertia hub assembly, 5 – rotating torque sensor, 6 – brake disk, 7 – brake mechanism, 8 – flow meter, 9 – pressure sensor.

The brake disk is rotated by the drive system with maximum velocity up to 2500 rpm. The controlled (input) pressure can be generated by two ways. *Option 1:* by air-hydraulic actuating mechanism with pressure up to 21 MPa. *Option 2:* using an external hardware-in-the-loop test rig [9]. The measurement system includes a flow meter (accuracy 0.15 cm^3), pressure sensor (accuracy +/- 0.105 MPa) and torque sensor (accuracy 55 Nm). The flow meter is located before the investigated brake mechanism and it measures the amount of volumetric displacement. The pressure transducer is used to measure actual pressure entering to the test brake. The torque sensor is connected with inertia hub assembly. The measured torque signal is acquired by telemetry and it represents the output brake torque.

The parameters of experiments are as follows: The pressure law is a pulse signal from 3.2 up to 5.5 MPa with the frequency of 6 Hz that corresponds to the ABS braking with two-phase algorithm under wet road conditions for the test car (see Table 2); replications – three cases for each test measurement; measurement sampling frequency - 10 kHz; initial velocity – 90 and 120 km/h; contact temperature of friction pair of wheel brake – 100 and 300 ºC; testing modes – "in-stop" mode (free rotation of brake disk, emulation of emergency braking) and "drag" mode (constant rotational velocity, emulation of service braking).

The examples of processed test characteristics together with simulation results are shown in Fig. 3. A general analysis of experimental results allows drawing a number of conclusions that are of relevance for subsequent development of analytical hysteresis model:

- Increase of initial braking velocity and contact temperature reduces the average area of hysteresis;
- The hysteresis loops have an asymmetric form being wider in the region of lower pressure;
- The hysteresis loops with the brake torque as the output parameter have nonlinear character with distinct non-ordered fluctuations of the brake torque and overlapping the pressure phases. This can be also explained by the apparent fluctuation of friction coefficient during braking [10].

The listed qualitative attributes of the obtained experimental hysteresis loops were used together with numerical test results for the analytical model presented in the next chapter.

MATHEMATICAL MODEL OF WHEEL BRAKE UNDER CONTROL ACTION

A mathematical model of hysteresis processes related to the operation of the wheel brake should (i) interpret the nonlinear and asymmetric form of the loops and (ii) take into account the dynamic character of the hysteresis under control actions. At the same time, the model should have a simplified character to be suitable for the embedding into algorithms of vehicle control systems. Taking into account these requirements, the Bouc-Wen method was chosen for the subsequent modelling. The Bouc-Wen hysteresis model is known for huge range of different objects such as magnetorheological dampers, piezoelectric actuators and others [11].

The mechanical model of wheel brake is shown in Fig. 2. In the subsequent discussion, it is assumed that the friction processes are the same for the primary and secondary brake pads. The elements, which have translational motion, are lumped to one equivalent mass m_{eq}. It is represented by the second-order differential equation:

$$m_{eq}\ddot{x} + c_1\dot{x} + \Phi(\dot{x}, x, z_1) = u(t) \tag{1}$$

x is the displacement of brake piston with the unit m; m_{eq} is the lumped mass of elements with translational motion (piston, pad, etc), kg; c_1 is the translational damping coefficient, $Nm^{-1}s^{-1}$; $u(t)$ is the control input force is equal to $u(t)=p(t)\frac{\pi d^2}{4}$; $p(t)$ is the input pressure, MPa; d is the piston/cylinder diameter, m.

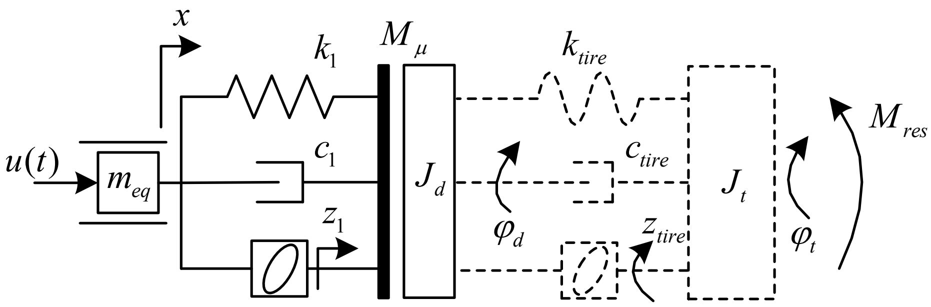

Figure 2: Schematic model of wheel brake under control action

The contact restoring force Φ(x), which has nonlinear and asymmetric behaviour in unloading-loading cycle, is described by the modified version of the Bouc-Wen model [11]:

$$\Phi(x)=k_1\left(x+z_1\right)$$

$$\dot{z}_1=\begin{cases}\alpha_1\dot{x}-\beta_1\left|\dot{x}\right|z_1\left|z_1\right|^{n_1-1}-\gamma_1\dot{x}\left|z_1\right|^{n_1}, \dot{x}\geq 0\\ \alpha_2\dot{x}-\beta_2\left|\dot{x}\right|z_1\left|z_1\right|^{n_2-1}-\gamma_2\dot{x}\left|z_1\right|^{n_2}, \dot{x}<0\end{cases} \tag{2}$$

k_1 is the translational contact stiffness, Nm^{-1}; $\alpha_1,\beta_1,\gamma_1,n_1,\alpha_2,\beta_2,\gamma_2,n_2$ are the coefficients describing the hysteresis in translational direction.

The contact friction torque for the brake mechanism with fixed caliper is given by

$$M_{br}=M_\mu=2\mu(n\nu,\eta)\Phi(x)r_{fr} \tag{3}$$

r_{fr} is the effective radius of friction, m; μ is the dynamic friction coefficient depending on the actual normal load, the actual tangential velocity and the temperature in the frictional contact. For this investigation, it is represented by the system of differential equations developed by Ostermeyer [12]:

$$\dot{\mu}=-a'\left(\left(n\nu+\varepsilon\right)\mu-\eta\right)$$

$$\dot{\eta}=-c'\left(\eta-\eta_0-\alpha' n\nu\right)-\gamma'\left(\eta^4-\eta_0^4\right) \tag{4}$$

To involve the tyre hysteresis (dashed part of Fig. 2), the system of equations can be expanded. In this case, the rotational dynamics is represented as:

$$J_d\frac{d^2\varphi_d}{dt}=M_\mu-c_{tyre}\Delta\dot{\varphi}-M\left(\Delta\dot{\varphi},\Delta\varphi,z_{tyre}\right)$$

$$J_t\frac{d^2\varphi_t}{dt}=c_{tyre}\Delta\dot{\varphi}+M\left(\Delta\dot{\varphi},\Delta\varphi,z_{tyre}\right)+M_{res} \tag{5}$$

$$\Delta\varphi=\varphi_d-\varphi_t$$

φ_d is the angular displacement of brake disk, rad; φ_t is the angular displacement of tyre, rad; J_d is the moment of inertia of brake disk, kgm^2; J_t is the moment of inertia of tyre*, kgm^2; c_{tyre} is the rotational damping coefficient, $kgm^2s^{-1}rad^{-1}$; M_{res} is the resistance torque from the road, Nm.
* At test rig experiments, the resistance torque is equal zero, and additional moment of inertia J_v, which is equivalent to vehicle mass, is attached to the required inertia.

The output torque (measured tyre torque) is equal to:

$$M_{tyre} = M\left(\Delta\dot{\varphi}, \Delta\varphi, z_{tyre}\right) = k_{tyre}\left(\Delta\varphi + z_{tyre}\right)$$
$$\dot{z}_{tyre} = \alpha\Delta\dot{\varphi} - \beta\left|\Delta\dot{\varphi}\right| z_{tyre}\left|z_{tyre}\right|^{n-1} - \gamma\Delta\dot{\varphi}\left|z_{tyre}\right|^{n} \tag{6}$$

k_{tyre} is the rotational contact stiffness, Nmrad^{-1}; α,β,γ,n are the coefficients describing the hysteresis in rotational direction.

Table 1. Model Parameters

Parameters of proposed model	Value	Parameters of proposed model	Value
mass of piston	0.18	n_1	0.75
mass of pad	0.24	$\alpha_1=\alpha_2$	2.0
piston/cylinder diameter	0.059	β_1	0.2
effective radius	0.13	γ_1	0.5
inertia moment of brake disk	0.11	n_2	0.65
actual inertia	47.46	β_2	4.2
required inertia	61.79	γ_2	0.7
static coefficient of friction	0.33	a'	0.1
translational contact stiffness, 10^7	1.35	c'	0.3
translational damping, 10^5	2.0	α'	0.8
initial velocity, rads^{-1}	90.42	γ'	0.35
initial disk temperature, ^{0}C	96	η_0	0.1

For the simulation purposes, the model was parameterized for the test specimen of wheel brake used during the experiments described above, Table 1. The results of model simulation and experiments are displayed in Fig. 3. The input pressure law (the same for experiment and simulation) is shown in Fig. 3a. The brake piston stroke (lumped mass displacement) vs. input brake pressure and output brake torque vs. input brake pressure are shown in Fig. 3b and 3c. The temperature is given in Fig. 3d, following rotational velocity of brake disk, Fig.3e, and dynamic friction coefficient, Fig. 3f. The influence of model parameter α (parameter α_1) on the hysteresis characteristic under constant friction coefficient, its area and width is shown in Fig. 4. The next chapter introduce the implementation of the developed hysteresis model for the quarter-model of the vehicle with the ABS control.

QUARTER-MODEL OF VEHICLE AND CONTROL ALGORITHMS

The quarter-model of vehicle with the fixed vertical force is considered to minimize influence of load transfer, changing of wheel radius and other factors on vehicle dynamics. The mathematical description of longitudinal dynamics is given below:

$$m_a\frac{dV_x}{dt} = -F_x$$
$$J_t\frac{d\omega_w}{dt} = -M_{br} + F_x r_w \tag{7}$$

m_a is the vehicle mass, kg; F_x is the longitudinal force, N; r_w is the wheel radius, m; V_x is the longitudinal velocity, m/s; ω_w is the angular wheel velocity, rad/s.
The longitudinal force is calculated by [13]:

$$F_x = F_z\left(k_{c1}\left(1 - e^{-k_{c2}\cdot s}\right) + k_{c3}s\right)sign(s) \tag{8}$$

F_z is the vertical force (in this model it is assumed to be constant), N; k_{c1}, k_{c2} and k_{c3} are the empirical parameters; s is the slip coefficient.

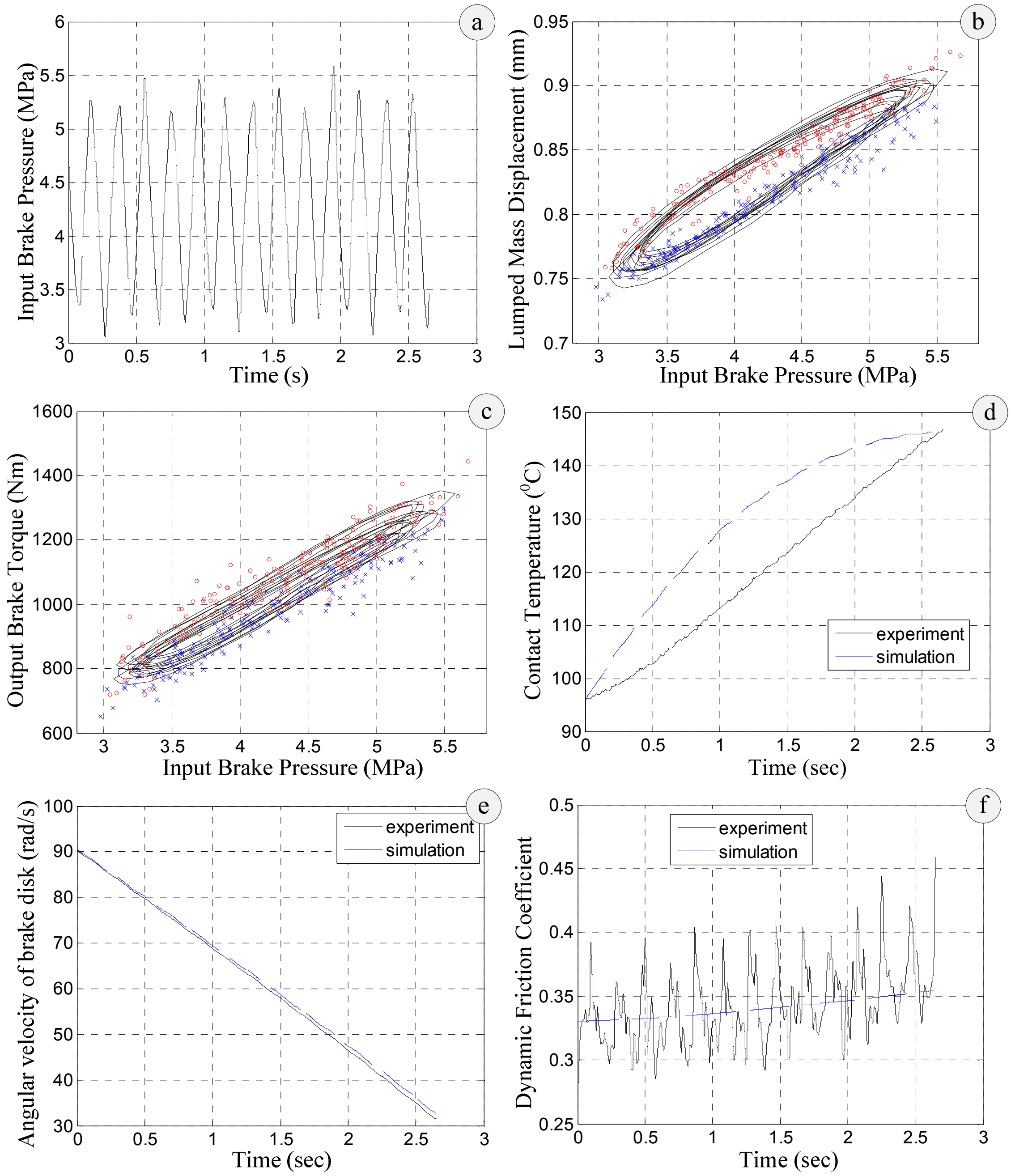

Figure 3: Experimental and simulation results:
x(blue) – phase of pressure increasing; o(red) – phase of pressure decreasing

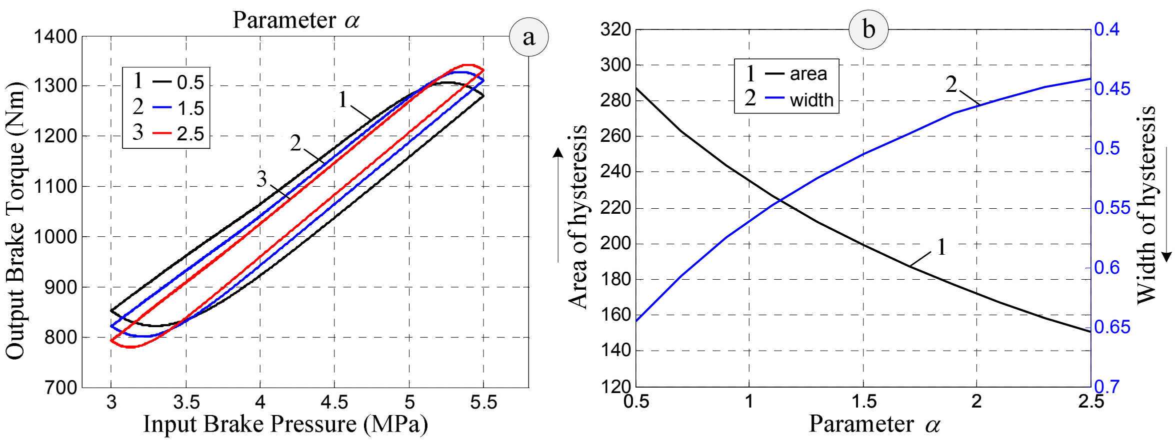

Figure 4: Characteristic and parameters of hysteresis loop under constant friction coefficient

The brake pressure is determined by the control algorithm and it is calculated ignoring brake actuator dynamics (for electro-hydraulic brake system the solenoid valve dynamics, the fluid behaviour and the compressibility in the hydraulic control unit, pipelines and others). It is represented as:

$$p = k_{pressure} \int_{0}^{p_{\max}} r dt \tag{9}$$

$k_{pressure}$ is the linear pressure gain, r is the control signal.

Further two control algorithms are considered: the classical PID control, which provides two phases of regulation (increase/decrease of the pressure), and sliding mode control (SMC). The applied SMC is based on the methodology from [14]. The parameters of simulation model are shown in Table 2. The next chapter presents the analysis of the simulation results.

INFLUENCE OF HYSTERESIS PARAMETERS ON VEHICLE DYNAMICS

The simulation for PID control was carried out for three types of road surfaces (dry and wet asphalt, snow) with different initial velocities. The parameters of hysteresis model were chosen as for the initial experimental test data from Table 1. Table 3 illustrates the obtained results, where $\overline{A}$ and $\overline{w}$ are the average area and width of hysteresis during braking; s_{br} is the brake distance, m; t_{br} is the stopping time, s; j is the average braking deceleration, m/s^2; r_{en} is the ratio between energy consumed in wheel slip and input wheel energy; f_{br} is the brake frequency, Hz. The type of hysteresis shows how the effect of hysteresis is taken into account: “zero” means that hysteresis is neglected, “model” means that the application of abovementioned model; “delay” means that the constant transfer delay for brake torque is applied; “linear” means that the hysteresis characteristic is linear. The parameters of delay and backlash were chosen such that average area of hysteresis is the same as for “model” type for wet asphalt.

Table 2. Parameters of simulation model

<table>
<tr><td colspan="4">Vehicle mass, kg</td><td colspan="2">480</td><td colspan="6">Road (dry asphalt)</td></tr>
<tr><td colspan="4">Wheel radius, m</td><td colspan="2">0.36</td><td colspan="2">k_{c1}</td><td colspan="2">k_{c2}</td><td colspan="2">k_{c3}</td></tr>
<tr><td colspan="4">Linear pressure gain</td><td colspan="2">50</td><td colspan="2">0.875</td><td colspan="2">34.638</td><td colspan="2">0.143</td></tr>
<tr><td colspan="6">Road (wet asphalt – 0.1 mm)</td><td colspan="6">Road (snow)</td></tr>
<tr><td colspan="2">k_{c1}</td><td colspan="2">k_{c2}</td><td colspan="2">k_{c3}</td><td colspan="2">k_{c1}</td><td colspan="2">k_{c2}</td><td colspan="2">k_{c3}</td></tr>
<tr><td colspan="2">0.58</td><td colspan="2">53.81</td><td colspan="2">0.1</td><td colspan="2">0.214</td><td colspan="2">110.118</td><td colspan="2">0.022</td></tr>
<tr><td colspan="5">PID controller</td><td>K_p</td><td>14.5</td><td>K_i</td><td>22.7</td><td>K_d</td><td colspan="2">0.02</td></tr>
</table>

The hysteresis has more influence on brake distance and time under conditions of low velocities and slippery roads. This is mainly due to the fact that the ABS frequency is increased and the total area of hysteresis is enlarged. Nevertheless, the average area and width are decreased as compared with the braking on the road with higher friction coefficient. For the dry asphalt, the difference between the “zero” and “model” cases is more than 5% and it is up to 8.5% for the slippery road.

The decrease of the hysteresis width by the parameter α (Fig. 5) causes simultaneously the reduction of brake distance and growth of the frequency. The effect has strong impact on the braking on a slippery road. For dry roads, the changing of width has indicated small influence on the brake distance and control frequency.

One hypothesis is that the hysteresis can be controlled by actuator frequency which is changed by control parameters. To check the validity of this hypothesis, the simulation with fixed parameters of hysteresis and variation of control parameters was carried out. Moreover, to check how the consideration of hysteresis characteristics influences on the vehicle braking, the above-mentioned tests for both control strategies were provided for one type of road (ice).

Table 3. Simulation results for PID control

Road	V_x	Hysteresis Type	$\bar{A}$	$\bar{w}$	s_{br}	t_{br}	j	r_{en}	f_{br}
Dry asphalt	90	zero	0	0	38.38	2.76	-8.18	0.173	8.3
		model	297	0.517	40.50	2.92	-7.75	0.181	6.9
		delay	311	0.601	40.53	2.92	-7.74	0.181	6.9
		linear	230	0.502	40.08	2.88	-7.83	0.179	6.6
Wet asphalt	75	zero	0	0	39.36	3.36	-5.49	0.121	8.9
		model	185	0.450	41.88	3.57	-5.16	0.127	7.6
		delay	185	0.502	41.90	3.58	-5.15	0.127	7.6
		linear	185	0.505	41.72	3.55	-5.18	0.127	6.8
Snow	45	zero	0	0	36.46	4.88	-2.07	0.062	11.1
		model	59	0.308	39.84	5.34	-1.89	0.067	9.0
		delay	59	0.305	39.79	5.33	-1.89	0.067	9.0
		linear	101	0.502	41.11	5.48	-1.84	0.070	7.1

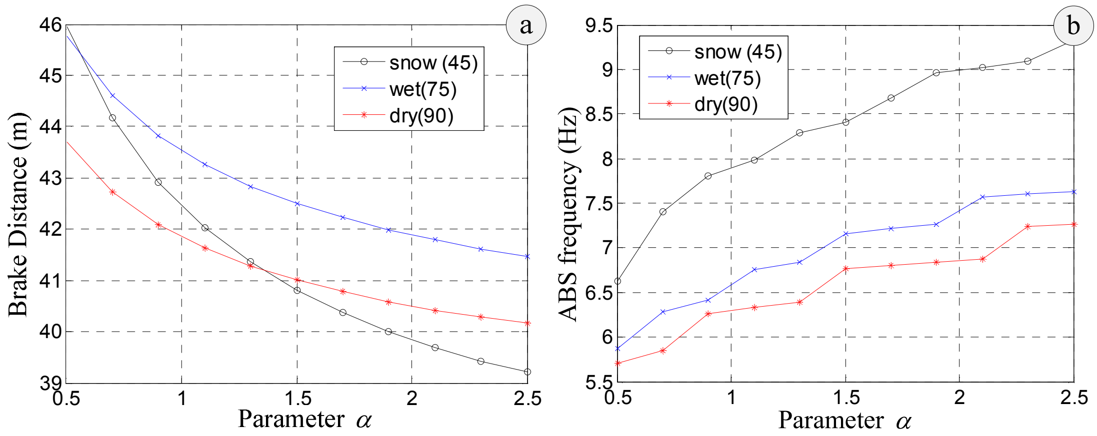

Figure 5: (a) Brake distance and (b) ABS frequency vs. parameter of hysteresis for the PID control

The minimum of average area of hysteresis for the PID control is reached under low proportional gains; however, the brake distance is increased in this case (Fig. 6a). The reason is that the settling time of control process is large. From the other side, high proportional gain induces a growth of average area of hysteresis by overshoot in control, which achieves significant value and starts to affect on the system dynamics. One of the possible ways to minimize this effect is to increase a derivative coefficient.

The neglect of hysteresis causes monotonic decreasing of the brake distance under increasing of proportional gain, thereby, the distorting of real system behaviour. It should be noted that for the SMC controller with saturation, the resulting brake torque is smooth without any overshooting compared with PID control. The instances of brake force vs. time for PID (K_p=17) and SMC (η=290) control are shown in Fig.7. Therefore, hysteresis does not have some high impact on vehicle braking dynamics under sliding mode control (Fig.6b).

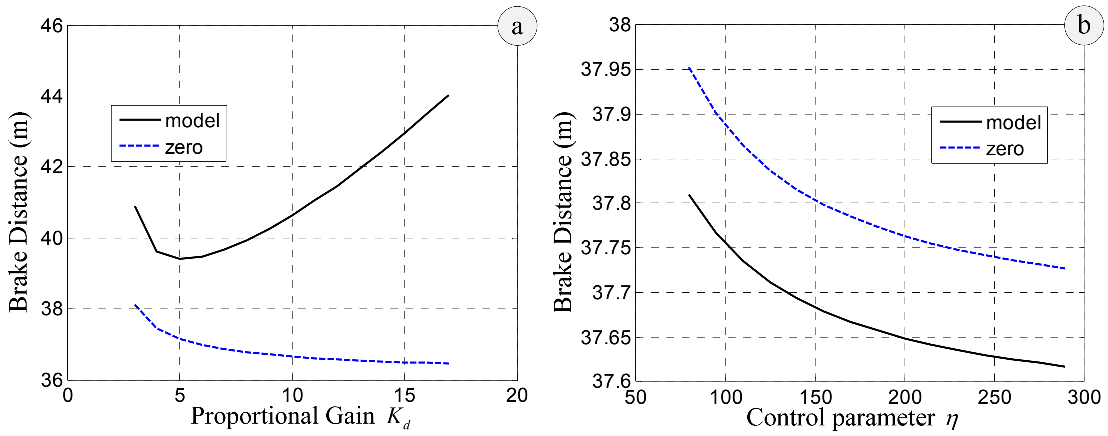

Figure 6: Brake distance vs. control parameter for the (a) PID and (b) SMC control

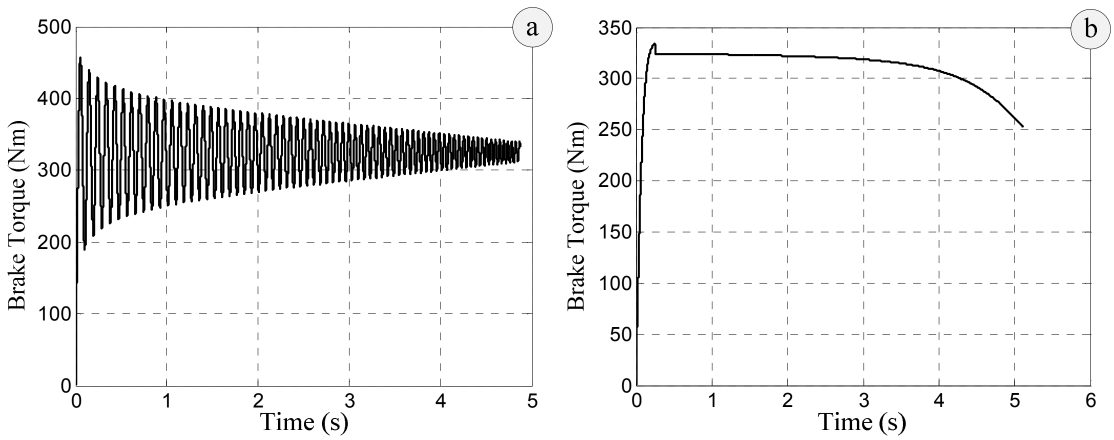

Figure 7: Brake force vs. times for the (a) PID and (b) SMC control (instance)

CONCLUSIONS

Experimental investigations of hysteresis processes of the wheel brake under control action confirm the nonlinear and asymmetric form of hysteresis. The computational model of the wheel brakes with hysteresis elements has been developed based on the Bouc-Wen method and verified with the obtained experimental results.

Using the developed model, the analysis of hysteresis influence on vehicle dynamics during the ABS straight-line braking has been provided taking into account two variants of the ABS controller: PID control and sliding mode control. The influence of variation of hysteresis parameters on the main vehicle brake characteristics such as stopping distance, time of braking, average acceleration are shown. The main recommendations of the presented research are:

1. Experimental analysis using dynamometer test rig allows qualitative assessment of the hysteresis processes in the wheel brake under control action, in particular, the nonlinear and asymmetric character of hysteresis loops.
2. Hysteresis-characterized processes influences complex vehicle dynamic maneuvers, such as the braking on slippery roads, at low velocities or by the sharp changing of tyre/road friction coefficient.
3. The analysis shows that the effect of hysteresis on stopping distance does not exceed 10% in most critical cases. Ignoring this fact can cause incorrect ABS control.

4. To simplify the initial procedures of the ABS and brake design, the influence of the brake hysteresis on vehicle dynamics can be represented by emulating the hysteresis as a signal delay or linear function.
5. Influence of brake hysteresis processes on vehicle dynamics can be controlled and minimized by the implementation of the sliding mode control that improves the adaptive properties of the ABS.

The proposed model can be used for the computational simulation of brake system to introduce the hysteresis effect on the process of brake torque development. The nonlinear behaviour of hysteresis requires a more complex control law to achieve the precise tracking of reference signals. The authors have marked the latter issue as the direction of future works.

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REFERENCES

(1) J. C. Gerdes and J. K. Hedrick, "Brake System Modeling for Simulation and Control," Journal of Dynamic Systems, Measurement, and Control, vol. 121, pp. 496-503, 1999.

(2) K. Augsburg et al., "Comparison Between Different Investigation Methods of Quasi-Static and Dynamic Brake Pad Behaviour", SAE Technical Paper Series, no. 2003-01-3340.

(3) A. Hildebrandt et. al., "Nonlinear control design for implementation of specific pedal feeling in brake-by-wire car design concepts", Proc. of American Control Conference, vol. 2, pp. 1464-1468, 2004.

(4) D. Tretsiak, et al., "Research in hydraulic brake components and operational factors influencing the hysteresis losses," IMechE Part D: J. Automobile Engineering , vol. 222, pp. 1633-1645, 2008.

(5) T. Jianmin and W. Baohua, "Study of the dynamic threshold and control logic on anti-lock brake system (ABS)," Vehicle Electronics Conference, pp. 82-85 vol. 1, 1999

(6) M. S. A. Hardy and D. Cebon, "Investigation of anti-lock braking strategies for heavy goods vehicles," IMechE Part D: J. Automobile Engineering, vol. 209, pp. 263-271, 1995.

(7) H. Raza, et al., "Modeling and control design for a computer-controlled brake system," Control Systems Technology, IEEE Transactions on, vol. 5, pp. 279-296, 1997.

(8) R. Hoseinnezhad, et al., "Real-Time Clamp Force Measurement in Electromechanical Brake Calipers," Vehicular Technology, IEEE Transactions on, vol. 57, pp. 770-777, 2008.

(9) K. Augsburg et al., "Investigation of Brake Control Using Test Rig-in-the-Loop Technique", SAE Technical Paper Series, no. 2011-01-2372.

(10) M. Eriksson and S. Jacobson, "Friction behaviour and squeal generation of disc brakes at low speeds," IMechE Part D: J. Automobile Engineering, vol. 215, pp. 1245-1256, 2001.

(11) Y. Q. Ni, et al., "Modelling and identification of a wire-cable vibration isolator via a cyclic loading test Part 1: And experiment and model development," IMechE Part I: J. Systems and Control Engineering, vol. 213, pp. 163-171, 1999.

(12) Ostermeyer, G. (2001). Friction and wear of brake systems. Forschung im Ingenieurwesen, 66(6), 267-272.

(13) U. Kiencke and L. Nielsen, "Automotive Control Systems for Engine, Driveline and Vehicle", Springer, 2005

(14) S. Zheng, et al., "Controller design for vehicle stability enhancement. Control engineering practice", 14(12), 1413-1421, 2006.