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SIMULATION AND ROBUST CONTOL OF ANTILOCK BRAKING SYSTEM ABS

SIMULACE A ROBUSTNÍ ŘÍZENÍ ANTIBLOKOVACÍHO SYSTÉMU ABS

Abstract

This paper deals with simulation and robust control of Antilock Braking System ABS. The briefly are described the main parts of ABS hydraulic system and control algorithm of ABS. Hydraulic system described here is BOSCH ABS 5.x series. The goal of ABS system is vehicle stability and vehicle steering response when braking. If during the braking occurred slip at one or more wheels from any reason, ABS evaluates this by "brake slip" controller. At this moment ABS is trying to use maximal limits of adhesion between tire and road. It means that is necessary control the differences between braking torque M_B and friction torque M_k, which reacts to the wheel via friction reaction tire-road surface. This is realized through the solenoid valves, which are controls (triggered) by U_{valve} on the base of PID controller described further in chapter 4. Presented concept is more or less standard for most of the existing ABS systems. The issue should be applied concept of robust ABS control algorithm, which is specific for every type of ABS.

Abstrakt

Příspěvek se zabývá simulací a robustním řízením antiblokovacího systému ABS. Jsou zde stručně popsány hlavní části hydraulického systému ABS a návrh regulačního algoritmu ABS. Hydraulický systém popsaný v tomto příspěvku je BOSCH ABS 5. x série. Hlavním úkolem ABS systému je udržení stability a řiditelnosti vozu během brzdění. Dojde-li při brzdění vozidla z jakéhokoliv důvodu k zablokování některého z jeho kol, vyhodnotí ABS „brake slip" regulátor na kole skluz. V tomto okamžiku se snaží ABS využít maximální hodnoty meze přilnavosti mezi pneumatikou a vozovkou. Znamená to, že je třeba udržet co nejmenší rozdíl mezi brzdným momentem M_B, a momentem tření M_k, který působí zpětně na kolo přes třecí dvojici pneumatika-vozovka. Regulace brzdného momentu na kole je realizována za pomocí elektro-magnetických ventilů řízeným napětím U_{valve} na základě vyhodnocení regulátoru PID, popsaného v kapitole 4. Prezentovaný koncept ABS systému je více méně standardem současných ABS systémů. Hlavním výstupem by měl být samotný koncept robustního ABS algoritmu, který je jedinečný pro každý typ ABS.

1 INTRODUCTION

Continuing developments in passenger car brake systems led to powerful and reliable systems capable of furnishing optimum retardation from high rates of speed. Under normal operating conditions these systems can provide fast and effective braking for the vehicle. But under more

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critical conditions such as wet or slippery road surfaces, driver panic reaction or mistakes committed by other drivers and pedestrians can lead to lock of the wheels during braking. The result is loss vehicle steering response as the vehicle loses traction and/or loss of the vehicle stability. This is the type of situation when ABS comes into play. The ABS system recognized incipient locking at one or more wheels in time to react by inhibiting further increases or initiating decreases in braking pressure. Result is vehicle steering response and vehicle stability on the road.

2 ABS CONTROL LOOP

The ABS control system consists of two main parts. Hydraulic system, which is described in chapter 3, and ABS controller, described in chapter 4. Controlled system consists of vehicle with wheel brakes, wheels and friction between tires and road surface. Controlled variables are wheel speed and the data derived from it like deceleration at the wheels, vehicle reference speed and brake slip. Controller consists of wheel speed sensors and ABS control unit. Disturbance factors are road-surface conditions, brake conditions and vehicle and tires conditions (tire pressure or tread depth). Reference input variable is pressure applied by driver to the brake pedal. Manipulated variable is brake pressure.

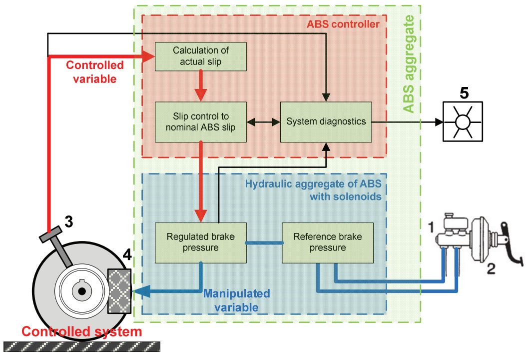

Fig. **1** Control system of ABS; 1-tandem master brake cylinder, 2-braking booster, 3-wheel speed sensor, 4-wheel brake cylinder, 5-ABS indicator lamp.

3 ABS HYDRAULIC SYSTEM

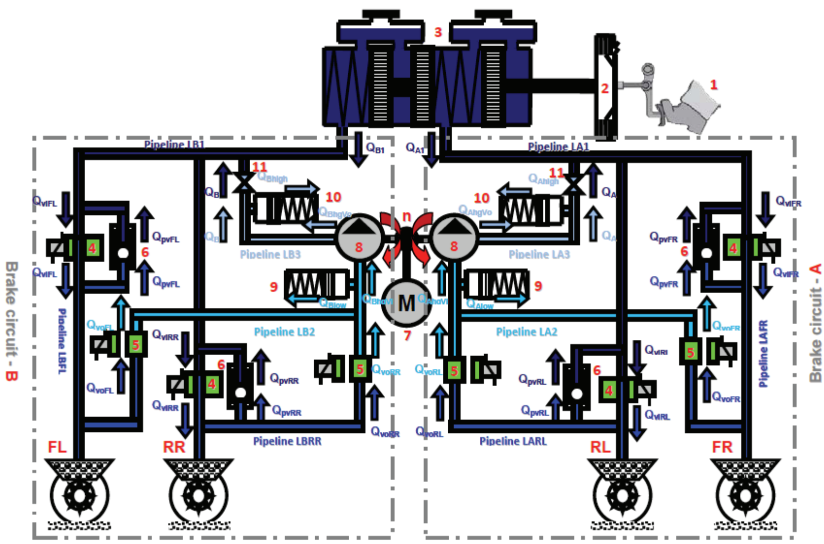

Fig. 2 ABS hydraulic system diagonal construction and 2/2 valve constructions.

The hydraulic system of ABS for X-variant braking force distribution and 2/2 solenoid valves consist of tandem master brake cylinder (3) with braking booster (2). Four input solenoid valves (4) and four output solenoid valves (5). To reduce and protect the pressure at the wheel brake cylinder is there also bleeder (6) as a part of each input solenoid valve. As "storage" is present low pressure hydraulic accumulator (9) for each of circuit. As a pulse and noise absorber is present high pressure accumulator (10). For backward delivery is used for each circuit pressure pump (8) driven by DC motor (7). For more damping effect is used also orifice in both circuits.

Braking starts by driver via brake pedal. The brake booster is an element, which amplifies the foot pressure applied by driver during braking. The functionality of braking booster is described by equation:

$$F_{Bb} = S_{diaphragm} \cdot p_{Bb} \tag{3.1}$$

Braking booster (2) force made by atmospheric pressure, which press to the surface of diaphragm applies to the tandem master brake cylinder. Pressing piston of tandem master brake cylinder is initiated flow of brake fluid from the cylinder to the pipeline. Next equations describe tandem master brake cylinder (3) with floating piston with two separate brake circuits A and B:

$$\ddot{x}_A(t) = \frac{1}{m_A}\left[F_{Bb} - S_{Ap} \cdot p_A(t) - k_A \cdot \left(x_A(t) - x_B(t)\right) - b_A \cdot \dot{x}_A(t)\right] \tag{3.2}$$

$$Q_A = S_{Av} \cdot \dot{x}_A(t) \tag{3.3}$$

$$\ddot{x}_B(t) = \frac{1}{m_B}\left[S_{Ap} \cdot p_A(t) + k_A \cdot \left(x_A(t) - x_B(t)\right) - S_{Bp} \cdot p_B(t) - k_B \cdot x_B(t) - b_B \cdot \dot{x}_B(t)\right] \tag{3.4}$$

$$Q_B = S_{Bv} \cdot \dot{x}_B(t) \tag{3.5}$$

The discharges Q_A and Q_B (continuity relation) in close pipeline are source of pressure, which is made by pressing the brake fluid. The main characteristic of pipeline is hydraulic capacity:

$$p_p = \frac{K_p}{V_p}\int\left[Q_{input} - Q_{output}\right]dt + p_{p0} \tag{3.6}$$

The parameter K_p is a compressibility of brake fluid and V_p is a volume of pipeline. Q_{input} is influent flow and Q_{output} is effluent flow of pipeline. Input (4) and output solenoid (5) valves are responsible for regulating of brake pressure in the hydraulic system. The equation of flow through the solenoid valve is representing by Bernoulli equation and continuity relation:

$$Q_v = S_v \cdot v_{v\_out} = \left(x_v(t) \pm x_{v0}\right) \cdot \frac{\pi \cdot D_v^2}{4} \cdot \sqrt{\frac{2}{\rho}} \cdot \sqrt{\left|p_{v\_in} - p_{v\_out}\right|} \cdot sign(\Delta p) \tag{3.7}$$

Dynamic of the slider is represented by proportional characteristic of second order with dumping coefficient:

$$T_v^2 \frac{d^2 x_v}{dt^2} + 2\xi_v T_v \frac{dx_v}{dt} + x_v = k_v u_v \tag{3.8}$$

Also check valve (6) is an important part of the ABS hydraulic system. A check valve is installed parallel to the input valve (4). When the brake is released the check valve opens supplementary, large diameter passage leading from the wheel brake cylinder to the brake master cylinder. It also ensures that it remains possible to release the brake in the event of a defect on the input valve. Equation of the flow through the valve is also representing by Bernoulli equation and continuity relation:

$$Q_{cv} = S_{cv} \cdot v_{out} = S_{cv} \cdot \sqrt{\frac{2}{\rho}} \cdot \sqrt{p_{cv\_in} - p_{cv\_out}} = \frac{\pi \cdot D_{cv}^2}{4} \cdot \left(x_{cv}(t) \pm x_{cv0}\right) \cdot \sqrt{\frac{2}{\rho}} \cdot \sqrt{p_{cv\_in} - p_{cv\_out}} \quad (3.9)$$

Simulated check valve is single stage valve. Control element is a check ball or valve plug controlled direct by the adjusted spring and input pressure. Motion equation of the safety valve is:

$$m_{cv} \cdot \ddot{x}_{cv}(t) = S_{cv} \cdot p_{cv\_in}(t) - S_{cv} \cdot p_{cv\_out}(t) - S_{cv} \cdot p_{cv\_\max} - b_{pv} \cdot \dot{x}_{pv}(t) - k_{pv} \cdot x_{pv}(t) \quad (3.10)$$

The hydraulic accumulators (9, 10) are used such a chamber and damper. They eliminate also the noise during the backward delivery of the brake fluid to the tandem master brake cylinder. The flow and motion equations of the piston are:

$$Q_{ha} = S_{ha_v} \cdot \dot{x}_{ha}(t) \quad (3.11)$$

$$\ddot{x}_{ha}(t) = \frac{1}{m_{ha}} \left[S_{ha_p} \cdot p_{ha} - b_{ha} \cdot \dot{x}_{ha}(t) - k_{ha} \cdot x_{ha}(t) \right] \quad (3.12)$$

The last parts responsible for backward delivery of brake fluid to the tandem master brake cylinder, during the phase of pressure release are pressure pump driven by DC motor. Piston position of the pressure pump is:

$$x_{hg}(t) = r_{hg} + e_{hg} \cdot \cos(\omega_m \cdot t) \quad (3.13)$$

Piston and eccentric cam haven't any fix connection. The load torque of pressure pump is based on loading from pressure force F_p during backward brake fluid delivery:

$$M_{hg} = e_{hg} \cdot \sin(\omega_m \cdot t) \cdot F_p \quad (3.14)$$

Electrical equation for DC motor with self actuating and permanent magnets is:

$$\frac{di_a}{dt} = \frac{1}{L_a} \cdot \left[u_a - \xi \cdot \omega_m - R_a \cdot i_a \right] \quad (3.15)$$

Equation for mechanical part of DC motor is:

$$\frac{d\omega_m}{dt} = \frac{1}{J_m} \left[\xi \cdot i_a - M_{hg} \right] \quad (3.16)$$

All these equations and something more is used to design mathematical model of ABS hydraulic system. Some of any parts were simplified, for example the dynamic equation of solenoid valve slider was substituted by transfer function. Same situation is used for braking booster. There is used pressure characteristic of braking booster.

4 ABS CONTROLLER

The goal of ABS controller is quickly response to slip, which can occurs at the wheels when braking. For the slip determination at the wheel is necessary a few information, but the main information is wheel speed v_{wheel} and vehicle reference speed v_x. Next equation describe wheel speed calculated from instantaneous braking force F_B and the tire rigidity C_{tire}.

$$v_{wheelfree} = v_{wheel} \cdot \frac{C_{tire}}{C_{tire} - \frac{F_B}{F_N}} \quad (4.1)$$

Using the yaw velocity $\dot{\psi}$ the steering angle δ and lateral velocity v_y together with the vehicle geometry, the (free rolling) wheel speed is transformed to the centre of gravity v_x.

The actual slip is calculated:

$$\lambda = 1 - \frac{v_{wheel}}{v_{wheelfree}} \tag{4.2}$$

The idea how control the braking torque to use the maximum of adhesion between tire and road surface is use stationary braking force F_{BS} and PID control low. Nominal torque at the wheel can be calculated as a function of nominal and actual slip deviation:

$$M_{WheelNo} = F_{BS} \cdot r_d + K_p(\lambda_{No} - \lambda) r_d + K_D\left(\frac{d}{dt} v_{wheel} - \frac{d}{dt} v_{wheelfree}\right) \frac{j_{wheel}}{r_d} + K_I \cdot C_p \int (\lambda_{No} - \lambda) dt \tag{4.3}$$

When the wheel nominal braking torque is actually known then is used inverse hydraulic model for determination of braking pressure and valve-triggering mode U_{valve}. The applied braking pressure at the wheels is adjusted by the brake hydraulic and actual valve-triggering mode.

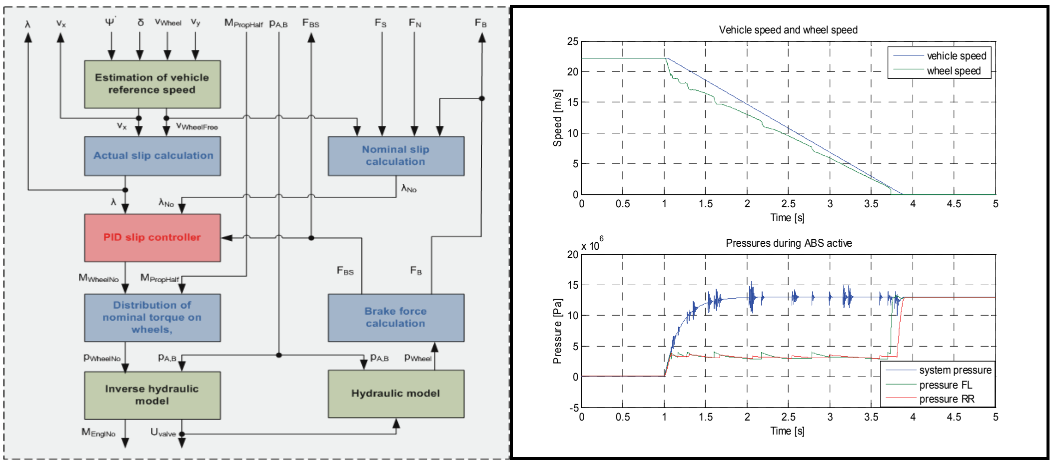

Fig. 3 Block diagram of ABS brake slip controller [3], simulation results for ABS tire slip algorithm.

5 SUMMARY

In this paper is introduced mathematical model of BOSCH ABS system 5.x with all hydraulic components and ABS control algorithm. In first part is briefly describing ABS control loop system with all controlled values. This concept is common for all currently ABS systems in automotive industry. Next chapter ABS hydraulic system describes mathematical equations all main parts of ABS hydraulic system. System was simulated in program environment MATLAB-Simulink. The variables and values for simulation system are matching with commercial vehicle class A1. Simulation step size is 1[ms]. In last chapter is description of ABS brake slip control algorithm. This control strategy includes also engine drag-torque control MSR. The ABS model described in this paper is extendable to traction control systems (TCS) and yaw moment stability as well. The paper is supported by grant project of MŠMT ČR SPECIFIC RESEARCH No. 2101/352.

Nomenclature:

Variables and Subscripts:

ABS	Antilock Braking System	Q_{pv}	Flow of safety valve
b_{ha}	Damping factor of hydraulic accumulator	Q_v	Flow of solenoid valve
b_{pv}	Damping factor of safety valve	R_a	Resistor at rotor of DC motor
C_P	Brake-torque ratio	r_d	Dynamic radius of the wheel
C_{tire}	Tire rigidity	r_{hg}	Radius of hydraulic generator cam

D_{pv}	Diameter of safety valve flow	$S_{Ap,Bp}$	Surface of TMBC piston
D_v	Diameter of solenoid valve flow	$S_{Av,Bv}$	Surface of TMBC piston rod
e_{hg}	Eccentricity of hydraulic generator cam	S_{ha}	Surface of hydraulic accumulator
F_B	Brake force at the wheel	S_{pv}	Surface of safety valve
F_{Bb}	Braking booster force	S_v	Surface of solenoid valve flow
F_{BS}	Filtered braking force	t	Time
F_N	Normal force at wheel	TCS	Traction Control System
F_p	Force from pressure	TMBC	Tandem master brake cylinder
i_a	Current of DC motor	T_v	Time constant of solenoid valve
J_m	Moment of inertia of DC motor	u_a	Voltage of DC motor
J_{Wheel}	Moment of inertia of the wheel	u_v	Voltage of solenoid valve
$k_{A,B}$	Spring rate coefficient of TMBC	U_{valve}	Valve-triggering mode
k_{ha}	Spring rate coefficient of hydraulic accum.	V_p	Capacity of pipeline
K_p	Module of brake fluid compressibility	v_{pv}	Flow speed of safety valve
K_P, K_D, K_I	Controller gains	v_v	Flow speed of solenoid valve
k_{pv}	Coefficient of spring rate	v_{Wheel}	Wheel speed
k_v	Gain of solenoid valve	$v_{WheelFree}$	Free wheel speed
L_a	Coil at rotor of DC motor	v_x	Longitudinal velocity
$m_{A,B}$	Weight of TMBC piston	v_y	Lateral velocity
m_{ha}	Weight of hydraulic accumulator piston	x_{ha}	Position of hydraulic accumulator
M_{hg}	Torque of hydraulic generator	x_{hg}	Position of hydraulic generator piston
m_{pv}	Weight of safety valve piston	x_{pv}	Safety valve position
MSR	Torque system regulation	x_v	Solenoid valve position
$M_{WheelNo}$	Nominal brake torque at the wheel	δ	Steering angle
$p_{A,B}$	Pressure in chamber A and B	λ	Slip
p_{Bb}	Pressure of braking booster	λ_{No}	Nominal slip
p_p	Pipeline pressure	ξ	DC motor Constant
p_{pv}	Pressure at safety valve	ξ_v	Damping factor of solenoid valve
p_v	Pressure of solenoid valve	π	Circular constant
Q	Flow	ρ	Density of brake fluid
$Q_{A,B}$	Flow in chamber A and B of TMBC	ω_{hg}	Angular velocity of hydraulic generator
Q_{ha}	Flow of safety valve	$\dot{\psi}$	Yaw velocity of vehicle

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