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SLIP CONTROL AT SMALL SLIP VALUES FOR ROAD VEHICLE BRAKE SYSTEMS

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Abstract

A control algorithm is presented here, that controls the brake forces of a two-axle road vehicle in order to achieve equal slips on the wheels. This slip control algorithm operates in the domain of small slip values, where the Anti Lock System (ABS) is inactive. In case of successful control, the wheels reach the limit of adhesion at the same instant. This results in a later activation of the ABS and higher achievable deceleration. Measurements on a two-axle lorry are also included.

Keywords: slip control, electropneumatic brake system, adaptive control.

1. Slip Control

The target of the slip control algorithm is to achieve equal slip on the wheels. In case of successful control the operating points of the wheels coincide in the adhesion diagram (*Fig. 1*). This results in the achievement of the stability limits at the same instant while increasing the brake forces.

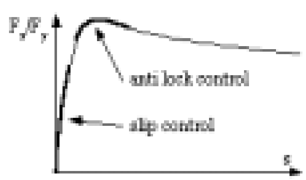

Fig. 1. Slip diagram: retardation (i.e. quotient of the horizontal and normal forces) versus wheel-slip. The operating ranges of the different control algorithms are indicated.

The slip is defined as the specific difference of the speed of rotation of the wheel and the moving speed of the contact point on the ground:

$$s_{\text{wheel}} = \frac{v_{\text{contact point}} - r_{\text{wheel}}\omega_{\text{wheel}}}{v_{\text{contact point}}}.$$

The slip corresponding to an axle having two wheels is defined as the average of the wheel slips:

$$s_{\text{axle}} = \frac{s_{\text{left wheel}} + s_{\text{right wheel}}}{2}.$$

The speed of the contact point is unknown, therefore the slip cannot be calculated. Nevertheless, the difference of the axle slips can be approximated from the wheel speed signals. The difference of the slips on axle 1 and 2 is:

$$s_2 - s_1 = \frac{r_1\omega_1 - r_2\omega_2}{v_{\text{contact point}}} \approx \frac{r_1\omega_1 - r_2\omega_2}{r_1\omega_1}.$$

This holds if the speeds of the contact points are the same for both axles, for example if the vehicle is in straight motion. The slip difference assumes the form:

$$\Delta s = \frac{r_1\omega_1 - r_2\omega_2}{r_1\omega_1}.$$

The target of the slip control algorithm is to achieve $\Delta s = 0$.

2. Processing Wheel Speed Signals

The feedback signal of the slip control algorithm is the Δs axle-wise specific speed difference. This is a small value usually in the range of 0.01–0.05. The effect of the wheel radii, the wheel-path difference while cornering, and the noise on the wheel speed of rotation signals are in the same order of magnitude as the slip difference itself.

2.1. Estimation of Wheel Radius Difference

The slip difference Δs is influenced by the ratio of the rolling radii, depending mainly on the load conditions, tire type, tire pressure and wear:

$$\Delta s = \frac{\omega_1 - \dfrac{r_2}{r_1} \cdot \omega_2}{\omega_2}.$$

The radius difference can be much higher than the above-mentioned order of magnitude of the slip difference (10–15% difference is usually due to different tire types, 1–2% due to load condition on the grounds of my experiences).

Because of these effects this ratio cannot be handled as a constant (end of line) parameter, but has to be estimated during operation.

The radius difference can be estimated in a special operation phase, during gearshifts. In such cases the clutch is open, the driven wheels are free rolling

($s_1 = s_2 = \Delta s = 0$), the measured difference of the speeds is caused by the radius difference:

$$\frac{\omega_1}{\omega_2} = \frac{r_2}{r_1}.$$

2.2. Gearshift Detection

As a consequence of the above facts, the gearshifts have to be detected. For this purpose the speed of rotation of the engine shaft and that of the driven wheels are compared. At closed clutch, the ratio of these signals is nearly constant, because a 'rigid' driving-gear connects them. In case of open clutch, the speeds of the shafts move away from each other:

$$\frac{\mathrm{d}}{\mathrm{d}t}\frac{\omega_{\text{engine}}}{\omega_{\text{driven axle}}}\begin{cases} = 0 & \Rightarrow \quad \text{clutch is not surely open} \\ \neq 0 & \Rightarrow \quad \text{clutch is open} \end{cases}$$

Observation of this derivative of the speed ratio recognizes the gearshifts. This numerical differentiation requires a good signal filtering, which smoothes the signal but does not cause great delay in the recognition, because the gearshifts have short time periods, usually in the range of 0.5 sec.

Fig. 2 shows a gearshift. The gear ratio calculated from the engine (top diagram) and wheel speed signals (second diagram) is not constant any more, its derivative (third diagram) is nonzero. This is the sign of the gearshift. The specific wheel speed difference (bottom diagram) gets the value caused by the wheel radius deviation.

2.3. Effect of Cornering

The paths of the front and rear wheels differ in case of cornering. This falsifies the calculated slip-difference signal. Cornering has to be detected therefore, and some routines, such as wheel radius correction have to be turned off in such case. Nevertheless, the calculation of the slip difference cannot be suspended, because the control algorithm has to operate in a curve as well.

Curve can be identified by observing the specific speed difference of the wheels on both sides of an axle. By using some geometrical parameters of the vehicle (wheel base l and track b) the specific speed difference between the axles can be estimated using a simple geometrical model with zero side slip angles and some linearization:

$$\left.\frac{v_1 - v_2}{v_1}\right|_{\text{free roll while cornering}} \approx \left(\frac{4 \cdot l \cdot \dfrac{v_{1\text{left}} - v_{1\text{right}}}{v_1}}{b}\right)^2 .$$

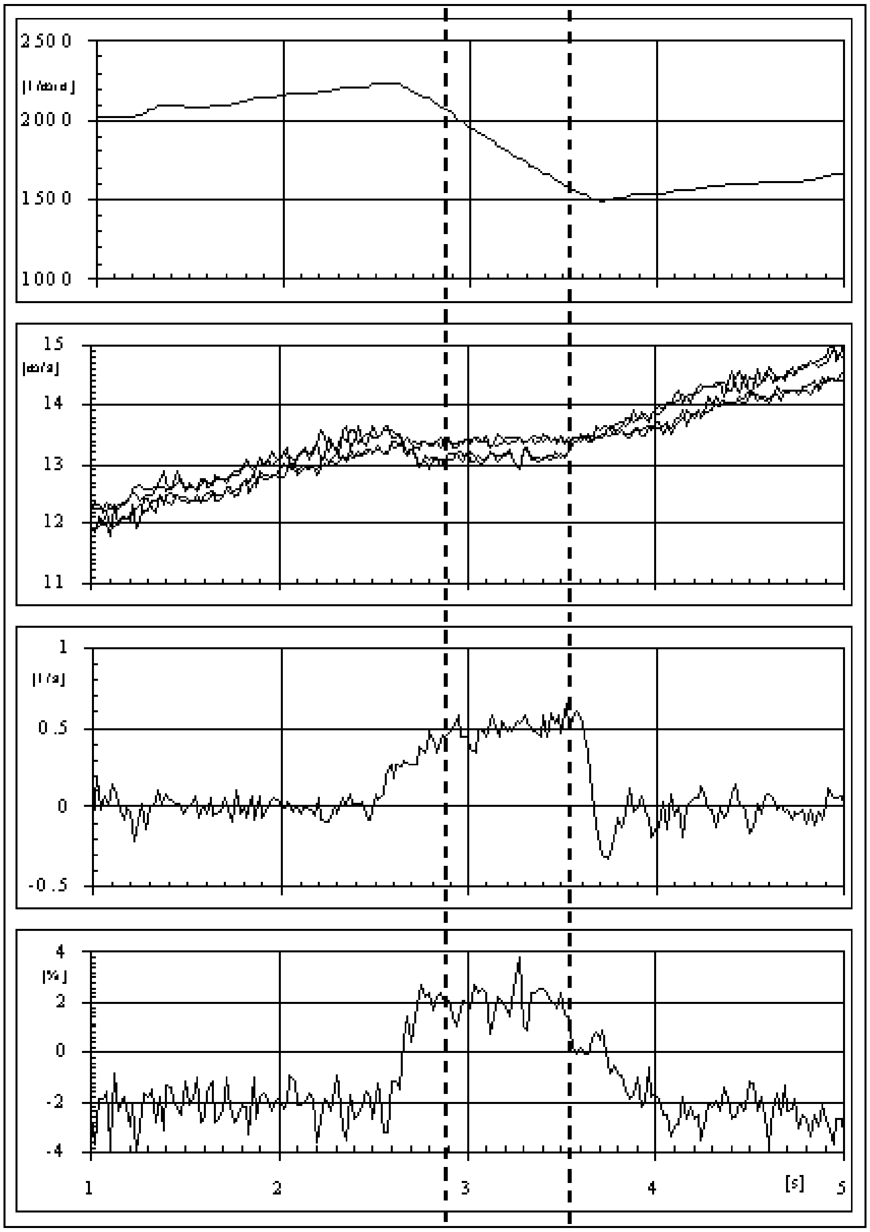

Fig. 2. Estimation of wheel radius difference during gearshift. Engine speed [1/min], wheel speeds [m/s], derivative of their ratio [1/s] and slip difference [%]. Two dotted lines indicate the instant of detected gearshift.

This specific speed difference is used for the estimation of the axle-wise slip difference.

3. Brake Force Distribution

The slip control algorithm changes the axle-wise ratio of the brake forces as a function of the slip difference. This algorithm does not modify the average level of the brake forces thus the deceleration of the vehicle is not influenced.

The steady state equations of motion of the vehicle are:

$$\begin{aligned} F_{2y} \cdot l &= m \cdot g \cdot l_1 - h \cdot (F_{1x} + F_{2x}), \\ F_{1x} + F_{2x} &= m \cdot a, \\ F_{1y} + F_{2y} &= m \cdot g, \end{aligned}$$

where

F	wheel force
M	vehicle mass
a, g	horizontal acceleration, gravitational acceleration
$x, y, 1, 2$	indices: horizontal, vertical direction and front, rear axle
l, l_1, h	geometrical parameters: wheelbase, horizontal, vertical CG-position (measured from the front wheel contact point)

After some rearrangement (division of the first equation with the third) we get:

$$\frac{F_{2y}}{F_{1y} + F_{2y}} = \frac{l_1}{l} - \frac{h}{lmg}(F_{1x} + F_{2x})$$

or

$$\frac{F_{2y}}{F_{1y} + F_{2y}} = c_0 - c_1(F_{1x} + F_{2x}),$$

thus the *rear axle part of the load* depends linearly on the total horizontal (braking) force. The parameters c_0 and c_1 describe the load conditions of the vehicle: c_0 is the dimensionless horizontal CG-position, while c_1 depends on the CG-height and the mass. The CG of a laden vehicle is usually closer to the rear axle than that of an empty one. The parameter c_1 does not change so significantly, because, in case of trucks with usual geometry, the increasing mass increases the CG-height also.

3.1. Adaptive Control

The algorithm consists of two parts: adaptive and feedback parts. The adaptive part identifies the geometrical parameters of the vehicle corresponding to the steady state equations, while the feedback part corrects the slip error resulting from dynamic

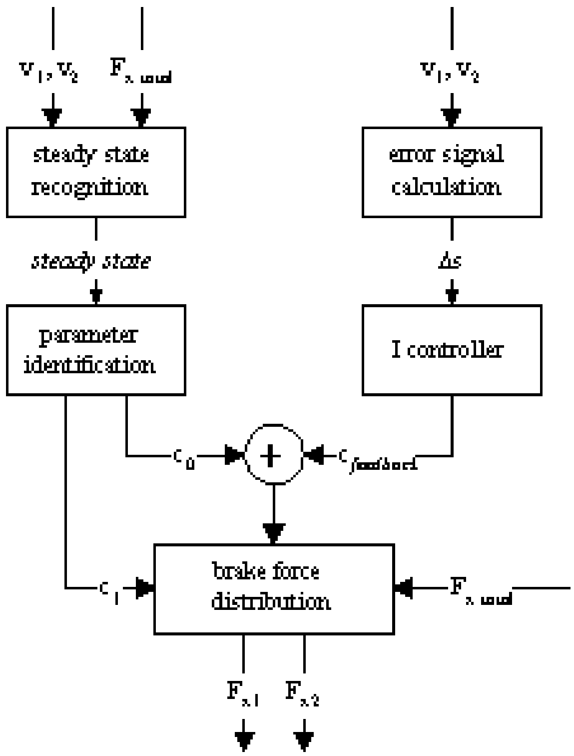

Fig. 3. Structure of the control algorithm

changes in the total braking force (changing drivers demand) and other disturbances, such as road surface change or unevenness.

The adaptive part observes the brake force and wheel speed signals and uses steady phases (with constant deceleration) to identify the above mentioned parameters c_0 and c_1. The feedback part is an I-controller. Its control signal is added to the static parameter c_0. The structure of the control algorithm is shown in *Fig. 3*.

A braking with slip control is shown in *Fig. 4*. Instead of brake forces, the brake pressures are shown here. These are proportional to the brake forces.

At the beginning of the braking, the rear axle gets bigger brake pressure than the front one. This causes a slip difference of about 1%. The algorithm reorders the brake forces in three steps beginning from the 3rd second. The front axle gets bigger brake pressure than the rear one after the fourth second, therefore the slip difference decreases to zero.

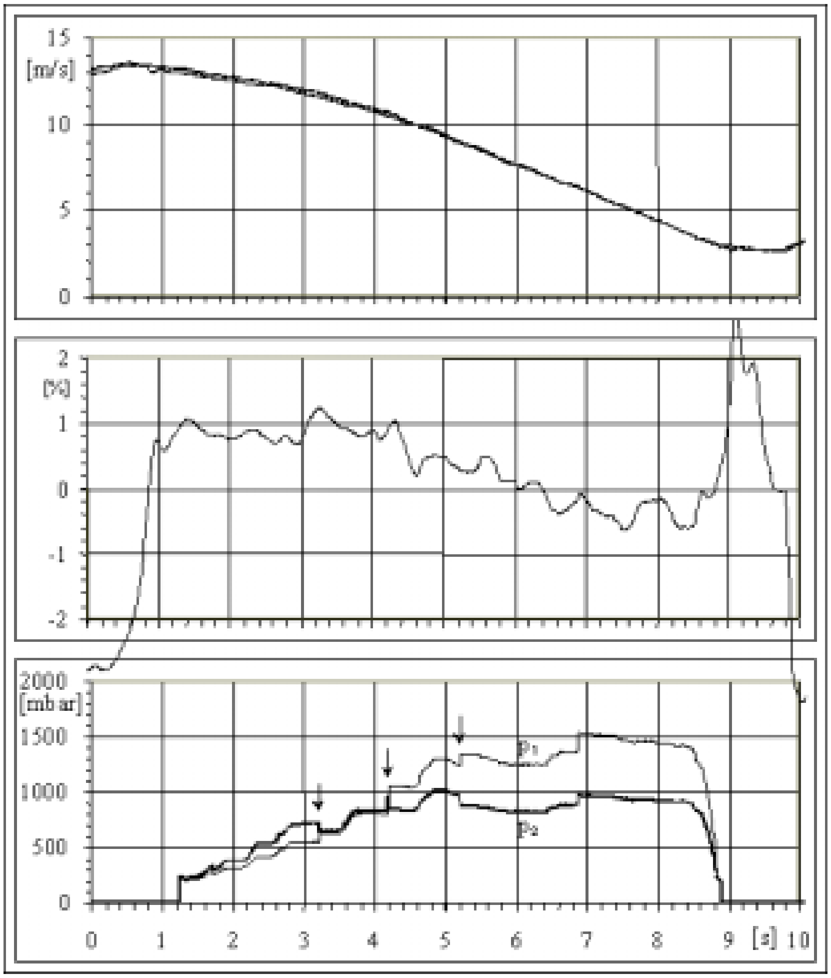

Fig. 4. Braking with slip control. Wheel speeds [m/s], slip difference [%] and brake pressures [mbar]. The control interventions are indicated.

4. Conclusions

The slip-control algorithm is able to control the brake force distribution of commercial vehicles equipped with electropneumatic brake systems.

The presented structure of the controller allows a fast achievement of the control target: the slip equality. The signal processing takes into account the disturbances caused by wheel radius differences, load distribution and cornering. Measurement results show that the algorithm sets the slip difference within a tolerance of 1%, resulting in a delayed approach of the stability limit.

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