

An Investigation Into New ABS Control Strategies

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ABSTRACT

An investigation into two new control strategies for the vehicle Anti-lock Braking System (ABS) are made for a possible replacement of current non-optimal slip control methods.

This paper applies two techniques in order to maximize the braking force without any wheel locking. The first considers the power dissipated by the brake actuator. This power method does not use slip to construct its reference signal for control. A heuristic approach is taken with this algorithm where one searches for the maximum power dissipated. This can open up easier implementation of regenerative braking concurrently with ABS on an electro-hydraulic braking system. Parameter scheduling is explored in this algorithm.

The second algorithm employs the use of perturbation based Extremum Seeking Control (ESC) to provide a reference slip and a Youla controller in a negative feedback loop.

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INTRODUCTION

In an emergency braking situation, there is a maximum braking force that can be applied that does not lock the wheel and cause the vehicle to lose steering control. This maximum braking force is dictated by a number of uncertain conditions and parameters e.g. road type, tire condition, tire temperature, etc. Current Anti-lock Braking Systems (ABS) in use today attempt to provide the maximum braking force by providing an increase in brake pressure until the wheel begins to lock up, then lets off the pressure to allow the wheel to rotate again, and repeats. This algorithm is not ideal and non-optimal as it simply reacts to wheel locking in order to provide the maximum braking force. Furthermore, for most drivers the subjective brake pedal feedback from this strategy is poor.

This paper applies two different ABS algorithms to find and apply the largest braking torque at the wheel without wheel locking. The first algorithm uses a heuristic set up in which the power dissipated at the wheel by the brake is maximized. The relationship between this power dissipated and the maximum braking force available is explored and tested in simulation.

The second algorithm uses ESC with a Youla based controller. The model is linearized to a first order transfer function from which a Youla controller is designed. Youla parameterization shows promising results as a practical controller that can be used over many of the feedback control strategies usually found in literature when considering ESC, such as Lyapunov based controllers.

System Modeling

Simulation begins with a proper model. A simple unicycle model (also known as a single corner model) is sufficient for preliminary testing of new algorithms [1]. Fig. 1 shows a simple unicycle model as a mass on a wheel constrained to only moving on a single axis. The model accepts no vertical displacement, has no suspension, and assumes the normal force to be constant. It only accepts a torque input T_b at the wheel to simulate braking actuation and a force F_b to represent the available braking force the road can provide to the tire.

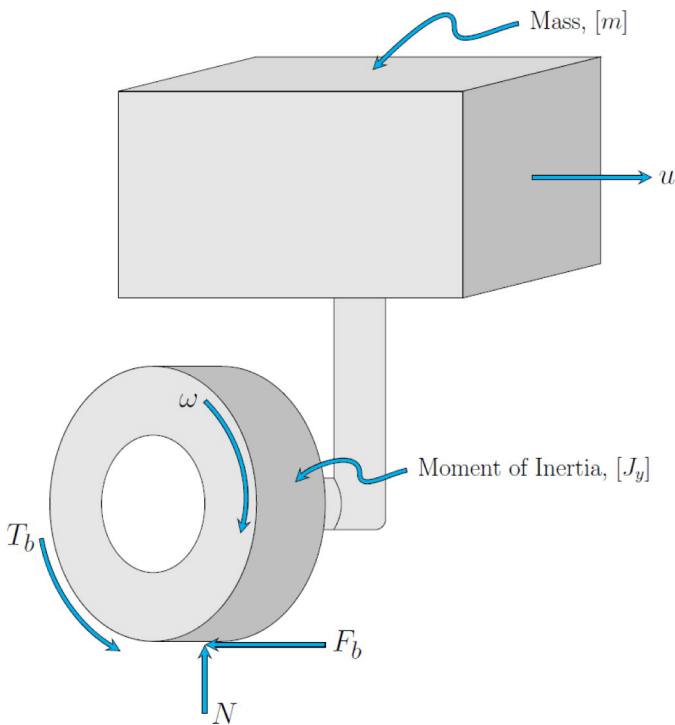


Figure 1. Simple Unicycle Model

This model is also represented by a bond graph as shown in Fig 2. This representation is a bit more intuitive when looking at power for the heuristic power approach. The braking torque applied at the wheel removes energy from the system and is represented as the effort source SE. The nonlinear braking force is represented as a resistive element, R. A linear resistive bearing friction is also included in the model for simulation [2] with a damping value of B.

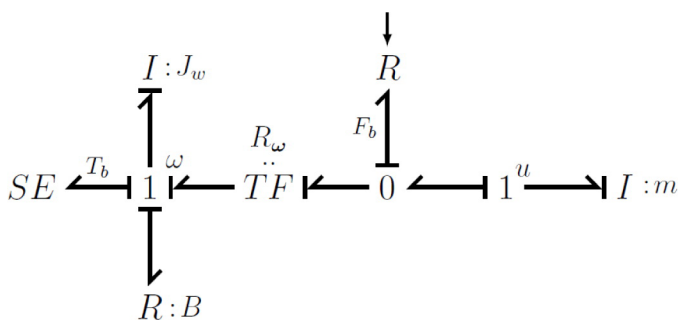


Figure 2. Bond graph representation of the unicycle model shown in Fig 1.

The equations of motion derived from the model are as follows

$$J_y \dot{\omega} = F_b R_{\omega} - \tau_b - B\omega \quad (1)$$

$$m\dot{u} = -F_b \quad (2)$$

where $\omega(\text{rad/s})$ is the wheel velocity, $u(\text{m/s})$ is the longitudinal velocity, $T_b(\text{Nm})$ is the brake torque applied at the wheel, and $F_b(N)$ is the brake force. Parameters used are defined and values given in Table 1. F_b is calculated as

$$F_b = \mu N \quad (3)$$

where N is the normal force.

Table 1. Parameter set for modeling

Parameter	Symbol	Value	Unit
Mass	m	400	Kg
Wheel Moment of Inertia	J_y	1.17	Kg-m ²
Wheel Radius	R_{ω}	.3	M
Bearing Friction Coefficient	B	.01	N-m-s
Gravity	g	9.81	m/s ²

Many tire models exist for calculating the tire forces, such as Pacejka, Dugoff, LuGre, and Burckhardt tire models. A simplified Burckhardt tire model is used in this study. The longitudinal coefficient of friction is calculated by Eq. 4 [3]

$$\mu = c_1(1 - e^{-c_2\lambda}) - c_3\lambda \quad (4)$$

where c_1 , c_2 , and c_3 are tire model parameters. These c values are changed depending on what type of road is represented. A list of these values and corresponding road types are available in Table 2. A depiction of the tire model can be seen in Fig 3.

The brake force F_b applied on the wheel by the road is determined by the tire slip λ . This paper only looks at the tire slip when braking, and is determined by

$$\lambda = \frac{R\omega - u}{u} \quad (5)$$

Forward velocity and acceleration is assumed to be measureable for the simulation.

Table 2. Burckhardt parameters for different road types [3].

Road Surface	C_1	C_2	C_3
Dry Asphalt	1.2801	23.99	0.52
Wet Asphalt	0.857	33.822	0.347
Dry Cobblestone	1.3713	6.4565	0.6691
Wet Cobblestone	0.4004	33.7080	0.1204
Snow	0.1946	94.129	0.0646

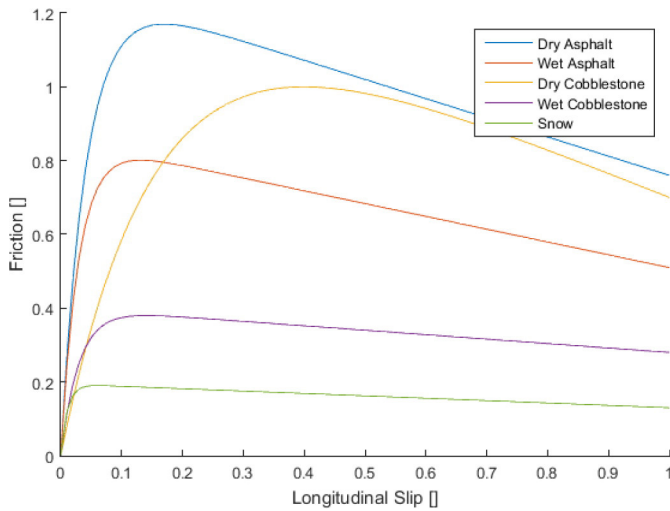


Figure 3. Burckhardt tire model for various road conditions.

ALGORITHM DESIGNS

Heuristic Power

An internal paper by Assadian, F. [4] explores a power approach to finding the maximum braking force. The brake torque is assumed to be directly controlled. Associating the maximization energy consumed by the brake system to the maximum power rate is done by looking at the power balance of the system, as shown in Eq. 6.

$$\dot{P}_j \omega + \tau_b \omega + F_b(u - R_\omega \omega) = F_b u \quad (6)$$

where $\dot{P}_j$ is the derivative of the rotational momentum of the tire. This derivation is simplified by neglecting the bearing resistance. This can now be simplified to the following:

$$\dot{P}_j \omega + \tau_b \omega = F_b R_\omega \omega \quad (7)$$

We now assume a steady state case which drops the inertial term. The relationship is now

$$\tau_b \omega = F_b R_\omega \omega \quad (8)$$

From Eq. 8 we see that if we can maximize our power from the brake actuation, we maximize the longitudinal brake force as long as the tire does not lock up and slide. A simple heuristic algorithm can be created [4] which can be seen in Fig 4.

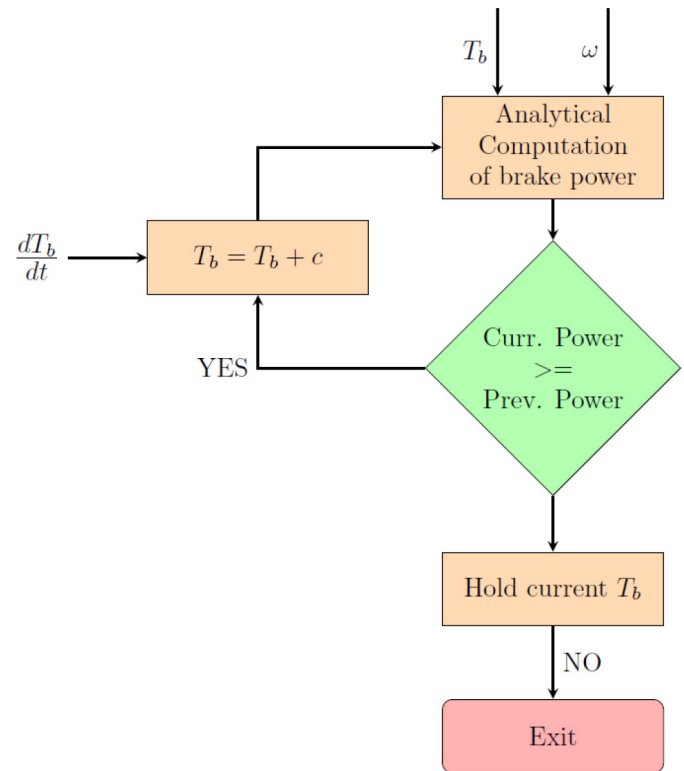


Figure 4. Flow chart of the power approach.

The algorithm requires an input dT_b/dt , the torque rate of change that is applied at the wheel. Choosing an appropriate value that would allow the heuristic algorithm to find the maximum power is difficult at first look. Choosing some constant values show how the algorithm is sensitive to an arbitrary choice. Fig 5 shows the power for various constant torque rate of change inputs. If dT_b/dt is too high, the maximum power is overshoot, which in turn overshoots the torque applied at the wheel. The wheel locks up as a consequence and the power drops to a value of zero quickly. If dT_b/dt is too low, the final torque applied at the wheel is lower than the optimum torque which results in a braking force lower than the maximum.

A relationship for dT_b/dt can be derived from the model. A lengthy derivation can be found in [4], which gives the following:

$$\frac{dT_b}{dt} = \alpha(\mu_{max} - \mu) \quad (9)$$

$$\alpha = -\frac{\partial \mu(u) R_\omega^3 N^2}{\partial u J_\omega} \quad (10)$$

Eq. 9 reveals that knowledge of the road condition, or an estimation of the maximum coefficient of friction, would be needed. Access to these road uncertainties are difficult. Having them would mean one can directly apply them in a simple controller that can outperform anything available.