An Application of Nonlinear PID Control to a Class of Truck ABS Problems

Fangjun Jiang
Ford Motor Company, Product Development Center, GB-E65, MD 199
20901 Oakwood Blvd. Dearborn, MI 48124
Zhiqiang Gao
The Applied Control Research Laboratory
Department of Electrical and Computer Engineering
Cleveland State University, Cleveland, Oh 44115

Abstract: A new NPID (Nonlinear Proportional-Integral-Differential) control algorithm is applied to a class of truck ABS (Anti-lock Brake System) problems. The NPID algorithm combines the advantages of robust control and easy tuning. Simulation results at various situations using TruckSim show that NPID controller has shorter stopping distance and better velocity performance than the conventional PID controller and a loop-shaping controller.

Keywords: Nonlinear, PID, ABS.

1. Introduction

ABS for commercial vehicles appeared on the market in 1960s and began to grow fast in 1970s with the technologies of microcomputers and electronics [1]. ABS is recognized as an important contribution to road safety. It is now available in almost all types of vehicles. The automotive industry is continuously developing new generations of ABS. The technologies of ABS are also applied in TCS (Traction Control System) and VDSC (Vehicle Dynamic Stability Control)

It is well known that wheels will slip and lockup during severe braking or when braking on a slippery road surface (wet, icy, etc.). This usually causes a long stopping distance and sometimes the vehicle will lose steering stability. The objective of ABS is to prevent wheels from lockup and achieve minimum stopping distance while maintaining good steering stability during braking.

The wheel slip is defined as:

$$S = \frac{V - \omega R}{V} \tag{1.1}$$

where S, ω, R and V denote the wheel slip, the wheel angular velocity, the wheel rolling radius, and the vehicle forward velocity, respectively.

In normal driving conditions, $V \approx \omega R$ therefore $S \approx 0$. In severe braking, it is common to have $\omega = 0$ while $V \neq 0$, or $S = 1$, which is called wheel lockup. Wheel lockup is undesirable since it prolongs the stopping distance and causes the loss of direction control.

1.1 A Class of Truck ABS Problems

The objective of ABS is to manipulate the wheel slip so that a maximum friction force is obtained and the steering stability (also known as the lateral stability) is maintained. That is, to make the vehicle stop in the shortest distance possible while maintaining the directional control. It is well known that the friction coefficient, μ, is a nonlinear function of the slip, S. The ideal goal for the control design is to regulate the wheel velocity, ω, such that an optimal slip, which corresponds to the maximum friction, is obtained. For the sake of simplicity, however, it is very common in industry to set a desired slip to .2. Given the vehicle velocity, V, and the wheel radius R, the ABS control problem becomes regulating ω such that the slip in (1.1) reaches a desired value, such as .2.

In this paper, the control design is focused on a class of truck ABS problems, which pose a few unique challenges, different from passenger cars.

1. The actuator of the truck ABS is a pneumatic brake system, which is typically slower in response and harder to control than a hydraulic brake system. The control action of the brake system is discrete. The brake pressure is controlled by discrete valves (open or close). The brake pressure can be controlled to increase, hold constant or decrease. Through PWM (Pulse Width Modulation), the actions of the discrete valves are mapped into a continues analog control signal ranging from *–1* to *+1*, where *–1* means fully exhausting pressure, +1 means fully building up pressure and 0 means holding pressure as constant.
2. The measurement of the brake pressure is not available, which makes the control of the pneumatic brake system even more difficult. The ABS controller must deal with the brake dynamics and the wheel dynamics as a whole plant.
3. The measurement of the vehicle velocity or vehicle acceleration is not available. The only feedback signals are two or four channels of wheel angular velocity. It poses a challenging problem for the vehicle velocity estimation since the vehicle velocity is necessary to set the wheel reference velocity. A separate study was carried out to resolve this issue in [2].
4. The complex dynamics of the tractor/trailer system and the large variations of the truck operation condition set a very stringent requirement for the ABS controller. The tuning and testing of a truck ABS are also much more difficult than an ABS for passenger cars.

1.2 Current Technology

Various control strategies have been implemented in real ABS products or discussed in publications. Since the technologies used in commercial ABS products are usually kept as trade secrets, it is very difficult to determine their detailed control algorithms. From the literature available [3, 4, 5, 6], a few algorithms use an approach similar to "bang-bang" control. They usually have two or more threshold values for the wheel deceleration or the wheel slip. Once the calculated wheel deceleration or wheel slip is over one of the threshold values, the brake pressure is commanded to increase, hold constant or decrease. This algorithm will result in a peak-seeking strategy in the μ–slip curve or forcing the wheel deceleration/slip to be within a particular range.

Finite state machine methods are also widely applied in the industry. Based upon the measured signals such as wheel velocity, vehicle deceleration and/or brake pressure, the operation of the vehicle is characterized by a set of different states, such as normal driving, lockup, free rolling, etc. The brake pressure is then controlled to increase, hold constant or decrease based on the state the vehicle is in and other design logic.

These two methods heavily rely on the experience of the designers and drivers. It is fairly difficult to analyze the controller's performance during the design stage. The tuning of the controller is done purely on trial and error basis. The needs for a systematic design approach for the ABS development are quite evident in this industry. Such needs motivated the research efforts that result in [9].

In particular, the truck ABS problems are reformulated as a closed-loop control problem. A cascade loop structure, as shown in Figure 1, as well as various control algorithms are proposed. The outer loop, which includes the vehicle velocity estimation and desired slip calculation, provides the command signal, V_{wd} ,for the inner wheel velocity loop. The separation of the outer and inner loop designs, similar to the separation principle in linear system theory, are only made possible in the framework of Figure 1.

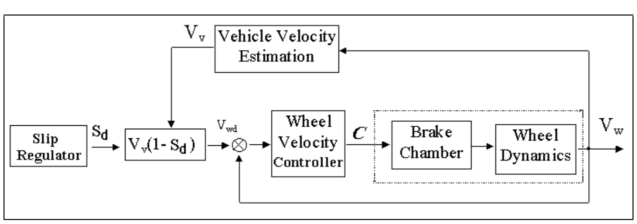

Figure 1: A Cascade Structure for ABS

The vehicle velocity estimation and the wheel velocity controller are the key design issues. A nonlinear filter approach, based on the work in [10], to vehicle velocity estimation problems was developed and proved to be quite effective [2,9]. For inner loop control, three methods were explored in [9], including the PID, the loop-shaping, and the NPID algorithms. The PID is easy to design and tune but is also limited in performance. The loop-shaping controller, designed based on the linear model of the plant and frequency response loop-shaping concepts, is quite capable in simulation. The drawback is the difficulty of tuning such controllers on a real industrial simulator, where the controller must be adjusted for nonlinearities and disturbances uncounted for in the model.

Similar tuning difficulties can also be seen in various other advanced control strategies such as fuzzy logic control, model reference control and neural network, which were also extensively discussed as possible candidates for ABS.

In the development of an ABS controller, one of the major issues is testing. The ABS controller needs to go through a series of software and hardware tests. Due to the complexity of the truck system and the large variations of operation conditions, on-site calibration or tuning of the controller is necessary. This requires the new control methods to be not only more powerful, but also easily tunable. The tuning of a fuzzy logic controller or model reference controller involves multiple rules or re-design of the controller. It is not convenient to be carried out on-site. The conventional PID controller is easy to tune but appears not to be adequate in performance.

Based on the above discussion, we propose a NPID control design strategy, based on the work of J. Han [11,12], that combines the advantages of robust performance and the ease of tuning. It is proved to be an effective controller for truck ABS.

2. Proposed Approach

The PID controller is simple and easy to implement. It is widely applied in industry to solve various control problems. Based on the conventional PID controller, a nonlinear PID controller (NPID) was investigated.

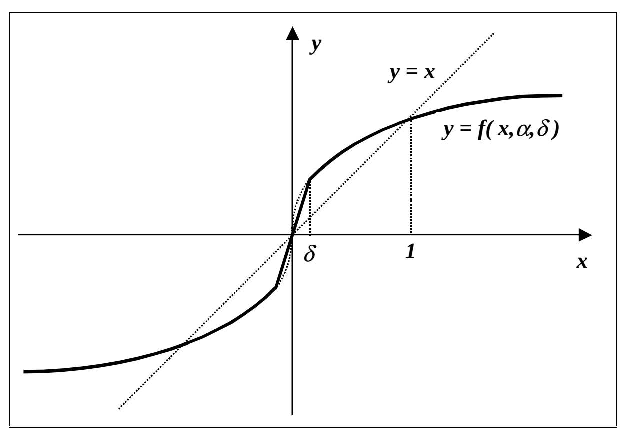

Figure 2: Illustration of the nonlinear function

The conventional PID (or specifically, linear PID) controller can be described as:

$$u = K_P(e + T_I \int e + T_D \dot{e}) \qquad (2.1)$$

where e, $\int e$ and $\dot{e}$ represent the error, the integration of the error, and the derivative of the error, respectively. K_P is the proportional gain, T_I is the integral time constant and T_D is the derivative time constant. u is the controller output.

The nonlinear PID (NPID) controller can be described as:

$$u = K_{NP}[\,f(e,\alpha_P,\delta_P) + T_{NI}\,f(\int e,\alpha_I,\delta_I) + T_{ND}\,f(\dot{e},\alpha_D,\delta_D)] \tag{2.2}$$

where e, $\int e$ and $\dot{e}$ are the same as in the linear PID controller. K_{NP}, T_{NI} and T_{ND} are three parameters, which have similar meanings to K_P, T_I and T_D in the linear PID controller. $f(*)$ is a nonlinear function (illustrated in Figure 2), which is defined as:

$$y = f(x,\alpha,\delta) = \begin{cases} sign(x)\cdot|x|^{\alpha}, & when\ |x| > \delta \\ \delta^{\alpha-1}\cdot x, & when\ |x| \le \delta \end{cases} \tag{2.3}$$

where x is the input and y is the output. The α and δ terms are two parameters of the nonlinear function. Usually, α is between 0 and 1 ($0<\alpha\le 1$). When $\alpha = 1$, it is equivalent to the linear function of $y = x$. δ is a small positive number applied to create a small linear area in this nonlinear function when x is around zero. This is to avoid excessive high gain in the neighborhood of the origin, which could lead to numerical problems.

The idea of the NPID controller is to use a nonlinear combination of e, $\int e$ and $\dot{e}$ in place of the linear one in the conventional PID controller. The $f(x, \alpha, \delta)$ function is an exponential function and α is the exponent. A commonly used value for α is 0.5, which gives a nonlinear mapping between x and y shown in Figure 2. Compared with the linear function $y = x$, the nonlinear function $f(x, \alpha, \delta)$ gives high gain for small x and small gain for large x.

When applied in the industry, a PID controller is usually implemented with varying parameters, which means that the gain, the integral time constant and the derivative time constant are all modified on-line based on the magnitude of the error. This is called gain-scheduling, which gives high gain for small errors and small gain for large errors. The NPID controller uses an exponential function to implement this idea simply and systematically. The tuning of a NPID controller is similar to tuning a PID controller, which can be done on-site to achieve better performance based on test results. Later simulation results show that a NPID controller for ABS achieves better performance than a PID controller, and has strong robustness dealing with the large variations of the road surface condition, the air supply pressure and the brake chamber dynamics.

3. Simulation

The purpose of the simulations is to compare the performance of three controllers: the PID, the loop-shaping controller, and the NPID. Six different operating conditions are used, including the nominal braking condition, braking on low-μ surface, braking with high and low air pressures, and braking with fast and slow brake dynamics. The tests were conducted using an industrial simulator. Every attempt was made to make test conditions are as realistic as possible.

3.1 Controllers and Conditions

The controllers designed and simulated are given as follows:

1. The PID controller:

$$G_{PID}(s) = K_{Pj}(1 + \frac{T_{Ij}}{s} + T_{Dj}s),\ j = 4, 5 \tag{3.1}$$

   where j denotes the parameter for the PID controller of axle 4 or axle 5. K_P, T_I and T_D represent the proportional gain, the integral time constant and the derivative time constant, respectively. $K_{P4}=-0.03$, $K_{P5}=-0.05$, $T_{I4}=T_{I5}=0.3$, $T_{D4}=T_{D5}=0.01$.
2. The loop-shaping controller is designed based on the linear transfer function obtained from the test data [9]:

$$G_{c2}(s) = \frac{-1.5\times 10^5 (s+5)^4}{s(s+100)^5} \tag{3.2}$$

   Note that the same controller is applied to both axle 4 and axle 5.
3. The NPID controller is designed and tuned as:

$$u = K_{NP}[\,f(e,\alpha_P,\delta_P) + T_{NI}\,f(\int e,\alpha_I,\delta_I) + T_{ND}\,f(\dot{e},\alpha_D,\delta_D)] \tag{3.3}$$

   where $\alpha_P = \alpha_I = \alpha_D = \alpha = 0.5$, $K_{NP}=-0.015$, $T_{NI}=T_{ND}=0.5$, $\delta_P = \delta_I = \delta_D = \delta = 0.1$. This controller is also implemented at both axle 4 and axle 5.

The six different cases reflect the variations of the road surface condition, the air supply pressure and the brake chamber dynamics. They are defined as follows:

S1: Nominal ($P_c=90\ PSIG$, $\mu=0.7$, nominal parameters for the brake chamber dynamics)
S2: Low μ surface ($\mu=0.4$)
S3: High air supply pressure ($P_c=120\ PSIG$)
S4: Low air supply pressure ($P_c=60\ PSIG$)
S5: Fast brake dynamics (the smallest time constant and the smallest damping ratio)
S6: Slow brake dynamics (the largest time constant and the largest damping ratio)

The simulation time is 20 seconds for case S2 and 15 seconds for the rest of the cases. The step size of the simulation is 2.5ms. The 15ms sample and hold is applied for the brake chamber control signal and the wheel velocity signal.

3.2 The Industrial Simulator

The software used to simulate the truck dynamics with

ABS controllers above is TruckSim. It is a realistic industry simulation package developed by Mechanical Simulation Corporation (MSC) for heavy vehicle dynamic simulations.

TruckSim is an easy-to-use software package designed to simulate and analyze heavy vehicle dynamics. It is specialized in the braking and handling behavior of trucks, busses and tractor-trailer combinations under various testing conditions. TruckSim performs virtual tests by replacing the testing vehicle with a mathematical model. It solves motion equations to predict the response of the vehicle under braking and steering. TruckSim uses detailed nonlinear tire models and nonlinear spring models. It includes major kinetic and compliance effects in the suspension and steering system for trucks, busses and other highway vehicles with solid-axle suspension and asymmetric steering system. For control inputs, TruckSim accepts time history of the braking and steering angle inputs (open-loop control). TruckSim also has closed-loop control options for the steering (driver model) and speed control.

TruckSim software has three forms available. Its Matlab CMEX function file can be used in the MathWorks' Simulink environment and makes it very easy for rapid prototype controller development. Details can be found in reference [7].

Due to the uniqueness of the ABS problems, it is difficult to fully evaluate the performance of an ABS controller in simulations. The objective of ABS is to reduce the stopping distance and improve the lateral stability. The stopping distance can be obtained through simulations and it is easy to compare. However, it should not be the only criterion to evaluate an ABS controller. On a high μ surface, a braking with many wheel lockups could still achieve a short stopping distance, but the lateral stability usually is poor and it could cause damages to the tires, or even a vehicle rollover. In our simulations, however, since the lateral motion could not be simulated in TruckSim for the braking on a straight path, only the stopping distance and the 2-norm of the wheel velocity error are used for comparison.

3.3 Simulation Results

Two measures of performance are applied to evaluate the simulation results of three controllers. One is the stopping distance and the other is the 2-norm of the wheel velocity error. The stopping distance *SD* is defined as:

$$SD = \int_{t_0}^{t_1} V_V \, dt \qquad (3.4)$$

where V_V is the actual vehicle velocity , t_0 and t_1 are the start time and the end time of the braking. In the simulations, we set $t_0 = 0$, $t_1 = 15$ sec (or *20 sec*) and SD is computed iteratively.

The 2-norm of the wheel velocity error is defined as:

$$NM_j = \sqrt{\sum_{1}^{i} e_i^2}\ ,\ i = 1, \ldots, 6000\ (\text{or } 8000),\ j=4,5 \qquad (3.5)$$

where $e=V_{Wd}-V_W$ is the difference between the desired wheel velocity (or the wheel reference velocity) and the wheel velocity. e_i is the value of the error in the i^{th} simulation step. For a simulation step size of *2.5ms*, *6000* or *8000* steps are used for *15* or *20* seconds of simulation. Two 2-norms (NM_4 for the wheel L4 and NM_5 for the wheel R5) are calculated. The final result *NM* is the average of NM_4 and NM_5:

$$NM=(NM_4+NM_5)/2 \qquad (3.6)$$

The average norm *NM* reflects the overall performance of the wheel velocity controller. The smaller the value, the better the performance.

Table I shows the comparison of the stopping distance, *SD*, and Table II the comparison of the wheel velocity response. The symbols $^{\Delta}$ and * denote the best and the second best performances in the simulations. A sample of simulation curves is shown in Figure 3 to Figure 5.

Table I: Stopping distance

SD (Stopping distance)		PID	Loop-shaping	NPID
Nominal	S1	86.3	61.3^{Δ}	65.9^{*}
Low μ	S2	154.1	139.9	114.6^{Δ}
High air pressure	S3	91.0	63.4^{Δ}	66.4^{*}
Low air pressure	S4	83.3	62.5^{Δ}	66.7^{*}
Fast brake response	S5	78.5	80.4	64.3^{Δ}
Slow brake response	S6	94.7	67.6^{Δ}	70.8^{*}

Table II: Wheel velocity response

NM (Velocity error 2-norm)		PID	Loop-shaping	NPID
Nominal	S1	740.7	216.1^{Δ}	286.1^{*}
Low μ	S2	945.1	694.2^{*}	603.2^{Δ}
High air pressure	S3	770.4	360.2^{Δ}	394.5^{*}
Low air pressure	S4	631.7	241.8^{Δ}	256.8^{*}
Fast brake response	S5	594.6	492.7	229.3^{Δ}
Slow brake response	S6	818.6	518.2	377.0^{Δ}

From the tables and figures we can see that the wheel velocities under PID control usually have two lockups at the beginning of the braking, and the *SD* and the *NM* calculated from all six cases for the PID controller are larger than those of the loop-shaping controller and the NPID controller. This means less satisfactory performance. On the other hand, based on the two criteria, the NPID controller performs well in all six cases. The loop-shaping controller has an overall better performance than the PID controller and in some cases it is even better (in numeric) than the NPID controller. However, its performances on a low μ surface or under the variations of the brake chamber dynamics are less satisfactory.

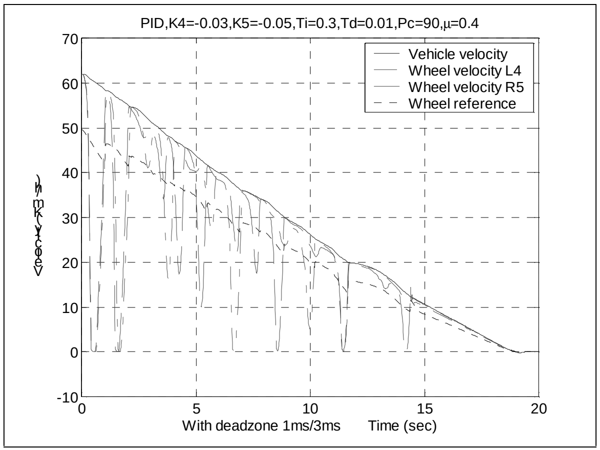

Figure 3: Simulation of PID in case S2

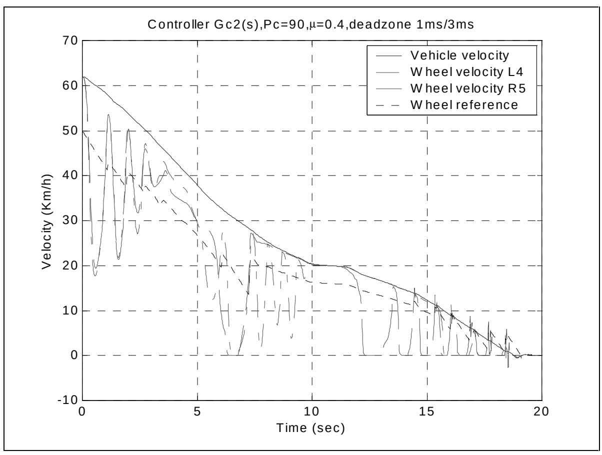

Figure 4: Simulation of loop-shaping controller in case S2

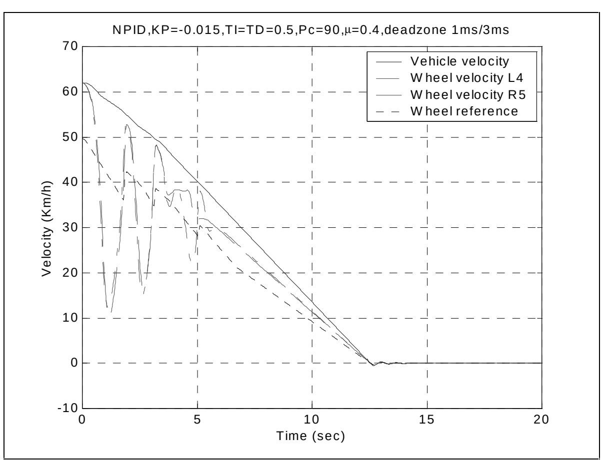

Figure 5: Simulation of NPID in case S2

3.4 Observations

From the simulation results shown in the previous sections, it is shown that the PID controller, the loop-shaping controller and the NPID controller are all viable solutions for the truck ABS problems. No total lockup or total free-rolling was observed.

The PID controller is simple and easy to implement. The tuning of the PID controller is intuitive and is well accepted by practitioners. Its performance for wheel velocity control would be enhanced if the initial wheel lockups could be eliminated through more elaborate tuning.

The loop-shaping controller has strong regulations on the wheel velocities. The wheel velocities are controlled to follow the wheel reference velocity closely and smoothly, which is ideal for the ABS control. However, this may cause a problem for the vehicle velocity estimation since a smooth wheel velocity does not contain much information of vehicle velocity. The performances of the loop-shaping controller under the variations of the brake chamber dynamics or on a low μ surface need to be improved. Once again, the main problem with the loop shaping controller is the difficulty of tuning it on the fly.

The nonlinear PID controller also poses strong regulations on the wheel velocities. It has an overall better performance than the PID controller and the loop-shaping controller. The problem of the vehicle velocity estimation also exists when the wheel velocities are smooth. The nonlinear PID controller is easy to tune, which makes it suitable for on-line testing and adjustment.

4. Concluding Remarks

A nonlinear PID design strategy is proposed as a solution to a class of truck ABS problems. The new controller proves to be more powerful than the existing controllers but retains the ease of tuning and intuitions from the standard PID controller. The simulation results obtained from an industrial simulator, Trucksim, demonstrate that the NPID is a promising technology for ABS applications.

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5. Appendix

The simplified model of the brake system dynamics and the single wheel dynamics is provided below. More details can be found in [8] and [9].

$$G_q(s) = G_b(s)\cdot G_w(s)$$
$$= \frac{P_c G_I}{s(\tau^2 s^2 + 2\tau D s + 1)} \cdot \frac{-K_b}{J}\frac{1}{s}$$
$$= -\frac{P_c G_I K_b}{J}\frac{1}{s^2(\tau^2 s^2 + 2\tau D s + 1)}$$

where $G_b(s)$ is the transfer function of the brake chamber dynamics and $G_w(s)$ is the transfer function of the single-wheel dynamics. The input of this transfer function is the control signal ranging from *–1* to *+1*, the output is the wheel angular velocity in rad/s.

P_c: Air supply pressure, *60~120 PSIG*
G_I: Brake System integration gain, *6.3~12.5*
K_b: Gain constant from pressure to torque, *157Nm/PSIG*
J: Wheel moment of inertia, $21.75\ Nms^2$
R: Tire radius, *0.52m*
τ: Time constant of the second order system *0.04~0.22*
D: Damping ratio of the second order system *0.55~1.0*

6. References

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