Design of Antilock Braking System Based on Wheel Slip Estimation

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Abstract. The goal in this research is to develop a brand new ABS set of rules the use of doubtlessly available statistics approximately wheel forces. A novel ABS set of rules that uses both force and wheel slip measurements for manipulate is designed. In this study, a new integrated Nonlinear tracking Control (NTC) is evolved that includes the dynamic evaluation of hydraulic braking systems. A longitudinal dynamic behaviour of vehicle model under straight manoeuvre is simulated, which includes the angular wheel speed, vehicle velocity, wheel slip variation, brake pressure modulation and stopping distance. Mathematical modelling of Vehicle Braking System has been carried out in simulink, which employs a car pitch optimization using nonlinear control, when vehicle undergoes a surprising acceleration/deceleration and for the duration of panic braking scenario.

1. Introduction

The first ABS dates again to 1929 and is credited to the French automobile and aircraft pioneer Gabriel Voisin. Later in 1945, the primary set of mechanical ABS brakes were implanted on a Boeing B-47 to prevent spin outs and tires. In 1952, Dunlops Maxaret automatic and fully mechanical brake manipulate was one of the most vital gadgets in the history of the aviation safety. Back then, vast development in braking efficiency and the elimination of the pilots fear of over-braking, which could bring about skidding and burst of tires has ended in a marked discount in touchdown distances (as much as 30%). ABS brakes have been usually set up in airplanes thenceforth. Though already acknowledged as a revolutionary system in the aviation scenery during the 1960s, fully mechanical systems saw limited automobile use on high end automobiles only. In 1972, ”The British Jensen Interceptor automobile became the first production car to offer a Maxaret-based ABS” [1]. Low reliability of system electronics and low public awareness, allied to the additional cost to the buyer, led to their quiet withdrawal from the market in the
The development of digital electronics changing from analogy to integrated circuits and microprocessors resulted in a major milestone in automobile manufacture with the introduction of the Bosch ABS on the Mercedes-Benz Class passenger car in 1978. It was the first completely electronic four-wheel multi-channel ABS. BMW and others succeeded shortly. Japanese brake and vehicle manufactures introduced ABS brakes based totally at the Bosch machine in addition to their personal designs by way of the middle Nineteen Eighties. The Bosch device become used in 1986 in corvette and Cadillac Allante, followed by way of Ford in 1987. Since the late 1980s and early 1990s, ABSs were determined on almost all top models of each manufacturer. By the late 1990s, nearly all passenger automobiles and mild vans were prepared with four-wheel ABSs, either as an choice or well known equipment.

2. Related work

"The Magic Formula (MF) tire model provides a set of mathematical formulae from which the forces and moment acting from road to tire can be calculated at longitudinal, lateral and camber slip conditions, which may occur simultaneously”. "The Magic Formula concept is an elegant, empirical method of fitting tire data for inclusion [2]in vehicle dynamics models”. "The formula gives a good representation of measured tire characteristics while certain coefficients of the model retain a physical significance and therefore be expected to respond to road surface variations in a meaningful manner”.

2.1. General Form of Magic Formula

"The general shape of the formula that holds for a given cost of vertical load and camber perspective is given by” :

\[
y(x) = D \sin C \arctan \left[ B x(1-E) + E \arctan(Bx) \right]
\]

\[
Y(x) = y(x) + S_y
\]

\[
X = x + S_x
\]

Where, Y stands for either forces or moments X may represent the slip angle or longitudinal slip \(S_x\), \(S_y\), B, C, D and E are the anti-symmetrical shape coefficients.

The shape coefficients have to be diagnosed from the experimental information the usage of the nonlinear-curve-becoming algorithms[3]. For speedy execution-of-the iteration procedure during becoming and to make certain convergence, it is critical to generate preliminary estimates of the six coefficients which might be close to their very last values as shown in Figure 1.

"Where, \(S_x\), \(S_y\) can be estimated with reasonable accuracy directly from the experimental data for a tire”

"BCD corresponds to the slope at the origin”

D describes the peak value of the force or moment (y)

C can be estimated[4] corresponding to a large value of x

B relates approximately to the dupe in the linear range

"E controls the slip at which the peak occurs”.

2.2. Formulae of the Magic Tire Model for Braking

A. General formula for pure slip

\[
y(x) = D \sin C \arctan \left[ B x(1-E) + E \arctan(Bx) \right]
\]

\[
Y(x) = y(x) + S_y
\]

\[
X = x + S_x
\]

Where, D=peak factor= \(y_m ax\)

C= shape factor =2/\(\pi \arcsin (y_s/D)\)

B= stiffness factor=(dy/dx)

E=curvature factor = \((Bx_m-\tan(\pi/2C))/(Bx_m-\arctan(Bx_m))\)
Figure 1. Typical Tire Characteristic of Magic Formula

Figure 2. Magic Formula Optimization

\( S_x = \) Horizontal shift
\( S_y = \) Vertical shift.

B. The Lateral Force
\( Y_y = F_y \) (Lateral Force)
\( X_x = \alpha \) (side slip angle)
\( D_y = \mu_y m \)
\( F_z = (\alpha_1 F_z + \alpha_2) F_z \)
\( \text{BCD} = a_3 \sin(2 \arctan(F_z/a_4)) (1 - a_5 |\gamma|) \)
\( C_x = a_6 \)
\( B_y = \text{BCD} y / C_y D_y \)
\( E_y = a_6 F_z + a_7 \)
\( S_{xy} = a_8 \gamma + a_9 F_z + a_10 \)

C. Longitudinal Force
\( Y_x = F_x \) (Longitudinal Force)
\( X_x = s \) (Longitudinal slip)
\( D_z = \mu_x m \)
\( F_z = (b_1 F_z + b_2) F_z \)
\( \text{BCD} = (b_3 F_z^2 + b_4 F_z) \exp(-b_5 F_z) \)
\( C_z = b_6 \)
\( B_z = \text{BCD} x / C_x D_x \)
Figure 3. Magic Longitudinal Force vs. Slip on Wet Surface

\[ E_x = b_6 F_z^2 + b_7 + b_8 \]
\[ S_{x,x} = b_9 F_z + b_{10} \]
\[ S_{y,x} = 0 \]

Figure 4. Longitudinal Force vs. Slip on Dry Concrete Surface
3. Magic Formula Optimization

The desired largest function force could be more exactly derived from MF tire model. The greatest friction force will occur [5] at the desired slip $x^*$. Therefore, one can find

$$\frac{dy}{dx} = 0$$

under the condition $x = x^*$. (2)

From Magic Formula, one has

$$\frac{dy}{dx} = \frac{BCD}{[Bx(1-E)+E\arctan(Bx)]^2}$$

(1-E+E/(1+B^2x^2)) \cos(C \arctan[Bx(1-E)+E \arctan(Bx)]) (3)

Then the Equation (1) and (2) yield

$$\cos(C \arctan[Bx^*(1-E)+E \arctan(Bx^*)]) = 0$$

Hence, "the optimized slip can be expressed as"

$$x^* = \frac{(\tan(0.5\pi/C)-E \arctan(Bx^*))}{B(1-E)}.$$ (5)

The control architecture is used only when $x^*$ finally gets convergent as shown in Figure.2, which is a close loop control system.

On Comparing sliding model optimization and magic formula optimization, results of both of them yield very close to optimal tracking of the desired trajectory. The optimal peak slip controlled within MF [6,7] model is known without error. However, MF optimization is strongly dependent on the road conditions because the required shape coefficients differ with road conditions change.
Figure 6. Longitudinal Force vs. Slip on Slippery Ice Surface

Figure 7. Lateral Force vs. Slip Angle in Cornering
4. Road Surface Correction of Magic Formula
A series of -slip profiles for the different road surfaces are presented in Figure.1 generated by the tire model with optimized system variables. Further, the sensitivity of the magic formula on
different road conditions is investigated for a selected tire. Thus, data of a tire on various road surfaces is measured. "The MF tire model was employed because it gives an amazing illustration of measured tire characteristics and since certain coefficients of the version hold a bodily significance, is therefore predicted for them to react to street surface variations in a meaningful manner". "In the outcomes of the measurements data on different avenue surfaces are offered as display in Table.1".

The results of lateral force measurements were tested "on laboratory as well as road surfaces". "The different road surfaces of "A", "B", "C" and "Lab" present the different lateral force distributions dependent to the wheel slip with the use of a same structure tire as described in the paper". "The measurement results show that parameters of $a_1$, $a_2$ and $a_3$ in the MF change significantly from surface to surface". "From all of above, also it was found that a tire model could be transferred from one surface to another by including two weighting parameters $C_1$, $C_2$ in the MF with respect to a theoretical road surface as follows":

$$D^* = C_1 D$$

$$(BCD)^* = C_2 BCD$$
where, \( D = a_1 F_z^2 + a_2 F_z \)

\[ \text{BCD} = a_3 \sin \left( 2 \tan^{-1} \left( \frac{F_z}{a_4} \right) \right) \left( 1 - a_5 \, | \gamma | \right) \]

The weighting parameters \( C_1, C_2 \) different\[8,9\] road surfaces are given as Table.2.
Further investigations, proved that the equal assumptions are also suitable for the longitudinal dynamics. In an empirical result, become proposed consequently, the parameters in MF can be written\cite{10,11} as a function of vertical load on the tire as

\begin{align*}
C &= 1.8 \\
D &= b_1 F_z^2 + b_2 F_z \\
B &= (b_3 F_z^2 + b_4 F_z) / (C D) (e^{b_5 F_z}) \\
E &= b_6 F_z^2 + b_7 F_z + b_8
\end{align*}
Figure 17. Estimated Error between Real and Estimated

Figure 18. Estimated Relative Velocity

The coefficients corresponding to a wet road that were used in the model and simulations, are given in Table.3. The braking force vs. slip is shown in Figure.3. The maximum braking
Figure 19. Estimated and Real Longitudinal velocity

Table 1. Lateral coefficients for different surfaces

| Surface Coefficients | Road A | Road B | Road C | Lab    |
|----------------------|--------|--------|--------|--------|
| $a_1$                | -43.707| -42.026| -39.925| -35.408|
| $a_2$                | 1329.305| 1280.406| 1215.734| 1086.310|
| $a_3$                | 1285.465| 1378.389| 1306.211| 1502.732|
| $a_4$                | 8.875  | 8.847  | 8.832  | 8.792  |
| $a_5$                | 0.017  | 0.016  | 0.016  | 0.014  |
| $a_6$                | -0.021 | -0.025 | -0.032 | -0.025 |
| $a_7$                | -0.513 | -0.428 | -0.378 | -0.451 |
| $a_8$                | -0.111 | -0.132 | -0.136 | -0.141 |
| $a_4t$               | 8.210  | 7.977  | 7.542  | 8.122  |

Table 2. Road Correction factors $C_1$ and $C_2$

| Surface | Road A | Road A | Road B | Road B | Road C | Road C |
|---------|--------|--------|--------|--------|--------|--------|
| Tire A  | $C_1$  | $C_2$  | $C_1$  | $C_2$  | $C_1$  | $C_2$  |
| Tire B  | 1.224  | 0.855  | 1.186  | 0.917  | 1.119  | 0.869  |

Table 3. Longitudinal coefficients for wet road

| $b_1$  | $b_2$  | $b_3$  | $b_4$  | $b_5$  | $b_6$  | $b_7$  | $b_8$  |
|--------|--------|--------|--------|--------|--------|--------|--------|
| -21.3  | 744.0  | 49.6   | 226.0  | 0.3    | -0.006 | 0.056  | 0.486  |
force corresponds to wet asphalt with fiction\cite{12,13} coefficient ranging from 0.4 to 0.5 when the normal loads equal to 2kN, 4kN, 6kN and 8kN.

5. Longitudinal Force for Straight Line Braking
The typical experimental test results for the straight line braking was given in Figure 3, Corresponding to this typical plot, the MF coefficients corrected to different road surfaces are

Table 4. Longitudinal coefficients for different surfaces

| Surfaces  | Dry Concrete | Wet Asphalt | Snow    | Slippery Ice |
|-----------|--------------|-------------|---------|--------------|
| $b_1$     | 33.015       | -21.3       | -6.56   | -3.28        |
| $b_2$     | 1153.2       | 744.0       | 229.152 | 114.576      |
| $b_3$     | 113.398      | 49.6        | 9.92    | 4.96         |
| $b_4$     | 516.593      | 226.0       | 45.2    | 22.6         |
| $b_5$     | 0.3          | 0.3         | 0.3     | 0.3          |
| $b_6$     | -0.006       | -0.006      | -0.006  | -0.006       |
| $b_7$     | -0.056       | -0.056      | -0.056  | -0.056       |
| $b_8$     | -0.486       | -0.486      | -0.486  | -0.486       |

Table 5. Correction factors for different surfaces

| Surfaces  | Dry Concrete | Wet Asphalt | Snow    | Slippery Ice |
|-----------|--------------|-------------|---------|--------------|
| $c_1$     | 1.55         | 1           | 0.308   | 0.154        |
| $c_2$     | 2.286        | 1           | 0.2     | 0.1          |

Figure 20. vehicle and wheel angular velocity
given in Table 4. Using Alleyne’s empirical results (Table 3), the two corrections factors $C_1$ and $C_2$ for different road [14,15] surfaces are results as in Table 5. Note that the large variation with lateral force of parameter $b_4$ is due to the large changing of $C_2$. 

Figure 21. Vehicle and Wheel Speed in Absence of Controller

Figure 22. Normalized Relative Slip in presence and absence of controller
Therefore, the braking force vs. slip ratio for the corresponding coefficients of the dry concrete, snow and slippery ice surfaces, are shown in Figure 3 to Figure 5, which are used in the simulation. From the results of Figure 4 to Figure 6, optimal slips are 5%-15% for dry concrete surface, 15%-35% for snow surface, 10%-30% for slippery ice surface and the road adhesion are around 1, 0.2 and 0.08, which are in good correlation with the typical experimental test results as shown in Figure 3.

6. Lateral and Longitudinal Forces while Turning
Of all the manoeuvres encountered in everyday driving, one of the most critical and most significant for motor vehicle mean is braking during cornering. The vehicle’s response pattern must represent the optimum compromise between steering response, stability and braking effectiveness[18,19]. Improvement of cornering performance with consideration of braking is important for automobile safety. Therefore, it is necessary to analyze vehicle dynamics and cornering characteristics with consideration of both friction and braking forces. From the typical experiment test results given in Figure 3 and based on Jagt’s empirical results (Table 1), the typical lateral coefficients for dry concrete normal road condition that are further used in this study are given in Table 6. Therefore, the normal lateral force vs. side slip angle corresponding to the correction factor coefficients further used in the braking system are shown in Figure 6, which also is good correlation in well with the typical experimental test results as shown in Figure 7.

Figure 23. Vehicle and wheel Speed
7. Simulation and Results

The block diagram of the control system is shown in Figure.8. The nonlinear tracking control law requires estimates of the value of friction coefficients and braking performance.

7.1. Block Diagrams of the Simulation System

The optimal ABS control system block diagrams and functions are described below. The controller is conceived as a system that\cite{22,24} identifies the design force in conjunction with the displacement of the actuator. The detailed description of the simulation system is given in Figure.9 to Figure.14. From the simulation results it is often ensured that an upright parameter trailing is obtained as long because the relative contact speed is totally different from zero. The nonlinear observer also successfully estimates the vehicle longitudinal velocity. Figure.8, shows the\cite{25,26} Simulink Model of vehicle model consisting of wheel and vehicle dynamics. Figure 15. Shows the Accelerating Torque, Figure 16, shows the Road Condition Parameter and its Estimation, Figure 17 shows the Estimated Error between Real and Estimated, Figure 18 shows the Estimated Relative Velocity and Figure 19 shows the Estimated and Real Longitudinal velocity, Figure 20 shows the vehicle and wheel angular velocity and the vehicle slip and wheel speed in absence of controller is depicted in Figure 21. Normalized Relative Slip in presence and absence of controller as shown in Figure 22 and Figure 23 shows the Vehicle and wheel Speed in presence of controller and the vehicle stopping distance in presence of bang bang controller and mode controller is shown in Figure.24.

8. Conclusion

The stopping distances of the proposed system are smaller than that when implementing without a control, mainly at high speed. The proposed system performances shown better representation for vehicle stopping distance and deceleration on dry roads. The reason is the proposed system is based upon the dynamic analysis of the hydraulic braking system, hence it is more stable and
exactly tracks the desired trajectory. From the simulation results it is able to be ensured that, an awesome parameter monitoring is obtained, so long as the relative touch velocity is not like zero. The nonlinear observer also successfully estimates the car longitudinal velocity.

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