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# A FUZZY LOGIC CONTROLLER DESIGN FOR VEHICLE ABS WITH A ON-LINE OPTIMIZED TARGET WHEEL SLIP RATIO

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**ABSTRACT**−For a vehicle Anti-lock Braking System (ABS), the control target is to maintain friction coefficients within maximum range to ensure minimum stopping distance and vehicle stability. But in order to achieve a directionally stable maneuver, tire side forces must be considered along with the braking friction. Focusing on combined braking and turning operation conditions, this paper presents a new control scheme for an ABS controller design, which calculates optimal target wheel slip ratio on-line based on vehicle dynamic states and prevailing road condition. A fuzzy logic approach is applied to maintain the optimal target slip ratio so that the best compromise between braking deceleration, stopping distance and direction stability performances can be obtained for the vehicle. The scheme is implemented using an 8-DOF nonlinear vehicle model and simulation tests were carried out in different conditions. The simulation results show that the proposed scheme is robust and effective. Compared with a fixed-slip ratio scheme, the stopping distance can be decreased with satisfactory directional control performance meanwhile.

**KEY WORDS :** Anti-lock braking system, Fuzzy logic controller, Wheel slip, Real-time, Optimal

## NOMENCLATURE

$a$ : distance between c.g. and front axle
$a_x$ : vehicle longitudinal acceleration
$a_y$ : vehicle lateral acceleration
$b$ : distance between c.g. and rear axle
$e$ : distance between c.g. and roll center
$h_g$ : height of c.g.
$f_0$ : rolling resistance coefficient
$p$ : brake pressure
$p_k$ : braking line pressure
$p_0$ : thresh hold value
$t_r$ : track width between rear wheels
$t_f$ : track width between front wheels
$A_f$ : front section area of vehicle
$C_d$ : air drag coefficient
$B_x$, $B_y$, $B_z$ : *Pacejka* model coefficients
$C_x$, $C_y$, $C_z$ : *Pacejka* model coefficients
$D_x$, $D_y$, $D_z$ : *Pacejka* tire model coefficients
$F_f$ : rolling resistance force
$F_x$ : tire longitudinal force in combined condition
$F_y$ : tire lateral force in combined condition
$F_{x0}$ : tire longitudinal force in single operation condition
$F_{y0}$ : tire lateral force in single operation condition
$F_w$ : aerodynamic force
$G_1$ : effective friction coefficient of braking contacts
$G_2$ : section area of braking cylinder
$G_3$ : average radius of friction contacts
$I_z$ : inertia of vehicle to $z$-axis
$I_x$ : inertia of vehicle to $x$-axis
$I_i$ : inertia of $i$th tire ($i$=1, 2, 3 and 4)
$K$ : constant in the calculation of peak value of $\lambda$
$K_{\dot{\phi}}$ : roll damper coefficient
$K_{\phi}$ : roll stiffness
$K_{mph}$ : converting coefficient in calculation for rolling resistance
$LF$ : thermal fraction reduction coefficient
$M$ : vehicle mass
$M_b$ : sprung mass
$M_x$ : moment for $x$-axis
$M_z$ : moment for $z$-axis
$V_{x0}$ : vehicle initial speed while braking
$V_x$ : vehicle longitudinal velocity
$V_y$ : vehicle lateral velocity
$T_{b_i}$ : braking torque
$T_{fi}$ : rolling resistant torque

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$R_r$ : wheel radius
$\gamma$ : yaw velocity
$\alpha$ : tire slip angle
$\lambda$ : slip ratio
$\sigma$, $\sigma_x$, $\sigma_y$ : distribution coefficients of tire longitudinal and lateral forces
$\phi_x$, $\phi_y$, $\phi_z$ : functions of tire normal force and slip angle
$\phi$ : roll angle
$\delta_1$, $\delta_2$: steering angle

## 1. INTRODUCTION

The safe operation of an automobile requires continuous adjusting of its speed to changing traffic conditions. With the increase of the speed and density of road vehicles, the safety performance is obviously demanded and Anti-lock Braking Systems (ABS) have been developed and now are used in more and more vehicles in recent decades (Si, 1996; Park *et al.*, 1999; Taheri, 1990; Bowman, 1993). Generally, for an ABS system, the control target is to maintain friction coefficients within maximum range in corresponding operating condition in order to obtain the maximum braking force from road surface, and meanwhile ensuring vehicle stability. However, in the condition of vehicle braking while turning, total tire traction available has to be shared by braking friction and side traction.

When braking is present during a steering maneuver, the tire side friction coefficient decreases with increasing values of tire slip, indicating that braking slip significantly reduces the side friction coefficient for a given slip angle. The typical tire side and braking friction coefficients $\mu$ as a function of braking slip $\lambda$ and slip angle $\alpha$ is shown in Figure 1. The figure shows the braking friction coefficient decreases with increasing values of slip angle. But the optimum slip at which tire peak friction occurs, i.e., $\lambda_{\text{peak}}$, increases with higher values of tire slip angle. Although the presented tire characteristics are elementary, they clearly indicate the complex relationships with braking and side force production of a tire. Hence, to ensure minimum stopping distance, directional stability and stable braking while turning, an optimal compromise between the different performance requirements must be needed (Park *et al.*, 1999). The optimal target slip ratio for ABS controller deigned should be on-line calculated considering that fact of complex road-tire characteristics and changeable braking conditions.

Tuheri (Taheri, 1990; Bowman, 1993) proposed two algorithms off adjusting the target longitudinal slip ratio accounting for the effects of lateral maneuvering, i.e., taking the desired slip ratio as function of steering angle and vehicle yaw rate respectively. Two empirical equations are respectively given to define the two algorithms respectively. However, the simulation tests showed poor robust performance and adaptive ability. The two algorithms worked well only for some particular conditions.

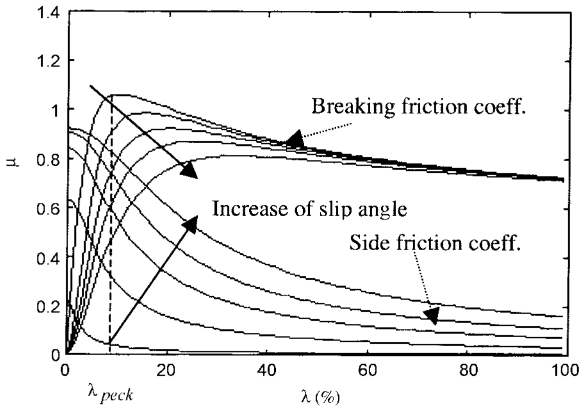


Figure 1. Effects of slip angle $\alpha$ and slip ratio $\lambda$ on coeff. μ.

Focusing on combined braking and turning operation conditions, this paper presents a new control scheme for an ABS controller design. The following issues were investigated.
(1) The scheme is implemented using an 8-DOF nonlinear vehicle model in Matlab/Simulink software environment, and dynamic simulation model is described.
(2) In different conditions, simulation tests were carried out and its feasibility is proved.
(3) Based on the previous researches, compared with a fixed-slip ratio scheme, the effectiveness of the new scheme is examined.

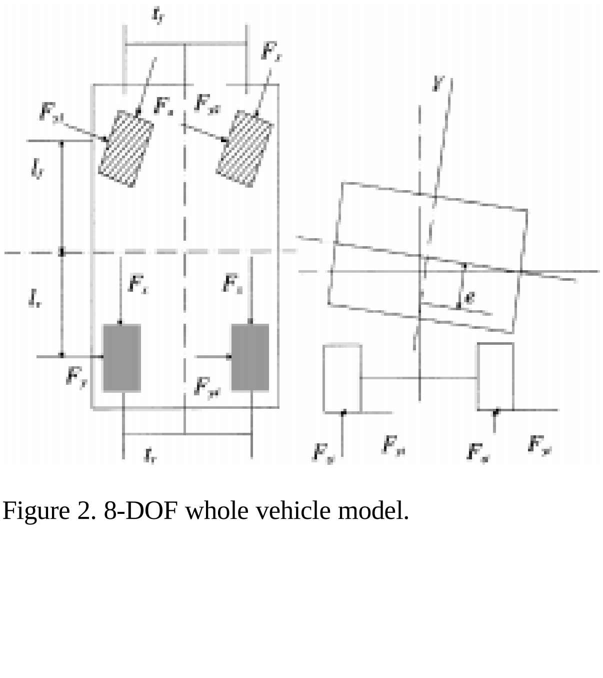

Figure 2. 8-DOF whole vehicle model.

## 2. VEHICLE SYSTEM MODEL

### 2.1. Vehicle Model

An 8-DOF whole vehicle model, shown in Figure 2, is used in the present study, which includes four vehicle body motion states, i.e., lateral displacement, longitudinal displacement, yaw and roll, and four degrees of freedom respectively for each wheel spin. Considering aerodynamic force and tire rolling resistance, the equation of motion are written as below,

$$-F_w - F_f - \sum F_x = (\dot{V}_x - V_y \cdot \gamma) \quad (1)$$

in which

$$F_w = \frac{1}{4} C_d A_f V_x^2$$

$$F_f = M_b \cdot g \cdot (f_0 + 3.24(K_{mph} V_x)^{2.5})$$

$$\sum F_y = M \cdot (\dot{V}_y + V_x \cdot \gamma) \quad (2)$$

$$\sum M_z = I_z \cdot \dot{\gamma} \quad (3)$$

$$\sum M_x = -K_\phi - K_{\dot{\phi}} \dot{\phi} + M_b \cdot g \cdot e \cdot \sin\phi = \quad (4)$$

$$I_x \ddot{\phi} - M_b \cdot e \cdot (\dot{V}_y + V_x \cdot \gamma)$$

$$I_i \cdot \dot{\omega}_i = F_{x_i} \cdot R_r - T_{b_i} - T_{f_i} \quad i=1,\,2,\,3,\,4 \quad (5)$$

Where the notations through the whole paper is presented in the nomenclature.

### 2.2. Tire Model

In this study, Pacejka's nonlinear tire model is used (Bakker *et al*.). Based on tire normal force, slip angle and slip ratio, the outputs can be respectively calculated which includes longitudinal force, lateral force and self-aligning moment.

$$F_x = \frac{\sigma_x}{\sigma} \cdot F_{x0} \quad (6)$$

$$F_y = \frac{\sigma_y}{\sigma} \cdot F_{y0} \quad (7)$$

$$M_{zz} = D_z \cdot \sin(C_z \tan^{-1}(B_z \cdot \phi_z)) \quad (8)$$

where

$$F_{x0} = D_z \cdot \sin(C_x \tan^{-1}(B_x \cdot \phi_x))$$

$$F_{y0} = D_y \cdot \sin(C_y \tan^{-1}(B_y \cdot \phi_y))$$

$$\sigma = \sqrt{\sigma_x + \sigma_y} \quad \sigma_x = -\frac{\lambda}{1+\lambda} \quad \sigma_y = -\frac{\tan\alpha}{1+\lambda}$$

During a vehicle steering maneuver, tire slip angle can be estimated by the following equations, i.e.,

$$\alpha_1 = \delta_1 - \tan^{-1}\left(\frac{V_y + a \cdot \gamma}{V_x + t_f \cdot \gamma}\right) \quad (9)$$

$$\alpha_2 = \delta_2 - \tan^{-1}\left(\frac{V_y + a \cdot \gamma}{V_x + t_f \cdot \gamma}\right) \quad (10)$$

$$\alpha_3 = -\tan^{-1}\left(\frac{V_y - b \cdot \gamma}{V_x + t_r \cdot \gamma}\right) \quad (11)$$

$$\alpha_4 = -\tan^{-1}\left(\frac{V_y - b \cdot \gamma}{V_x + t_r \cdot \gamma}\right) \quad (12)$$

### 2.3. Brake Model

Taking a disk brake as example, braking torque can be calculated as,

$$T_b = \begin{cases} 0 \quad \text{when} \quad p - p_0 < 0 \\ \frac{1}{6}(p - p_0) \cdot G_1 \cdot G_2 \cdot G_3 \cdot (LF) \quad \text{when} \quad p - p_0 \geq 0 \end{cases} \quad (13)$$

## 3. OPTIMAL TARGET SLIP RATIO

It is obvious that the target slip ratio should be control corresponding range of maximum braking traction coefficient to obtain maximum braking force from road. In the case without tire slip, i.e., in zeroled within value of tire slip angle condition, the range of maximum braking traction coefficient is about 8~10% as shown in Figure 1. However, in combined braking and turning condition, when tire slip angle increases, the range of slip corresponding to maximum braking traction coefficient changes. From Figure 1, it can be seen that peak $\mu_b$ moves to left as slip angle $\alpha$ increases and the relationship between the both appears non-linear. In straight line braking conditions, a fixed target slip ratio can normally be used. However, in combined braking and turning cases, if the fixed target value is still used, the resulting braking performances will not be optimal. Hence, the potential improvement for decreasing stopping distance may be lost.

In this study, the proposed control strategy is to respectively calculate practical wheel slip and optimal target slip ratio based on sensed information on present vehicle motion states firstly, and then based on the calculations to modulate braking line pressure via solenoid to keep the actual slip around target value as possible. Therefore, the real time calculation of target slip ratio is one of the most important steps to ensure the control objective. The target slip ratio should be chosen with higher friction value particularly at larger slip angle.

A simple approach can be used to implement the scheme. Based on road-tire characteristics a fitting curve can be obtained to describe the relationship between the peak values of braking traction coefficient $\mu_{peak}$ and tire slip angle $\alpha$. However, this implementation is based on the idea of minimizing stopping distance without considering side stability performance and therefore not suitable for practical application. Hence, in order to ensure an overall vehicle performance, i.e., considering vehicle side stability, control error, vehicle motion states, the changes of tire characteristics and road surface condition, etc., an optimal target slip ratio should be real time calculated, using the equation as below (Taheri 1990; Bowman, 1993),

$$\lambda_{target}=\frac{K \cdot a_y \cdot g}{a_x \cdot V_{x0}^2} \tag{14}$$

The equation above describes the relationship between target slip ratio and tire slip angle in braking while turning process. The term $a_x/g$ represents road surface condition, $a_y$ accounts for the effect of side stability and $V_{x0}$ for the effect of initial braking speed. Simulation tests show the approach is feasible and can adapt road condition satisfactorily; the best compromise can be obtained between braking distance and side stability.

Since the previous studies (Bowman, 1993) showed that the adaptation of rear wheel slip ratio is not significant to vehicle performances, hence, in order to maintain the side force of rear wheels and side stability, a fixed target slip ratio is still used for rear wheel control, while the real-time optimal target slip ratio is only used for front wheels.

## 4. FUZZY LOGIC CONTROLLER DESIGN

The control objective for the ABS controller design is to maintain wheel slip ratio as close as possible to the optimal target value to minimize stopping distance meanwhile ensuring side stability within acceptable range. Since slip ratio is continuous variable, many control theories can be used for its controller design, such as conventional PID or optimal control, etc. Although a PID controller is practical and simple to be implemented, its system parameters need to be adapted corresponding to changing vehicle operation conditions. And the complexity of vehicle system, particular for tire non-linearity, could result in the difficulties in the parameter adaptation. On the contrary, a fuzzy logic controller can easily adapt to the complex, changeable operation condition and non-linearity existing in the vehicle suspension system and tires, also with good robust performance (Bauer and Tomizuka 1996; Li *et al.* 2001, Akey 1995). The scheme for the ABS fuzzy logic controller is shown in Figure 3.

The inputs of the fuzzy controller are the difference between actual slip value and target value, i.e., $e_{error}$, and the changing rate $d\lambda_{error}/dt$, which are respectively described as (Li, 1996),

$$\lambda_{error}=\lambda-\lambda_{target}$$

$$\frac{d\lambda_{error}}{dt} \approx \frac{\Delta\lambda_{error}}{\Delta t} \tag{15}$$

The output is the expected difference of braking pressure $\Delta p$. Hence the adapted pressure is: $p_{k+1}=p_k+\Delta p$ $k=1, 2, 3, ..., n$.

The all membership functions of the input language variables and the output language variable are taken as triangle format, and typically shown in Figure 4.

The control rules are described for the ABS fuzzy logic control system based on optimal target slip ratio, shown in Table 1.

Here PB denotes Positive Big, NS for Negative Small and ZO for Zero, etc. In this study the *Mandain* algorithm is used for fuzzy logic operation and the *Gravity* algorithm is used for anti- fuzzilization calculation (Li, 1996).

## 5. SIMULATION RESULTS

The fuzzy control simulation system is implemented in

Table 1. Fuzzy rules for ABS controller.

| $\Delta p$ | | $\lambda_{error}$ | | | | |
|---|---|---|---|---|---|---|
| | | PB | PS | ZO | NS | NB |
| $\frac{d\lambda_{error}}{dt}$ | PB | *NB* | *NB* | *NB* | *NS* | *ZO* |
| | PS | *NB* | *NB* | *NS* | *PS* | *PS* |
| | ZO | *NB* | *NS* | *ZO* | *PS* | *PB* |
| | NS | *NS* | *NS* | *ZO* | *PB* | *PB* |
| | NB | *NS* | *NS* | *PS* | *PB* | *PB* |

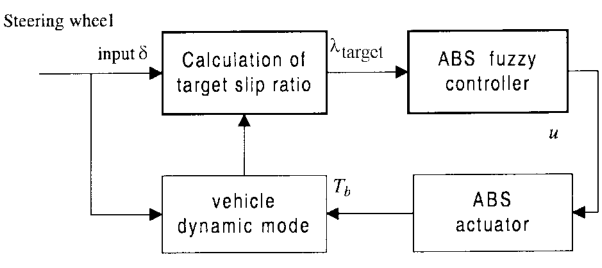


Figure 3. Fuzzy logic control system based on optimal target slip ratio.

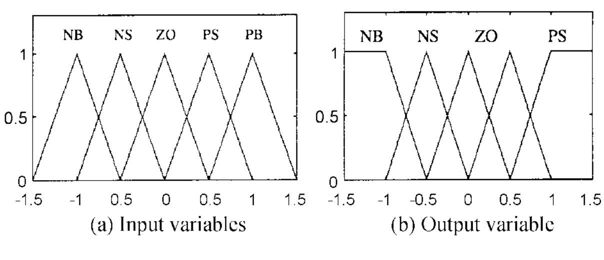


Figure 4. Membership functions of input/output anguage variables.

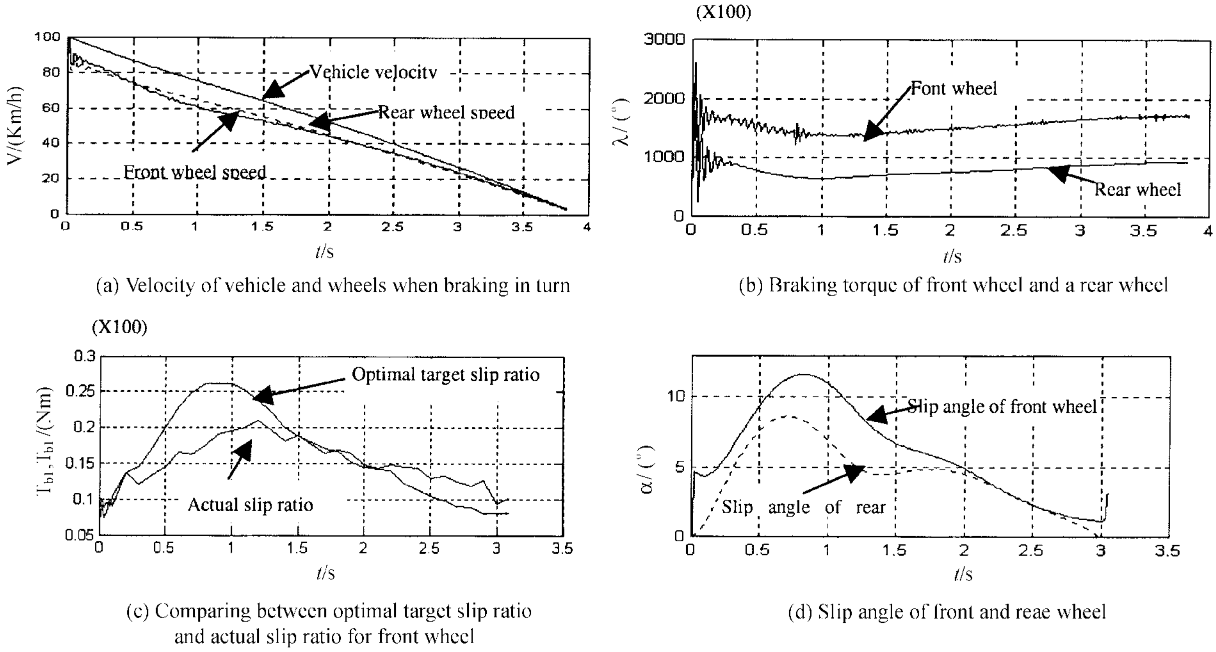


Figure 5. Braking process in a given road condition with steering wheel step input of 5 degrees.

the environment of Maltlab/Simulink software (Natick, 1990), and the main program mainly includes ABS fuzzy controller module, hydraulic and actuation system module, non-linear whole vehicle module. With the data file of a tested vehicle loaded, simulation testes can be carried out for the whole braking process of the vehicle. A monitor, i.e., Scope, is designed to graphically demonstrate the output results for each subsystem in real-time.

Some typical simulation results are presented in this paper. Figure 5 shows the performances in time domain in the simulation condition of a braking maneuver with a steering wheel step input of 5 degrees at braking start. The performances include vehicle speed and wheel spin speeds presented in Figure 5a, braking line pressure in Figure 5b, and optimal target slip ratio in Figure 5c, slip angles of front and rear wheels in Figure 5d. From the figures, it can be seen that, with the steering input, the slip angles for the front wheels and rear wheels begin to increase, and therefore the resulted front wheel target slip ratio increases.

Because of the significant of reduction of braking friction coefficient, braking pressure becomes less. And with the reduction of tire slip angle, the real-time calculated optimal target slip ratio decreases. This leads to a rise of braking friction coefficient, thus resultant braking pressure regains. Because of the employed fixed slip ratio control, the braking pressure of rear wheels is modulated less frequently than that of front wheels.

Figure 6 demonstrates the trajectory of the center of gravity (c.g) of the vehicle during braking process with different steering inputs and on different road surfaces. In the simulations, the vehicle speed at the beginning of braking is 100 kilometer/hour. Figure 6a shows the comparison of the c.g. trajectory between optimal target slip ratio and fixed slip ratio schemes in a high friction road surface condition with a steering input of 5 degrees when braking start. The significant improvements can be seen for stopping distance, which is reduced from 55.66 meters to 49.82 meters compared with fixed slip ratio results. Although the side stability is slightly worse, it is still in the acceptable range. Figures 6b, 6c, and 6d show the results for different steering inputs and in different road surface conditions. Similar significant benefits for stopping distance are obtained with the reduction about up to 15%, indicating that the overall performance of the optimal target slip ratio control is better than the fixed ratio scheme.

Simulations in different conditions also show the robustness of the system. The resulted changing frequency of braking line pressure become less by using the fuzzy logic controller, and is also beneficial for actuation mechanism, with longer use life, etc.

## 6. CONCLUSIONS

This paper proposed a new control scheme for an ABS controller design based on real-time optimal target wheel slip ratio. A fuzzy logic controller was designed to

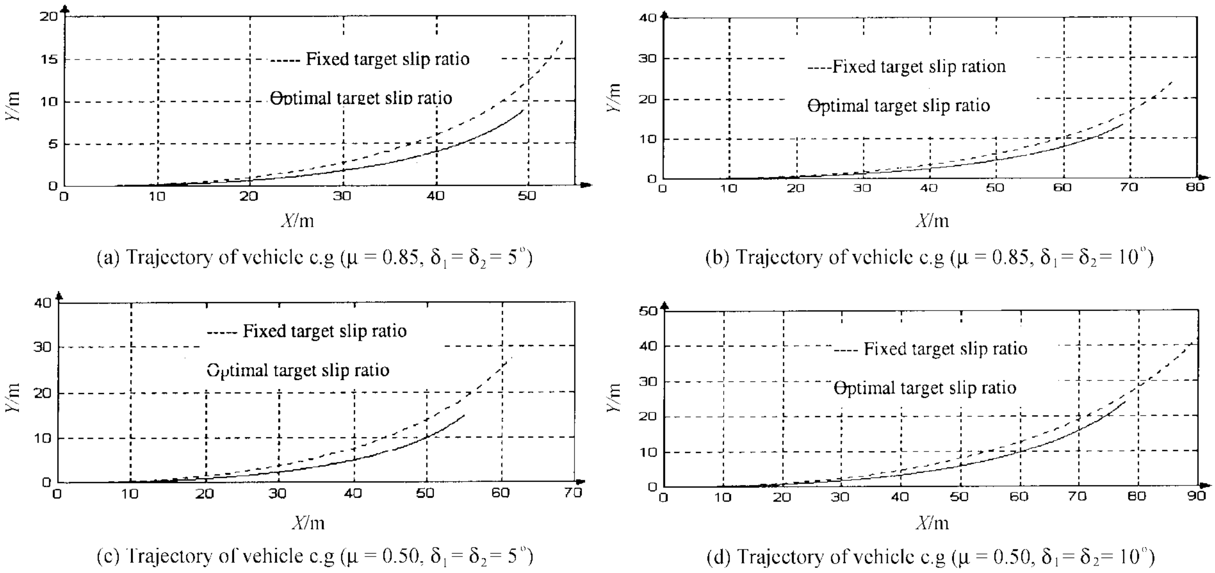


Figure 6. Simulation result comparison between the optimal target slip ratio scheme and fixed target scheme.

maintain the optimal target slip ratio so that the best compromise between braking deceleration, side stability and direction control performances could be obtained for the vehicle. The scheme was implemented using an 8-DOF nonlinear vehicle model in Matlab/Simulink software environment, and simulation tests were carried out in different conditions.

The simulation results showed that the proposed optimal target slip ratio control is a feasible control scheme. Its robustness is proved by simulation results. The best compromise between braking stopping distance and direction stability performances for the vehicle can be obtained. Compared with a fixed-slip ratio scheme, the stopping distance can be decreased significantly, up to 15%. The controller also showed good adaptive ability to the changes in variable road friction conditions. The idea based on real-time optimal target slip ratio control can be used in traction control (TCS) as well and the further work on TCS /ABS fuzzy control system is being carried out.

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