Research Article

Antisideslip and Antirollover Safety Speed Controller Design for Vehicle on Curved Road

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When the drivers cannot be aware of the existing of forthcoming curved roads and fail to regulate their safety speeds accordingly, sideslip or rollover may occur with high probability. The antisideslip and antirollover control of vehicle on curved road in automatic highway systems is studied. The safety speed warning system is set before entering the curved road firstly. The speed adhesion control is adopted to shorten the braking distance while decelerating and to guarantee the safety speed. The velocity controller when decelerating on the straight path and the posture controller when driving on curved road are designed, respectively, utilizing integral backstepping technology. Simulation results demonstrate that this control system is characterized by quick and precise tracking and global stability. Consequently, it is able to avoid the dangerous operating conditions, such as sideslip and rollover, and guarantee the safety and directional stability when driving on curved road.

1. Introduction

The amount of traffic accidents taking place on curved roads is obviously bigger than that on the straight roads per kilometer, especially when the drivers cannot be aware of the existing or forthcoming curved roads and fail to regulate their safety speeds. According to NHTSA's 2009 Fatality Analysis Reporting System (FARS), the majority of vehicles in single- and two-vehicle crashes were going straight prior to the crash, while the next most common maneuver was negotiating a curve, which resulted in 14.2% of total fatalities. The Road Traffic Accident Annual Census Report of China Ministry of Public Security reported that more than 65,225 people died in at least 219,521 accident cases in 2010. The statistics results on different road alignments indicated that the fatalities on general curved roads accounted for 7.6% of total fatalities, ranking only second to the straight roads.

The current advanced safety driver assistance system (ADAS) provides a good solution to curve road safety by increasing the comfort and safety of traffic participants by sensing the environment, analyzing the situation, and signaling relevant information to the driver [1]. The ultimate

target of such systems is to drive a vehicle automatically, such as intelligent vehicle [2]. As to the curve road assistance, both the realization of curve antisideslip or antirollover speed warning and curve lane keeping are key technologies. There are several systems suggested to reduce the danger of unexpected accidents and to warn the driver of driving under the appropriate speed. Sung et al. [3] converged telematics and wireless sensor networks technologies to develop a curved road collision warning system. Moon et al. [4] described the design, tuning, and evaluation of a full-range adaptive cruise control (ACC) system with collision avoidance (CA). To provide more highly detailed and precise maps for ADAS, Jiménez et al. [5] presented an on-board inertial system based on a speed sensor and a gyroscopic platform. They evaluated measurement uncertainty by quantifying the accumulative error of the system and segmented the road into sections with homogeneous geometric properties to fit geometric curves with the experimental data. Warner and Åberg [6] compared different intelligent speed adaptation (ISA) systems. When driving on curved roads besides highways, the motorcyclists endure the same or even more serious danger than the vehicle driver. In accordance with the identified need to support

riders in safely negotiating curves, the curve warning (CW) system has been designed to detect incorrect, insufficient, or missing rider actions with regard to the longitudinal control of the vehicle when approaching a curve [7, 8].

Commonly occurring accidents on curved roads are sideslip or rollover. The presence of horizontal curves is associated with increased probabilities of high-severity outcomes in a median rollover crash [9], particularly those curved to the right [10]. So far, several national and international researches on antisideslip or antirollover speed warning and lane keeping concerning curved roads have attained valuable achievements. For example, Chen and Peng [11] proposed an antirollover control algorithm based on the time-to-rollover (TTR) metric, during which the steering and direct yaw moment control inputs were constructed to calculate the TTR in real-time. Lewis and El-Gindy [12] investigated rollover prevention strategies for a truck/semitrailer combination based on a sliding mode controller and differential braking. Lee and Deng [13] designed a speed control system to assist the driver achieving safer and more comfortable curve following performance. They determined a desired curve speed profile for the incoming curve according to a map database and regulated the vehicle speed toward the desired speed. Chiu et al. [14] presented a robust controller design methodology for vehicle rollover prevention utilizing active steering. Their controller was based on keeping the magnitude of the vehicle load transfer ratio (LTR) below a certain level in the presence of driver steering inputs, which can reduce the transient magnitude of the LTR while maintaining the steady state steering response of the vehicle.

Taking into account the real-time requirement for antisideslip or antirollover and the vehicle nonholonomic constraints, this paper proposes a new antisideslip and antirollover controller on the curved road with safety speed for automatic highway systems. The organization of this paper is as follows. The design scheme of the curve antisideslip and antirollover automatic control system is described in Section 2. Section 3 designs the velocity controller during the braking deceleration phase and the trajectory tracking controller during curve lane keeping phase utilizing backstepping technology. Several simulations are conducted to verify the effectiveness and robustness of the control methods in Section 4. Finally, Section 5 draws some conclusions and summarizes future work.

2. System Overview

The curved road antisideslip and antirollover automatic control system is shown in Figure 1, which includes four modules, such as curvature recognition, safe speed calculation, safe status judgment, and automatic control system.

The function of the curvature recognition module is to realize the real-time curvature detection of the curved road ahead. There are mainly two curvature detection methods at present, among which, one is to estimate the road curvature according to road coordinates obtained by the on-board GPS/GIS [15], and another is to calculate the road curvature based on the road model fitted by image processing

technology [16]. This paper adopts the second method to get the curvature; a detailed algorithm is described in [17]. The safe speed calculation module is used to derive the critical speed for vehicle negotiating the curve safely, which satisfied the sideslip speed restraint and the rollover speed restraint on the curved road. Just as its name implies, the safe status judgment module is to judge the current safe state of the automatic vehicle according to its current movement. Finally, the automatic control system is to regulate and control the vehicle running states appropriately. It includes two parts, as shown in Figure 1; dashed line block (1) indicates the schematic of velocity controller when driving on a straight path and block (2) describes the schematic of posture control system structure for following curve road path.

2.1. Safe Speed Calculation. When the speed exceeds a certain value while driving on the curve with a constant yaw angle like lane keeping or a variable yaw angle like lane changing, road adhesion force could not overcome the influence of centrifugal force, so the rear axle of the automatic vehicle can easily sideslip. Meanwhile, the body can easily roll, even rollover, because the normal load on each wheel changes greatly, which is caused by the transfer of both longitudinal and lateral mass [18].

When driving on a curved road, the condition for no sideslip is that the centrifugal force should not be more than the road adhesion force, which is formulated as follows:

$$F \leq F_x, \quad m \frac{v^2}{R} \leq \varphi mg, \quad (1)$$

and it is equal to

$$v \leq \sqrt{\varphi g R}, \quad (2)$$

where F is defined as centrifugal force, F_x indicates road adhesion force, m is vehicle mass, g is the gravity acceleration, φ denotes lateral adhesion coefficient, v represents vehicle speed, and R indicates the curvature radius of the curve, which is set as the turning radius of the vehicle in a general sense. When (2) is to be equal, it means the critical state of automobile sideslip.

If the heeling moment is lower than the aligning torque on curved road, the vehicle will not roll over, which can be described in the formula below as follows:

$$\begin{aligned} \sum M_\varphi &\leq \sum T, \\ m \frac{v^2}{R} h &\leq mg \frac{b}{2}, \end{aligned} \quad (3)$$

and it is equivalent to

$$v \leq \sqrt{\frac{b}{2h} g R}, \quad (4)$$

where $\sum M_\varphi$ denotes the heeling moment caused by centrifugal force, $\sum T$ indicates aligning torque, h is the height of vehicle centroid, b represents track width, and other

FIGURE 1: Schematic of curve antisideslip and antirollover automatic control system.

parameters are the same as above. It represents the critical state of rollover with the equality of (3).

In order to guarantee the safety when driving on curved road and to avoid the dangerous operating conditions, sideslip and rollover, the speed should satisfy those two constraints, which is to say

$$v \leq \min \left[\sqrt{\varphi g R}, \sqrt{\frac{b}{2h} g R} \right], \quad (5)$$

where v denotes the critical safe speed if (5) is to be equal.

If a given vehicle drives around a specific curve, vehicle parameters m , b , and h and road parameters R are all constants, so the safe speed is only related to the lateral adhesion coefficient φ . If the speed has already exceeded the critical safe speed before turning, the automatic control system will make the vehicle brake and reduce the speed to a safe speed; then the vehicle will drive on the curved road safely, which could avoid traffic accidents. Therefore, enough distance before turning is needed for braking and deceleration, which requires it to determine braking deceleration accurately according to braking distance on the straight path.

2.2. Braking Deceleration Planning. In the whole process of braking and deceleration, the ideal deceleration is continuously increasing while in the high-speed section, and the ideal deceleration will not change until the speed reduces to safe speeds while in the low-speed section in order to restrict maximum deceleration, which is shown in Figure 2. This

FIGURE 2: Braking deceleration module.

braking and decelerating module is called speed adhesion control [19], which could shorten the braking distance while decelerating and could guarantee the speed is reduced to safe speed driving on curved road on the straight path.

The longitudinal braking deceleration $\ddot{x}$ is shown below according to the braking deceleration module above;

$$\ddot{x} = \begin{cases} -\frac{5}{36}t^2 & 0 < t \leq t_1 \\ -5 & t_1 < t \leq t_2, \end{cases} \quad (6)$$

where $0 \sim t_1$ indicates the high-speed section and $t_1 \sim t_2$ denotes the low-speed section.

Integrating (6) yields

$$\dot{x} = \begin{cases} v_0 - \frac{5}{108}t^3 & 0 < t \leq t_1 \\ v_0 - \frac{5}{108}t_1^3 - 5(t - t_1) & t_1 < t \leq t_2, \end{cases} \quad (7)$$

where $0 \sim t_1$ corresponds to high-speed section, $t_1 \sim t_2$ corresponds to low-speed section, and v_0 is the initial value of vehicle longitudinal speed.

3. Controller Design

3.1. Model Description. Due to the directionality of the vehicle motion, the vehicle's position and orientation are described by two independent coordinates, which are world coordinate system (WCS) Oxy and local coordinate system (LCS) Mij . This paper takes the center of the vehicle rear driving wheels as the origin of LCS and the front wheels conduct the steering ability. Regarding the vehicle as a rigid body and neglecting the effect of suspension, position, and orientation, namely the posture of the vehicle, is shown in Figure 3.

This paper focuses on designing a safety speed controller and proving its stability and makes some simplifications. If the vehicle is doing the motion of pure rolling and there is no relative sliding between wheels when driving, then the vehicle nonholonomic constraint can be described as

$$\dot{y} \cos \theta - \dot{x} \sin \theta = 0. \quad (8)$$

Assuming that $p_r = [x_r \ y_r \ \theta_r]^T \in R^3$ is the vehicle's reference posture, $N(x_r \ y_r)$ indicates the vehicle's reference position and θ_r denotes its reference moving orientation. Taking $p_c = [x_c \ y_c \ \theta_c]^T \in R^3$ as the vehicle's current posture and θ_c as the vehicle's current moving orientation along the x -axis anticlockwise, namely, it is the angle between the coordinate system of WCS and LCS. The relationship between the reference posture and the current posture is

$$\dot{p}_c = \begin{bmatrix} \dot{x}_c \\ \dot{y}_c \\ \dot{\theta}_c \end{bmatrix} = \begin{bmatrix} \cos \theta_c & 0 \\ \sin \theta_c & 0 \\ 0 & 1 \end{bmatrix} \begin{bmatrix} v_c \\ \omega_c \end{bmatrix} = J u_c, \quad (9)$$

where $J(p) \in R^{3 \times 2}$ and $u_c = [v_c \ \omega_c]^T \in R^{2 \times 1}$ are the velocity transformation matrix and velocity vector, respectively. Here, $v_c \in R$ denotes the linear velocity of the mass center and $\omega_c \in R$ indicates the angular velocity of the vehicle.

In order to track a given trajectory smoothly, a path must be computed from a given initial location and heading to some target point on the desired trajectory. A tracking error function is generally defined by a vector between the predictive reference vector and a controlled vehicle traveling vector. The error functions are a velocity equation that is calculated by current posture, which is derived from the velocity equation [20]. Therefore, the relationship between control vector and the posture error should be explained. To achieve the tracking performance of $p_c \rightarrow p_r$ when $t \rightarrow \infty$,

FIGURE 3: Nonholonomic vehicle model.

the tracking error vector $p_e = [x_e \ y_e \ \theta_e]^T \in R^3$ in LCS can be explained as follows:

$$p_e = \begin{bmatrix} x_e \\ y_e \\ \theta_e \end{bmatrix}_{(Mij)} = \begin{bmatrix} \cos \theta_c & \sin \theta_c & 0 \\ -\sin \theta_c & \cos \theta_c & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x_r - x_c \\ y_r - y_c \\ \theta_r - \theta_c \end{bmatrix}_{(Oxy)} = J_1(p_r - p_c)_{(Oxy)}, \quad (10)$$

where (x_e, y_e) is a phasor coordinate of MN in LCS and J_1 denotes the posture error transfer matrix, which transfers the posture error from WCS to LCS. Obviously, the tracking error vector $(x_e, y_e, \theta_e) = 0$ if and only if $(x, y, \theta) = (x_r, y_r, \theta_r)$.

Differentiating the above tracking error (10) and substituting $\dot{p}_c$ by (9), the differential of the vehicle posture error can be derived as

$$\dot{p}_e = \begin{bmatrix} \dot{x}_e \\ \dot{y}_e \\ \dot{\theta}_e \end{bmatrix} = \begin{bmatrix} \omega_c y_e - v_c + v_r \cos \theta_e \\ -\omega_c x_e + v_r \sin \theta_e \\ \omega_r - \omega_c \end{bmatrix}, \quad (11)$$

where v_r, ω_r are reference linear and angular velocities of the vehicle, respectively.

3.2. Controller Design. In order to avoid the dangerous operating conditions, sideslip and rollover, and to guarantee safety when driving on curved road, the desired safe running state should be tracked not only when decelerating on the straight path but also when driving on the curved road. The nonlinear control method based on smooth static feedback could not reach the requirement of balance and stabilization because the vehicle has the nature of nonholonomic constraints [21]. So a new nonlinear system stability design theory called integral backstepping, which is a system recursive design method that need not realize the linearization of the nonlinear system, is applied to derive tracking controllers which have characteristics of global stability [22, 23]. Therefore, the issues of trajectory tracking when decelerating on the straight path and driving on curved road based on the system kinematics model could be transformed to find a bounded

control input $u_c = [v_c \ \omega_c]^T$, such that the tracking error vector $p_e = [x_e \ y_e \ \theta_e]^T$ is abounded and satisfies

$$\lim_{t \rightarrow \infty} [|x_e(t)| + |y_e(t)| + |\theta_e(t)|] = 0. \quad (12)$$

Above all, the posture controller is derived via the integral backstepping technology based on the vehicle kinematics model in this paper. In the tracking error model (11), the lateral position error y_e cannot be directly controlled, so we define a new variable $\bar{x}_e$ as follows:

$$\bar{x}_e = x_e - k_1 \sin(\arctan(\delta\omega_c)) y_e, \quad (13)$$

where k_1 and δ are all positive constants and $k_1 \sin(\arctan(\delta\omega_c))$ is the virtual feedback parameter.

$$u_c = \begin{bmatrix} v_c \\ \omega_c \end{bmatrix} = \begin{bmatrix} v_r \cos \theta_e - k_1 \cos(\arctan(\delta\omega_c)) \frac{\delta}{1 + \delta^2 \omega_c^2} \dot{\omega}_c y_e + k_1 \omega_c \sin(\arctan(\delta\omega_c)) x_e \\ -k_1 v_r \sin(\arctan(\delta\omega_c)) \sin \theta_e + k_2 x_e - k_1 k_2 \sin(\arctan(\delta\omega_c)) y_e \\ \omega_r + 2k_3 v_r y_e \cos \frac{\theta_e}{2} + k_4 \sin \frac{\theta_e}{2} \end{bmatrix}, \quad (17)$$

where k_1, k_2, k_3, k_4 are all positive constants whose value could determine the control directly.

3.3. Stability Analysis. Note that for $\forall x \in R$ and $|x| < \infty$, $\varphi(x) = x \sin(\arctan x) \geq 0$ and $\varphi(x) = 0$ if and only if $x = 0$. Obviously, $\dot{V}_y \leq 0$. By the Lyapunov stability theorem, system (15) will be asymptotically stabilized as $t \rightarrow 0$. If $v_r, \dot{v}_r$ and $\omega_r, \dot{\omega}_r$ are bounded for $\forall t \in (0, +\infty)$, as well as v_r and ω_r are equal to zero with asynchronism, the tracking controller (17) will make (11) globally stable; meanwhile, $\lim_{t \rightarrow \infty} [|x_e(t)| + |y_e(t)| + |\theta_e(t)|] = 0$ as $v_r \neq 0$. To analyze the stability of the controller, a candidate Lyapunov function is as follows:

$$V = \frac{1}{2} \bar{x}_e^2 + \frac{1}{2} y_e^2 + \frac{2}{k_3} \left(1 - \cos \frac{\theta_e}{2}\right). \quad (18)$$

Obviously $V \geq 0$ due to k_3 being a positive constant. The Lyapunov function $V = 0$ ($\theta_e \in [-\pi, \pi]$) if and only if $(\bar{x}_e \ y_e \ \theta_e)^T = 0$. Taking the time derivation of V along the trajectory of (11), (13), and (14) yields

$$\dot{V} = \bar{x}_e \dot{\bar{x}}_e + y_e \dot{y}_e + \frac{1}{k_3} \sin \frac{\theta_e}{2} \dot{\theta}_e$$

Differentiating (13) yields

$$\begin{aligned} \dot{\bar{x}}_e &= \dot{x}_e - k_1 \cos(\arctan(\delta\omega_c)) \frac{\delta}{1 + \delta^2 \omega_c^2} \dot{\omega}_c y_e \\ &\quad - k_1 \sin(\arctan(\delta\omega_c)) \dot{y}_e. \end{aligned} \quad (14)$$

In this case, when $x_e \rightarrow k_1 \sin(\arctan(\delta\omega_c)) y_e$ and $\theta_e \rightarrow 0$, according to system (8), it can be obtained that

$$\dot{y}_e = -x_e \omega_c = -k_1 \omega_c \sin(\arctan(\delta\omega_c)) y_e. \quad (15)$$

Supposing a Lyapunov function $V_y = (1/2) y_e^2$ and differentiating this Lyapunov function, we get

$$\dot{V}_y = y_e \dot{y}_e = -k_1 \omega_c \sin(\arctan(\delta\omega_c)) y_e^2. \quad (16)$$

According to the proposed preliminary, the purpose of tracking controller design is to find an appropriated control input $u_c = [v_c \ \omega_c]^T$, such that $x_e \rightarrow k_1 \sin(\arctan(\delta\omega_c)) y_e$ and $\theta_e \rightarrow 0$, so a theorem can be given as follows:

$$\begin{aligned} &= \bar{x}_e \left(\dot{x}_e - k_1 \cos(\arctan(\delta\omega_c)) \frac{\delta}{1 + \delta^2 \omega_c^2} \dot{\omega}_c y_e \right. \\ &\quad \left. - k_1 \sin(\arctan(\delta\omega_c)) \dot{y}_e \right) \\ &\quad + y_e (-\omega_c x_e + v_r \sin \theta_e) \\ &\quad + \frac{1}{k_3} \sin \frac{\theta_e}{2} (\omega_r - \omega_c) \\ &= \bar{x}_e \left[(\omega_c y_e - v_c + v_r \cos \theta_e) \right. \\ &\quad \left. - k_1 \cos(\arctan(\delta\omega_c)) \frac{\delta}{1 + \delta^2 \omega_c^2} \dot{\omega}_c y_e \right. \\ &\quad \left. - k_1 \sin(\arctan(\delta\omega_c)) \dot{y}_e \right] \\ &\quad + y_e [(-\omega_c (\bar{x}_e + k_1 \sin(\arctan(\delta\omega_c)) y_e) \\ &\quad + v_r \sin \theta_e)] + \frac{1}{k_3} \sin \frac{\theta_e}{2} (\omega_r - \omega_c) \end{aligned}$$

FIGURE 4: Deceleration curve and trajectory tracking simulation curves with different φ .

$$= \bar{x}_e [v_r \cos \theta_e - v_c$$

$$- k_1 \cos(\arctan(\delta \omega_c)) \frac{\delta}{1 + \delta^2 \omega_c^2} \dot{\omega}_c y_e$$

$$- k_1 \sin(\arctan(\delta \omega_c)) (-\omega_c x_e + v_r \sin \theta_e)]$$

$$- k_1 \sin(\arctan(\delta \omega_c)) y_e^2$$

$$+ \frac{1}{k_3} \sin \frac{\theta_e}{2} \left[(\omega_r - \omega_c) + 2k_3 v_r y_e \cos \frac{\theta_e}{2} \right]. \quad (19)$$

Substituting the control law (17) into (19) indicates that

$$\dot{V} = -k_2 \bar{x}_e^2 - k_1 \frac{2\delta \omega_c^2}{1 + \omega_c^2} y_e^2 - \frac{k_4}{k_3} \sin^2 \frac{\theta_e}{2}. \quad (20)$$

Noting that $\dot{V}$ is a negative semidefinite and uniformly continuous function and k_1, k_2, k_3, k_4 , and δ are all positive constants, obviously $\dot{V} \leq 0$. According to Lyapunov's stability criterion [24] and Barbalat's lemma [25], it can be concluded that tracking errors $p_e = [x_e \ y_e \ \theta_e]^T$ is globally, uniformly, and ultimately bounded. That is to say $\lim_{t \rightarrow \infty} [|x_e(t)| + |y_e(t)| + |\theta_e(t)|] = 0$ under controller (17). Therefore, both the

FIGURE 5: Trajectory tracking errors with different φ .

velocity controller when decelerating on the straight path and the posture controller have characteristics of global stability.

4. Simulation Results

In order to verify the effectiveness of the control method above, massive simulation experiments are performed. The simulation environment is based on Matlab/Simulink and the sampling time is 0.14 s. Assuming that the basic parameters of the automatic vehicle are as follows, $b = 3.2$ m and $h = 0.9$ m, the gravity acceleration $g = 9.8$ m/s². The numerical value of

the lateral adhesion coefficient φ depends on many factors, such as road material, road condition, the tire's construction, tread pattern and material, vehicle speed, and so on, and $\varphi = 0.2 \sim 0.7$ in general case [26]. Parts of the vehicle parameters are shown in Table 1.

The initial values of longitudinal displacement error x_e , longitudinal velocity $\dot{x}$, lateral displacement error y_e , lateral velocity $\dot{y}$, yaw angle θ_e , and yaw rate $\dot{\theta}$ are shown in Table 2, while the initial values of both lateral deceleration and yaw deceleration are all 0, and the longitudinal deceleration is as (6).

(a) Control input of linear velocity v_c (b) Control input of angular velocity ω_c FIGURE 6: Control inputs curves with different φ .

TABLE 1: parameters of the vehicle.

M (kg)	I (kg·m ²)	a/b (m)	C_f/C_r (kN/rad)	d (m)	f_R	c_x/c_y
2010	2280	1.335/1.265	40/40	1.395	0.02	0.35/0.4

TABLE 2: Initial state values of vehicle.

x_e (m)	$\dot{x}$ (m/s)	y_e (m)	$\dot{y}$ (m/s)	θ_e (rad)	$\dot{\theta}$ (rad/s)
-1	30	1	0	$-\pi/4$	0

TABLE 3: Computed results under different φ .

φ	0.2	0.3	0.4	0.6
v (m/s)	28	34.3	39.5	48.4
u (m/s)	12.5	15	17.5	20

TABLE 4: Parameters of the controller.

k_1	k_2	k_3	k_4	δ
2	1.5	2.5	5.2	1.5

Obviously, we can get from (5) that the critical safe speed v is not only related to vehicle parameters b and h but also concerned with the lateral adhesion coefficient φ and the curvature radius R . Here, b and h are all constants about a given vehicle. So the simulation should be performed with the change of parameters φ or R to verify the effectiveness of the proposed control method in this paper.

4.1. Results with Different φ . Assuming that the curvature radius of the lane is set to $R = 400$ m, we compare the simulation results under different values of φ in the following. As it can be seen, the larger the φ , the larger the road adhesion force, and the higher the safety when driving on curved road in the general case. The value of φ is set to 0.2, 0.3, 0.4, and 0.6, respectively, and the corresponding critical safe speed v can be derived according to (5). The safe speed $u = K_v v$ when driving on curved road. Here, the function of K_v is to improve safe coefficient, which could be set to $K_v = 0.6 \sim 0.9$.

Additionally, according to China Design Specification for Highway Alignment, the design speed of the road must be lower than 80 km/h when the usual minimum the radius of curve is 400 m. On the basis of these, the corresponding safe speed u can be determined. All the computed results are shown in Table 3.

Suppose that the speed of 30 m/s~20 m/s belongs to the high-speed section and 20 m/s~10 m/s belongs to the low-speed section in this paper. Here, the initial value of vehicle longitudinal speed is $v_0 = 30$ m/s. We can get the time needed for decelerating to safe speeds t_2 from (7). Then the automatic vehicle could drive through 1/4 round corners with the curvature radius $R = 400$ m at a safe speed evenly. So the corresponding simulation time for the φ above is 57 s, 48 s, 42 s, and 37 s, respectively. The control law (17) is used here, and the control parameters for the simulation are show in Table 4. The corresponding results are shown from Figure 4 to Figure 6.

Figure 4(a) describes the expected deceleration curve under different values of φ ; Figure 4(b) is shown as the trajectory tracking simulation curves of both deceleration on the straight path and lane keeping on the curve road aiming at different φ ; Figure 4(c) indicates the trajectory tracking results when $\varphi = 0.2$.

It can be seen from Figure 4(a) that the corresponding time t_2 while braking and decelerating on the straight path is 7.5 s, 7 s, 6.5 s, and 6 s, respectively, when the value of φ is set to 0.2, 0.3, 0.4, and 0.6. Meanwhile, during 0~6 s, the longitudinal deceleration is increased by exponential curves from 0 m/s² to 5 m/s²; during 6~7 s, the speed is reduced to corresponding safe speed at a constant deceleration 5 m/s².

FIGURE 7: Deceleration curve and trajectory tracking simulation curves with different R .

Figure 4(b) is shown as the trajectory tracking simulation curves of deceleration on the straight path and driving on curved road with the curvature radius $R = 400$ m safely under different φ . Figure 4(c) indicates the trajectory tracking results when $\varphi = 0.2$, where the dotted lines are the desired states of lane changing and the solid lines denote the actual states. From the local enlarging graphs of this figure, we can get that the desired running state could be tracked effectively after longitudinal driving about 5 m under the initial posture error $x_e = -1$ and $y_e = 1$.

Figure 5 represents changes of tracking errors x_e , y_e , and θ_e with process time t about different φ . As is shown in Figure 5, these 3 tracking errors describing the vehicle system all converge to zero asymptotically at about 4 s, that is, the braking and deceleration stage, under the function of the controllers above, including velocity controller and posture

controller. Simulation results demonstrate that the controller has characteristics of quick convergence and global stability under different lateral adhesion coefficients φ .

Figure 6 denotes the corresponding system control inputs. Figure 6(a) indicates the linear velocity v_c input and Figure 6(b) denotes the angular velocity ω_c input. From Figure 6(a), it can be seen that the speed is reduced to safe speeds at about 7 s during the process of braking and deceleration under different φ , and the vehicle can run on curved road at a safe speed. From Figure 6(b), we can get that during 0~5 s, a slight fluctuation of the angular velocity is produced around the zero mark with the amplitude of 0.01 rad/s when the value of φ is set to 0.3, 0.4, and 0.6, during 5~7.5 s, it converges to zero asymptotically and, respectively. And at a later stage of driving on curved road, there is also a slight drop of the angular velocity with the amplitude of

FIGURE 8: Trajectory tracking errors with different R .

0.005 rad/s when the value of φ is set as above. But the tiny fluctuation above could not destroy the directional stability when driving on the straight path under these φ above. After about 7 s, the automatic vehicle is driving on curved road with the curvature radius $R = 400$ m at a uniform speed, and the difference between the theory angular velocity and the actual angular velocity at this stage which is gotten from the figure is less than 0.01 rad/s about different φ . So it could realize curve lane keeping and achieve better performances of directional stability when steering on curved road no matter how high or low the lateral adhesion coefficient is.

4.2. Results with Different R . Assume that the lateral adhesion coefficient is set to $\varphi = 0.2$ and the simulation results are compared via the change of the curvature radius R . The curvature radius of curved lanes is set to 100 m, 200 m, 400 m, and 1000 m, respectively, so the corresponding critical safe speed v can be derived according to (5) simultaneously. Here, based on China Design Specification for Highway Alignment, the corresponding design speed of the road must be lower than 40 km/h, 60 km/h, and 100 km/h when the usual minimum radius of the curve is 100 m, 200 m, and 1000 m, respectively. Therefore, the corresponding safe speed

FIGURE 9: Control inputs curves with different R .TABLE 5: Computed results under different R .

R (m)	100	200	400	1000
v (m/s)	14	19.7	28	44.2
u (m/s)	10	15	20	30

u can be determined in the same way, and the computed results are shown in Table 5.

It can be gotten from Table 4 that the safe speed when driving on curved road with the curvature radius $R = 1000$ m is 30 m/s, which is the initial value of vehicle longitudinal speed, so it does not need braking and deceleration under these circumstances. The simulation process is as above when R is set to other values, including $R = 100$ m, $R = 200$ m, and $R = 400$ m, which consists of decelerating on the straight path and driving through 1/4 round corners with the curvature radius $R = 400$ m. So the corresponding simulation time for the R above is 24 s, 28 s, 37 s, and 52 s, respectively. Similarly, the control law (17) is used here, and the control parameters for the simulation are shown in Table 3.

In this condition, the corresponding results are shown from Figure 7 to Figure 9. Figure 7(a) indicates the expected deceleration curve under different R ; Figure 7(b) is shown as the trajectory tracking simulation curves of both deceleration on the straight path and lane keeping on the curve road aiming at different R ; Figure 7(c) describes the trajectory tracking results when $R = 200$ m, where the dotted lines are the desired states of lane changing and the solid lines denote the actual states. Figure 8 represents changes of tracking errors x_e , y_e , and θ_e with process time t about different R . Figure 9 denotes the corresponding system control inputs.

As can be seen from the simulation results above, the control method in this paper could realize the tracking of running state effectively and it could perform auto anti-sideslip and antiroll-over and control of vehicle on curved

road in automatic highway systems however large or small the curvature radius is. Above all, it could guarantee the safety and directional stability when driving.

5. Conclusions and Future Work

In this paper, the kinematics model of the vehicle with the nonholonomic constraint is established, and the velocity controller when decelerating on the straight path and the posture controller when driving on curved road are designed via integral backstepping technology, respectively, which realizes the antisideslip and antiroll-over control of the vehicle on curved road in automatic highway system. The controller can ensure better performance of safety and stability and the vehicle will not roll over or sideslip when driving on curved road, so the method above has certain valuable significance and applications. Additionally, simulations are carried out under different lateral adhesion coefficients and under different curvature radius, respectively, which demonstrates the effectiveness of the proposed control method in this paper.

Though the automatic vehicle system is described from the perspectives of kinematics, the dynamic control law of the vehicle has not been studied in this paper. Currently, we are working on verifying those researches using the realistic vehicle dynamics simulator, such as Carsim, Adams, and so forth; we will present the results in the near future once we get meaningful outcomes. Additionally, the improved control method should be applied into the vehicle processor system to perform real road tests by getting the position of the vehicle, as well as the linear velocity of the mass center and the angular velocity of the vehicle.

Conflict of Interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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