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Research Article

Minimum Time Approach to Emergency Collision Avoidance by Vehicle Handling Inverse Dynamics

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Vehicle driving safety is the urgent key problem to be solved of automobile independent development while encountering emergency collision avoidance with high speed. And it is also the premise and one of the necessary conditions of vehicle active safety. A new technique of vehicle handling inverse dynamics which can evaluate the emergency collision avoidance performance is proposed. Based on optimal control theory, the steering angle input and the traction/brake force imposed by driver are the control variables; the minimum time required to complete the fitting biker line change is the control object. By using the improved direct multiple shooting method, the optimal control problem is converted into a nonlinear programming problem that is then solved by means of the sequential quadratic programming. The simulation results show that the proposed method can solve the vehicle minimum time maneuver problem, and can compare the maneuverability of two different vehicles that complete fitting biker line change with the minimum time and the correctness of the model is verified through real vehicle test.

1. Introduction

With the continuous development of the automobile industry, the number of car accidents grows accordingly, especially in traffic accidents involving pedestrians and cyclists. In some cases, car accidents can be seen as a collision between vehicles and obstacles. And then the emergency avoidance problem is proposed to avoid the accidents. Today, people pay more and more attention to the problem of high-speed emergency avoidance [1, 2]. When vehicles traveling at high speed meet obstacles, the driver will often choose the emergency brake parking or the bypass passing to avoid obstacles. Both the maneuvers require the driver to avoid obstacles in minimum time [3].

The research methods of vehicle handling dynamics usually include open-loop and closed-loop method. The two methods are called "forward problem" method of vehicle handling dynamics research. Open-loop research method does not consider the function of the driver's feedback and obtains vehicle response under the condition of mathematical model of vehicle and driver input. But closed-loop method obtains vehicle motion which follows the ideal path based on driver vehicle closed-loop control system model [4]. In order to avoid building driver model which is a difficult and important problem for vehicle model, the method of vehicle handling inverse dynamics is proposed [5–7]. The vehicle handling inverse dynamics can be reversed to obtain the driver's handling input based on the known model and vehicle motion (vehicle response). Then, the vehicle handling inverse dynamics can analyze what kind of handing is easily accepted by the driver with the safest and most rapid way [8–10]. Google is developing self-driving technology that combines data collected by sensors installed on a car with existing mapping software to speed up, brake, and steer to a destination. First, the driver's handling input was obtained in the Google car. According to the handling input, self-drive mode controls the car. So Google car apply the principle of the vehicle handling inverse dynamics. The Mercedes Benz system of PRE-SAFE and BAS-PLUS are designed to help to avoid accidents. The effectiveness is a measure for the efficiency, with which a safety system succeeds in achieving this target within its range of operation in vehicle. The process is also the application of the vehicle handling inverse dynamics [11].

In emergency avoidance research, the vast majority of research focused on the shortest path in the process of emergency avoidance. Sundar and Shiller (1997) proposed a method producing the shortest path based on the Hamilton-Jacobi-Bellman equation in a cluttered environment. The method attributed the emergency avoidance problem of shortest distance to the optimal control problem of shortest time, generated the shortest path through the function of negative gradient, and achieved good results [12]. Hattori et al. (2006) optimized the vehicle trajectory control for obstacle avoidance problem. A new control algorithm for obstacle avoidance within the shortest possible distance is proposed [2]. Mukai et al. [13] transformed the problem of generating an optimal path without a collision between an automobile and obstacles as a mixed integer programming problem [2]. However, the minimum time required to complete the emergency avoidance was very little to research. The maneuvers under the emergency avoidance require the driver to avoid obstacles in minimum time. So the minimum time approach to emergency collision avoidance is very important. In the paper, the inverse dynamics method was introduced to research the minimum time problem of the vehicle emergency collision avoidance.

Vehicle handling inverse dynamics can evaluate the driver's handling input by the specified handling performance and improve the performance of high-speed vehicle emergency avoidance. The handling performance of different vehicle can be compared with the most efficient way by the vehicle handling inverse dynamics [14–16].

In the paper, the optimal control theory is used in the field of vehicle handling inverse dynamics. In order to simplify the problem, the ideal driver handling inputs are considered without consideration of driver response lag and the forward-looking role.

2. Vehicle Steering Motion Model

2.1. The Mathematical Model of Vehicle Steering Wheel Torque Input. Assuming tire cornering properties in the linear range and considering rotational inertia of the steering system, the vehicle steering motion model is simplified as shown in Figure 1, which is a linear vehicle model with four degrees of freedom (DOF). The four degrees are lateral movement, horizontal pendulum movement, longitudinal motion, and steering system turning. A 4 DOF vehicle steering motion model is built as shown in Figure 1. The differential equations of motion are expressed as

$$\begin{gathered}
\dot{v} = -u\omega_r + \frac{F_{yf}\cos\delta + F_{yr} + F_{xf}\sin\delta}{m}, \\
\dot{\omega}_r = \frac{aF_{yf}\cos\delta - bF_{yr} + aF_{xf}\sin\delta}{I_z}, \\
\dot{u} = v\omega_r + \frac{F_{xf}\cos\delta - F_{yf}\sin\delta + F_{xr} - F_f - F_w}{m}, \\
\dot{\delta} = p \\
\dot{p} = -\frac{k_1\xi_1}{I_w u}v - \frac{k_1\xi_1 a}{I_w u}\omega_r + \frac{(k_1\xi_1 - k_w)}{I_w}\delta - \frac{c_w}{I_w}p + \frac{T_{\mathrm{sw}}i}{I_w},
\end{gathered} \tag{1}$$

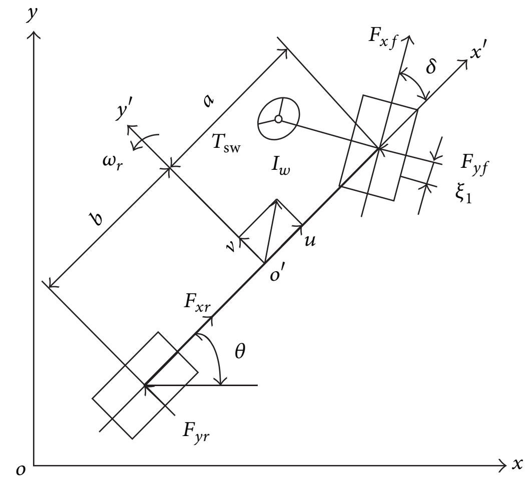


Figure 1: 4 DOF vehicle steering model.

where v is the lateral velocity, u is the longitudinal velocity, ω_r is the yaw rate, m is the total mass of vehicle, I_z is the vehicle moment of inertia around the vertical axis, I_w is the steering system moment of inertia, ξ_1 is the returnable arm of front wheel, p is the state variables, c_w is the resistance coefficient of steering system, k_w is the composite stiffness of steering system, i is transmission ratio of steering system, T_{sw} is the torque of steering wheel, a, b are the distance from the whole vehicle centroid to front and rear axle, k_1, k_2 are the comprehensive cornering stiffness of the former and rear wheels, δ is the rotation angle of the former and rear wheels, F_{yf} is the cornering force of the front wheel, F_{yr} is the cornering force of the rear wheel, F_{yr} is the driving/braking force of the front wheel (when $F_{xf} \geq 0$, F_{xf} is driving force; when $F_{xf} < 0$, F_{xf} is braking force), F_{xr} is the rear wheel driving/braking force, F_f is the rolling resistance ($F_f = mgf$; f is the rolling resistance coefficient), and F_w is the air resistance ($F_w = C_D A(3.6u)^2/21.15$; C_D is the air resistance coefficient and A is the windward area). The vehicle runs on smooth surfaces, no slope resistance. In order to simplify the problem, the air resistance and rolling resistance are considered, not considering acceleration resistance.

If driving force/braking force is considered to impact the cornering force, it is

$$\begin{gathered}
F_{yf} = k_1\left(\frac{v + a\omega_r}{u} - \delta\right)\sqrt{1 - \left(\frac{F_{xf}}{F_{zf}}\right)^2 + \left(\frac{F_{xf}}{k_1}\right)^2} \\
F_{yr} = k_2\left(\frac{v - b\omega_r}{u}\right)\sqrt{1 - \left(\frac{F_{xr}}{F_{zr}}\right)^2 + \left(\frac{F_{xr}}{k_2}\right)^2},
\end{gathered} \tag{2}$$

where φ is the friction coefficient of pavement, F_{zf} is the vertical force of front wheel, and F_{zr} is the vertical force of rear

wheel. Take the longitudinal load transfer into consideration; it is

$$F_{zf} = \frac{mgb - \left(F_{xf} + F_{xr}\right) h_g}{a+b},$$
$$F_{zr} = \frac{mga + \left(F_{xf} + F_{xr}\right) h_g}{a+b}, \tag{3}$$

where h_g is the height of vehicle mass center.

The x, y coordinates of the vehicle mass center in the xoy coordinate system have the following relations:

$$\dot{x} = u\cos\theta - v\sin\theta$$
$$\dot{y} = v\cos\theta + u\sin\theta. \tag{4}$$

2.2. The Optimal Control Model of Steering Wheel Torque Input. Control variable $Z(t)$ is the steering wheel torque $T_{\text{sw}}(t)$ and wheel driving force/braking force $F_{xf}(t)$; the control goal is the minimum time through a given path. Therefore, the performance indicators of time are

$$J(Z) = \int_{t_0}^{t_e} \mathrm{d}t, \tag{5}$$

where t_0, t_e are the initial time and terminal time.

According to (1), state equation can be expressed as

$$\dot{\mathbf{X}} = f\left[X(t), Z(t)\right]. \tag{6}$$

In the equation, the state variables are

$$X(t) = \left[v(t) \;\; \omega_r(t) \;\; u(t) \;\; \delta(t) \;\; p(t) \;\; x(t) \;\; y(t) \;\; \theta(t)\right]^{\mathrm{T}}. \tag{7}$$

Longitudinal velocity u is bounded by the vehicle maximum speed, lateral displacement is bounded by track around the border, the angle of the steering wheel δ_{sw} is bounded by driver's physiological limit, and control variable F_{xf} is bounded by road adhesion. When the vehicle is under a front wheel driving, it is

$$F_{xf} \le \frac{\varphi mgb}{a+b+\varphi h_g},$$
$$F_{xr} = 0. \tag{8}$$

When the vehicle is under a braking force and the front and rear wheels are assumed in lock state, it is

$$F_{xf} \ge -\frac{\varphi mg\left(b+\varphi h_g\right)}{a+b},$$
$$F_{xr} = \frac{a-\varphi h_g}{b+\varphi h_g} F_{xf}. \tag{9}$$

Control variable F_{xf} is bounded by the maximum driving force which is provided by the power transmission system.

According to the connection between engine speed and the velocity of vehicles and the connection between the engine output torque and the driving force, the regulation between the maximum driving force and the velocity of vehicles can be obtained by the engine external characteristic curve.

According to the literature [17], the constraints preventing rollover in the course of the vehicle driving are

$$\frac{u^2\delta}{(a+b)\left(1+Ku^2\right)g} \le \frac{L}{2h_g}, \tag{10}$$

where L is wheel tread and K is stability factor.

All the constraints are shown by the following equation:

$$\psi\left[\mathbf{X}(t), Z(t)\right] = 0. \tag{11}$$

2.3. The Transformation of State Variables. In the process of the vehicle tracking the desired trajectory, the ultimate elapsed time is difficult to determine. In order to solve this problem conveniently, the free terminal time can be transformed into the fixed terminal time for optimal control problem with the following ways.

Longitudinal displacement variable $\overline{x}$ which is defined unitization is

$$\overline{x} = \frac{x - x_0}{x_e - x_0} \quad (0 \le \overline{x} \le 1), \tag{12}$$

where x_0 is the initial longitudinal displacement and x_e is the terminal longitudinal displacement.

According to (4) and the time derivative of (12), the following equation can be obtained by taking a derivative with respect to time t in (12). Consider

$$\frac{\mathrm{d}\overline{x}}{\mathrm{d}t} = \frac{1}{x_e - x_0}\dot{x} = \frac{u\cos\theta - v\sin\theta}{x_e - x_0} \quad (0 \le \overline{x} \le 1). \tag{13}$$

Make state variable $\overline{\mathbf{X}}(\overline{x}) = \left[v(\overline{x}) \;\; \omega_r(\overline{x}) \;\; u(\overline{x}) \;\; \delta(\overline{x}) \;\; p(\overline{x}) \;\; y(\overline{x}) \;\; \theta(\overline{x})\right]^{\mathrm{T}}$.

According to (13), performance index of (5) can be transformed into

$$\mathrm{d}t = \frac{x_e - x_0}{u\cos\theta - v\sin\theta}\mathrm{d}\overline{x} \quad (0 \le \overline{x} \le 1)$$
$$J(Z) = \int_{\overline{x}_0}^{\overline{x}_e} \lambda\, \mathrm{d}\overline{x}, \tag{14}$$

where $\lambda = (x_e - x_0)/(u\cos\theta - v\sin\theta)$.

Similarly, (6) is

$$\frac{\mathrm{d}\overline{\mathbf{X}}}{\mathrm{d}\overline{x}} = \lambda f\left[\overline{\mathbf{X}}(\overline{x}), Z(\overline{x})\right]. \tag{15}$$

2.4. Nonlinear Programming Method of Improved Direct Multiple Shooting. The state variable, control variable, and time of nodes are assumed at the same time in the direct multiple shooting algorithm. It will increase variable numbers of the transformed nonlinear programming problem, thus making it more difficult to get the answer. Therefore, this paper puts

forward an improved direct multiple shooting method; in other words, only control variables of nodes are assumed.

(1) The Original Problem Is Converted into a Fixed Terminal Time Mayer Problem [18]. The state space expands into $n+1$ dimensions; new state variables are introduced to satisfy the following two equations:

$$\dot{\overline{X}}_{n+1} = \lambda, \tag{16}$$

$$\overline{X}_{n+1}(0) = 0. \tag{17}$$

The performance index of (14) can be translated into

$$J(Z) = \lambda \overline{X}_{n+1}(1). \tag{18}$$

Therefore, as long as (16) is incorporated in to the system state equation (15) and (17) is merged into constraint equation (11), the original problem is translated into standard Mayer problem with given terminal.

(2) The Optimal Control Problem Is Transformed into Finite Dimensional Nonlinear Programming Problem. The optimization method used in the paper is one of the sequential quadratic programming method (SQP): Wilson-Han-Powell method. The method is based on the common Lagrange-Newton method. The basic theory of SQP converts the nonlinear programming problem to a series of quadratic programming problems, so it is called SQP. Specifically, the approximate solution x_k and approximate multiplier vector λ_k were assumed to be known when the kth iteration starts. In this way, the kth quadratic programming subproblem P_k can be given, the new approximate solution x_{k+1} can be obtained by solving the problem P_k, and the corresponding Lagrange multiplier vector λ_{k+1} is determined too. The above-mentioned process is repeated until the approximate optimal solution of nonlinear programming problem is obtained. Assuming $d_k = x_{k+1} - x_k$, then, getting x_{k+1} by solving P_k can be converted to getting the d_k by solving the subproblem P_k.

Considering the common nonlinear constrained optimal control problem,

$$\begin{aligned} &\min_{x\in R^n} && f(x) \\ &\text{s.t.} && c_i(x) = 0, \quad i \in E \\ & && c_i(x) \geq 0, \quad i \in I, \end{aligned} \tag{19}$$

where $f(x)$, $c_i(x)$ are all real-valued continuous functions and at least one of them is nonlinear, $E = \{1, 2, \ldots, m_e\}$, $E = \{1, 2, \ldots, m_e\}$. The subproblem is constructed. Consider

$$\begin{aligned} &\min_{d\in R^n} && g_k^T d + \frac{1}{2} d^T B_k d \\ &\text{s.t.} && a_i(x_k)^T d + c_i(x_k) = 0, \quad i \in E \\ & && a_i(x_k)^T d + c_i(x_k) \geq 0, \quad i \in I. \end{aligned} \tag{20}$$

In the above equations, $A(x_k) = [a_1(x_k), \ldots, a_m(x_k)] = \nabla c(x_k)^T$, g_k is the gradient of $f(x)$ in the x_k point, and B_k is the approximation of Hesse matrix of Lagrange function. The solution of above subproblem is d_k; the d_k is used as the search direction of kth iteration in the method of Wilson-Han-Powell. It is the descent direction of penalty function.

The procedures of sequential quadratic programming are given as follows.

(1) Give

$$x_1 \in R^n, \quad \sigma > 0, \quad \delta > 0, \quad B_1 \in R^{n\times n}, \quad \varepsilon \geq 0, \quad k = 1. \tag{21}$$

(2) One gets d_k by solving the above subproblem. If $\|d_k\| \leq \varepsilon$, then stop the iteration. Solve $\alpha_k \in [0, \delta]$, which makes

$$P(x_k + \alpha_k d_k, \sigma) \leq \min_{0\leq\alpha\leq\delta} P(x_k + \alpha_k d_k, \sigma) + \varepsilon_k. \tag{22}$$

(3) $x_{k+1} = x_k + \alpha_k d_k$; calculating B_{k+1}, $k = k+1$; return to step (2).

In formula (22), the penalty function $P(x, \sigma)$ is the precise penalty function of L_1; ε_k is a nonnegative series and satisfies the following condition:

$$\sum_{k=1}^{\infty} \varepsilon_k < +\infty. \tag{23}$$

B_{k+1} is obtained by using quasi-Newton formula: take

$$s_k = x_{k+1} - x_k,$$
$$y_k = \nabla f(x_{k+1}) - \nabla f(x_k) - \sum_{i=1}^{m} (\lambda_k)_i [\nabla c_i(x_{k+1}) - \nabla c_i(x_k)]. \tag{24}$$

Then calculate B_{k+1} by using BFGS correction formula:

$$B_{k+1} = B_k - \frac{B_k s_k s_k^T B_k}{s_k^T B_k s_k} + \frac{y_k y_k^T}{s_k^T y_k}. \tag{25}$$

For the optimal control problem of time-varying system in this paper, it can be converted to the finite dimensional nonlinear programming problem by using the improved direct multiple shooting method.

(1) The interval $\overline{x} \in [0, 1]$ is divided into n uniform. $n+1$ nodes are obtained.

(2) A set of vectors e_j $(j = 0, 1, \ldots, n-1)$ are introduced as estimated values of control variable at the node. Control variable values between nodes are gotten by linear interpolation of two adjacent values. If the node place control variables e_j are known, each state variable can be gotten one by one by integration. Thus $\overline{X}_{n+1}(1)$ can be obtained, and then performance index is gotten. Therefore, it can be argued that the solution of differential equation and performance indicators are only the function of each node control variable.

The gating finite dimensional nonlinear programming problem can be solved by using the sequential quadratic programming (using the fmincon function in the optimization toolbox of MATLAB).

Table 1: Two vehicles parameter list.

Parameter	Model A	Model B
m/kg	1500	1265
I_z/(kg·m^2)	2080	1800
a/m	1.185	1.170
b/m	1.283	1.195
k_1/(N·rad^{-1})	−60533	−60042
k_2/(N·rad^{-1})	−110185	−109295
i	20	20
I_w/(kg·m^2)	17.53	16.38
c_w/(N·m·s·rad^{-1})	160	140
k_w/(N·m·rad^{-1})	0	0
ξ_1/m	0.028	0.021
φ	0.8	0.8
h_g/m	0.53	0.53
i_g	0.97	0.914
i_0	4.12	4.38
r/m	0.289	0.280
u/km·h^{-1}	72	72

i_g is the transmission gear ratio, i_0 is the main reducer gear ratio, and r is the wheel radius.

3. Numerical Simulation

The biker line performance of two vehicles is researched. The vehicle specific parameter values are shown in Table 1. The steering wheel torque is controlled between ±8 N·m. The initial time is

$$\begin{aligned}&[v(0)\ \ \omega_r(0)\ \ u(0)\ \ \delta(0)\ \ p(0)\ \ x(0)\ \ y(0)\ \ \theta(0)]^{\mathrm{T}}\\ &\quad=[0\ \ 0\ \ 20\ \ 0\ \ 0\ \ 0\ \ 0\ \ 0]^{\mathrm{T}}.\end{aligned} \tag{26}$$

The size of biker line test road is shown in Figure 2. Parameter value in Figure 2 is $s_0 = L = 2u$, $s = 3u$. Benchmarking width is $B = 2.46$ m.

In the actual driving process, driver's ideal target track should be as shown in Figure 3. It is a low order continuous smooth curve. Three-order curve of a continuous first derivative is gotten after three spline fits. Consider

$$f(t)=\begin{cases}0 & t\in t_1\\ \dfrac{B(1.0+\sin\omega t)}{2.0} & t\in t_2\\ B\cos\omega t & t\in t_3\\ \dfrac{B(-1.0+\sin\omega t)}{2.0} & t\in t_4\\ 0 & t\in t_5.\end{cases} \tag{27}$$

Parameter Model A and Model B are as in Table 1.

As is shown in Table 1, the curb weight of A vehicle was bigger than B vehicle. A vehicle was better than B vehicle in the configuration and space.

After 14 iterations, the minimum time in which model A passes the biker line after optimization is 15.7 s. After 16 iterations, the minimum time in which model B passes the biker line after optimization is 16.2 s. Therefore, the minimum time in which model A passes the biker line after optimization is shorter than that of model B. Figures 4~7 show simulation results of some state variables and control variables in the process of biker line when $u = 72$ km/h.

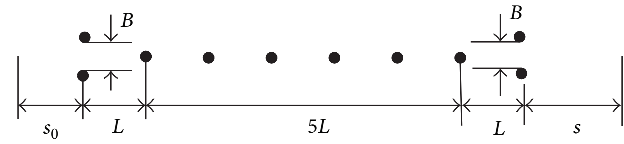


Figure 2: Biker line test road.

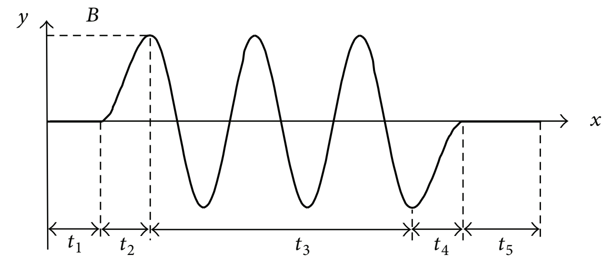


Figure 3: Fitting biker line.

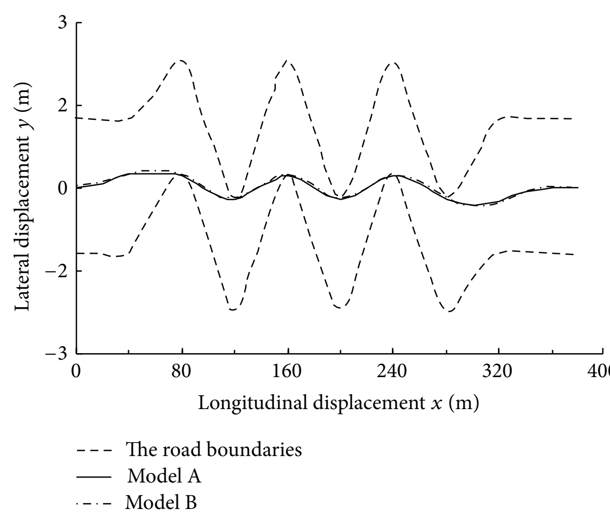


Figure 4: Lateral displacement simulation results.

Figure 4 shows the simulation results about lateral displacements of two types of models. It can be seen that the two kinds of models' lateral displacement are almost coincidence. The motion law for vehicle between the road boundaries is that vehicles move almost in straight line.

Figure 5 shows the simulation results about steering wheel torque of two vehicles. It can be seen from the several steering wheel torque amplitudes that the steering wheel torque amplitude of vehicle A is larger than that of vehicle B.

Figure 6 shows the simulation results about the wheel driving force of two vehicles. It can be seen that the driving force decreases at first and then increases in the serpentine

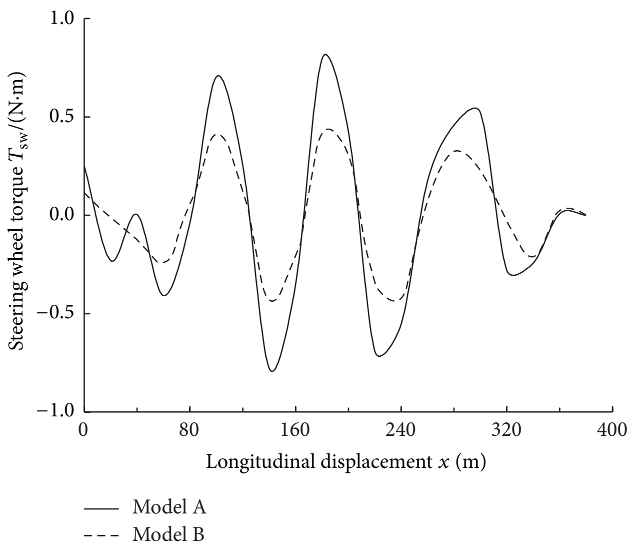


Figure 5: The steering wheel torque simulation results.

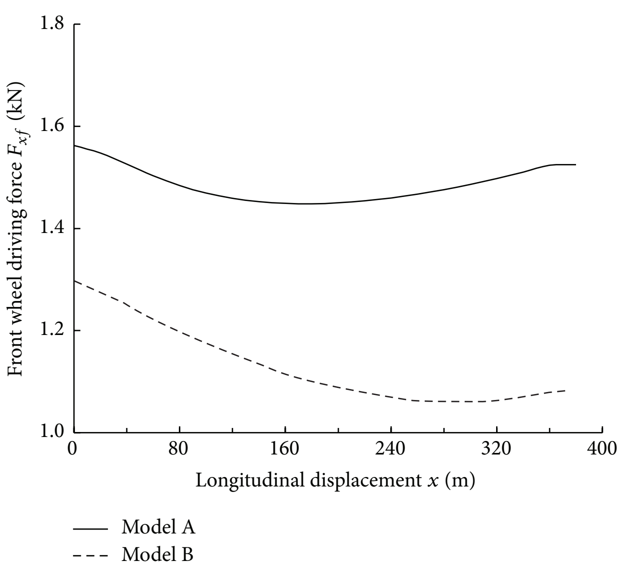


Figure 6: Wheel driving force simulation results.

line performance process; it also can be seen that the vehicle A's driving force is larger than that of the vehicle B.

Figure 7 shows the two models' longitudinal velocity simulation results. It can be seen that in serpentine line performance process, Model A increases rapidly from 20 m/s to 30.7 m/s and Model B increases rapidly from 25 m/s to 29.2 m/s. Therefore, the acceleration performance of Model A is better than that of B.

When the vehicle travels at 25 m/s high speed initially, after 14 iterations, the minimum time in which Model A passes the biker line after optimization is 14.9 s. After 18 iterations, the minimum time in which model A passes the biker line after optimization is 15.2 s. Therefore, when the vehicle run at the fast speed, it drive through the serpentine in short time. But as the vehicle's speed increases, the driver's

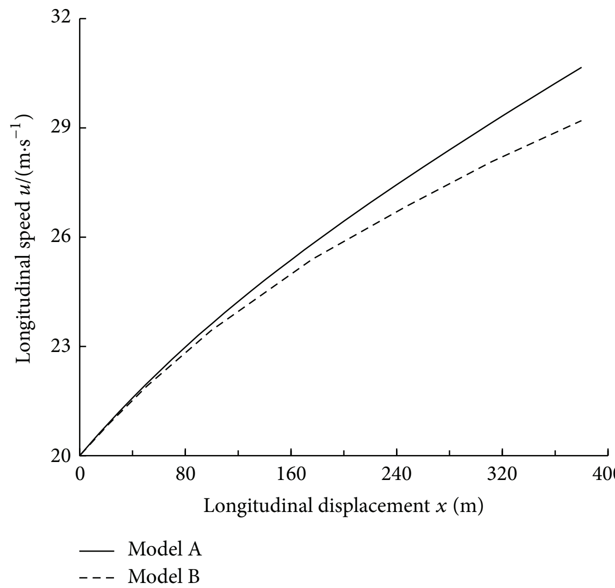


Figure 7: Longitudinal velocity simulation results.

burden increases and the safety reduces. The driver has to reduce vehicle's speed to a certain level to ensure his safety before the vehicle passes the biker line.

4. Experimental Verification

In this paper, two types of off-road vehicles mentioned above are used to test vehicle handling stability. Real vehicle test is very dangerous in high speed. In order to consider the driver's safety, the method of pavement design point is taken in the test.

The test procedures are as follows.

(1) In the test site, stake position marker is designed as in Figure 5.

(2) Connect the test instruments; switch instruments power on in order towarm the instruments to normal operating temperature.

(3) The vehicle passes the test section with an initial speed of 72 km/h. Running over the marker is not allowed in the running process. At the same time, the time history curve of the measured variables (steering wheel angle and longitudinal velocity) is recorded by the computer.

(4) Repeat steps (3) process 12 times (the times of press the marker is not considered). Two vehicle types' experimental data are obtained by the same test methods above if the vehicle type is changed.

The test site is built as shown in Figure 8.

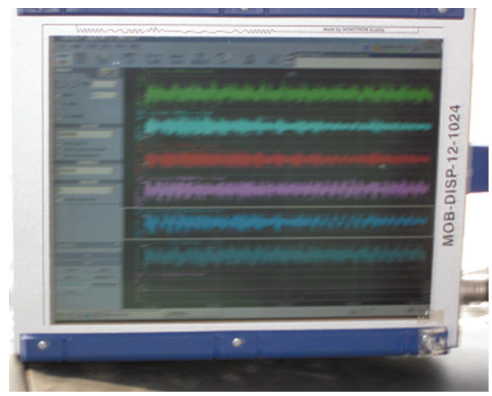
The experimental procedures and protocols are built as shown in Figure 9.

12 groups of test time were, respectively, 17.8 s, 18.1 s, 17.9 s, 18.5 s, 18.8 s, 18.3 s, 17.9 s, 18.3 s, 18.2 s, 18.9 s, 19.0 s, and 18.8 s. Due to considering a lot of factors, such as driver's reaction

(a) The bollard of biker line

(b) Car running track

(c) The instrument of steering wheel torque



(d) DEWESoft data signal acquisition system

Figure 8: Vehicle biker line test.

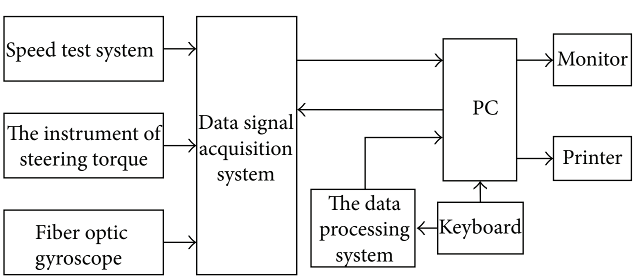


Figure 9: Experimental procedures and protocols.

time and road conditions, the experimental test time was generally longer than the time of optimal control. The mean and standard deviation were, respectively, 18.27 s and 0.3743.

Comparison between the simulation value and the experimental value is shown in Figures 10 and 11. As shown in Figures 10 and 11, there are some errors between simulation value and experimental value, mainly because every driver's subjective feelings and driving skills is different. In addition, the test instrument also has some errors. But the change tendency of simulation value and experimental values is consistent. So the correctness of the optimal control model is proved.

5. Conclusion

In the field of automotive engineering, many researchers are focusing on the development of self-driving technologies. Self-driving vehicles promise to bring a number of benefits to society, including prevention of road accidents, optimal fuel usage, comfort, and convenience. Vehicle handling inverse dynamics is form of the self-driving technologies. The steering wheel torque can be obtained by the vehicle handling inverse dynamics and used to determine the vehicle steering problems in the emergency collision avoidance. So the vehicle handling inverse dynamics can promote the self-driving vehicle development.

In this paper, minimum time approach to emergency collision avoidance is researched by the method of vehicle handling inverse dynamics. Firstly, the optimal control model of the vehicle emergency collision avoidance problem was established. And then the optimal control problem was changed into a nonlinear programming problem using the improved direct multiple shooting method. Finally, the transformed nonlinear programming problem was solved by using sequential quadratic programming method. The correctness of the optimal control model is verified by using real vehicle test. The results show that this method can successfully solve the minimum time problem of vehicle emergency

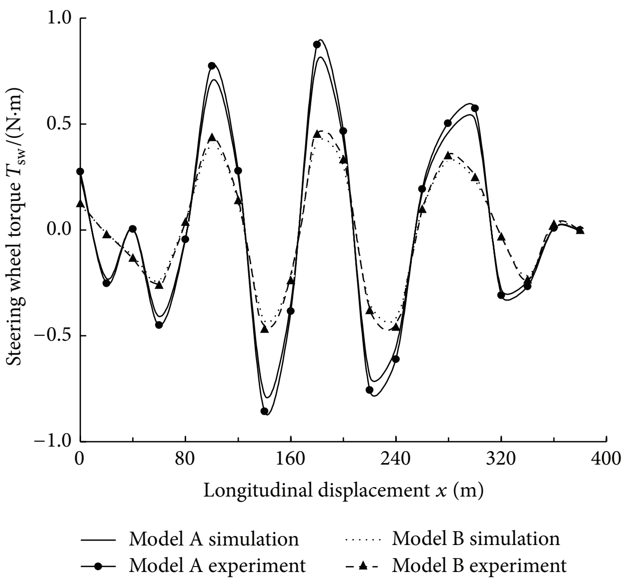


Figure 10: The steering wheel torque simulation results.

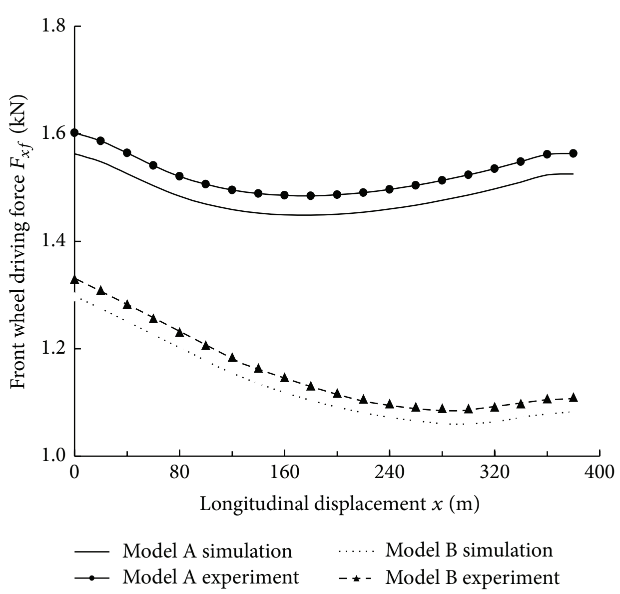


Figure 11: Wheel driving force simulation results.

collision avoidance and compare different vehicles in the minimum time through a given path control performance. It can provide guidance for the self-drive research. Intelligent vehicle driving also has certain reference value.

Conflict of Interests

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

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References

[1] A. Mas, F. Merienne, and A. Kemeny, "Lateral control assistance and driver behavior in emergency situations," *Advances in Transportation Studies*, pp. 149–158, 2011.

[2] Y. Hattori, E. Ono, and S. Hosoe, "Optimum vehicle trajectory control for obstacle avoidance problem," *IEEE/ASME Transactions on Mechatronics*, vol. 11, no. 5, pp. 507–512, 2006.

[3] J.-M. Park, D.-W. Kim, Y.-S. Yoon, H. J. Kim, and K.-S. Yi, "Obstacle avoidance of autonomous vehicles based on model predictive control," *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, vol. 223, no. 12, pp. 1499–1516, 2009.

[4] H. Hatwal and E. C. Mikulcik, "Some inverse solutions to an automobile path-tracking problem with input control of steering and brakes," *Vehicle System Dynamics*, vol. 15, no. 2, pp. 61–71, 1986.

[5] J. Andreasson and T. Bunte, "Global chassis control based on inverse vehicle dynamics models," *Vehicle System Dynamics*, vol. 44, no. 1, pp. 321–328, 2006.

[6] C.-S. Liu and H. Peng, "Inverse-dynamics based state and disturbance observers for linear time-invariant systems," *Journal of Dynamic Systems, Measurement and Control, Transactions of the ASME*, vol. 124, no. 3, pp. 375–381, 2002.

[7] J. Cho, J. C. Principe, D. Erdogmus, and M. A. Motter, "Modeling and inverse controller design for an unmanned aerial vehicle based on the self-organizing map," *IEEE Transactions on Neural Networks*, vol. 17, no. 2, pp. 445–460, 2006.

[8] C.-F. Lin and Y. Chuang, "Energy management strategy and control laws of an inverse differential gear hybrid vehicle," *World Electric Vehicle Journal*, vol. 4, no. 1, pp. 98–103, 2011.

[9] I. Han and S.-U. Park, "Inverse analysis of pre- and post-impact dynamics for vehicle accident reconstruction," *Vehicle System Dynamics*, vol. 36, no. 6, pp. 413–433, 2001.

[10] F. Boyer and S. Ali, "Recursive inverse dynamics of mobile multibody systems with joints and wheels," *IEEE Transactions on Robotics*, vol. 27, no. 2, pp. 215–228, 2011.

[11] H. Schittenhelm, "Design of effective collision mitigation systems and prediction of their statistical efficiency to avoid or mitigate real world accidents," in *Proceedings of the 32nd FISITA World Automotive Congress*, vol. 7, pp. 14–23, September 2008.

[12] S. Sundar and Z. Shiller, "Optimal obstacle avoidance based on the Hamilton-Jacobi-Bellman equation," *IEEE Transactions on Robotics and Automation*, vol. 13, no. 2, pp. 305–310, 1997.

[13] M. Mukai, T. Kawabe, H. Nishira, Y. Takagi, and Y. Deguchi, "On vehicle path generation method for collision avoidance using mixed integer programming," in *Proceedings of the 16th IEEE International Conference on Control Applications*, pp. 1371–1375, Singapore, October 2007.

[14] J. D. Trom, M. J. Vanderploeg, and J. E. Bernard, "Application of inverse models to vehicle optimization problems," *Vehicle System Dynamics*, vol. 19, no. 2, pp. 97–110, 1990.

[15] J. P. M. Hendrikx, T. J. J. Meijlink, and R. F. C. Kriens, "Application of optimal control theory to inverse simulation of car

handling," *Vehicle System Dynamics*, vol. 26, no. 6, pp. 449–461, 1996.

[16] J. Sridhar and H. Hatwal, "Comparative study of four wheel steering models using the inverse solution," *Vehicle System Dynamics*, vol. 21, no. 1, pp. 1–18, 1992.

[17] J. Bernard and M. Pickelmann, "An inverse linear model of a vehicle," *Vehicle System Dynamics*, vol. 15, no. 4, pp. 179–186, 1986.

[18] D. Kraft, "Algorithm 733: TOMP—fortran modules for optimal control calculations," *ACM Transactions on Mathematical Software*, vol. 20, no. 3, pp. 262–281, 1994.