Advanced Engineering Forum *Online: 2011-12-22*
ISSN: 2234-991X, Vols. 2-3, pp 33-38
doi:10.4028/www.scientific.net/AEF.2-3.33

Directional Control of a Driver-Heavy-Vehicle Closed-Loop System

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Keywords: driver-heavy-vehicle closed-loop system, preview driver model, lateral vehicle dynamics, Gim tire model

Abstract. A two degree of freedom (DOF) lateral dynamic model for a three-axe heavy vehicle is set up and the vehicle ordinary differential equations of motion are derived. The nonlinear lateral tire forces are obtained by Gim model with vertical loads, slip angles and cornering performances of front and rear tires being input parameters. A revised closed-loop single-point preview method is proposed to model the driver's directional control performance. In this proposed method, the steering angle of front wheels is calculated in real time according to the track error between a certain point ahead of the vehicle and the required route. Then the steering angle is input into the vehicle model to gain the dynamic responses and position of the vehicle in next time step. Thus the driver-heavy-vehicle closed-loop system is built. The dynamic responses of the system are simulated on the condition of double lane change and the effects of system parameters on the path following behavior of the vehicle are researched. Then the advice on how to improve the vehicle directional control ability can be brought forward.

Introduction

In vehicle dynamics, the directional control and lateral dynamic responses of two-axle saloon cars were widely studied. A lot of directional control methods have been proposed, such as the single point preview optimal curvature model [1-3], the self-paced preview tracking control model [4], the time-invariant optimal control method [5], the multi-point preview model [6] and so on. However, as far as we are concerned, the investigation on directional control approaches of three-axle heavy vehicle is seldom found.

This work gives a description on how to model the driver-heavy-vehicle closed-loop system and brings up a revised closed-loop single-point preview method for calculating the steering angle of front wheel in real time. The dynamic responses of the system are simulated on the condition of double lane change by numerical integration including the lateral velocity, the yaw rate, the vehicle track and the steering angle. The effects of system parameters such as preview distance, critical value of directional control, and vehicle running speed on the path following behavior of the vehicle are analyzed.

Model building

Vehicle lateral dynamic model. The moving vehicle coordinate system (x,y,z) and the static ground coordinate system (X,Y,Z) are built according to right hand rule, as shown in Figure 1. A two degree of freedom (DOF) lateral dynamic model for a three-axe heavy vehicle is set up and the ordinary differential equations of motion in vehicle coordinate system may be expressed by

$$\begin{cases} m(\dot{V}_y + V_x\omega_r) = \sum_{i=1}^{6}[F_{\alpha i}\cos(\delta_i)] \\ I_z\dot{\omega}_r = \sum_{i=1}^{6}[F_{\alpha i}\cos(\delta_i)l_{xi} + F_{\alpha i}\sin(\delta_i)l_{yi} + M_{zi}] \end{cases} \tag{1}$$

where, m, I_z, V_x, V_y,ω_r are vehicle mass, vehicle inertia around z axial, longitudinal, lateral and yaw velocity of the vehicle respectively.δ_i, $F_{\alpha i}$ and M_{zi} are steering angle, lateral tire force and self-aligning torque of wheels respectively. l_{xi} and l_{yi} are the distance from wheel center to the center of vehicle mass in longitudinal and lateral direction respectively.

The vehicle coordinate system (x,y,z) can be transferred to the ground coordinate system (X,Y,Z) by the following function

$$\begin{cases} \dot{X} = V_x\cos\psi - V_y\sin\psi \\ \dot{Y} = V_x\sin\psi + V_y\cos\psi \end{cases} \tag{2}$$

where, ψ is the course angle of the vehicle mass center.

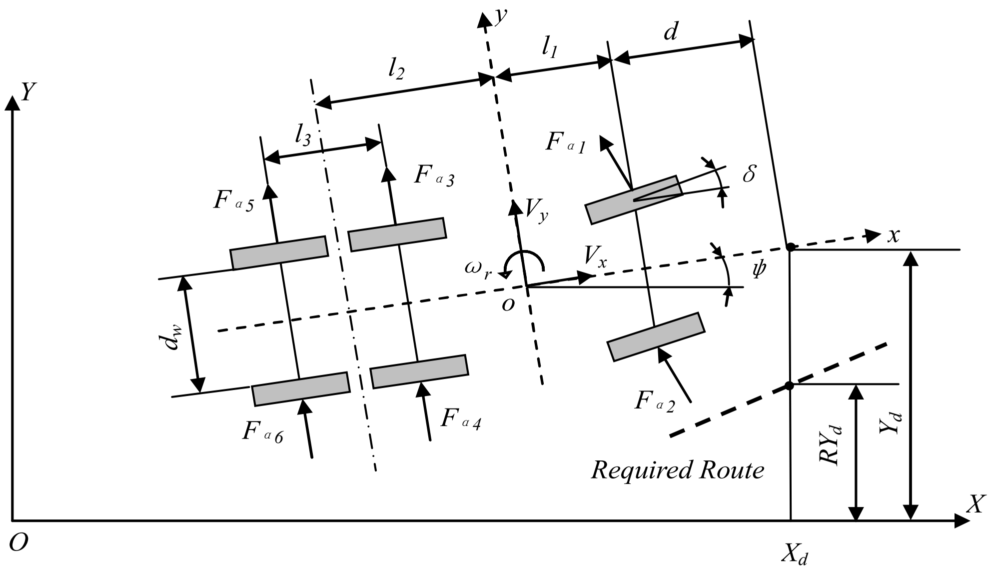


Fig. 1. The lateral dynamic model for a three-axe heavy vehicle

Tire model. The nonlinear lateral tire forces are described by the Gim tire model [7],

$$F_\alpha = \begin{cases} C_\alpha S_\alpha l_n^2 + \mu F_z(1 - 3l_n^2 + 2l_n^3) & S_\alpha < S_{\alpha c} \\ \mu F_z & S_\alpha \geq S_{\alpha c} \end{cases} \tag{3a}$$

$$M_z = \begin{cases} C_\alpha S_\alpha l(1 - S_n)^3 / 6 & S_\alpha < S_{\alpha c} \\ 0 & S_\alpha \geq S_{\alpha c} \end{cases} \tag{3b}$$

where F_z, C_α, S_α, μ,$S_{\alpha c}$ *and* l are vertical load, cornering stiffness, lateral slip ratio, friction coefficient, critical value of slip ratio and the length of tire footprint respectively.

$$S_\alpha = |\tan\alpha|,\ \ \mu = \mu_0(1 - \frac{1-\mu_1/\mu_0}{S_1}S_\alpha),\ \ S_n = \frac{C_\alpha S_\alpha}{3\mu F_z},\ \ l_n = 1 - S_n, S_{\alpha c} = \frac{3\mu F_z}{C_\alpha},\ \ l = 4R(\frac{F_z}{2C_r R})^{0.55}. \tag{4}$$

Here,αstands for the slip angle. R is the tire rolling radius. C_r is the tire radial stiffness.

Driver model. In the traditional single-point preview method [1], the optimal steering angle of front wheel is expressed by

$$\delta_{fw} = 2L/d^2[f(t+T) - y(t) - T\dot{y}(t)] \tag{5}$$

where, d is the preview distance. T is the preview time. L is the axle distance of vehicle. $f(t+T)$ is the required lateral position at the time of $t+T$. y(t) is the lateral displacement of vehicle at the time of t. It is obvious that the traditional single-point preview method calculates the optimal steering angle based on vehicle coordinate (x,y,z) and neglected the course angle of vehicle. Moreover, it is difficult to derive $f(t+T)$ from the required route if the vehicle running speed is not a constant.

Hence, a revised closed-loop single-point preview method is proposed here to model the driver's directional control performance. The optimal steering angle of front wheel is formulated by

$$\begin{cases} \delta_{fw}(t) = 2(l_1 + l_2)/d^2[RY_d - Y_d(t)] & |RY_d - Y_d(t)| > e_{cr} \\ \delta_{fw}(t) = \delta_{fw}(t - dt) & |RY_d - Y_d(t)| \le e_{cr} \end{cases} \tag{6}$$

where, RY_d, Y_d e_{cr} and dt are the required lateral position and real lateral displacement of the preview point in the ground coordinate system, the critical value of directional control and the integration time step respectively. Y_d can be gained from the position of vehicle mass center,

$$\begin{cases} X_d = X(t) + (d + l_1)\cos\psi \\ Y_d = Y(t) + (d + l_1)\sin\psi \end{cases} \tag{7}$$

where X(t), Y(t) are the position of vehicle mass center in the ground coordinate system. According to X_d and the required route function, RY_d is easily obtained.

Then the steering angle is input into the vehicle model to gain the lateral responses and positions of vehicle in next time step. Thus the driver-heavy-vehicle closed-loop system is built, as shown in Fig.2.

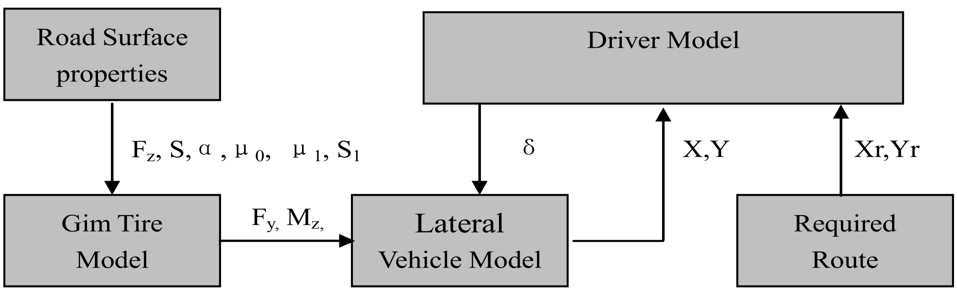


Fig. 2. The driver-heavy-vehicle closed-loop model

Simulation Result

The parameters of vehicle and road surface properties are selected as follows [1,9]
m=11685kg, I_z =8.63×10^3 kg m^2. l_1=3.64m, l_2=2.71m, l_3=1.3m, d_w=1.9m, R=0.42m, S_s=0.01, μ_0=0.01, μ_1=0.9, S_1=0.15, $C_{\alpha i}$=227.3×10^{3}N/m (i=1,2), $C_{\alpha i}$=455×10^{3}N/m (i=3~6), C_{byi}=730×10^{3}N/m (i=1,2), C_{byi}=1460×10^{3}N/m (i=3~6), C_{ri}=1460×10^{3}N/m (i=1,2), C_{ri}=2920×10^{3}N/m (i=3~6)。

Lateral responses of the system are simulated on the condition of double lane change by numerical integration. Fig.3 shows the lateral and yaw velocity of the vehicle, track of vehicle mass center, the required route and steering angle of front wheel.

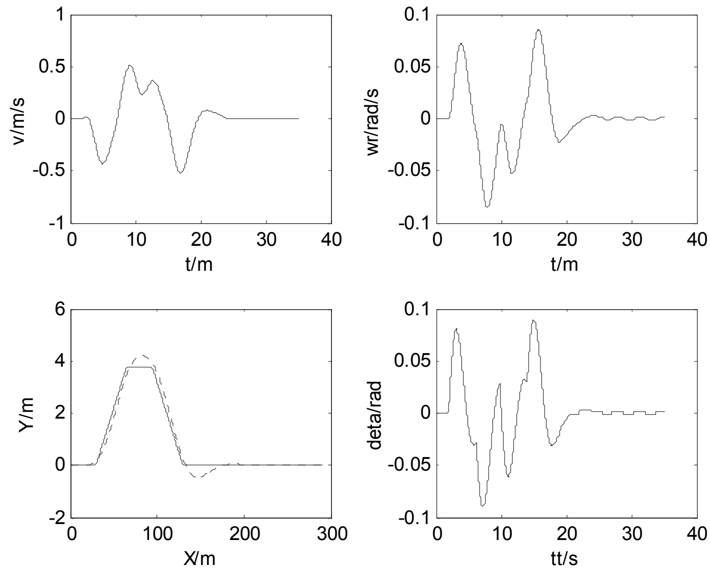


Fig. 3. Lateral responses of vehicle (V_x=30km/h, d=10m)

The effect of preview distance *d* on vehicle running route and path following error are shown in Fig.4. It can be seen from Fig.4 that there exists an optimal preview distance for controlling the motion track of the vehicle. With the rise of vehicle running speed, the optimal preview distance increased. Too small or too big preview distance will deteriorate the directional control effect.

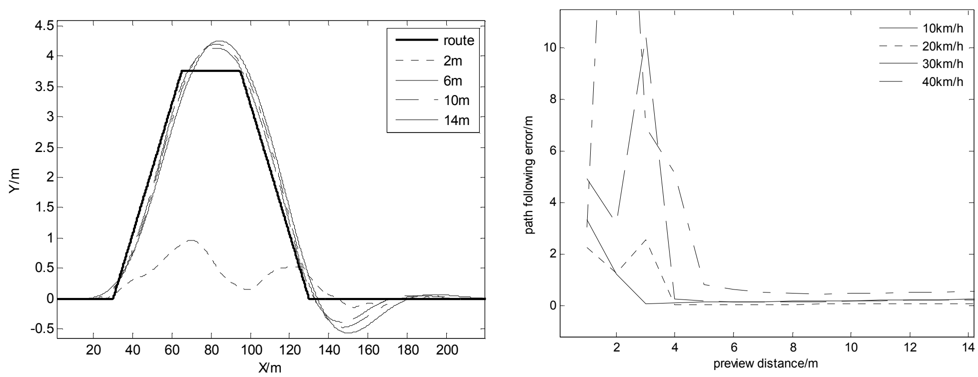


Fig. 4. The effect of preview distance on vehicle track and path following error

The effect of critical value of directional control e_{cr} on vehicle running route is also simulated and shown in Fig.5. It is found that the critical value of directional control influences the vehicle track slightly and a too small e_{cr} may not improve the path following accuracy of the revised single-point preview method.

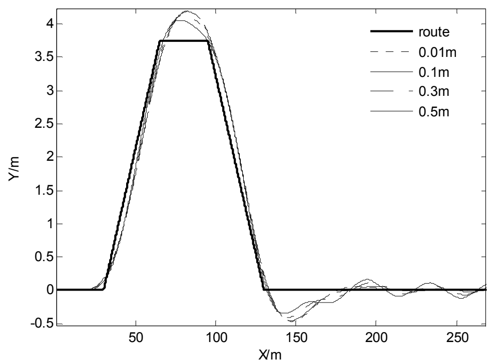
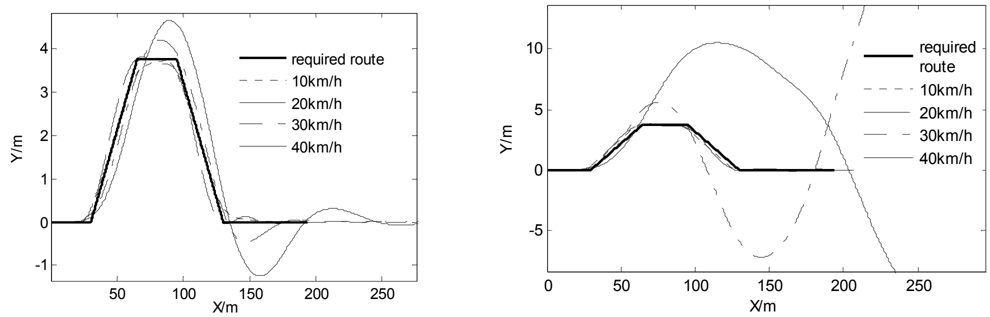


Fig. 5. The effect of critical value of directional control on vehicle track

The directional control results of the revised model and the traditional single point preview model on different running speed are compared in Fig.6. It is obvious that the revised model is more suitable than the traditional model on higher speed.



(a) The revised model (b) The traditional model

Fig. 6. Comparison of the revised model and the traditional model

Summary

A driver-heavy-vehicle closed-loop system is built in this work and a revised closed-loop single-point preview method is proposed to model the driver's directional control performance. On the condition of double lane change, the effects of system parameters such as preview distance, critical value of directional control, and vehicle running speed on the path following behavior of the vehicle are analyzed. It may be concluded from this work that

(1) There exists an optimal preview distance for controlling the motion track of the vehicle. Too small or too big preview distance will deteriorate the directional control effect.
(2) A small critical value of directional control may not improve the path following accuracy of the revised single-point preview method.
(3) The revised model is more suitable than the traditional one on higher speed.

Acknowledgements

This work is supported by the National Natural Science Foundation of China under Grant No. 10932006, 11072159 and the Scientific Research Program from the Education Department of Hebei Province under Grant No.2009472.

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