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Lateral Control of Autonomous Vehicle by Yaw Rate Feedback

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In the autonomous vehicle, the reference lane is continually detected by machine vision system. And then the vehicle is steered to follow the reference yaw rates which are generated by the deviations of lateral distance and the yaw angle between a vehicle and the reference lane. To cope with the steering delay and the side-slip of vehicle, PI controller is introduced by yaw rate feedback and tuned from the simulation where the vehicle is modeled as 2 DOF and 79 DOF and verified by the results of an actual vehicle test. The lateral control algorithm by yaw rate feedback has good performances of lane tracking and passenger comfort.

Key Words: Lateral Control, Yaw Rate Control, Machine Vision

1. Introduction

In the field of the intelligent vehicle systems (IVS), there are driving assisting system, longitudinal control, and lateral control. Ackermann (1997) implemented a driving assisting system for abrupt lateral forces such as wind and change of road friction. Cruise control (K. Yi et al., 2001) and Platoon of vehicles in PATH project correspond to the longitudinal control. The lateral control or steering control is autonomously required to track the reference lane. It can be applied to buses or transport vehicles in factories and ships in docks. There are two types of lateral control; look-down method or look-ahead method. In the look-down method, inductive cables were used for marking the reference lane. In PATH project (J. K. Hedrick et al., 1994), permanent magnets were used to mark the reference roadways and four magnetic sensors detected markers when the vehicle passed on them. In the look-ahead method, machine vision (S. Tsugawa, 1999; K. B. Han, 2001), radar, or ultrasonic sensor(H. Makela and T. V. Numers, 2001) were used. Machine vision systems come under a passive reference and a passive sensing system. Contrasting with the look-down method, it is easy to mark the reference lane. But the detected information are heavily affected by the weather condition.

In this paper, the lateral control of autonomous vehicle is considered. The error between the reference yaw rate and the measured one is used as controller input, and the reference yaw rates are generated from the lane information. The vehicle models, i. e as 2 DOF single-track model and 79 DOF multibody model (D. H. Han et al., 2000; B. H. Lee et al., 2001), are developed and used for the simulation for the proposed lateral control.

This paper is organized as follows: The lateral control algorithm and single-track model are given in Sec. 2. In Sec. 3, the multibody models of vehicle are explained. Section 4 presents the results of simulations for the lateral control.

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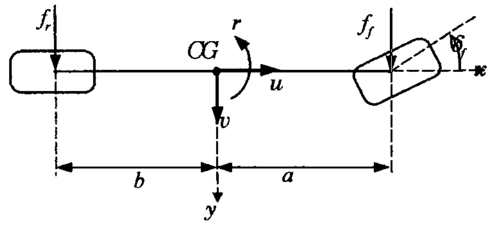


Fig. 1 Single-track model

Finally, the discussion and the main conclusions are given in Sec. 5.

2. Lateral Control

2.1 Vehicle models for simulation

The classical single-track model (Ohnuma A. and Metz. L. D, 1989) is obtained by lumping the two front wheels into one wheel in the centerline of the vehicle, the same is done with the two rear wheels as shown in Fig. 1. This model is described with the following variables and parameters:

- a : distance from the vehicle C. G. to front axle [m]
- b : distance from the vehicle C. G. to rear axle [m]
- c_f/c_r : front/rear tire cornering stiffness [N/rad]
- I_{zz} : yaw moment of inertia [kgm^2]
- l : wheel base (i.e. $l=a+b$)
- m : vehicle total mass [kg]
- r : yaw rate [rad/sec]
- u : longitudinal velocity of the vehicle at C. G. [m/s]
- v : side-slip velocity of the vehicle at C. G. [m/s]
- f : front steering angle input [rad]

The vehicle models are given by :

$$\begin{bmatrix}\dot{v}\\ \dot{r}\end{bmatrix}=\begin{bmatrix} -\dfrac{2(C_f+C_r)}{mu} & -u-\dfrac{l(aC_f-bC_r)}{mu} \\ -\dfrac{2(aC_f-bC_r)}{I_{zz}u} & -\dfrac{2(a^2C_f+b^2C_r)}{I_{zz}u}\end{bmatrix}\begin{bmatrix}v\\ r\end{bmatrix}+\begin{bmatrix}\dfrac{2C_f}{m}\\ \dfrac{2aC_f}{I_{zz}}\end{bmatrix}\delta_f \qquad (1)$$

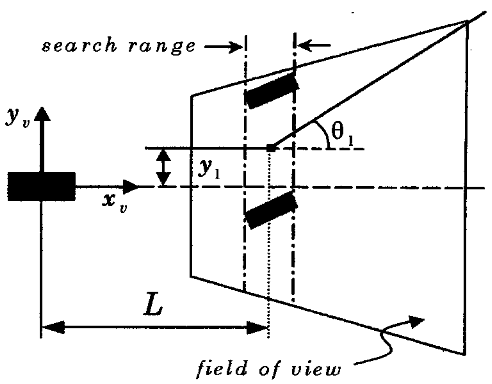


Fig. 2 A field of view of a vehicle

The front wheel steering actuator is assumed to be dominated by the first order delay (J. Ackermann et al., 1994).

$$\theta_m(s)=\frac{1}{Ts+1}U(s) \qquad (2)$$

where T is a time constant of motor, θ_m motor (or handle) angle, and U control input. From the IFAC benchmark example (J. Ackeramann and W. Darenberg, 1990), the steering angle is limited to $|\delta_f|\leqq 40$ deg and the steering angle rate is limited to $|\dot{\delta}_f|\leqq 23$ deg/s. Steering system from motor (or handle) angle θ_m to wheel angle δ_f is modeled as a gear, with the gear ratio of n.

$$\delta_f=n\theta_m \qquad (3)$$

Combining the dynamics of the actuator and the vehicle dynamics, a 3rd order state space model with states $X=[v\ r\ \theta_m]^T$ is obtained as :

$$\dot{X}=\begin{bmatrix} a_{11} & a_{12} & b_{11}\cdot n \\ a_{21} & a_{22} & b_{21}\cdot n \\ 0 & 0 & -1/T\end{bmatrix}X+\begin{bmatrix}0\\0\\1/T\end{bmatrix}u \qquad (4)$$

2.2 Lateral control algorithm

In Fig. 2, where the position of the vehicle is at the origin and its heading is zero, let (L, y_1) be the present target point and θ_1 be the heading angle. M. Yanagiya (1999) found the optimal distance L of the view from simulation with a single-track model. The cubic curve that goes through the two points is uniquely defined as follows:

$$y=Ax^3+Bx^2 \qquad (5)$$

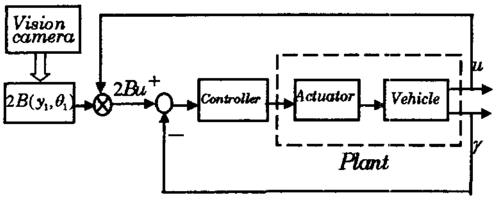


Fig. 3 The lateral control by the yaw rate feedback

where $A=\dfrac{L \tan \theta_1 - 2y_1}{L^3}$ and $B=\dfrac{3y_1 - L \tan \theta_1}{L^2}$.

The desired trajectories are independent of the longitudinal velocity u and the desired yaw rate r is given by

$$r=2Bu \tag{6}$$

In research of S. Tsugawa (1994), the vehicle is considered as no slip model and the steering control input is presented by

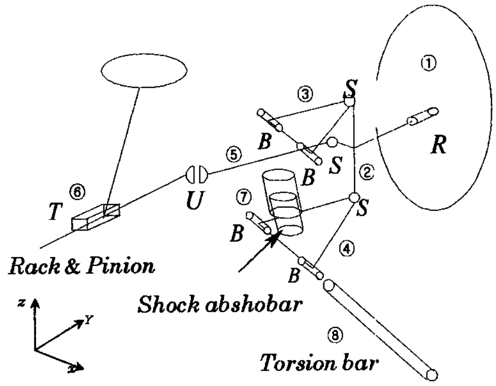
$$\delta_f = arctan(2lB) \tag{7}$$

But in the case of the actual vehicle, the delay of steering angle and the tire slip can't be ignored. To cope with them, the feedback of yaw rate is introduced where the yaw rate is measurable by yaw rate sensor such as gyroscope. The PI controller is tuned to minimize the error between the desired yaw rate and the measure one. Figure 3 shows the overall structure of the lateral control.

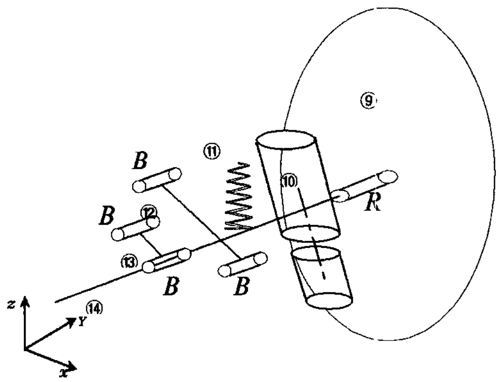
3. Multibody Dynamics of Vehicle

3.1 Suspension systems and tire model

To improve the accuracy of the vehicle model in simulation, bushings are considered between suspension systems and chassis. The bushing made by rubber is modeled as each 3-dimensional TSDA (translational spring damper actuator) and RSDA (rotational spring damper actuator). Totally there are six stiffness-parameters and six damping parameters (E. J. Haug, 1989). The front suspensions are independent suspensions with double wishbone type where shock absorbers are modeled TSDA. To reduce the rolling of vehicle, a stabilizer bar is considered. On the other hand, the rear left and right suspensions are linked like that four links type. Figure 4 shows the front and rear suspension systems. Where R, S, T, U, and B represent



(a) Double wishbone type front right



(b) Four links type rear suspension

Fig. 4 Front and rear suspension systems

revolute joint, spherical joint, translational joint, universal joint, and bushing. Each bodies are tire ①/⑨, front right knuckle ②, front upper/lower arm ③/④, front tie rod ⑤, Rack & pinion ⑥, front shock absorber ⑦/⑩, front torsion bar ⑧, rear spring ⑩, lower/upper arm link ⑫/⑬, and rear axle ⑭.

Since tire is the only mediator transferring the actuating, breaking, steering forces between vehicle and road, it is the most important component in the field of vehicle dynamics. So the dynamic characteristics of vehicle are dependent on the type of tire model. Generally, tire is experimentally or theoretically modeled. The carpet plot model calculating the tire forces and moments by data interpolation is one of the experimental models. Another experimental one is the magic formula model which generates a function from the input-output relations and determines forces and moments. While the theo-

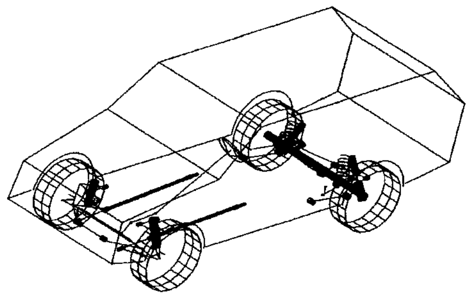


Fig. 5 Full vehicle model

Table 1 Specifications of actual and model vehicles

	Actual Vehicle	Modeled Vehicle
Drive	2 WD	Initial velocity
Tire Pressure	28psi(FRT) 32psi (RR)	38psi(FRT, RR)
Corner Load(Kg)	LF:462 RF:452 LR:537 RR:568	LF:469 RF:445 LR:552 RR:553
Speed	80Km/h (J-turn)	80Km/h (J-turn)

retical tire models are based on theoretical calculation of the vertical, longitudinal, and lateral forces. In this paper, tires are modeled as the theoretical one assuming a point contact on the road (DADS user's manual). The theoretical model has restrictions for low longitudinal velocity, when the calculated slip ratio may diverge.

From the suspension systems including busing and theoretical tire model, the overall vehicle is constructed as shown in Fig. 5 where the degrees of freedom are 79.

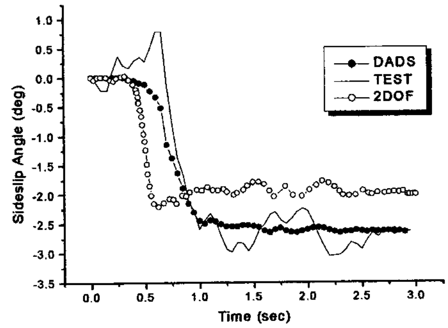


Fig. 6 Side-slip angle at the front wheel in step steering test (22m/s)

3.2 Verifications of the modeled vehicle

To verify the modeled vehicle, its dynamic characteristics are compared with the actual vehicle's. Table 1 presents the specifications of the actual vehicle and modeled vehicle. Commonly J-turn test is carried out to verify the steering performances such as the rollover or the riding performance. In simulation, the handle angle is stepped up to 34° within 0.2 second when the longitudinal velocity is 22m/s. Figures 6-8 show the results of J-turn for the single-track model, the multibody model, and the actual vehicle. In Fig. 6, the side-slip of the single-track model is a little larger then the actual slip. So its yaw rate and lateral acceleration are larger then others as shown in Fig. 7, 8. The multibody model shows almost similar characteristics to the actual vehicle's. Therefore, it is possible to replace the actual vehicle with the single-track model or the multibody model in the simulation of lateral control algorithm.

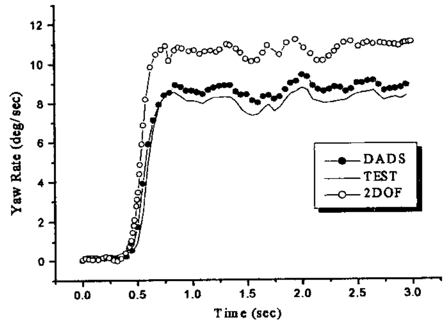


Fig. 7 Yaw rate in step steering test (22m/s)

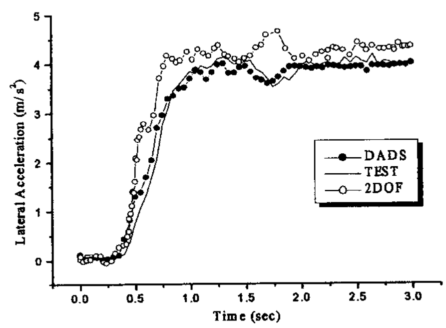
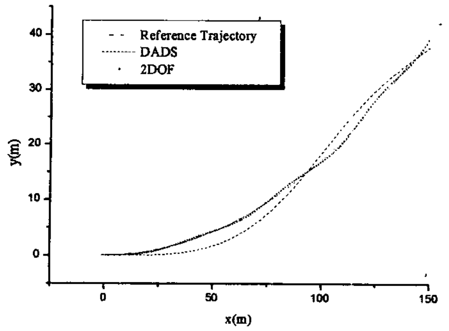
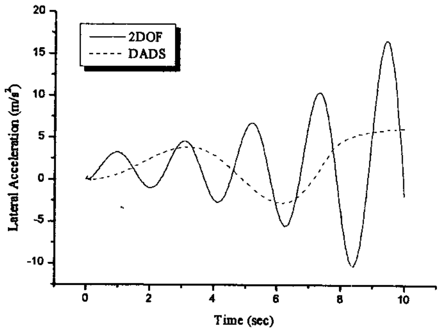


Fig. 8 Lateral accelerations in step steering test (22m/s)

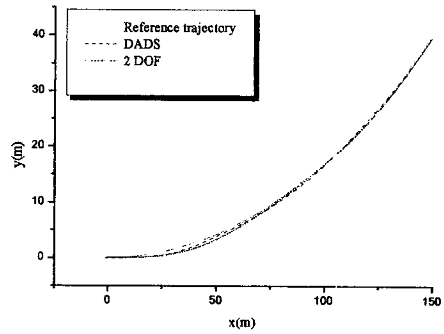


(a) Trajectories of vehicle C. G.



(b) Lateral accelerations of vehicle

Fig. 9 The results of lateral control by Target Point Following algorithm when vehicle tracks the circular lane (r=300m, 22m/s)



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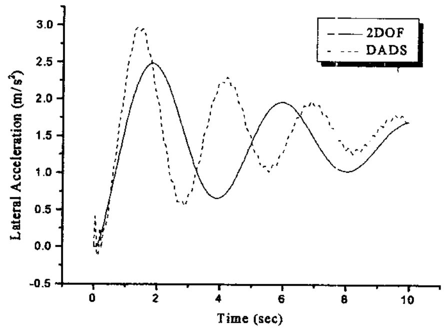

(b) Lateral accelerations of vehicle

Fig. 10 The results of lateral control by proposed algorithm when vehicle tracks the circular lane (r=300m, 22m/s)

4. Simulation for Lateral Control

Simulations for the lateral control of vehicle are performed to the single-track model and multibody model where the reference lane is a circle with 300m radius. The distance of view is set up to 20m (i.e. L=20m) which values comes from M. Yanagiya et al. (1999). Figure 9 shows the trajectories of the C. G. and the lateral accelerations when vehicle tracks the circle by Tsugawa method. The differences of results between two models are occurred by actuator model. Because the actuator was considered with the first order delay with time-constant for simplicity, the reaction forces from tire or another bodies aren't transfer to handle in the single-track model. In general, the lateral accelerations should not exceed $4m/s^2$ for the rollover and $2m/s^2$ for passenger comfort in J. Ackermann et al. (1995). In Fig. 9(b), since the lateral accelerations rise above $4m/s^2$, it is impossible to realize the lateral control algorithm of Tsugawa when a vehicle has the side-slip and the delay of steering actuator. Figure 10 shows the results of PI controlled vehicle by yaw rate feedback. Although the peak of lateral acceleration is approximately $3m/s^2$, lateral acceleration is settled in $1.5m/s^2$ where the initial tracking error is caused by the discontinuity of lane curvatures from infinity (straight lane) to 1/300 (circular lane). Therefore, the PI controller by yaw rate feedback shows a good performance for the lane tracking of an autonomous vehicle.

5. Conclusion

An autonomous vehicle was considered for the lateral control with a PI controller. The controller was designed to minimized the error between the reference yaw rate and the measured one. The lane information detected by vision system were generated to the reference yaw rate. When the delay of steering actuator and the side-slip of vehicle was occurred, the lateral control algorithm by the kinematical relationship could not be used. So the feedback control of yaw rate had been introduced and simulated on the single-track model and the multibody model which were verified by results of actual vehicle. When the vehicle followed the circle trajectory, the lateral accelerations and the tracking errors by the proposed feedback were settled down while they by the Tsugawa method were diverged.

There are two future problems; one is a research for the delay of lane information caused by image processing and the other is integration of the lateral control and the longitudinal control to satisfy the limit of lateral acceleration against small curvature.

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