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Design and experiment of fuzzy PID yaw rate controller for an electrically driven four wheel vehicle without steering mechanism

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ABSTRACT

Design and experimental results of yaw rate controller is described for electrically driven four wheel vehicle without steering mechanism. Yaw rate controller has been known to be necessary to cope with nonlinear characteristics of the wheel/road conditions with respect to different road condition and steering angle. For an effective yaw rate control, a fuzzy PID gain scheduler is considered with changing control parameters. In order to apply proposed algorithm to the system, a downsized four wheel drive electrically driven vehicle without steering mechanism was manufactured. With these techniques, the proposed yaw rate controller is shown by experiment results to be obtained sufficient performance in the whole steering regions.

1. Introduction

Automatic vehicle control is considered to play a key role in increasing traffic capacity and relieving drivers load. Major research activities in automatic vehicle control are devoted to longitudinal and lateral control. In automatic lateral control, vehicle must follow a given path, normally at the center of the lane. Moreover, vehicle direction must be controlled as desired. For example, in a high speed lane change situation, the yaw rate of the vehicle should be kept small in order to avoid a spin. Therefore, the lateral motion and yaw motion of the vehicle should be controlled independently [1]. In case of engine driven vehicle, present study explores the use of the differential driving torque in right and left wheels for the lateral velocity and yaw rate control system [2].

For the control of the vehicle steering, a yaw rate signal has been mainly employed as a feedback signal in many research works [3,4]. Also, it should be noted that yaw motion dynamics of a vehicle are heavily influenced by vehicle parameters such as vehicle velocity and road-tire interaction. These vehicle parameters vary during operation, and thus it must be addressed in controller design.

Recently, a great deal of attention has been focused on the research of electrically driven vehicle systems which has a variety of application areas and usefulness [5]. Also, electrically driven vehicle makes it possible to independently control each wheel, which has been positively used to develop a control system for special purpose vehicles such as unmanned vehicle and off-road vehicle [6].

In this paper, a yaw rate control system is designed for electrically driven four wheel vehicle without steering mechanism which is assumed to have 3 degrees of freedom motion. Vehicle reference trajectory is determined by considering vehicle kinematics. For the control of vehicle steering, differential steering method is employed, where each wheel is controlled by velocity servo. It is noted that for the case of our vehicle, steering can be made by vehicle side slip angle given by road condition. And this requires the control of yaw rate of the vehicle. However for an effective yaw rate control, it should be considered that characteristics of the wheel/road conditions are nonlinear with respect to both road condition and steering angle. And such conditions also depend on several uncontrollable external physical factors, such as road/tire interface properties including road condition, weariness of tire and tire pressure, which are difficult to sense or estimate [7,8]. Thus, a simple linear controller with fixed gains cannot be applied for the purpose of yaw rate control, since such a controller cannot always effectively control nonlinear systems with changing parameters. Thus, yaw rate control parameters may be required to be adapted on-line based on parameter estimation, which requires certain knowledge of vehicle systems.

To overcome such difficulties, a fuzzy PID gain scheduler is here considered. In order to apply the proposed algorithm to the system, a downsized electrically driven four wheel vehicle without steering mechanism is manufactured. Here, the center of gravity of the vehicle is assumed to be located on road level. Then, there are no couplings on pitch and roll motions. Thus, vehicle kinematics can be derived by employing a simple planar model with 3 degrees of freedom. Proposed overall control system consists of a wheel velocity controllers, vehicle velocity controller and yaw rate controller.

The remainder of this paper is organized as follows; In Section 2, kinematics of our four wheel drive/steering vehicle is briefly described. A fuzzy PID gain scheduler is designed for yaw rate controller in Section 3. The proposed control algorithm has been tested on various yaw rate control experiments in Section 4. Finally, in Section 5, some concluding remarks are given.

2. Vehicle Kinematics

Consider an electrically four wheel drive/steering vehicle as shown in Fig. 1, where two electric motors are attached to each wheel for steering and driving, respectively. And consider the standard vehicle body centered reference frame for the four wheel vehicle in Fig. 1, where v_{fr}, v_{fl}, v_{rr}, v_{rl} and v imply the local velocity vectors in front-right, front-left, rear-right, rear-left

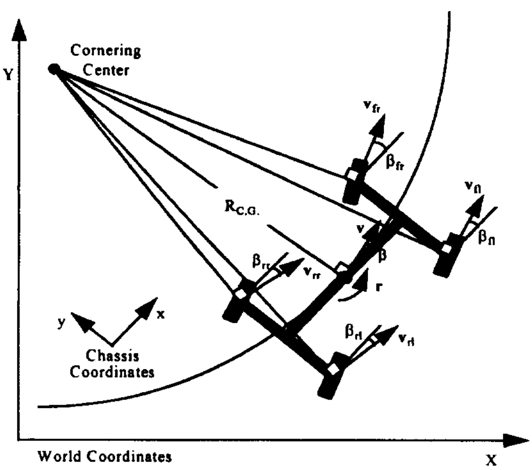


Fig. 1. Coordinates of 4WS/4WD vehicle

wheels and center of gravity(C.G.), respectively. These velocity vectors are perpendicular to the lineconnecting with the cornering center, usually located far away from C.G. of the vehicle in world coordinates. Cornering radius($R_{C.G.}$) implies the distance between center of gravity and cornering center. And, β_{fr}, β_{fl}, β_{rr}, and β_{rl} imply angles between the longitudinal center line and the direction of velocity vectors of each wheel, respectively. Let β and r be the vehicle side-slip angle and the yaw rate in C.G., respectively.[9]

Then, we can obtain local velocity vector from kinematics relation with vehicle velocity (v), side-slip angle (b) and the yaw rate(r) in center of gravity as follows;

$$v\cos\beta = v_{fr}\cos\beta_{fr}-l_f r = v_{fl}\cos\beta_{fl}+l_f r \\ = v_{rr}\cos\beta_{rr}-l_r r = v_{rl}\cos\beta_{rl}+l_r r, \tag{1}$$

and

$$v\sin\beta = v_{fr}\sin\beta_{fr}-dr = v_{fl}\sin\beta_{fl}-dr \\ = v_{rr}\sin\beta_{rr}+dr = v_{rl}\sin\beta_{rl}+dr, \tag{2}$$

where l_f and l_r imply the distances from C.G. to the front axle and rear axle, respectively, and d denotes the half vehicle width.

Now, from Eqs. (1) and (2), side slip angle and vehicle velocity of each wheel can be given as

$$\beta_{fr} = \tan^{-1}\left(\frac{v\sin\beta + l_f r}{v\cos\beta + dr}\right), \tag{3}$$

$$\beta_{fl} = \tan^{-1}\left(\frac{v\sin\beta + l_f r}{v\cos\beta - dr}\right), \tag{4}$$

$$\beta_{rr} = \tan^{-1}\left(\frac{v\sin\beta - l_f r}{v\cos\beta + dr}\right), \tag{5}$$

$$\beta_{rl} = \tan^{-1}\left(\frac{v\sin\beta - l_f r}{v\cos\beta - dr}\right), \tag{6}$$

$$v_{fr} = \sqrt{(v\sin\beta + l_f r)^2 + (v\cos\beta + dr)^2}, \tag{7}$$

$$v_{fl} = \sqrt{(v\sin\beta + l_f r)^2 + (v\cos\beta - dr)^2}, \tag{8}$$

$$v_{rr} = \sqrt{(v\sin\beta - l_f r)^2 + (v\cos\beta + dr)^2}, \tag{9}$$

and

$$v_{rl} = \sqrt{(v\sin\beta - l_f r)^2 + (v\cos\beta - dr)^2}. \tag{10}$$

Let α_w and λ_w be the wheel side slip angle and wheel slip ratio, respectively. Here, subscript w means one of wheels among front right (fr), front left (fl), rear right

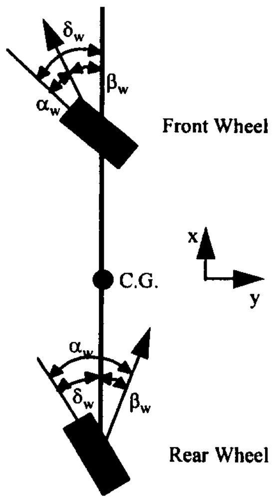


Fig. 2. Wheel slip angle.

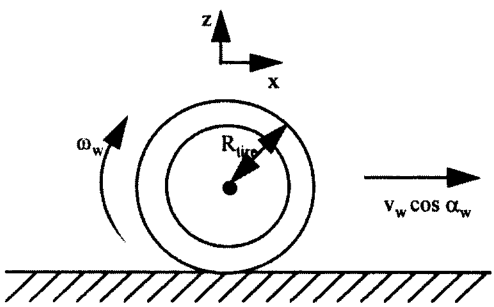


Fig. 3. Wheel slip ratio.

(rr), rear left (rl). Then, from Figs. 2 and 3, α_w and λ_w can be obtained by

$$\alpha_w = \delta_w - \beta_w, \tag{11}$$

and

$$\lambda_w = \frac{v_w \cos\alpha_w - \omega_w R_{tire}}{v_w \cos\alpha_w}. \tag{12}$$

And, ω_w, δ_w and R_{tire} imply wheel angular velocity, commanded steering angle, tire radius, respectively.

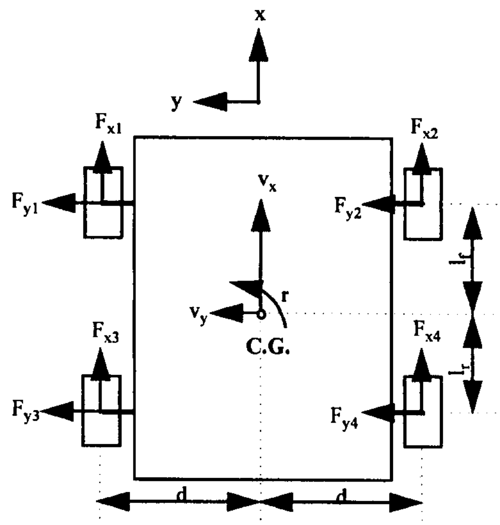


Fig. 4. Structure of 4WD vehicle without steering mechanism.

And $v_w\cos\alpha_w$ means longitudinal direction velocity of each wheel.[8,10]

In case of no steering mechanism as shown in Fig. 4, wheel slip angle becomes equal to zero, and thus, from Eq.(11), commanded steering angle will be the same as the vehicle side slip angle. In other words, steering can be possible by controlling side slip angle. For an effective steering control, it should be considered that side slip angle will be affected by both road parameters and the steering angle. But, such parameters also depend on several uncontrollable external physical factors, such as road/tire interface properties including road condition, weariness of tire and tire pressure, which are difficult to sense or estimate

3. Design of Fuzzy-PID yaw rate controller

As shown in Fig. 5, our proposed yaw rate controller,

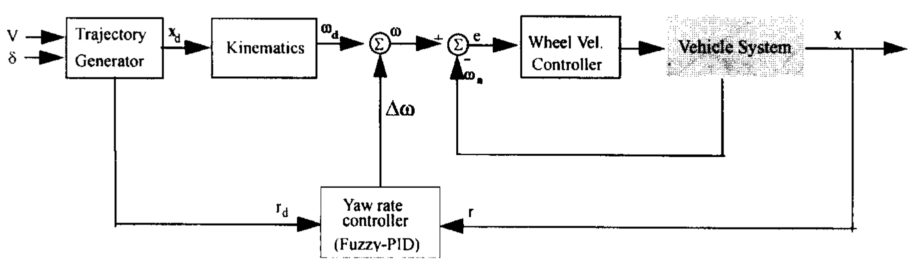


Fig. 5. Block diagram of yaw rate control system.

computes desired state to be required for vehicle kinematics. The kinematics relation is applied to obtain desired wheel angular velocity. Then, a fuzzy-PID yaw rate controller is designed to compensate the yaw rate error which caused by several uncontrollable external physical factors, such as road/tire interface properties, weariness of tire and tire pressure. Here, wheel velocity controller is designed to be a velocity servo.

It is desired that a vehicle must be controlled by proper steerings to move along the tangential line of circular arcs. If the vehicle moves along the tangential line of circular arcs, then lateral velocity of the vehicle must be zero, and longitudinal velocity of the vehicle must be the same as desired vehicle velocity, where yaw rate is given by $V/R_{C.G.}$ Therefore, desired state can be obtained as

$$v_{xd} = V, \tag{13}$$
$$v_{yd} = 0, \tag{14}$$

and

$$r_d = \frac{v}{R_{C.G}} = \frac{v}{(l_f/\tan\delta)+d} \tag{15}$$

where $R_{C.G}$, δ imply cornering radius, commanded steering angle, respectively.

It is noted that desired angular velocity of the wheel can be directly obtained from kinematics relation. If there is no wheel slippage, wheel angular velocity can be written as

$$v_{id} = R_{tire}\omega_{id}. \tag{16}$$

Thus, desired angular velocity of each wheel can be obtained as

$$\omega_{1d} = \frac{1}{R_{tire}}\sqrt{(v_{xd}-r_d d)^2+(v_{yd}+r_d l_f)^2}, \tag{17}$$

$$\omega_{2d} = \frac{1}{R_{tire}}\sqrt{(v_{xd}+r_d d)^2+(v_{yd}+r_d l_f)^2}, \tag{18}$$

$$\omega_{3d} = \frac{1}{R_{tire}}\sqrt{(v_{xd}-r_d d)^2+(v_{yd}-r_d l_f)^2}, \tag{19}$$

and

$$\omega_{4d} = \frac{1}{R_{tire}}\sqrt{(v_{xd}+r_d d)^2+(v_{yd}-r_d l_f)^2}. \tag{20}$$

Now, yaw rate controller is designed for yaw rate error compensator. But, accurate compensations are difficult due to side slip angle which shows nonlinearities and uncertainties. Therefore, instead of using a simple linear controller as yaw rate compensator, a fuzzy PID gain scheduler is here employed in such a way that yaw rate compensator output and desired angular velocity are combined and transferred to the wheel velocity controller. Thus, compensated desired angular velocity can be obtained as

$$\hat{\omega}_{1d} = \omega_{1d}+\Delta\omega_1, \tag{21}$$
$$\hat{\omega}_{2d} = \omega_{2d}+\Delta\omega_2, \tag{22}$$
$$\hat{\omega}_{3d} = \omega_{3d}+\Delta\omega_3, \tag{23}$$

and

$$\hat{\omega}_{4d} = \omega_{4d}+\Delta\omega_4, \tag{24}$$

where $\Delta\omega_i$(i=1,2,3,4) imply the angular velocity compensation terms.

Consider the discrete-time PID control law whose form is given as

$$u(k) = K_p e(k) + K_i T_s \sum_{i=1}^{n} e(i) + \frac{K_d}{T_s}\Delta e(k)' \tag{25}$$

where u(k), e(k) and T_s imply control input, error and sampling period for the controller, respectively. The parameters of the PID controller can be manipulated to produce various response curves from a given process. Finding optimum adjustments of a controller for a given process is not trivial. PID control system with a fuzzy gain scheduler is to exploit fuzzy rule and reasoning to generate controller parameters.[11] PID parameters are determined based on error e(k) and its first difference Δe(k). Specifically it is assumed that K_p, K_d are in prescribed ranges [$K_{p,min}$, $K_{p,max}$] and [$K_{d,min}$, $K_{d,max}$], respectively. The appropriate ranges are determined experimentally. For convenience, K_p and K_d are normalized into the range between zero and one by the following linear transformation

$$\hat{K}_p = \frac{K_p - K_{p,min}}{K_{p,max}-K_{p,min}} \tag{26}$$

$$\hat{K}_d = \frac{K_d - K_{d,min}}{K_{d,max}-K_{d,min}} \tag{27}$$

In the well-known Ziegler-Nichols PID tuning rule, the integral time constant T_i is always taken four times as large as the derivative time constant T_d·[12] For the optimum adjustments of a controller, the integral time constant is determined with reference to the derivative time constant, i.e.,

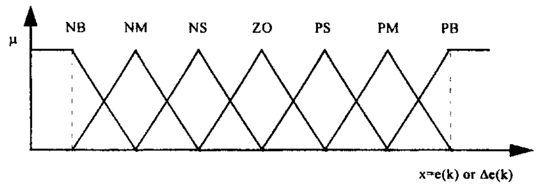


Fig. 6. Membership functions for e(k) and Δe(k).

$$T_i = \alpha T_d \tag{28}$$

and the integral gain is thus obtained by

$$K_i = K_p/(\alpha T_d) = K_2^{P}(\alpha K_d) \tag{29}$$

The parameters $\hat{K}_p$, $\hat{K}_d$ and α are determined by a set of fuzzy rules of the form

If e(k) is A_i and Δe(k) is B_i, then $\hat{K}_p$ is C_i and $\hat{K}_d$ is D_i and $\alpha = \alpha$, i=1, 2, 3, ..., m, (30)

where A_i, B_i, C_i and D_i are fuzzy sets on the corresponding supporting sets, and α_i is a constant. These fuzzy rules may be extracted from operators expertise. Here, we derive the rules based on step response of the process. At the beginning of a desired time response, we need a big control signal in order to achieve a fast rise time. To produce a big control signal, the PID controller should have a large proportional gain, a large integral gain, and small derivative gain. And around the desired value, we expect a small control signal to avoid a large overshoot. That is, the PID controller should have a small proportional gain, a large derivative gain, and a small integral gain. The membership functions of these fuzzy set for e(k) and Δe(k) are shown in Fig. 6.

In this Fig. 7 linguistic value, for each variable are used as follows ; NB(negative big) NM(negative medium) NS(negative small) ZO(zero) PS(positive small) PM (positive medium) PB(positive big).

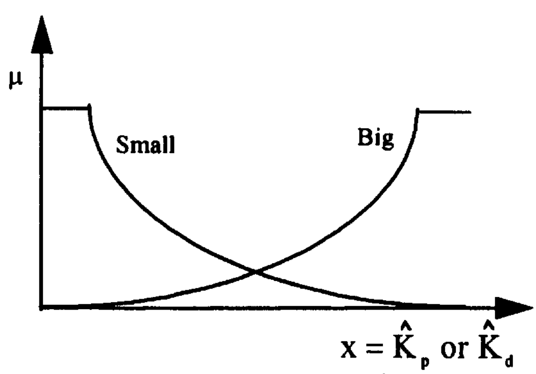


Fig. 7. Membership functions for $\hat{K}_p$ and $\hat{K}_d$.

The fuzzy sets C_i and D_i may be either Big or Small and are characterized by the membership functions shown Fig. 7, where the grade of the membership functions μ and the variable x(= $\hat{K}_p$ or $\hat{K}_d$) have the following relation;

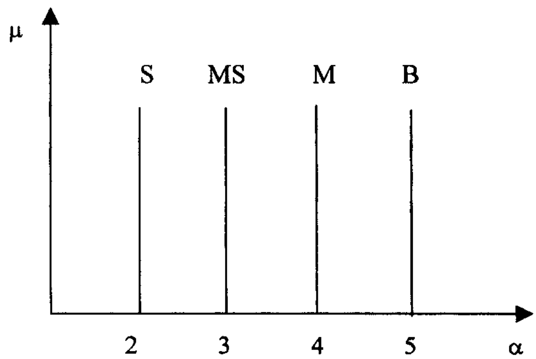


Fig. 8. Singleton membership functions for α.

Table 1. Fuzzy rule for $\hat{K}_p$, $\hat{K}_d$ and α.

$\hat{K}_p$		Δe(k)						
		NB	NM	NS	ZO	PS	PM	PB
e(k)	NB	B	B	B	B	B	B	B
	NM	S	B	B	B	B	B	S
	NS	S	S	B	B	B	S	S
	ZO	S	S	S	B	S	S	S
	PS	S	S	B	B	B	S	S
	PM	S	B	B	B	B	B	S
	PB	B	B	B	B	B	B	B

$\hat{K}_p$		Δe(k)						
		NB	NM	NS	ZO	PS	PM	PB
e(k)	NB	S	S	S	S	S	S	S
	NM	B	B	S	S	S	B	B
	NS	B	B	B	S	B	B	B
	ZO	B	B	B	B	B	B	B
	PS	B	B	B	S	B	B	B
	PM	B	B	S	S	S	B	B
	PB	S	S	S	S	S	S	S

α		Δe(k)						
		NB	NM	NS	ZO	PS	PM	PB
e(k)	NB	2	2	2	2	2	2	2
	NM	3	3	2	2	2	3	3
	NS	4	3	3	2	3	3	4
	ZO	5	4	3	3	3	4	5
	PS	4	3	3	2	3	3	4
	PM	3	3	2	2	2	3	3
	S	2	2	2	2	2	2	2

$$x_{Small}(\mu) = e^{-4\mu}, \text{ for small,} \tag{31}$$
$$x_{Big}(\mu) = 1-e^{-4\mu}, \text{ for Big.} \tag{32}$$

Here, it is noted that α may be also considered as a fuzzy number which has a singleton membership function as shown in Fig. 8. Thus a set of rules, as shown Table 1, may be used to adapt the $\hat{K}_p$, $\hat{K}_d$ and α.

The truth table value of ith rule in (30) μ_i is obtained by the product of the membership function value in the antecedent part of the rule;

$$\mu_i = \mu_{Ai}[e(k)] \cdot \mu_{Bi}[\Delta e(k)], \tag{33}$$

where μ_{Ai} is the membership function value of the fuzzy set A_i given a value of e(k), and μ_{Bi} is the membership function value of the fuzzy set B_i given a value of Δe(k). The defuzzification is applied to center of area method. Then, defuzzification yield the following;

$$\hat{K}_p = \frac{\sum_{i=1}^{m} \mu_i \hat{K}_{p,i}}{\sum_{i=1}^{m} \mu_i}, \tag{34}$$

$$\hat{K}_d = \frac{\sum_{i=1}^{m} \mu_i \hat{K}_{d,i}}{\sum_{i=1}^{m} \mu_i}, \tag{35}$$

$$\alpha = \frac{\sum_{i=1}^{m} \mu_i \alpha_i}{\sum_{i=1}^{m} \mu_i}. \tag{36}$$

Here, $\hat{K}_{p,i}$ is the value of $\hat{K}_p$ corresponding to the grade μ_i for the ith rule. $\hat{K}_{d,i}$ is obtained in the same way. Once $\hat{K}_p$, $\hat{K}_d$ and α are obtained, the PID controller parameters can be obtained as

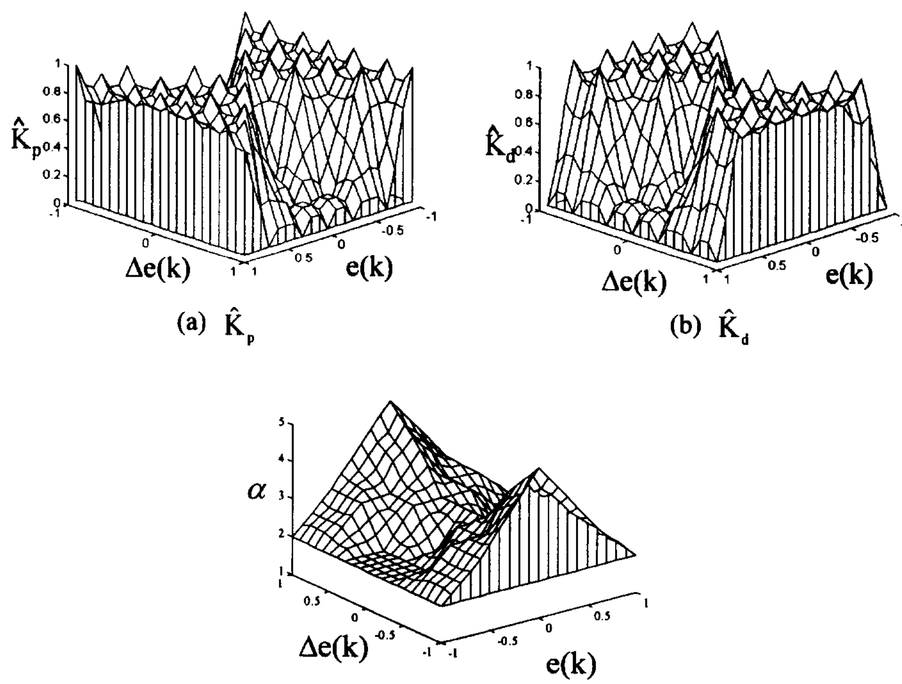
$$K_p = (K_{p,max} - K_{p,min})\ \hat{K}_p + K_{p,min}, \tag{37}$$
$$K_d = (K_{d,max} - K_{d,min})\ \hat{K}_d + K_{d,min}, \tag{38}$$
$$K_i = K_p^2/(\alpha K_d). \tag{39}$$

Fig. 9 shows fuzzy rule look-up table for $\hat{K}_p$, $\hat{K}_d$ and α.

4. Experimental results

4.1 Implementation of downsized laboratory vehicle

A downsized laboratory vehicle without steering mechanism has been built to evaluate the control algorithm experimentally. A photo and controller block diagram of the experimental vehicle are shown Fig. 10 and Fig. 11, respectively. Major technical specifications and functions are shown Table 2.



(c) α

Fig. 9. Fuzzy rule look-up table.

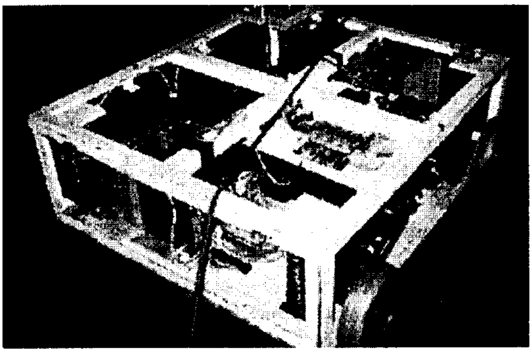


Fig. 10. Structure of downsized electrically driven vehicle.

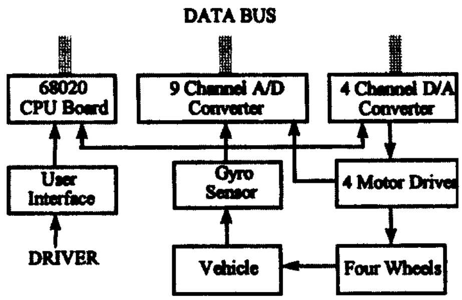


Fig. 11. Block diagram of vehicle controller.

Table 2. Specifications and function of electrically driven vehicle

Items	Specifications and Function
Vehicle Dimension	1200 * 1060 * 400 mm
Vehicle Mass	256.6 kg
Wheel diameter	260 mm
Max. vehicle velocity	20 km/h
Power	3Phase-220V
Motor	Induction Type Servo Motor withencoder 800W * 4
Functions	Forward/Backward, Differential steering, Pivoting, Spinning

The controllers are equipped with MC68020 CPU board, A/D converter board to process the I/O signals and D/A converter board to generate the wheel velocity command. Also, an angular velocity sensor (Murata Gyrostar ENV05S) is installed to measure the yaw rate of the vehicle. Induction servo motor drivers (Sumtak VID-001-308062) are applied to velocity servo control.

4.2 Experimental results

To show the validity of the proposed control algorithm, several experiments are illustrated that (i) in case of yaw rate controller is not employed (ii) in case of yaw rate controller is employed fixed PID gain (iii) in case of yaw rate controller is employed well tuned PID gain (iv) in case of yaw rate controller is employed fuzzy PID gain scheduler. In experiments, vehicle velocity and yaw rate feedback are obtained from rotary encoder in the wheel and angular velocity sensor, respectively.

In these experiments, the vehicle is desired to be steered 5 degree, 10 degree, 15 degree while traveling at the speed about 5 Km/h (about 1.4 m/sec). Therefore, desired yaw rates are 0.18 rad/sec, 0.35 rad/sec, 0.5 rad/sec, respectively.

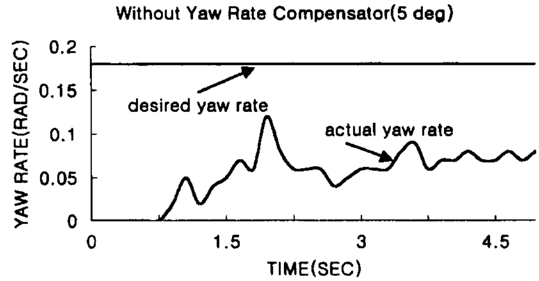


Fig. 12. Experimental result without yaw rate controller (5km/h,5 degree).

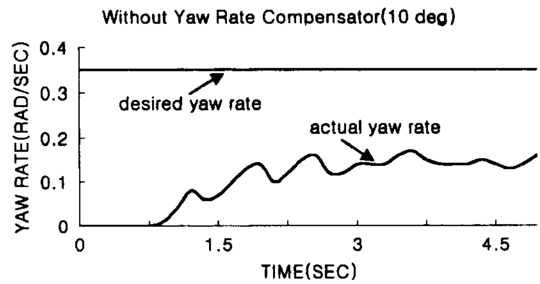


Fig. 13. Experimental result without yaw rate controller (5 km/h, 10 degree).

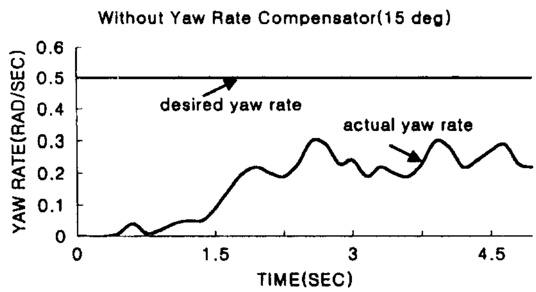


Fig .14. Experimental result without yaw rate controller (5 km/h, 15 degree).

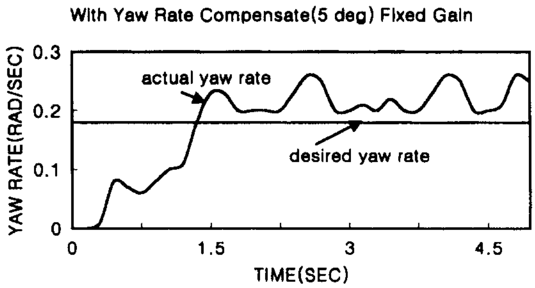


Fig. 15. Experimental result for the yaw rate controller is employed with fixed PID gain (5 km/h, 5 deg).

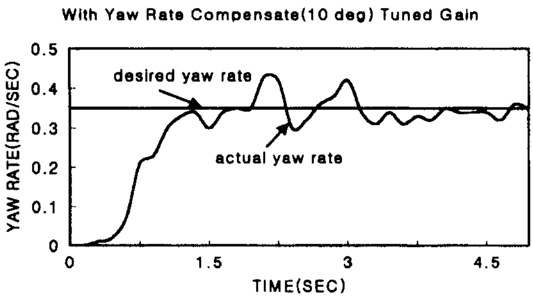


Fig. 16 Experimental result for the yaw rate controller is employed with fixed PID gain (5 km/h,10 deg).

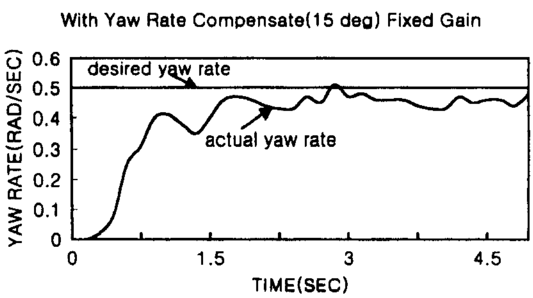


Fig. 17. Experimental result for the yaw rate controller is employed with fixed PID gain (5 km/h, 15 deg).

Case (i) : Fig. 12, Fig. 13 and Fig. 14 shows the output response for without yaw rate control. It is observed that the magnitude of yaw rate becomes smaller than the desired values in the whole steering region. This implies that understeer can be met when the vehicle steer. Therefore, it can be conclude that the yaw rate control is necessary to obtained desired steering performance.

Case (ii) : Fig. 15, Fig. 16 and Fig. 17 shows the output response for the fixed PID gain control. It is observed from Fig. 15 that the magnitude of yaw rate becomes larger than the desired value. This implies that oversteer can be met when the vehicle steer. In Fig. 16, the magnitude of yaw rate becomes to the desired value. This implies that neutral-steer can be met when the vehicle steer. In Fig. 17, the magnitude of yaw rate becomes smaller than the desired value. This implies that under-steer can be met when the vehicle steer. Therefore, it can be conclude that fixed PID gain control can not be adapted to each commanded steering angle.

Case(iii) : Thus, Fig. 18, Fig. 19 and Fig. 20 shows the output response for the tuned PID gain control. It is observed from Fig. 18, Fig. 19 and Fig. 20 that the magnitude of yaw rate becomes to the desired value. This implies that tuned PID gain control can be

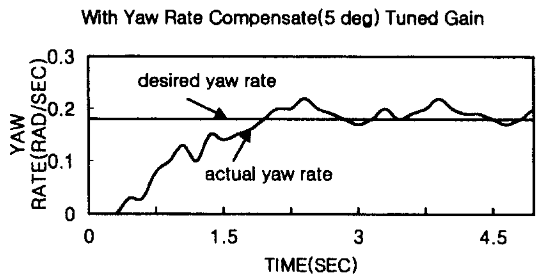


Fig. 18. Experimental result for the yaw rate controller is employed with tuned PID gain (5 km/h, 5 deg).

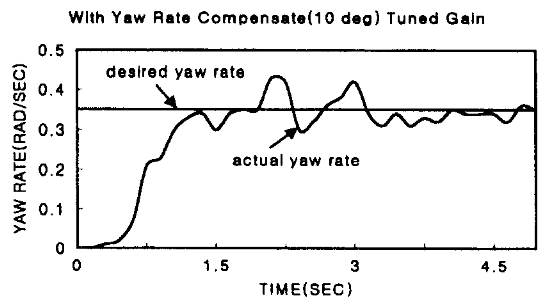


Fig. 19. Experimental result for the yaw rate controller is employed with tuned PID gain (5 km/h, 10 deg).

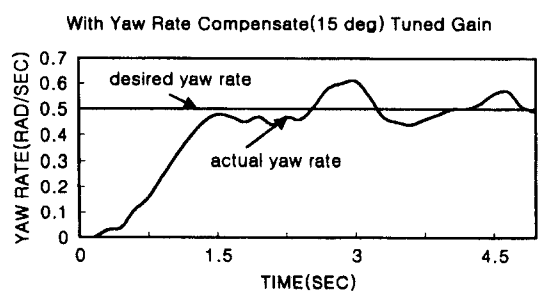


Fig. 20. Experimental result for the yaw rate controller is employed with tuned PID gain (5 km/h, 15 deg).

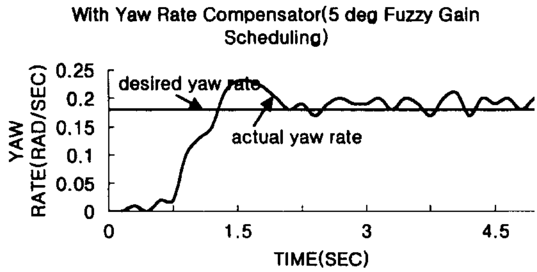


Fig. 21. Experimental result for the yaw rate controller is employed with fuzzy PID gain scheduler (5 km/h, 5 deg).

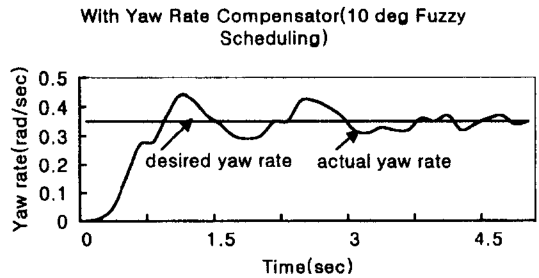


Fig. 22. Experimental result for the yaw rate controller is employed with fuzzy PID gain scheduler(5 km/h, 10 deg).

adapted to each commanded steering angle. It can be conclude that vehicle steering performances are heavily affected by control parameters. But, gain tuning process is required for many preceding experi-

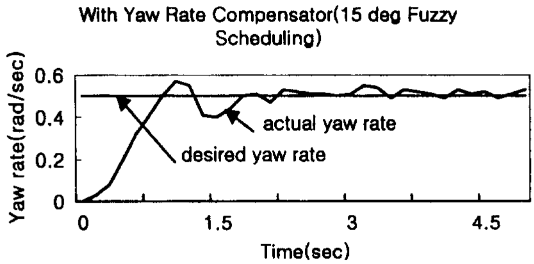


Fig. 23. Experimental result for the yaw rate controller is employed with fuzzy PID gain scheduler(5 km/h, 15 deg).

ments and it can not be obtained sufficient performance in whole steering region.

Case (iv): Fig. 21, Fig. 22 and Fig. 23 shows the output response for the fuzzy PID gain scheduler. It is observed from Fig. 21, Fig. 22 and Fig. 23 that the magnitude of yaw rate becomes to the desired value. This implies that fuzzy PID gain scheduler can be adapted to each commanded steering angle and obtained sufficient performance in the whole steering region.

The above experimental results show that a variety of steering angles can be satisfactorily controlled by fuzzy PID gain scheduler. It yields better control performance than fixed PID gain controller dose, which is confirmed by performance indexes such as the percent maximum overshoot, rising time and settling time. Table 3 shows the representative experimental results of the without yaw rate compensator, yaw rate com-

Table 3. Summary of performance indexes for experimental results

		5 Degrees	10 Degrees	15 Degrees
Without yaw rate compensator	Max. overshoot	?	?	?
	Rising time	?	?	?
	Settling time	?	?	?
Yaw rate compensator with fixed PID gain	Max. overshoot	27.77%	22.85%	2%
	Rising time	0.9 sec.	0.6 sec.	1.2 sec.
	Settling time	?	3.8 sec.	?
Yaw rate compensator with tuned PID gain	Max. overshoot	22.2%	22.85%	22%
	Rising time	1.29 sec.	0.6 sec.	0.82 sec.
Yaw rate compensator with tuned PID gain	Max. overshoot	22.2%	22.85%	22%
	Rising time	1.29 sec.	0.6 sec.	0.82 sec.
	Settling time	4.2 sec.	3.8 sec.	4 sec
Yaw rate compensator with fuzzy PID gain scheduler	Max. overshoot	27%	25.7%	14%
	Rising time	0.825 sec.	0.6 sec.	0.6 sec.
	Settling time	2.55 sec.	4.05 sec.	3.75 sec

*The ? marks are not available for the performance indexes.

*Yaw rate compensator with fixed PID gains are used to tuned PID gains at the 10 degrees of desired steering angle.

pensator with fixed PID gain, yaw rate compensator with tuned PID gain and yaw rate compensator with fuzzy PID gain scheduler.

5. Conclusions and remarks

In this paper, a yaw rate controller was designed for electrically driven four wheel vehicle without steering mechanism which was assumed to have 3 degrees of freedom motion. Vehicle reference trajectory was determined by trajectory generator which was generated trajectory from vehicle kinematics. Yaw rate controller should be considered that characteristics of the wheel/road conditions are nonlinear with respect to different road condition and steering angle. For an effective yaw rate control, a fuzzy PID gain scheduler was considered with changing control parameters. In order to apply proposed algorithm to the system, downsized electrically driven four wheel vehicle without steering mechanism was manufactured. From experimental results, it can be concluded that the proposed yaw rate controller showed relatively good performances in the whole steering region. In our future research work, we are investigating the effective technique for state measurement and/or estimation of the vehicle to apply dynamics based control algorithm to the vehicle.

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