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DEVELOPMENT OF AN ACTIVE FRONT STEERING SYSTEM

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ABSTRACT−We have developed an active front steering system (AFS) with a planetary gear train, which can vary the steering gear ratio according to the vehicle speed and improve vehicle stability by superimposing steering angle. We conducted vehicle tests showing that co-operated control of AFS with ESP can improve vehicle stability by direct control of tire slip angle and that steering reaction torque during AFS intervention can be compensated by torque compensation using electric power steering.

KEY WORDS : Type steering control, Electric power steering, Active front steering, Electronic stability program

1. INTRODUCTION

Recently steering technology focuses on not only driver's convenience, fuel efficiency, environment friendliness but also improved vehicle stability. However, EPS (Electric Power Steering) basically has the constant steering ratio. Therefore, it has limitations to control vehicle behavior. Meanwhile, SBW (Steer-by-Wire) could offer all of the advantages of EPS and control vehicle behavior with better controllability, but SBW cannot reach to mass-production due to reliability in case of failures and a legal restriction which states steering wheel must be mechanically linked to the front wheel. Therefore, AFS (Active Front Steering) which can provide variable gear ratio and improved vehicle stability, has been widely studied in the automotive industry (Schwarz *et al.*, 2003; Koehn *et al.*, 2004). Furthermore, AFS guarantees mechanical link in any cases. We have developed AFS and conducted vehicle tests resulting in improved vehicle stability on-split maneuver by cooperated control with ESP.

2. CONCEPT OF ACTIVE FRONT STEERING SYSTEM

AFS has many functions such as, Variable steering gear ratio, Direct yaw moment control, Applicability to advanced driver assist system.

AFS produces fast (direct) steering gear ratio at low speed so that drivers are able to steer vehicles with less steering maneuvers for parking or driving on city streets at low speeds. On the contrary, slow (indirect) gear ratio is produced at high speeds in order to provide drivers more solid and stable steering feel.

AFS enhances vehicle stability by making additional road wheel angle based on vehicle state information such as vehicle speed, steering angel, yaw rate and lateral acceleration.

AFS is applicable to advanced driver assist systems such as lane keeping system, crash avoidance system, etc.

3. SYSTEM DESIGN

3.1. Mechanism

Various methods have been contrived such as cam, link and differential mechanism. A planetary gear train is widely known among these mechanism due to its high degree of control, which makes it easy to vary steering

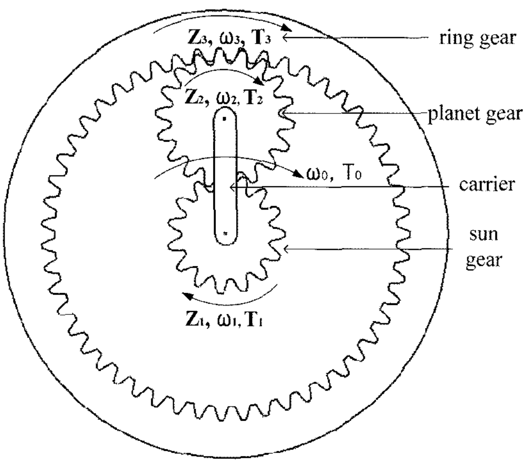

Figure 1. A simple planetary gear train.

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ratio and produce superimposed steering angle. A simple planetary gear mechanism is proposed as shown Figure 1. It consists of a sun gear, planet gears, an internal gear and a carrier. The variable characteristics are realized by controlling one of the above elements using an electric motor. In case of failure, the electric motor is locked and AFS sets to fixed steering gear ratio by keeping mechanical connection. The relations between four elements are shown in Equation (1), Equation (2) (Aoki and Hori, 2003).

$$Z_3\omega_3-(Z_1+Z_3)\omega_0+Z_1\omega_1=0 \quad (1)$$

$$T_1=\frac{Z_1}{Z_3}T_3=-\frac{Z_1}{Z_1+Z_3}T_0 \quad (2)$$

There can be many variations according to input-output choice but we chose the following arrangement shown in Figure 2 to have variable reduction ratio of steering input to output of 1:0.5~2.

The total number of steering turns from lock-to-lock varies according to vehicle speed as shown in Figure 3. It is possible to park with less steering turns and vehicle shows more agile maneuvers even in low speed driving. At the speed of 70 kph or higher the number of steering wheel turn becomes the same level of conventional steering.

3.2. Motor Power

Variable gear ratio is realized by rotating carrier with an electric motor. The power of motor should be taken into account in the design process. It is checked to see if it satisfies the following conditions.

(1) To satisfy the necessary response speed of steering system

In general, power steering system should meet steering response speed criteria for which power steering assists enough power in certain conditions. In case of AFS it should change gear ratio fast enough to reach target gear ratio in the same condition.

(2) Initialization of neutral position of steering wheel and front wheel

It is necessary to always keep track of input and output angle respectively to obtain target gear ratio according to the vehicle speed. If either input angle or output angle is turned arbitrarily somehow after system switched off, AFS should turn steering wheel automatically to the right position according to the current tire angle as a driver switches on the system.

From the specifications of steering system we chose following condition 1 shown in Table 1. For example at zero speed if a driver turns the steering wheel at the speed of 540 deg/sec AFS should be able to change the gear ratio without increasing steering effort more than 30 kgf-cm. Condition 2 is chosen from several vehicle information.

3.3. Motor Control

Since gear ratio varies according to the position of sun gear, it is required to control the position of motor, not torque. As shown in Figure 4, the position control system consists of a map of profile generation and a PID controller G(s) of cascade implementation.

As shown in Figure 5 the PID controller G(s) is composed of a position controller Gpc(s) and a speed

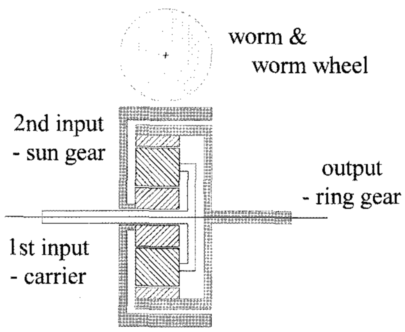


Figure 2. AFS mechanism.

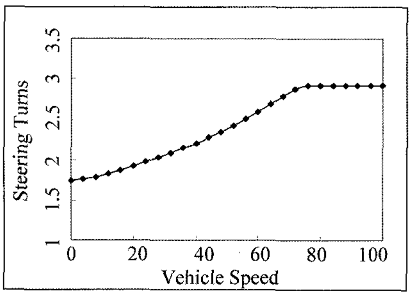


Figure 3. Change of steering turns.

Table 1. Conditions for AFS responsibility.

	Condition	Handwheel speed [deg/sec]	Steering effort [kgf-cm]
1	Steering while standing	Max 540	Max 30
	Steering while driving	Max 680	Max 80
2	Initialization while standing	Max 120	Max 100

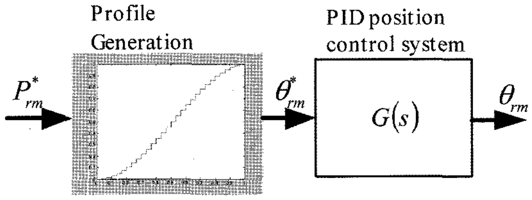


Figure 4. Motor position control system.

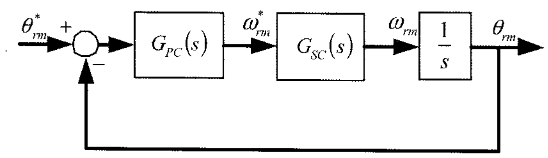


Figure 5. PID controller of motor speed.

controller Gsc(s) with a current controller for each component. The speed control block Gsc(s) secures stability and improves dynamic characteristics.

Because the command of motor control is determined with vehicle speed, the motor position should be able to respond fast enough to chase the speed rate. The maximum speed rate occurs during drastic deceleration and this is estimated to be 0.8G. The time to decrease vehicle speed at the rate of 0.8G, the maximum deceleration rate, is calculated to be 34.43 ms.

$$0.8G = 0.8\times 9.81 m/s^2 = 7.84 m/s^2$$
$$\frac{1kph}{7.84 m/s^2} = \frac{0.27 m/s}{7.84 m/s^2} = 34.43ms$$

This means the controller should reach to steady-state within 34.43 ms from the command. Based on the minimum response time, loop times of each controller are selected as follows.This is an example of the consideration.

$$34.43ms >> T_{PC} \cong 3 \sim 4ms$$
$$3ms \sim 4ms >> T_{SC} \cong 500us$$
$$500us >> T_{CC} \cong 62.5us$$

Tpc, Tsc and Tcc stand for the loop time of position, speed and current controller respectively. However shorter loop time is desirable in aspects of steering responsibility and cooperated control with ESP. The feedback variables, the position and current are measured values from sensors and the speed is estimated by state observer without an additional sensor. Even though it is possible to acquire speed signal from the differentiation of position it is not robust to noise in various speed conditions (Kim *et al.*, 2004).

A difference equation is presented in Equation (3) to realize the state observer with an embedded processor and its block diagram is shown in Figure 6.

$$\begin{bmatrix} \hat{\theta}_{rm}[n]-\hat{\theta}_{rm}[n-1] \\ \hat{\omega}_{rm}[n]-\hat{\omega}_{rm}[n-1] \\ \hat{T}_L[n]-\hat{T}_L[n-1] \end{bmatrix} = dt\cdot\begin{bmatrix} \hat{\omega}_{rm}[n-1]+\ell_1\left(\theta_{rm}[n-1]-\hat{\theta}_{rm}[n-1]\right) \\ -\frac{\hat{B}}{J}\hat{\omega}_{rm}[n-1]-\frac{1}{J}\hat{T}_L+\frac{1}{J}T_e+\ell_2\left(\theta_{rm}[n-1]-\hat{\theta}_{rm}[n-1]\right) \\ \ell_3\left(\theta_{rm}[n-1]-\hat{\theta}_{rm}[n-1]\right) \end{bmatrix} \quad (3)$$

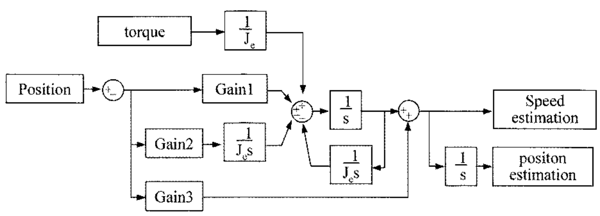


Figure 6. Block diagram of motor speed observer.

$\theta(angle): rad$
$t(time): s$
$\omega(anglespeed): rad/s$
$J(inertia): N\cdot m/rad\cdot s^{-2}$
$B(friction): N\cdot m/rad\cdot s^{-1}$

4. ESP & AFS CONTROL STRATEGY

4.1. ESP Control

The ESP system controls vehicle motion in emergency situations such that vehicle 'drifts out' from its intended driving course or 'spins out' when cornering radius is reducing sharply. In these situations ESP controls braking and traction force irrelevant to driver's braking operations to compensate cornering moment of vehicle (Van Zanten *et al.*, 1996).

The grip force of tire is made of cornering force and braking force resulting from driver's steering input and braking/acceleration. Balancing these forces allow vehicle to stay with the adhesion between the tire and road surface. As shown in Figure 7 braking force is

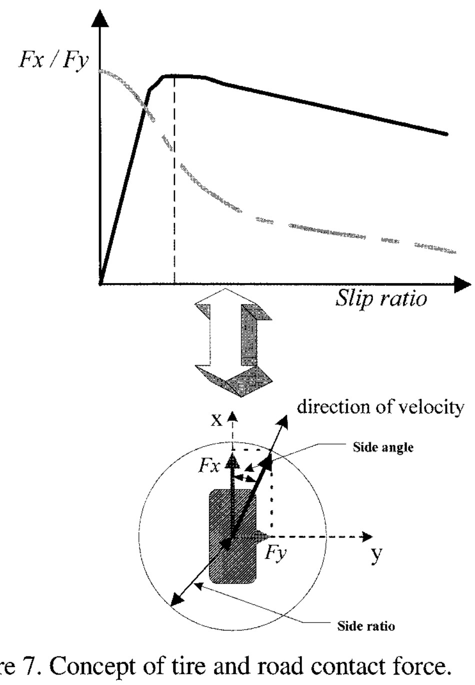


Figure 7. Concept of tire and road contact force.

controlled directly by braking and traction slip while the cornering force is controlled indirectly by slip ratio generated from tire distortion. ESP controls the slip rate of wheels in the longitudinal direction and body slip angle using vehicle yaw moment from longitudinal slip, thus enhancing handling stability. The control of braking and traction forces only deals with tire's slip rate and cannot reach to direct control of slip angle that generates cornering force. Furthermore excessive braking control can cause irregularity of control and deterioration of acceleration due to lowered engine power.

The ESP stabilizes vehicles by controlling the wheels of vehicles with the difference between the actual vehicle motion measured from sensors and target motion form reference vehicle model.

AFS controls the tire's slip angle by changing the steering angle as shown Equation (4), thus directly controlling cornering force of tire. Therefore it improves control performances by continuous control from an early stage.

$$\alpha_f = \beta + \frac{l_f r}{v_x} - \delta_f \tag{4}$$

α_f : slip angle of front wheel
β : sideslip angle
l_f : distance from CG to front wheel
v_x : vehicle speed
r : yaw rate
δ_f : steering angle at the front wheel

4.2. ESP/AFS Cooperation Control

The block diagram of EPS/AFS cooperated control is shown in Figure 8. Oversteer or understeer is determined with the error between the desired yaw rate and measured yaw rate. If the sign of error is positive it is judged to be oversteer and vice versa. If the magnitude of error is within a predetermined threshold, only AFS produces superimposed steering angle and controls the cornering force directly.

From this error target control moment is calculated and then distributed into required amount of braking and cornering force. From the cornering force superimposed steering angle is calculated. There should be limits and dead-zones of superimposed angle in order not to give strange steering feel to a driver according to vehicle states, such as braking situation and vehicle speed.

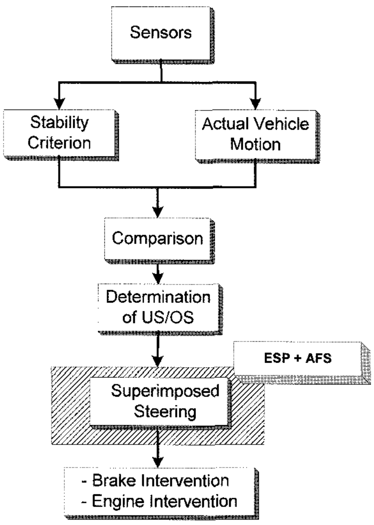


Figure 8. ESP /AFS- Cooperated control.

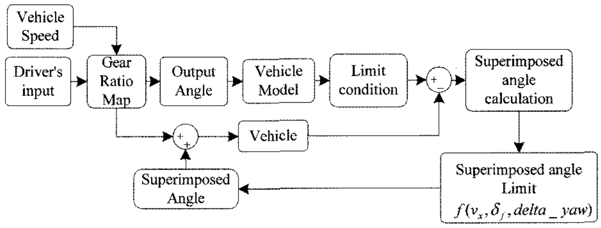


Figure 9. AFS-block diagram of superimposed steering.

5. VEHICLE STABILITY TEST

5.1. μ-split Road Test

To test the enhanced stability, AFS system is fitted into a test vehicle. Rack type EPS serves as a steering assist means and ESP cooperates with AFS. AFS ECU receives information of vehicle state such as vehicle speed, steering angle, yaw rate and lateral acceleration from REPS and ESP ECU via CAN.

The performance of cooperated control of AFS is evaluated in terms of vehicle responsibility and handling stability on asymmetric and irregular road conditions. We can evaluate the vehicle stability by yaw rate built up. In this μ-split situation, braking control focuses on limiting the yaw rate which means vehicle stability. To evaluate the effects of AFS cooperated control, vehicle is tested

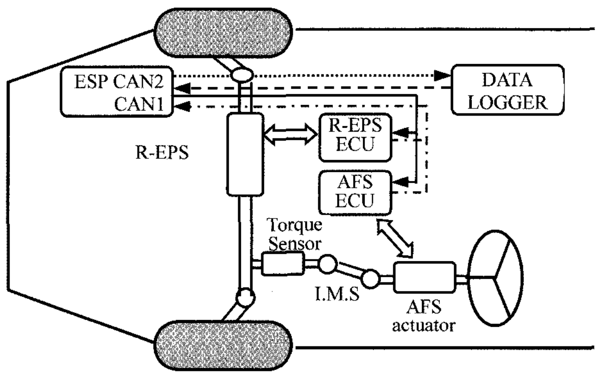
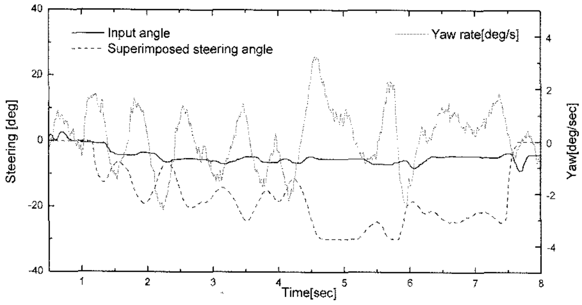
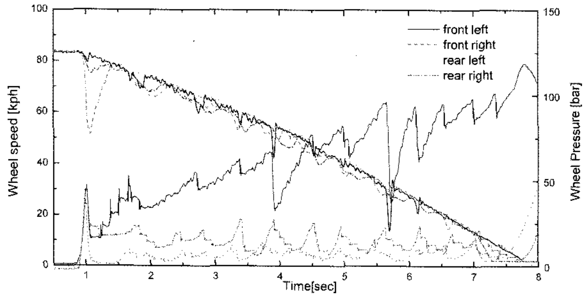


Figure 10. System configuration.



(a) Steering angle & yaw rate



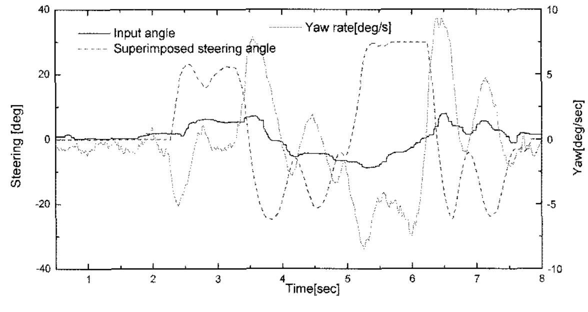
(b) Wheel speeds & pressures

Figure 11. μ-split braking with AFS-cooperation

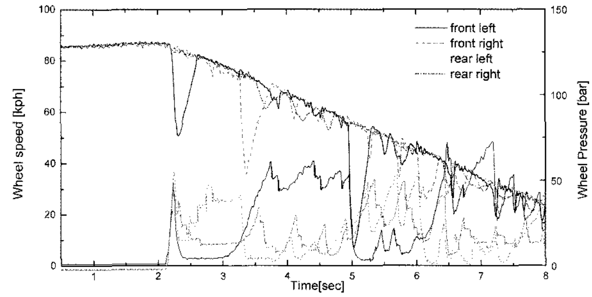
under such a condition that driver holds steering wheel and only superimposed steering angle added by AFS is acting. Figure 11 shows vehicle behavior in braking maneuver and AFS cooperation control on μ-split road.

Figure 11(a) shows superimposed steering angle in controlling vehicle behavior without driver's steering input. It shows that the yaw rate generated by this superimposed steering is up to 4 degee and thus the vehicle stays in a stable range. By this control the vehicle behavior keeps its track without deviating from its course on asymmetric roads as shown in Figure 11(b) and 12. In the same test, an ESP-only vehicle has limitations in the early control stage because it has a threshold to stabilize vehicle.

Figure 13 is the test result of variant m conditions. As shown in 13(a) driver's steering input fluctuates within 10 degree according to the sudden changes of road conditions.



(a) Steering angle & yaw rate



(b) Wheel speeds & pressures

Figure 13. Chess road (m-jump) braking with AFS-cooperation.

The vehicle is stabilized by superimposed steering which is alternating from positive to negative sign according to the vehicle behavior by the difference of friction coefficient on each side. Figure 13(b) shows that vehicle keeps track of narrow range by changing the brake pressure.

In this manner steering control with superimposed steering directly controls tire slip angle. Therefore novice drivers have quicker and safer braking maneuvers on m-split and variant m conditions than an ESP- only equipped vehicle.

5.2. Steering Torque Compensation

In planetary gear trains input torque and output torque

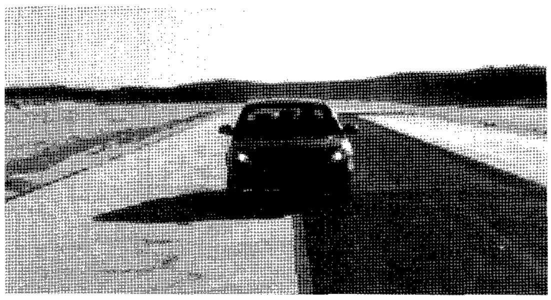


Figure 12. Vehicle test on μ-split road.

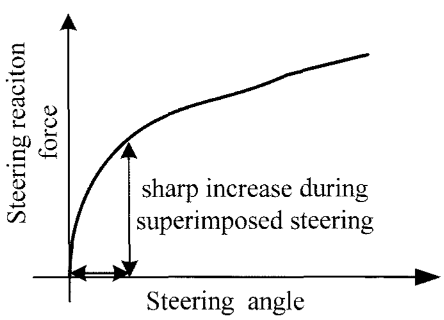


Figure 14. Reaction force during superimposed steering.

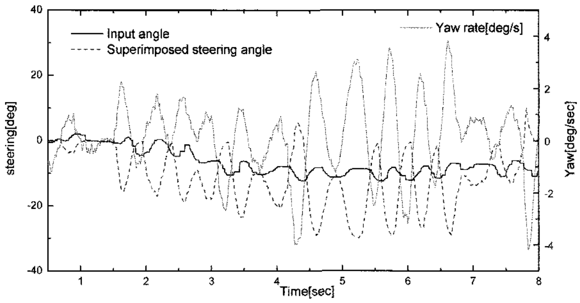


Figure 15. μ-split test without torque compensation.

which have constant ratio regardless of reduction ratio in Equation (2). The output torque increases during superimposed steering because it makes reaction force from front wheel increased sharply during straight driving as shown in Figure 14.

Consequently unexpected change of steering effort is induced. And this affects the vehicle behavior and ends up with superimposed angle changes.

Figure 15 shows driver's input swings for several times before it stops completely due to superimposed steering. To solve this problem, it is required to reduce steering output load in case of superimposed steering by increasing steering assist of EPS in order not to make steering effort change. Figure 11(a) of the previous tests is the result of steering torque compensation showing that the input angle is less affected by superimposed angle. Likewise, vehicle is stabilized fast with less fluctuation.

6. CONCLUSIONS

A planetary gear train for active front steering mechanism is proposed and cooperated control strategy of AFS with ESP is described. The vehicle tests show that superimposed steering control of AFS have effects in stabilizing vehicle behavior on μ-split roads. And steering reaction torque can be compensated with EPS compensation during superimposed steering control.

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