International Journal of Automotive Technology, Vol. 7, No. 2, pp. 201−208 (2006)

1229−9138/2006/026−12

CONTROL PHILOSOPHY AND ROBUSTNESS OF ELECTRONIC STABILITY PROGRAM FOR THE ENHANCEMENT OF VEHICLE STABILITY

D. S. KIM* and I. Y. HWANG

R&D Center, Mando Corporation, 413-5 Gomae-dong, Giheung-gu, Yongin-si, Gyeonggi 449-901, Korea

(Received 8 September 2005; Revised 30 November 2005)

ABSTRACT−This paper describes the control philosophy of ESP (Electronic Stability Program) which consists of the stability control the fault diagnosis and the fault tolerant control. Besides the functional performance of the stability control, robustness of control and fault diagnosis is focused to avoid the unnecessary activation of the controller. The look-up tables are mentioned to have the accurate target yaw rate of the vehicle and obtained from vehicle tests for the whole operation range of the steering wheel angle and the vehicle speed. The wheel slip control with a design goal of wheel slip invariance is implemented for the yaw compensation and the target wheel slip is determined by difference between the target yaw rate and actual yaw rate. Since the ESP has a high severity level and the robust control is required, the robustness margin for the stability control is determined according to several uncertainties and the robust fault diagnosis is performed. Both computer simulation and test results are shown in this paper.

KEY WORDS : Stability, Oversteer, Understeer, Robustness, Fault diagnosis, Fault tolerant control

1. INTRODUCTION

A survey (GDV, 2004) says that skidding is the major reason for severe and fatal accidents. 25% of all accidents with severe injuries are caused by skidding, 60% of all accidents with fatal injuries happen through side crashes caused mainly by skidding. Once a vehicle begins to spin out, the driver is not able to control the vehicle and finally a severe accident take places. Its solution by suspension or steering controllers is not effective since the controllers can't generate enough tire force to prevent the spin-out of the vehicle in such an emergency situation.

The ESP system makes use of the brake and engine intervention to keep the vehicle on its intended track under most road conditions. Although ABS and TCS control the wheels of a vehicle at the limit of adhesion between the tire and road surface in the longitudinal direction of the wheel, the ESP system deals with the longitudinal and lateral tire dynamics beyond the nonlinear motion of the vehicle.

Demand for ESP is expected to increase from around 9.8 million units in 2004 to 18.4 million units in 2008 worldwide (Strategy Analytics, 2004). Fitment rate is currently the highest in Europe and although adoption in North America is slower to take off, GM recently announced that GM SUVs and vans will have ESP standard by the end of 2007 and ESP will be standard on all GM cars and trucks sold to retail customers by the end of 2010 (PR Newswire, 2005).

Hartmann and van Zanten (1994) and Ehret and Hartman (1995) have suggested the ESP control concepts. In order to determine the driver's desired yaw rate, they devide the steering wheel angle into three operating range (small, medium and large) and calculate the corresponding target yaw rates for each range. Wanke (1998) use the two vehicle models - steady state and transient models - as the reference yaw model and the two linear models for transient model are applied in the entire nonlinear vehicle motion. Nakashima *et al.* (1999) control the vehicle stability using vehicle side slip and its rate although the estimation of side slip angle is difficult to be reliable in the whole vehicle motion. Chung *et al.* (2004) utilize the sliding control law for stability of the vehicle.

In this paper, the mothod to determine the desired yaw rate available on the entire nonlinear vehicle maneuvering is mentioned and additionally robustness of the calculated reference yaw rate with the inevitable uncertainties is also emphasized. And the software structure to implement the control concept has the form so as to use the already produced wheel slip control modules as many as possible since the reuse of the verified modules is

**Corresponding author.* e-mail: dskim@mando.com

strongly recommended in the aspects of system reliability and reduction of development time.

2. CONTROL PHILOSOPHY

2.1. Stability Control

2.1.1. Statement of problem

ESP system controls the dynamic vehicle motion in the emergency situation such as the final oversteer/understeer and allows the vehicle to follow the course as desired by the driver. The actual vehicle motion is nonlinear as shown in equations (1) and (2) (Wong, 1993).

$$m(\dot{V}_y + V_x\Omega_z) = F_{yr} + F_{yf}\cos\delta_f + F_{xf}\sin\delta_f \quad (1)$$

$$I_z\dot{\Omega}_z = l_1F_{yf}\cos\delta_f - l_2F_{yr} + l_1F_{xf}\sin\delta_f \quad (2)$$

Considering the uncertainties of parameters and the maneuvering on the various kinds of road surfaces such as snow, ice and split-mu, the nonlinearity is so high that it is not enough to use just nonlinear controller with the simplified linearized vehicle model. Actually the range of vehicle motion to safisfy the equations (1) & (2) on the low-m surface conditions is very limited and the alternative reference vehicle models should be developed.

Therefore, the vehicle stability control problems can be divided into two aspects. The first is the determination of vehicle reference model on the various road surfaces using several look-up tables which are appropriate to compensate the nonlinearity and the second is the design of nonlinear controller. In this paper, a gain scheduled controller of the several nonlinear controllers is implemented. Although the gain scheduled controller is an open-loop adaptation without intelligence and a time-consuming technology, it is very adequate to apply to ESP system in that 1) the parameters can be changed as quickly as measurement in reponse to changes in the vehicle dynamics. 2) it is convenient since the vehicle dynamics depend on a few easily measureable variables in a well-known fashion. 3) it can be implemented without loss of insight of vehicle motion. To decide and adapt a lot of parameters to various kinds of maneuvering situations requires an enomous vehicle testing and this research can be regarded as an experimental study.

2.1.2. Vehicle reference models

It is very important to determine the desired course, which is decided by the vehicle reference models. The vehicle reference models should be changed according to driving situations and the first reference model is based on the fundamental physics. Equations (3) and (4) show the 2 DOF linear equations of motion according to the yaw rate and side slip angle used for the first reference model (Millikan, 1995).

$$I_z\dot{r} = N_\beta\beta + N_r r + N_\delta\delta_{sw} \quad (3)$$

$$mV(r + \dot{\beta}) = Y_\beta\beta + Y_r r + Y_\delta\delta_{sw} \quad (4)$$

However, since the vehicle motion is highly nonlinear, the simple linear equations are not enough to describe the whole vehicle motion in the various kinds of situation, especially unstable motion in the tire adhesion limit. For the nonlinear system design, it is well known that the look-up table is very effective and the tables of the vehicle reference model should be determined by the actual vehicle tests on high and low friction road surface. The look-up table for the first vehicle reference is composed of 20 by 20 matrix according to the steering wheel angle and vehicle speed. The vehicle tests such as steady state circular run and step steering test for each applied vehicle should be carried out to complete the look-up table.

On the other hand, when the steering wheel is turned excessively in relation to the existing vehicle speed on the low-m surface, the desired yaw rate which is predetermined by the first reference model is not adequate to consider as the criterion for the vehicle control because if the vehicle follows the first model, its side slip angle increases and the vehicle pushes out in a strong understeering manner. The control of the vehicle motion makes sense only if the adhesion of the wheels on the road surface permits the required yaw moment to act on the vehicle. Therefore, the reference model should be prevented from being selected as the preset value under all circumstances according to the first reference model.

And the maximum yaw rate should be limited to the value which is available on the road surface. The limitation value is determined by the vehicle speed and the coefficient of friction between the tire and road surface.

In case of stationary motion which has a small or zero steering angle rate on the low-μ surface, the first reference model is still effective as long as the difference between the measured lateral acceleration and the target limited acceleration is below a threshold, because the difference of the first reference model on high-μ surface and measured yaw rate on the low-μ surface is to say that the vehicle slips sideways. A look-up table, therefore, is required to define the enter condition for the application of the yaw limitation and should be determined by the actual vehicle tests. Figure 1 shows the look-up table.

There exist some special conditions such as U-turn and J-turn, in which the above first reference model and yaw limitation don't have enough information to control the understeering of the vehicle. So, an additional reference model is needed using the side slip information since the most of unstability of the vehicle in U- turn and J- turn results from the slip side-ways.

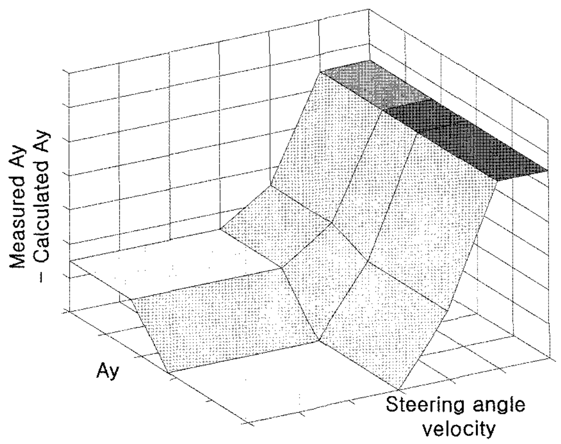


Figure 1. A look-up table for the enter condition of the yaw limitation.

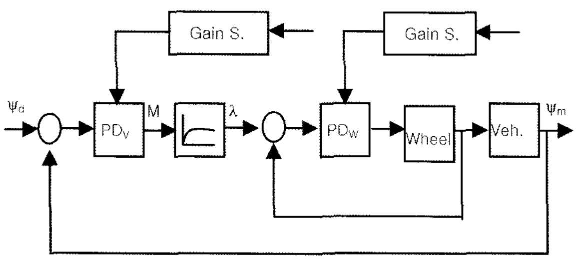


Figure 2. Block Diagram of Stability Controller.

2.1.3. Design of controller

Figure 2 shows the whole system block diagram. The outer feedback loop is for vehicle yaw control and inner loop for the wheel slip control.

The yaw controller has the performance index as equation (5) and its design goal is to achive the fast yaw response and small side slip angle irrespective of road friction and vehicle speed. The required yaw moment on the low-μ surface and at low vehicle speed should be physically smaller than that on the high-μ surface and at high vehicle speed, respectively.

$$J = \int \left(\begin{array}{l} K_P(\dot{\psi}_{measured} - \dot{\psi}_{desired}) \\ + K_D \dfrac{d(\dot{\psi}_{measured} - \dot{\psi}_{desired})}{dt} \end{array} \right) dt \qquad (5)$$

Therefore the scheduling variables are vehicle speed, road friction coefficient, side slip angle and under/oversteering. Although the side slip can be explicitly expressed in the performance index, it is dealt with a scheduling variable for easy implementation of controller.

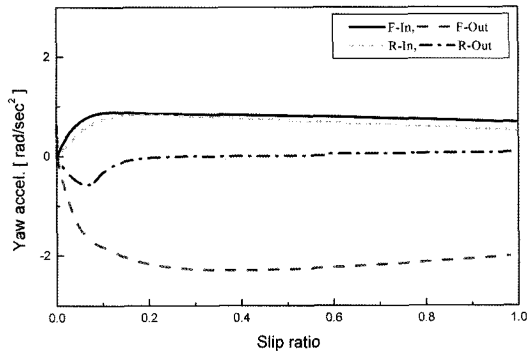
As first step of determination of the PD controller gains, the vehicle tests are carried out in the understeering and oversteering irrespective of road surface conditions as only the vehicle speed varies. Thus selected gains make the vehicle have the yaw stability, but the driver's feeing is likely harsh as anticipated. So, the effect of the road surface is needed to be considered and the gains are changed according to road surfaces. Since the amount of yaw moment compensation should be changed according to the coefficient of friction of the road surface, the estimation process for determining the coefficient of friction of the road surface is inevitable. The coefficient of friction can be determined only after the incipient instability of the vehicle begins. Since no estimated coefficient of friction is not available before the side slip of the vehicle is built up, the coefficient of friction μ equals to one at the onset. The incipient instability is detected by the comparison between the first reference vehicle model and measured yaw rate.

The controller gains determined by vehicle speed and road surface are desirable in the aspects of controllerability and driver's feeling. But in the special cases – J-turn and circular run with a small or zero steering angle rate which has the characteristics of slowly developing the side slip and small differences between target and measured yaw rates, the vehicle instability can not be controlled without considering side slip angle. So, additionally the side slip should be considered as a scheduling variable and can be estimated by the well-known observer.

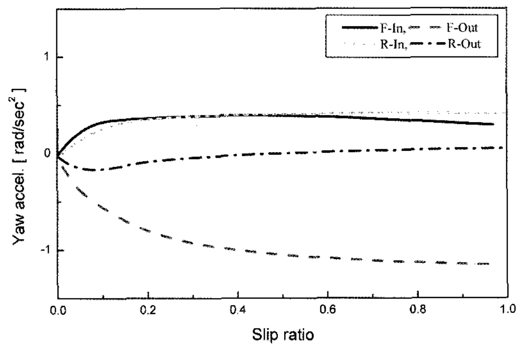
Consider the final scheduling variable, under/oversteering situation. When reaching the maximum lateral acceleration achievable on a road surface, the tire adhesion limit will be exceeded first at the front axle for an understeering vehicle and first at the rear axle for an oversteering vehicle, which means the slip angle increases uncontrolled at the respective axle. For understeering vehicle this results in a decrease of the side slip angle, which causes a relative decrease of the rear side slip angle. The corresponding reduction of the rear cornering force leads to a stabilization of the vehicle on a larger path radius with a smaller lateral acceleration. For oversteering vehicle in contrast, this results in an increase of the side slip angle and therefore a relative increase of the front slip angle. The corresponding increase at front axle leads to the dangerous spin-out. So, the PD gains in the performance index J are less weighted for the understeering case since the excessive compensation of understeering can make the vehicle undesired oversteering.

The wheel slip controller is used for the generation of tire forces and its design goal is a wheel slip invariance around the optimal slip ratio irrespective of vehicle speed and road surface. In case of the brake intervention, if the brake pressure is increased too much, the wheel goes to lock-up, which makes the driving comfort decrease and controllability reduce, especially on ice road surface.

Therefore we have to monitor and limit the wheel slip value for the vehicle body control. Figure 3 shows the

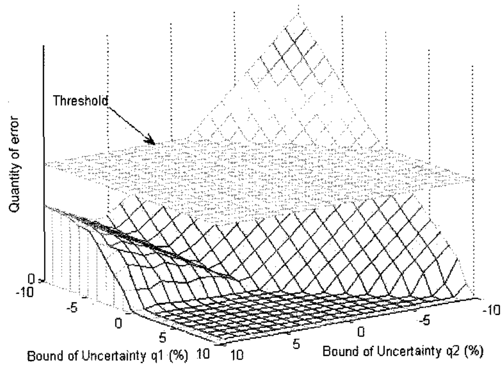


(a)

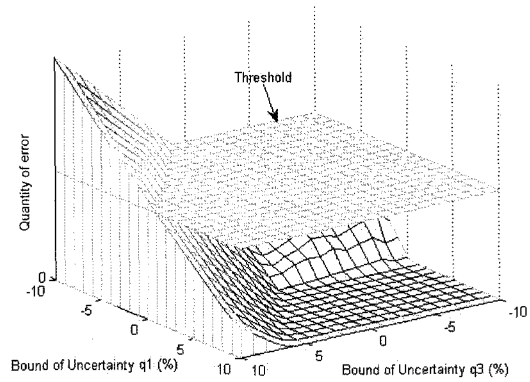


(b)

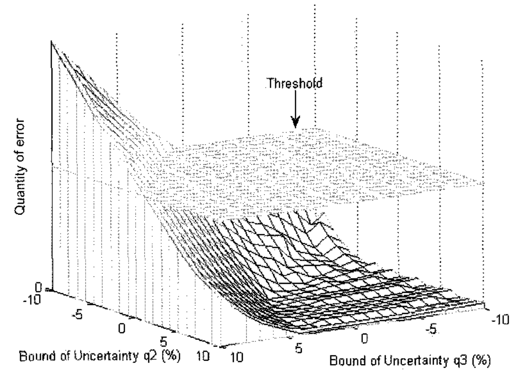
Figure 3. Relationship between yaw acceleration (moment) and wheel slip ratio (a) Dry asphalt; (b) Ice.



(a)



(b)



(c)

Figure 4. Quantity of error within the bound of uncertainties q_1, q_2 and q_3.

relationship between the yaw moment (or yaw acceleration) compensated by one wheel brake intervention and wheel slip ratio. These are obtained by step steering input and show the limitation values of the controlled wheel slip. The figures say that even though the wheel slip increases, the compensated yaw moment doesn't increase after the front outside wheel slip reaches approximately 30% on the dry asphalt, and rear inside wheel slip does approximately 10%, respectively.

On the ice road surface the saturation points are around 10% for rear, 60% for front wheel, respectively.

For the enhancement of vehicle stability, the brake intervention is the most effective method, but the engine intervention also helps reduce the instability of the vehicle although its response is slow. For the engine intervention in the stationary motion such as driving on the circle pad, we have to consider the driven axle, that is, FWD and RWD cases. For FWD vehicle, the engine torque reduction makes the vehicle oversteering condition and for RWD vehicle, on the contrary, understeering condition, which makes the brake intervention minimized or not to be needed any more. In the continuous slalom maneuvering, the engine torque reduction decreases the vehicle speed and the vehicle becomes stabilized although its rate is slow.

2.1.4. Robustness issues

The robustness of a controlled system should be analyzed and tested to avoid the intusive and unnecessary control. There are many uncertainties in the system, that is, parametric uncertainty like road friction coefficient and uncertainty due to factors such as unmodeled dynamics, nonlinearity of the vehicle motion, sensor offset and so on. Concerning the road friction coefficient, the vehicle motion on the low-μ surface (snow and ice road) is totally different from that on the high-mu surface in the nonlinear range. Within bounds on the uncertain quantities, therefore, the family of target yaw rates should be determined by vehicle test since the controllerbility and stability of the system have to be satisfied within the given ranges of friction coefficient.

For the unmodeled dynamics and nonlinearity of vehicle motion, the family of reference models for the entire bound of the uncertainties can be decided, but here a yaw reference model with the variable gains is used. Instead, the robustness margin is determined. The target yaw rate can be expressed as equation (6) from equations (3)–(4), and time constant τ is considered as a factor which determines the controller gain according to steering angle rate and can be ignored here for simplicity.

$$\frac{\psi_d}{\delta_{sw}/i} = \frac{\dfrac{\tau s+1}{l/V+K_{us}V/g}}{s^2/\omega_n^2+2\varsigma s/\omega_n+1} \tag{6}$$

, where ψ_d = desired yaw rate, δ_{sw} = steering wheel angle, i = steering gear ratio, τ = time constant, l = wheelbase, V = vehicle speed, K_{us} = understeering coefficient, ω_n = yaw natural frequency and ζ = yaw damping ratio.

Considering equation (6), we can select K_{us}, $2\zeta\omega_n$ and ω_n^2 as the uncertainties, that is, the uncertainties, q_1, q_2 and q_3 are as follows, $q_1 = K_{us} - K_{us,no}$, $q_2 = 2\zeta\omega_n - 2\zeta\omega_{n,no}$ and $q_3 = \omega_n^2 - \omega_{n,no}^2$, where subscription $_{no}$ means a nominal value. Figure 4(a)–(c) show the quantity of errors (compensated yaw moment) calculated by the differences between the designed and the actual yaw rates when the uncertainties vary in the range of ±10% of each nominal value.

The z-axis is determined by the maximum yaw responses for step steering input. The threshold plane shows the start condition of activation.

When q_1 and q_2 reduce by the amount of approximately 5% and more from the nominal values as in Figure 4(a), the unnecessary activation occurs since the decrease of q_1 and q_2 makes the actual yaw rate increase and the control for reducing oversteer is implemented. On the contrary, if q_1 and q_2 increase, the intrusive activation doesn't occur. This phenomena result from that the yaw stability controller is designed so as to repond earlier in case of oversteering and to prevent the excessive control for understeering. Figure 4(b)–(c) show the effect of w_n. Since the high value of ω_n means the fast response of motion, the unnecessary intervention happens when q_3 is above approximately +5% and q_1 and q_2 below −5%. These characteristics can be explained using the factors, such as suspension stiffness and vehicle mass which vary relatively easily. As the stiffness of front suspension increases (e.g. adaption of anti-roll bar), K_{us} and $2\zeta\omega_n$ increase, which makes the quantity of error smaller. On the contrary, when the stiffness of rear suspension increases, K_{us} and $2\zeta\omega_n$ decrease, which causes the bigger quantity of error and the unnecessary activation can occur. If the stiffness of front and rear suspension increases simultaneously, already existing under/oversteer tendency is intensified, so the quantity of error is reduced, which yields no intrusive control. Decrease of suspension stiffness has a reversed effect on the vehicle motion. The robustness margin can be different more or less for each vehicle and should be checked by vehicle tests.

Although the controlled system guaranties its robustness within the robustness margin, an adaptive target yaw rate needs to be developed for the accurate intervention within the whole uncertainty bound. The adaptive target yaw rate can be determined by parameter estimation method and monitoring the vehicle motion under stable driving condition. This robustness margin is also used to determine the admissible range of controller gains for tuning. Too big controller gain can exert an adverse influence on the robustness as well as stability of the system.

2.2. Fault Diagnosis and Fault Tolerant Control

2.2.1. Fault diagnosis

Fault diagnosis in the safety-related control system is very important and composed of fault detection, isolation and identification. For fault detection, there are three phases. The first phase is the electrical diagnosis of sensors using limit checking with mathematical model-free method. The second phase is the self-test for the intelligent sonsor. And the third phase is the model based approach. The model based diagnosis is based on the analytical (functional) redundancy which uses redunadant analytical relations between various measured variables of the monitored process rather than hardware (physical) redundancy.

There are several robust fault diagnosis methods and here the robust fault diagnosis using the optimal parity relations is used. The parity equations are suitable for the detection of the sensor (offset) faults, while the parameter estimation is appropriate for the process or system parameter faults (Isermann, 2000). For the determination of residuals using parity equations, it is necessary to consider two important aspects – Fault Detectabilty and Robustness. To apply these aspects to the ESP system, let us begin with the following general equation (7) for residuals, (Refer to (Chen and Patton, 1999) for detail.)

$$r(k) = v^T Z_t X(k) + v^T M_t F(k) \tag{7}$$

, where $r(k)$ = residual, v = residual generating vector, Z_t = matrix consisting of system model matrix and unknown disturbance matrix, $X(k)$ = state vector, M_t = fault matrix of sensor and actuators and $F(k)$ = fault input vector.

In order to detect faults, the residual signal $r(k)$ should become zero for the fault-free case and non-zero for the fault case; this requires that:

$$v^T Z_t = 0 \tag{8}$$

$$v^T M_t \neq 0 \tag{9}$$

In this system, direct redundancy exists among sensors whose outputs are algebraically related and one observation can be expressed as equation (10) (Chow and Willsky, 1984).

$$y_1(k) = \sum_{i=2}^{M} \alpha_i y_i(k) \tag{10}$$

, where α_i's depend on the vehicle characteristics such as wheel-track, steering gear ratio, vehicle speed and understeer coefficient. To satisfy the equation (8), the validity conditions of the residuals can be used for each measured variable. In this system, there are four residuals for each sensor and four measured variables and the validity conditions are 1) no wheel slip for wheel speed sensor, 2) small steering and no countersteering for the relation between steering angle and yaw rate sensor, 3) positive vehicle reference speed for the relation between steering angle and lateral acceleration sensor. Actually equation (8) is satisfied within a small tolerance. Concerning the equation (9), the residual will be greater than a predetermined threshold if a fault occurs in the sensor. There are well known methods for the determination of robust threshold, that is, adaptive threshold (Clark, 1989; Frank, 1995) and robust threshold selector (Emami-Naeini *et al.*, 1988) based on a nonlinear inequality whose solution defines the set of detectable sensor fault signal. Here, the adaptive threshold which varies according to the number of valid models is implemented. Thus we can achieve the robustness insensitive to model uncertainty and the detectability for the faulty case.

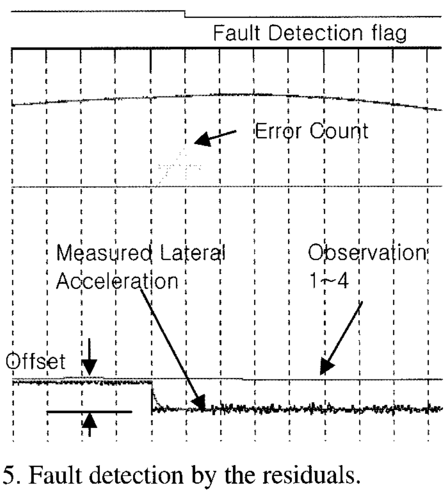


Figure 5. Fault detection by the residuals.

Figure 5 shows the fault detection when the offset of the lateral acceleration is generated on purpose. When the error count exceeds the predetermined threshold, a fault is detected.

To satisfy the equation (8)–(9) can be considered as finding an solution for an optimization problem, that is $\min\{v^T ZZ^T v\}$ and $\max\{v^T MM^T v\}$ (Chen and Patton, 1999). The simultaneous optimal solution for the multi-objective optimization does not actually exist and one needs to find a trade-off solution between robustness and detectability.

This redundancy is also used for the compensation of sensor offset and for the robustness check, the tremendous lab and vehicle tests should be performed especially at the extreme bounds of sensor uncertainty.

2.2.2. Fault tolerant control

As this is a new engineering field and to avoid the confusion on communication, it is particularly important to define the terminology (Blanke *et al.*, 2001).

- *Fail-operational*: a system, which is able to operate with no change in objectives or performance despite of any single failure
- *Fail-safe* : a system, which fails to state that is considered safe in the particular context
- *Fault-tolerance* : the ability of a controlled system to maintain control objectives, despite the occurance of a fault. A degradation of control performce may be accepted. Fault-tolerance can be obtained through fault accommodation or through system and/or controller reconfiguration.
- *Fault-accommodation* : change in controller parameters or structure to avoid the consequences of a fault. The input-output between controller and plant is unchanged. The original control objective is achieved although performance may degrade.
- *Reconfiguration* : change in input-output between the controller and plant through change of controller structure and parameters. The original control objective is achieved although performance may degrade.

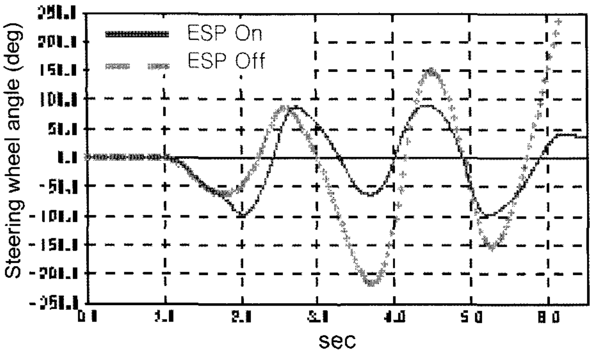
There are several fault tolerant control methods (Patton, 1993) and here, the simplest way, that is, pre-computed gain scheduling is used and the actual implementation is based on the concept that robustness recovery typically requires lowering the control gains in systematic fashion. To achieve a reasonable solution for robust fault tolerant control (FTC) it is needed to account a priori for the interaction between *robust* baseline controller and *robust* fault diagnosis by designing them together or by integrating their design. FTC logic monitors the system's control impairment status and is reconfigurable so as to replaces the fault sensor value into the appropriate redundancy and control with the redundancy. In case of steering angle sensor failure, the control is terminated gracefully since it is a reference signal. For the other sensor failures, the control keeps going with a redundancy until the yaw unstability is diminished.

2.3. Software Verification

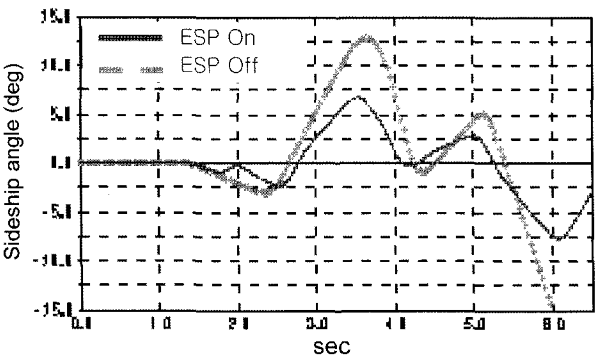
The verification of software is essential and its tests are composed of unit test, integration test, black box test and system tests. The unit test is carried out by software-in-the-loop (SIL) and the integration and black box test is by hardware-in-the-loop (HIL). For black box test, thousands of test cases should be developed – test cases by sweeping of input and output variables and the criteria for the evaluation of test results are also needed.

3. SIMULATION AND TEST RESULTS

For computer simulation, off-line and hardware-in- the-loop simulation (HILS) are conducted. Figure 6 shows the simulation results of the double lane change on the snow road surface, where the linear predictive driver model is used and the driver tries to avoid the collision. Without ESP, side slip angle becomes so large that the driver has to steer heavily to compensate the excessive vehicle motion.



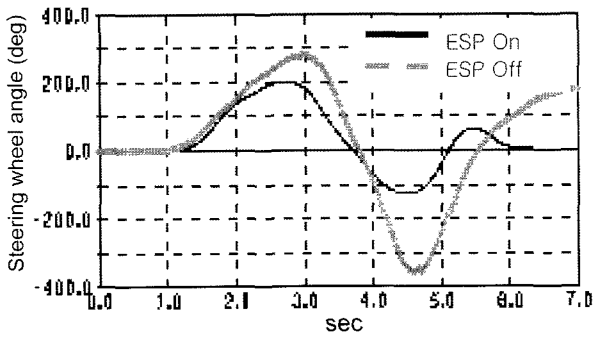
(a)



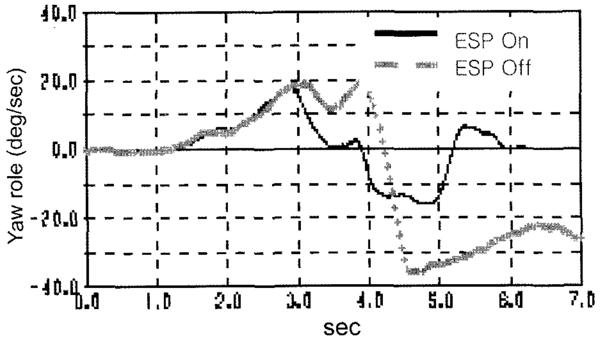
(b)

Figure 6. Simulation results for double lane change on the snow road surface.

The large countersteering causes another countersteering and the steering wheel angle becomes larger and larger and finally the vehicle goes to spin out. Figure 7 shows the lane change test results on the ice road surface and the vehicle with ESP can follow the desired course although the vehicle without ESP goes out of control.



(a)



(b)

Figure 7. Test results for lane change on ice road.

4. CONCLUSION

In addition to achieve the stability of vehicle in the emergency situation by determination of the reference yaw models and implementation of a nonlinear control method, a robustness is focused to avoid intrusive activation. Robustness should be considerd in the aspects of the control logic, fault diagnosis and fault tolerant control. First, it is ensured that the control logic to accomplish the functional performance is robust within the robustness margin. Second, fault diagnosis is performed on the basis of optimization concept and third FTC is a combination of *robust* control logic and *robust* fault diagnosis.

The performance of ESP can be different from car to car and from sedan to sport utility vehicle even though the control logic is same because the vehicle reference model is included in the control logic. Therefore, it takes a lot of efforts to set the vehicle reference model as accurate as possible. And the assessment of ESP can be different since the controller gains are finally determined by tuning based on subjective evaluation. Therefore the control logic structure should be made so as to adjust the appropriate parameters within the robustness margin as easily as possible.

REFERENCES

Blanke, M., Staroswiecki, M. and Wu, N. E. (2001). Concepts and methods in fault-tolerant control. *Proc. American Control Conf.*, 2606–2620.

Chen, Jie and Patton, R. J. (1999). *Robust Model-Based Fault Diagnosis for Dynamic Systems*. Kluwer Academic Publishers. Norwell. MA. USA.

Chow, E. and Willsky, A. (1984). Analytical redundancy and the design of robust failure detection systems. *IEEE Trans. Auto. Contr.* **AC-29**, **7**, 603–614.

Chung, T., Kim, J. and Yi, K. (2004). Human-in-the-loop evaluation of a vehicle stability controller using a vehicle simulator. *Int. J. Automotive Technology* **5**, **2**, 109–114.

Emami-Naeini, A., Akhter, M. M. and Rock, S. M. (1988). Effect of model uncertainty on failure detection – the threshold selector. *IEEE Trans. Automat. Control,* **33**, 1106–1115.

Ehret, T. and Hartman, U. (1995). *US Patent No. 5,402,342*, USPTO. Arlington. VA. USA.

Frank, P. M. (1995). Residual evaluation for fault diagnosis based on adaptive fuzzy thresholds. *IEE Colloquium on Qualitative and Quantitative Modeling Methods for Fault Diagnosis,* 4/1–4/11, London, UK.

GDV – German Insurance Association (2004). Berlin. Germany.

Hartmann, U. and van Zanten, A. (1994). *US Patent No. 5,332,300*, USPTO. Arlington. VA. USA.

Isermann, R. (2000). Diagnosis Methods for electronic controlled vehicles. *5th Int. Symp. Advanced Vehicle Control*, Ann Arbor, Michigan.

Millikan, W. and Millikan, D. (1995). *Race Car Vehicle Dynamics*. SAE. Warrendale. PA. USA.

Nakashima, H., Hamada, T. and Mihara, J. (1999). *US Patent No. 5,931,546*, USPTO. Arlington. VA. USA.

Patton, R. (1993). Robustness issues in fault-tolerant control. *Fault Diagnosis and Control System Reconfiguration, IEE Colloquium*, 1/1~1/25.

PR Newswire, Jan. (2005). New York. USA.

Strategy Analytics, July (2004). Newton Ctr. MA. USA.

Wanke, P. (1998). *US Patent No. 5,711,024*, USPTO. Arlington. VA. USA.

Wong, J. Y. (1993). *Theory of Ground Vehicles*. John Wiley and Sons. New York. USA.