DEVELOPMENT OF A TRANFER-CASE CONTROL SYSTEM FOR REAR WHEEL BASED 4WD VEHICLES

Woosung Park[a], Hyeongjin Ham[b],Kanghee Won[c], Hyeongcheol Lee[d]*

[(a),(b),(c)] Department of Electrical Engineering, Hanyang University, 17 Haengdang-dong, Sungdong-gu, Seoul 133-791, Korea
[(d)] Division of Electrical and Instrumentation Biomedical Engineering, Hanyang University, 17 Haengdang-dong, Seongdong-gu, Seoul 133-791, Korea

[(a)]Jacklos@hanyang.ac.kr, [(b)]postechj@hanyang.ac.kr, [(c)]wkh@hanyang.ac.kr, [(d)]hclee@hanyang.ac.kr

ABSTRACT

This paper presents a control method of torque distribution for Four Wheel Drive (4WD) vehicles. The 4WD system is a device which distributes optimal torque between front and rear propeller shafts to improve traction ability and stability of a vehicle. While the object of the traditional 4WD system is to improve only traction ability by distribution of the transmission torque, the recent 4WD system is used for to meliorate the traction ability, stability and fuel consumption efficiency. This paper proposes the control algorithm for the vehicle which is equipped the electronic controlled 4WD system to improve traction ability. In this paper, the mathematical equation-based control method by vehicle dynamics analysis is presented. The proposed control algorithm of the 4WD system is validated by real vehicle test using the Rapid Control Prototyping (RCP) and the mid-size Sport Utility Vehicle (SUV) equipped 4WD system of multi plate clutch type actuated by solenoid coil.

Keywords: Vehicle Dynamics, Power train & Drive train Control, Traction and Yaw control

1. INTRODUCTION

The 4WD system distributes the engine torque to the front or rear wheels according to the driving condition and the vehicle statement to improve traction ability and stability of the vehicle. Nowadays, there is a requirement of traction ability such as not only Sports Utility Vehicles (SUV) and Multi Purpose Vehicles (MPV) but also luxury sedan and sports car. Especially, the person who lives in the snowy region or country side needs the traction performance of the 4WD vehicles.

The 4WD system is classified by coupling of the Front Wheel Drive (FWD) vehicle and transfer-case of the Rear Wheel Drive (RWD) vehicle. Moreover according to the existence of controller, it is categorized by mechanical type and electronic type. The electronic type 4WD system is assorted again to hydraulic motor type and clutch type controlled by solenoid. Compared to the traditional mechanical 4WD, the electronically controlled 4WD system provides better controllability and fuel economy.

Controlling vehicle dynamics by the 4WD control system is a quite challenging area for the following reasons: [1]

a. The 4WD control is meaningful only when the engine generates driving torque or meaningful engine brake torque.

b. The 4WD control system can control only the maximum biasing torque that the coupling can support without slipping. The actual biasing torque can be any value under this maximum value, depending on the amount of engine torque, the tire–road surface condition, and the weight shift between the front and rear wheels.

c. The arbitration between the 4WD control system and the brake control system (e.g. ABS, TCS, and ESP) is not clear. Most of the time, the 4WD control system deactivates the coupling when the brake control system is activated.

Owing to these difficulties, the 4WD control algorithm has been frequently developed using the look-up table method. These types of controller proposed by [2], [3] have some problems, such as rough control (bad control accuracy with the large duty cycle increment) or slow response (with the small duty cycle increment).

In this paper, the Equation control strategy for the 4WD vehicles is proposed. For this, the 2 DOF vehicle model and the driveline model including torque biasing devices for RWD based 4WD are developed in section 2. A proposed 4WD control algorithm is given in section 3 and the simulation results are shown in section 4.

2. SYSTEM MODELING

The system modeling is provided in this section to figure out 4WD device and to design controller.

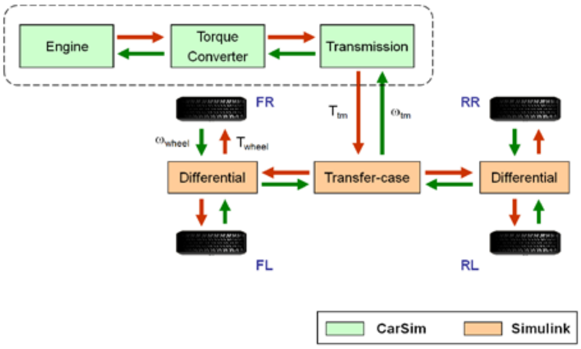


Figure 1: Vehicle Model

Fig. 1 shows a graphical description of the overall system configuration.

From this diagram, the modeling of transfer-case for the system is formulated in the following section 2.1.

2.1. Transfer-case Model

As explained previously in introduction, two different types of drive-line systems that one is coupling and the other is transfer-case, are typically used for the 4WD device model depending on the placement of a vehicle engine.

From Fig. 2, it implies that T_{bias} is the bias torque which is passed through the coupling device.

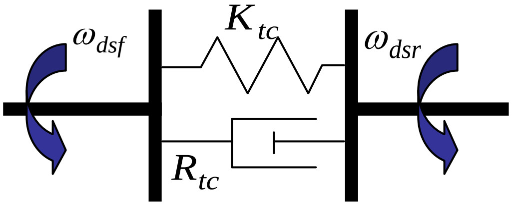


Figure 2: Torque biasing device

By assuming that the inertia is 0, the bias torque equation is derived as follow.

$$\mathrm{T_{bias}} = \mathrm{K_{tc}}(\phi_{\mathrm{dsf}} - \phi_{\mathrm{dsr}}) + \mathrm{R_{tc}}(\omega_{\mathrm{dsf}} - \omega_{\mathrm{dsr}}) \qquad (1)$$

where K_{tc}, R_{tc}, ϕ_{dsf}, ϕ_{dsr}, ω_{dsf} and ω_{dsr} are a spring coefficient, a damping coefficient, the displacements of the front and the rear drive shaft, and the angular velocities at the front and the rear drive shaft, respectively.

In this case, the bias torque defined by both the spring and the damper has an upper limit so that the bias torque can be always smaller than the input torque.

As shown in Fig. 3, if the speed difference between the front propeller shaft and the rear propeller shaft is detected, the electric signal is sent to solenoid by the control module so that the pressure is given to the ball/lamp and it give rise to the friction forces between clutch plates. As a result, the driving force is delivered to the front propeller shaft.

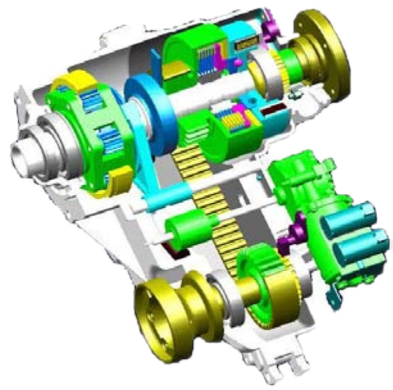


Figure 3: Solenoid type transfer-case device

By using the transfer-case device model as shown in Fig. 2, the model of the clutch plates can be expressed as equation 1, previously obtained.

However, since the torque for controlling clutch, provided by the control module, is not equal to the bias torque of equation 1, the lock and the slip should be analyzed in the different states.

If the clutch control torque provided by control module is smaller than the bias torque ($T_{control} < T_{bias}$), the clutch control torque transfers to the driven wheels. In this case, because the control torque exceeds the tolerance range of the transfer-case, the clutch slips. On the other hand, if the clutch control torque is larger than the bias torque ($T_{control} > T_{bias}$), the bias torque transfers to the driven wheels in the locking state of the transfer-case.

In other words, the bias torque is a criterion of conversion, whether lock state or slip state.

The torque relationship of each state is represented as follows:

$$* \left|\mathrm{T_{dsf\_lock}} + \mathrm{T_{bias}}\right| < \mathrm{T_{control\_c}} : \text{ Lock state}$$

$$\mathrm{T_{dsr}} = \frac{\mathrm{G_{taf}}}{\mathrm{G_{tar}}}\mathrm{T_{tm}} - \mathrm{T_{dsf}} \qquad (2\text{-}1)$$

$$\mathrm{T_{dsf}} = \mathrm{T_{dsf\_lock}} + \mathrm{T_{bias}}$$

$$* \left|\mathrm{T_{dsf\_lock}} + \mathrm{T_{bias}}\right| \geq \mathrm{T_{control\_c}} : \text{ Slip state}$$

$$\mathrm{T_{dsr}} = \mathrm{G_{tar}} \cdot \mathrm{T_{tm}} - \frac{\mathrm{G_{tar}}}{\mathrm{G_{taf}}}\mathrm{T_{dsf}} \qquad (2\text{-}2)$$

$$\mathrm{T_{dsf}} = \mathrm{T_{control\_c}}$$

where G_{taf}, G_{tar}, T_{dsf_lock} and T_{dsr_lock} are a gear ratio between transfer-case and front drive shaft, a gear ratio between transfer-case and rear drive shaft, lock torque of front drive shaft, lock torque of rear drive shaft, respectively.

2.2. Consideration Of The Road Condition

In general, a flow of the torque is represented from the engine to the wheels. The torque distribution neglected the road condition affect to the accuracy of vehicle drive-train model. In this paper, therefore, a limitation of transferring torque to the driven wheels is decided according to the road condition as follows equations.

A transmission torque equals summation of the front drive shaft torque and rear drive shaft torque.

$$T_{tm} = (T_{dsf} / G_{taf}) + (T_{dsr} / G_{tar}) \qquad (3)$$

Since the front and rear drive shaft torque equals product of longitudinal force of wheels and effective wheel radius, the difference between rear drive shaft and front drive shaft is derived as follow.

$$\begin{aligned} T_{dsr} - T_{dsf} = &\{(F_{rl} + F_{rr}) \times R_{wr} / G_{fdr}\} \\ &- \{(F_{fl} + F_{fr}) \times R_{wf} / G_{fdf}\} \end{aligned} \qquad (4)$$

where G_{fdf}, G_{fdr}, R_{wf} and R_{wr} are a final reduction gear ration of front and rear differential, a effective wheel radius of front and rear wheels, respectively.

From the equation (3) and (4), the limitation of front and rear drive shaft torque is derived using appropriate math operation.

$$\begin{aligned} T_{dsr\_Lock} = &\left[\begin{array}{l} G_{taf} \times G_{tar} \times T_{tm} + G_{tar} \\ \times \left\{ \begin{array}{l} (F_{rl} + F_{rr}) \times R_{wr} / G_{fdr} \\ - (F_{fl} + F_{fr}) \times R_{wf} / G_{fdf} \end{array} \right\} \end{array} \right] \\ &/(G_{taf} + G_{tar}) \\ T_{dsf\_Lock} = &(G_{taf} \times G_{tar} \times T_{tm} \\ &- G_{taf} T_{dsf}) / G_{tar} \end{aligned} \qquad (5)$$

3. CONTROL ALGORITHM

The controller of look-up table methods was used until from the now many Conventional vehicles. The method which it proposes from the present paper collects the strong point of Look-up table methods and Equation methods and it makes and they are Look-up table methods which formulate.

The purposes of the controller are to improve on traction ability, stability ability and robustness against model uncertainties. The traction performance and stability performance of the proposed control strategy are validated by vehicle simulation using MSC's CarSim®.

3.1. Traction Control

The 1st Traction Control strategy is the front wheel torque distribution which it follows in vehicle speeds and APS(Acceleration Pedal Signal). The 2nd Traction Control strategy is the front wheels torque distribution in order to reduce the difference between front and rear wheel speed.

The engine torque graph which it relationship with vehicle speed shown as follows:

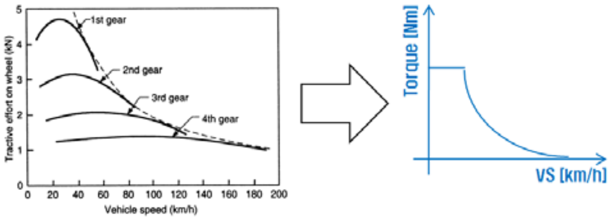


Figure 4: The torque limit with a engine performance

Because of this, transfer torque also has to decrease as the velocity of vehicle increases.

Next consideration such as APS and DeltaN(Wheel speed difference between Front wheel and Rear wheel) is the show of will and the condition of the car. Because of this, the output torque has to be above such level from the time when the input value is small. This could lead to the satisfaction of the driver and at the same time, can induce the car to a safe condition.

The relationship is defined as follows.

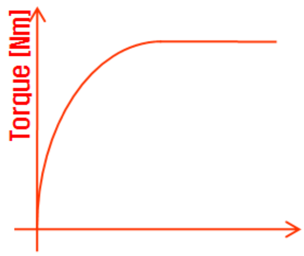


Figure 5: Torque of APS or DeltaN

The equation has been derived by considering all the factors above. The equation can be fine-tuned by varying the internal parameters.

First, the feed forward controller that could be determined by APS and vehicle speed are as follows.

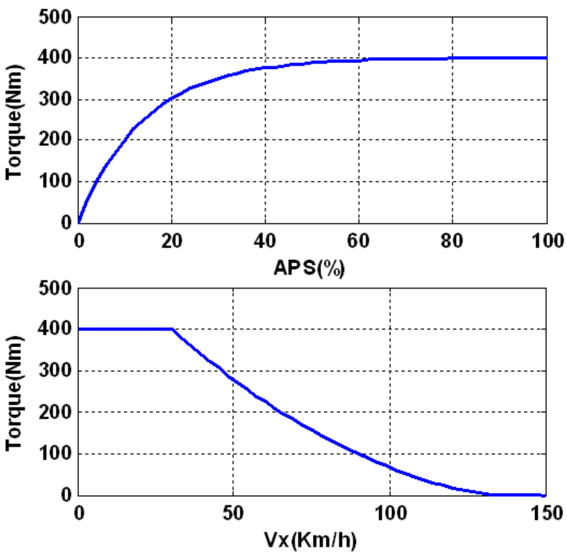


Figure 6: Feed-Forward equation graph

$$T_{TR\_FF} = C_{FF\_slope} \times T_{FF\_max} \times \left(1 - e^{(-APS \times p_{slope\_e})}\right) \quad (6)$$

$$C_{FF\_slope} = \frac{\left(p_{slope\_q} \times V_x - p_{sat}\right)^2}{p_{sat}^{\ 2}} \quad (7)$$

Seconds, feedback controller has been derived by using DeltaN and Vehicle speed. To achieve fast responses, instead of using feed forward control, we have decreased the vehicle speed and increased control amount.

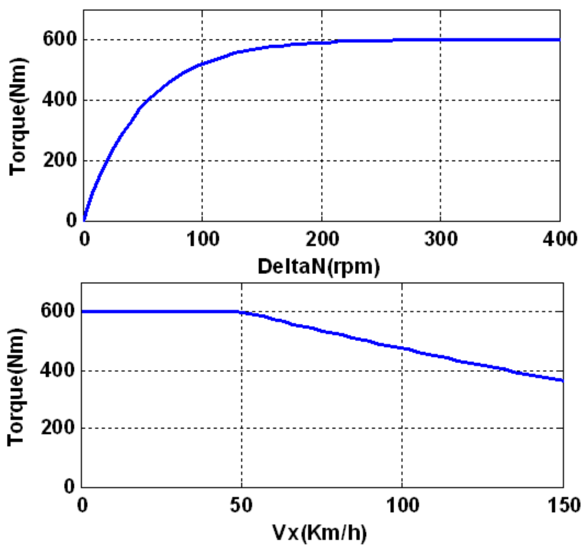


Figure 7: Feed-back equation graph

$$T_{TR\_FB} = C_{FB\_slope} \times T_{FB\_max} \times \left(1 - e^{(-DeltaN \times p_{slope\_e})}\right) \quad (8)$$

$$C_{FB\_slope} = \frac{\left(p_{slope\_q} \times V_x - p_{sat}\right)^2}{p_{sat}^{\ 2}} \quad (9)$$

We have developed equation controller based on the characteristic of the car and the will of the driver.

Compared to look-up table, control at more elaborate and dynamic range can be possible.

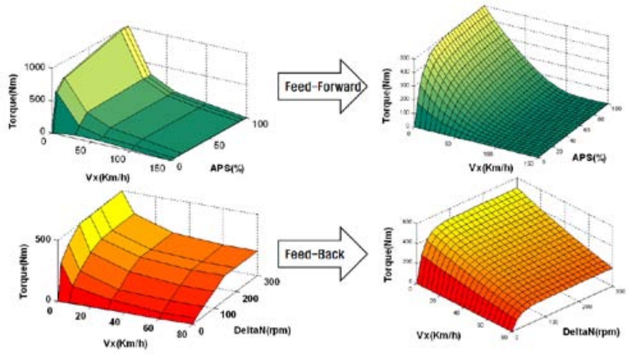


Figure 8: Comparison of Look-up table and Equation Control

3.2. Yaw Control

If the torque amount for Front and Rear wheel has been varied, the cornering force affecting the wheel can be changed and this could potentially lead to yaw moment in the car.

Yaw controller has been designed to decrease the yaw error using this method.

To find out yaw error which acts as input data for yaw control, desired yaw rate has been estimated by utilizing various methods shown below.

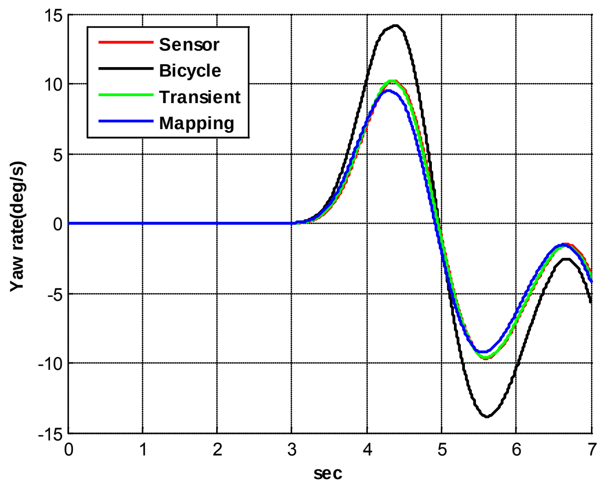


Figure 9: Double lane change desired yaw rate

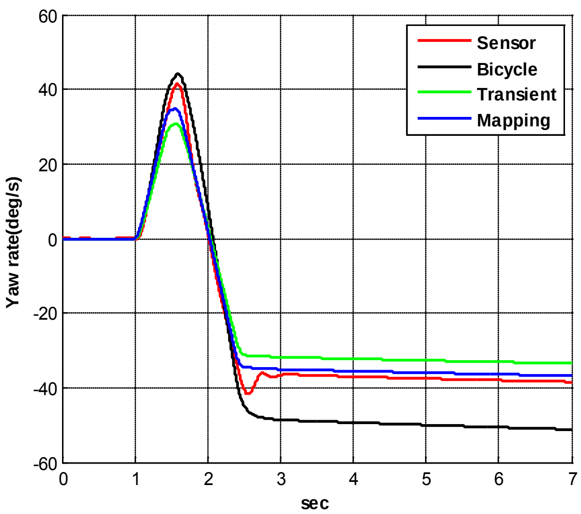


Figure 10: Fishhook desired yaw rate

Desired yaw rate can be estimated by using bicycle mode, transient model and simulation-based mapping look-up table. The following data is the comparison between the desired yaw rate and actual yaw rate.

Transient model which acts the most similar has been used from the simulation result.

Transient model has been accomplished by deriving space state eqution of Equation (10) from the bicycle model.
After Laplace transform has been applied, transfer function for equation (11) has been constructed.

$$\begin{pmatrix} \dot{v}_y \\ \ddot{\psi} \end{pmatrix} = \begin{bmatrix} -\frac{2(C_f + C_r)}{mv_x} & -v_x - \frac{2(C_f l_f - C_r l_r)}{mv_x} \\ -\frac{2(C_f l_f - C_r l_r)}{I_z v_x} & -\frac{2(C_f l_f^2 + C_r l_r^2)}{I_z v_x} \end{bmatrix} \begin{pmatrix} v_y \\ \dot{\psi} \end{pmatrix} + \begin{pmatrix} \frac{2C_f}{m} \\ \frac{2C_f l_f}{I_z} \end{pmatrix} \delta_f \qquad (10)$$

$$\Phi_{\delta_f \to \dot{\psi}} = \frac{b_2 s + (a_1 b_2 - a_3 b_1)}{s^2 + (a_1 + a_4)s + a_1 a_4 - a_2 a_3} \qquad (11)$$

$$a_3 = \frac{2(C_f l_f - C_r l_r)}{I_z v_x},\ a_4 = \frac{2(C_f l_f^2 + C_r l_r^2)}{I_z v_x},$$

Where,

$$b_1 = \frac{2C_f}{m},\ b_2 = \frac{2C_f l_f}{I_z}$$

By using yaw error derived, equation controller has been designed in a similar way for traction controller.

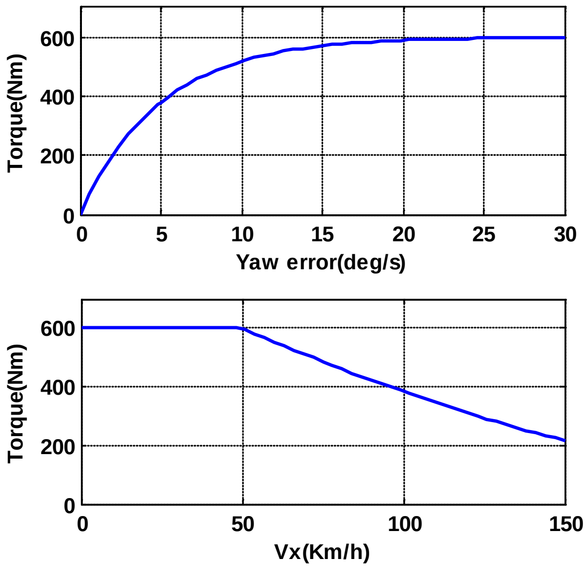


Figure 11: yaw control equation graph

$$T_{Yaw} = C_{Yaw\_slope} \times T_{Yaw\_max} \times \left(1 - e^{(-Yaw\_err \times p_{slope\_e})}\right) \qquad (12)$$

$$C_{Yaw\_slope} = \frac{\left(p_{slope\_q} \times V_x - p_{sat}\right)^2}{p_{sat}^{\ 2}} \qquad (13)$$

If above two equations are put together, yaw control equation can be derived below.

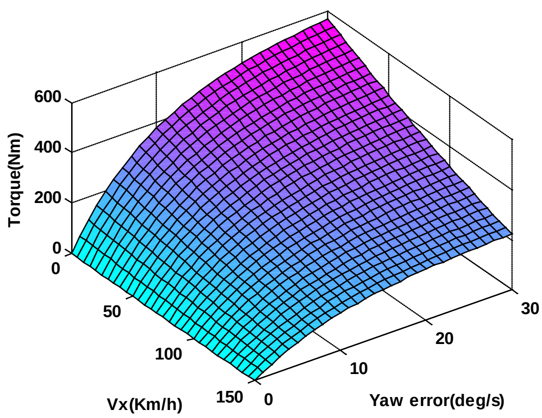


Figure 12: 3D Plot of Yaw control

3.3. Blending Control

Conventional controllers independently utilize traction control and yaw control, but in this paper, the two methods have been mixed together.

The concept of blending the traction and yaw controller is as follows.

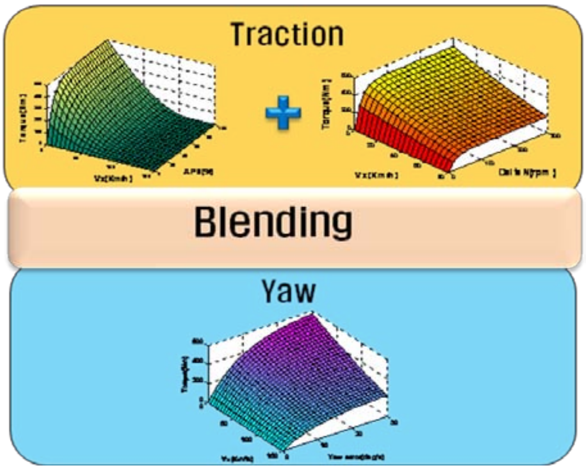


Figure 13: Blending Concept

Weighting functions varying along the error of traction control and yaw control can be multiplied. Each

Distributed output from the control algorithm can be used. The equation is shown below.

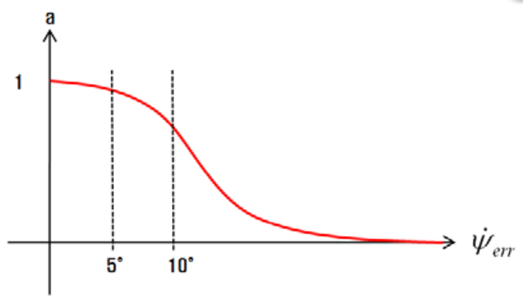


Figure 14: Weighting Function for Yaw control

$$T_{total} = a \cdot (T_{FF} + T_{FB}) + (1 - a) T_{yaw} \quad (14)$$

$$a = f(\dot{\psi}_{err}) \quad (15)$$

4. SIMULATION RESULT

This research has been preceded on top of method of "Model-based control design". To prove the performance of proposed control algorithm, computer simulation for driver performance and handling performance has been executed.

4.1. Simulation Configuration

To construct control logic for model base control design Matlab /simulink has been used. Carsim, which is fast in calculation but inexpensive, one of the powerful software in vehicle dynamics has been used. The following figure shows the interlinked simulation environment.

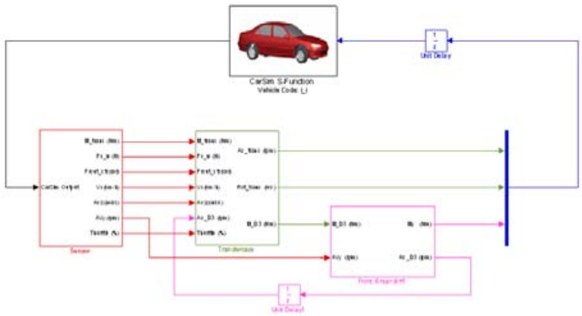


Figure 15: Overall system configuration

4.2. Traction control simulation

Since traction performance is one of the fundamental performances for 4WD, verifications were undergone at three difference conditions. To prove this, conventional look-up table method, PID control method and the proposed equation control methods were compared.

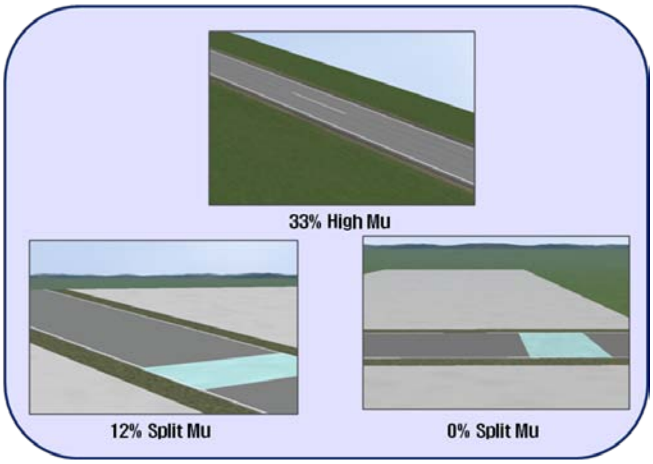
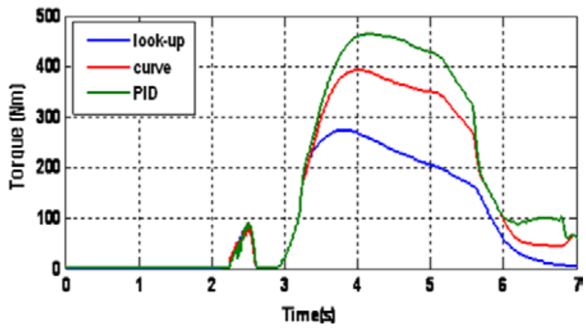


Figure 16: Traction Simulation Condition

The first condition is climbing experiment at 33% high mu. Without controlling the steering wheel, results shown below at Wide Open Throttle situation.

When at PID control in climbing situation, Front wheel torque had to be distributed as widely as possible,

But because of force loss from tire slip, controlling performance was very bad. Opposed, look up table control showed fairly bad performance compared to equation control due to the lack of control amount.



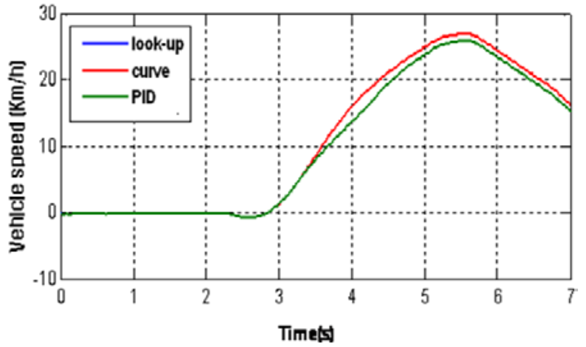


Figure 17: 33% straight simulation result

Second and third conditions are both split mu and the slope is 12% and 0% respectively.

Split mu condition at rear wheel has developed low mu (mu=0.05). So we have tested escape ability for Rear wheel based 4WD vehicle.

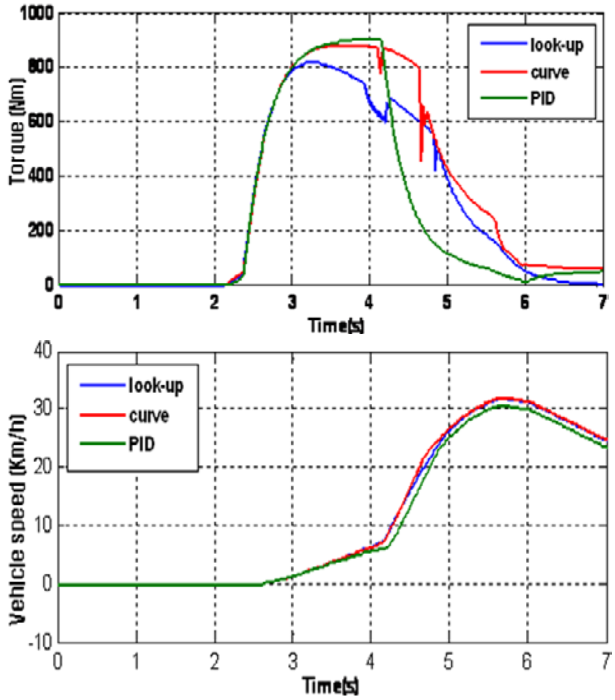


Figure 18: 12% Roller simulation result

As we can see from above, control amount for equation control is almost the same compared to PID and look up table.

The reason is because transfer torque limit is approximately 900Nm. To sum up, compared to vehicle speed, this shows better performance compared to different control method.

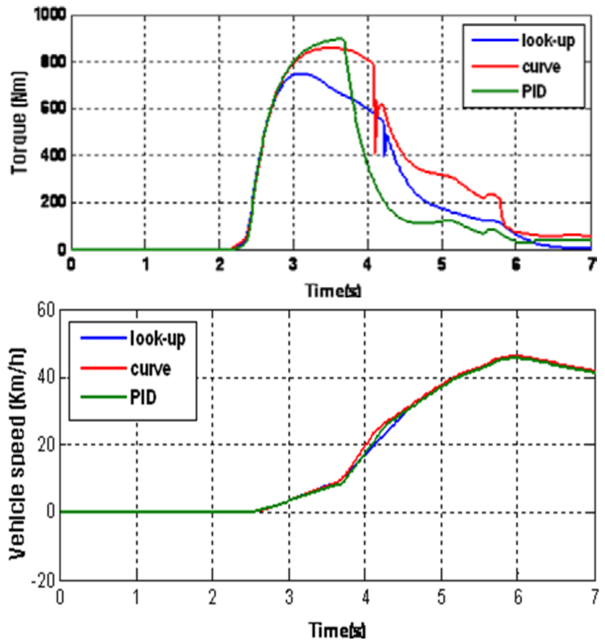


Figure 19: 0% Roller simulation result

We have come up with simulation result similar to that of 12% split mu. This shows the dominance of equation control.

4.3. Yaw control simulation

As mentioned earlier at yaw controller, yaw moment changes due to torque distribution for forward wheel.

This is why we have proved validity by comparing yaw rate and path under fishhook simulation condition.

Table 1: Yaw Simulation Condition

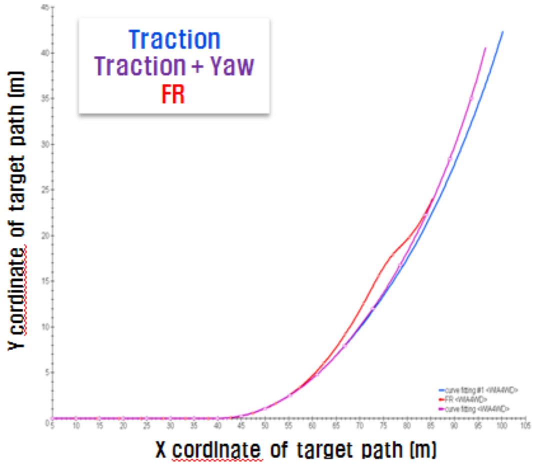
	Simulation 'a'	Simulation 'b'
Vehicle speed	40km/h ➡ 70km/h	40km/h ➡ 60km/h
Steering wheel angle	0˚ ➡ 60˚	0˚ ➡ 100˚
Road Condition	Mu = 0.5 , Slope = 0%	

Comparison results among back-wheel based 2WD model, only traction control model and traction+yaw control model is as follows.

If front-wheel torque increases for traction performance, the path for the vehicle goes under-steer.

Yaw rate also chatters as well. Opposed to this, model which considers both traction and yaw, the driver can follow the path by proper torque distribution.

Ahead of the proposed simulation 'a', 'b' conditions, both the Traction + Yaw model shows a more stable performance.



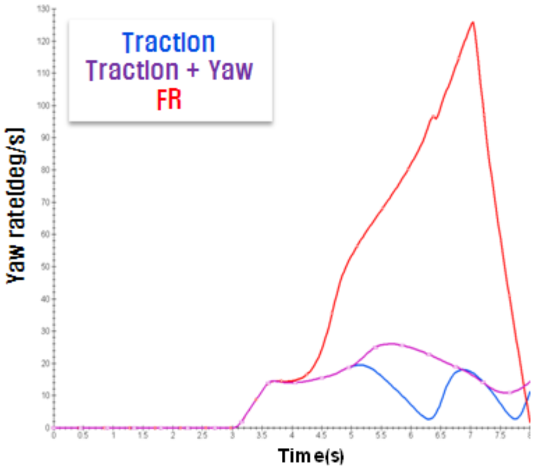
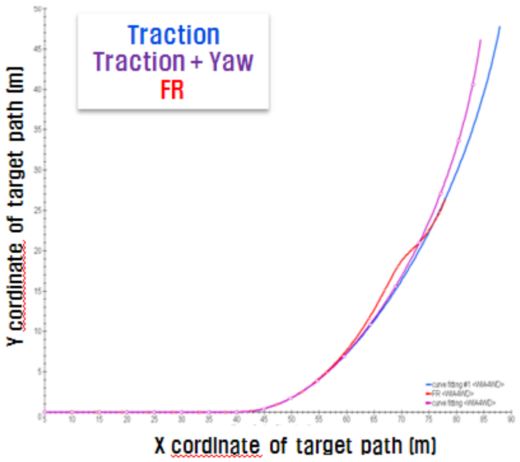


Figure 13: Fishhook simulation ‘a’ result



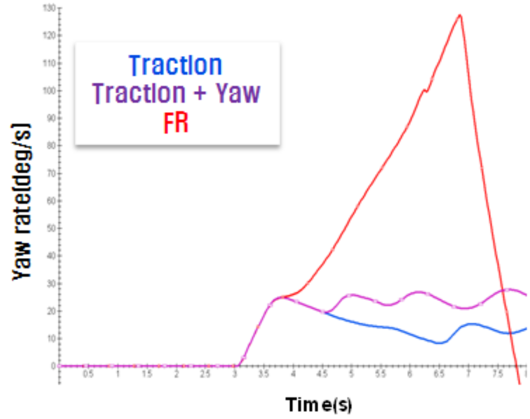


Figure 20: Fishhook simulation ‘b’ result

5. CONCLUSION

This paper presented a new traction and wheel slip control algorithm for the 4WD vehicle. A power-train model was derived for analyzing the torque flow of the 4WD vehicle. The proposed control algorithm was developed by using the Equation Method.

The control algorithm was also implemented and verified through the simulation. The simulation results showed that the proposed control algorithm can transfer the driving torque to the driven wheels and reduce the wheel slip. The proposed control algorithm can improve vehicle performance in terms of the ride comfort and power distribution.

REFERENCES

Hyeongcheol L., “Four-wheel drive control system using a clutchless centre limited slip differential”, Proc. IMechE Vol.220 Part D: J. Automobile Engineering, pp.665-681.

McKenzie, C. A. et al., "Ford Explorer control trac 4X4 system", SAE paper 952645, 1995.

Hiroyuki, A. et al., “AWD Vehicle Simulation with the Intelligent Torque Controlled Coupling as a Fully Controllable AWD System”, 2005 SAE World Congress.

Mitsuru, O. et al., “Development of a New Electronically Controlled 4WD System: Toyota Active Torque Control 4WD”, International Congress and Exposition Detroit, 1999.

Woojong Bong., “AWD Simulation with the Intelligent Torque Split Control Strategy for Improving Traction and Handling performance”, 2007 KSAE

E. Galvagno, “Differentials Modeling for Four Wheels Drive”, 2006 SAE World Congress