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TRANSFER CASE CLUTCH TORQUE ESTIMATION IN AWD VEHICLES

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ABSTRACT – AWD system transfers some portion of engine torque from the main drive shaft to the sub drive shaft. Recently, electronic motor actuated type AWD system has been emerged to control the vehicle more actively. Compared with a mechanical type AWD, electronic type AWD enables the variable torque distribution toward the sub drive shaft with the wet clutch friction material. This paper proposes the clutch torque estimation method of the electronic type transfer case. To compensate the uncertainty of clutch parameters, vehicle dynamics model is integrated in the torque estimator to utilize vehicle information with the sensor signal which is obtained from the vehicle can. By implementing carsim based simulation environment setup of AWD system, torque estimation is validated with several driving conditions.

1 INTRODUCTION

AWD has a roll to distribute engine torque to drive both front wheels and rear wheels, which makes a difference to conventional 2WD vehicle. Recently, the application of AWD system has been increasing in high performance passenger car due to the improvement of the engine power. Electronically actuated AWD system which is introduced in this paper has different hardware constitution from center differential 4WD and selective 4WD because it can control the amount of the distributed drive torque between the main drive shaft and the sub drive shaft by controlling the engaging force which is applied to the wet clutch of the transfer case. In control aspects, estimated vehicle states like slip ratio, slip angle, friction coefficient are commonly used as decisive factors in vehicle active safety system like ABS, ESC and AFS. For the vehicle state estimation, drive torque information is generally required in vehicle dynamics model. In case of 2WD system, engine torque which is obtained from ECU directly applies to derive the drive torque and warrants the high accuracy. However, engine torque information is not enough in AWD system because the drive torque of both front and rear shaft can be different depending on the amount of clutch engaging force. Variable clutch engaging force actuation of AWD is obviously different from short transient phenomenon like gear shifting, thus clutch torque estimation logic development in AWD system should be preceded for improving electronic type AWD vehicle performance by helping other principal vehicle states estimation in real time.

This paper suggests the novel estimation method of AWD clutch torque. Compared with the other previous works to estimate the transmitted clutch torque, vehicle dynamics model is integrated in the clutch model to utilize the vehicle sensor signal which helps the accuracy of the estimator as a compensation feedback. From the simulation environment setup by implementing AWD system as an external simulink model in carsim, torque estimation is validated with various driving scenarios.

2 SYSTEM MODELING

2.1 Drivetrain Dynamic Model

AWD system which is dealt in this paper can control the amount of torque by using multiple disc plates of wet clutch. And vehicle dynamic states with AWD system can change drastically depending on the amount of torque which is transferred to front or rear shaft. Thus simulation description should be available of variable distribution of front shaft torque depending on clutch engagement force especially between clutch slipping state and clutch lock up state. Schematic diagram of driveline model from transmission output shaft to front and rear differential gear can be shown in the figure 1 and mathematical models are as follows:

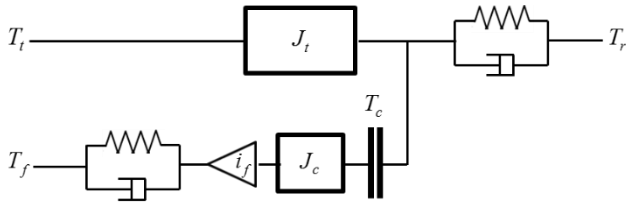

Figure 1: Schematic diagram of AWD transfer case system

$$T_t - T_c - T_r = J_t \dot{\omega}_t \tag{1}$$

$$T_c = f\left(F_c, \omega\right) \tag{2}$$

$$T_c - T_f / i_f = J_c \dot{\omega}_c \tag{3}$$

$$T_f = k_f\left(\theta_c - \theta_f\right) + b_f\left(\dot{\theta}_c - \dot{\theta}_f\right) \tag{4}$$

$$T_r = k_r\left(\theta_t - \theta_r\right) + b_r\left(\dot{\theta}_t - \dot{\theta}_r\right) \tag{5}$$

Where T_t is the transmission output torque, T_c is the clutch engagement torque, T_f and T_r are the front and rear shaft torque each. J_t is the main drive shaft inertia, J_c is the front drive shaft inertia and F_c is the clutch engagement force. ω is the clutch slip angular velocity. At the end of each drive shaft can be modelled as spring and damper system to represent the distortion of shaft. Transmitted clutch torque is expressed as a certain function of $f\ (F_c,\ \omega_{slip})$ and its detailed expression will be dealt in section 2.2.1.

2.2 Wet Clutch Model

To make the variable torque distribution by allowing clutch slipping for quite a long time, wet clutch is generally used in electronic AWD by overlapping multiple disc plates to transfer large amount of torque. To prevent the instantaneous disc wear, wet clutch absorbs the heat which is generated in friction plate by circulating oil continuously.

2.2.1 Generalized Maxwell Slip (GMS) Model

To guarantee the validity of simulation, appropriate friction model for the wet clutch system is important. GMS model [1,2], which was proved for friction model was selected for modelling transfer case system.

When clutch is fully lock up, differential equation of clutch transfer torque is as follows:

$$\frac{dM_i}{dt} = k_i\left(p, T_t\right) \cdot \omega \tag{6}$$

Where $k_i\ (p,\ T_t)$ is the function of clutch engagement force and transmission output torque. When clutch is slipping, differential equation of clutch transfer torque is as follows:

$$\frac{dM_i}{dt} = \operatorname{sgn}\left(\omega\right) C\left(\alpha_i - \frac{M_i}{s\left(\omega, p\right)}\right) \tag{7}$$

Where $s(\omega, p)$ is the stribeck equation about torque, which is expressed as follows:

$$s(\omega, p) = \mathrm{sgn}(\omega)\left(M_s + \left[M_c - M_s\right]e^{-\left|\frac{V_s(p)}{\omega}\right|}\right) \tag{8}$$

Wet clutch should consider viscous effect by the oil film between disc plates. Especially for wet clutch, drag torque contributes some part of transmitted torque so that this should be modelled carefully to improve the accuracy of transient response [4]. Drag torque of wet clutch is originated from the shear stress between plates and can be modelled by fluid dynamic analysis [4-6]. Then, viscous friction torque equation is obtained as follows:

$$M_{vis} = 2\pi\int_{r_1}^{r_o} \frac{\mu\omega r^3}{h}\left(1 + 0.0012\,\mathrm{Re}_h^{0.94}\right)dr \tag{9}$$

Where r_i is a clutch inner diameter, r_o is an effective diameter due to the centrifugal force, which is usually less than clutch outer radius. Re_h is a Reynolds number of Automatic Transmission Fluid (ATF), which is written as follows:

$$\mathrm{Re}_h = \frac{\rho\omega r h}{\mu} \tag{10}$$

Where μ is a viscous coefficient of ATF, ρ is a density of ATF and h is a clearance length of clutch plate.
Finally, transferred torque when clutch is slipping is the sum of the two effects:

$$T_c = M_i + M_{vis} \tag{11}$$

To guarantee the feasibility of wet clutch engagement simulation, clutch experimental friction coefficient data for commercial vehicle implementation was used.

3 ESTIMATOR DESIGN

3.1 Vehicle Dynamics Model

To improve the accuracy of the estimated clutch torque given parameter uncertainties, vehicle dynamics model which can connect clutch torque state to other states can be used. Since conventional vehicle dynamics states can be compensated by vehicle measurement signals, clutch torque can also be corrected indirectly due to the relation of equations.

By integrating vehicle longitudinal dynamics, lateral dynamics, yaw dynamics and wheel dynamics, clutch torque can be estimated by extended Kalman filter [7,8].

The equations of motion for the vehicle based on bicycle model can be categorized by three dynamics.
Longitudinal dynamics is:

$$\dot{v}_x = \frac{1}{m}\left\{F_{xf}\cos(\delta) - F_{yf}\sin(\delta) + F_{xr} - \frac{1}{2}\rho_{air}C_d A v_x^2 - mg\sin(\theta)\right\} + v_y r \tag{12}$$

Where v_x is the longitudinal velocity, m is the vehicle mass, F_{xf} is the front longitudinal tyre force, F_{xr} is the rear longitudinal tyre force, F_{yf} is the front lateral tyre force, δ is the tyre steer angle, ρ_{air} is the air density, C_d is the drag coefficient, v_y is the lateral velocity, r is the yaw rate and θ is the hill angle.

Lateral dynamics is

$$\dot{v}_y = \frac{1}{m}\left\{F_{xf}\sin(\delta) + F_{yf}\cos(\delta) + F_{yr}\right\} - v_x r \tag{13}$$

Where F_{yr} is the rear lateral tyre force.
Yaw dynamics is:

$$\dot{r} = \frac{1}{I_z}\left\{l_f\left(F_{xf}\sin(\delta) + F_{yf}\cos(\delta)\right) - l_r F_{yr}\right\} \tag{14}$$

Where I_z is the vehicle yaw moment of inertia, l_f is the distance from the center of mass to the front axle and l_r is the distance from the center of mass to the rear axle.

Another dynamics in which force terms are related is wheel dynamics that is expressed as follows:

$$\dot{\omega} = \frac{1}{I_w}\left\{T_d - R_e F_{xi} - R_e R_{ri}\right\} \tag{15}$$

Where ω is the wheel angular velocity, I_w is the wheel inertia, T_d is the wheel drive torque. In this paper, T_d is assumed to be known and does not consider braking situation. R_e is the wheel effective radius. F_{xi} and R_{ri} are the tyre longitudinal force and the tyre rolling resistance whose subscript i includes *fl* (front left), *fr* (front right), *rl* (rear left), *rr* (rear right) which is written throughout the paper.

3.2 Extended Kalman Filter Design

To combine vehicle dynamics model which is mentioned in section 3.1 with wet clutch model which is mentioned in section 2.1 to estimate the clutch torque, extended Kalman filter was adopted in this paper.

The states for Kalman filter are defined as follows:

$$\mathbf{x}(t) = \left[v_x, v_y, r, \boldsymbol{\omega}, T_c, \mu_\Delta\right]^T \tag{16}$$

Where

$$\boldsymbol{\omega} = \left[\omega_{fl} + \omega_{fr}, \omega_{rl} + \omega_{rr}\right] \tag{17}$$

Here, the amount of nominal clutch disc friction coefficient change was added as a state to deal with the uncertainty of clutch parameter value.

The measurements from vehicle signals are as follows:

$$\mathbf{y}(t) = \left[a_x, a_y, r, \boldsymbol{\omega}\right] \tag{18}$$

Where a_x and a_y are related as:

$$a_x = \frac{1}{m}\left\{F_{xf}\cos(\delta_f) + F_{yf}\sin(\delta_f) + F_{xr} - \frac{1}{2}\rho_{air} C_d A v_x^2 - mg\sin(\theta)\right\} \tag{19}$$

$$a_y = \frac{1}{m}\left\{F_{xf}\sin\left(\delta_f\right)+F_{yf}\cos\left(\delta_f\right)+F_{yr}\right\} \tag{20}$$

To sum up the above equations, the state space system is obtained as follows:

$$\dot{\mathbf{x}}(t)=\begin{bmatrix}\dot{v}_x\\ \dot{v}_y\\ \dot{r}\\ \dot{\omega}_{fl}+\dot{\omega}_{fr}\\ \dot{\omega}_{rl}+\dot{\omega}_{rr}\\ \dot{T}_c\\ \dot{\mu}_\Delta\end{bmatrix}=\begin{bmatrix}\frac{1}{m}\left\{F_{xf}\,\mathbf{cos}(\delta)-F_{yf}\,\mathbf{sin}(\delta)+F_{xr}-\frac{\rho_{air}C_d A v_x^2}{2}-mg\,\mathbf{sin}(\theta)\right\}+v_y r\\ \frac{1}{m}\left\{F_{xf}\,\mathbf{sin}(\delta)+F_{yf}\,\mathbf{cos}(\delta)+F_{yr}\right\}-v_x r\\ \frac{1}{I_z}\left(l_f\left[F_{xf}\,\mathbf{sin}(\delta)+F_{yf}\,\mathbf{cos}(\delta)\right]-l_r F_{yr}\right)\\ \frac{1}{I_w}\left(i_{ff}T_c-R_e F_{xf}-R_e R_{rf}\right)\\ \frac{1}{I_w}\left\{(T_t-T_c)i_{fr}-R_e F_{xr}-R_e R_{rr}\right\}\\ \mathbf{sgn}(\omega)\cdot C\cdot\left(1-\frac{T_c}{\mathbf{sgn}(\omega)\cdot F_c\cdot r_c\left(\mu_n+\mu_\Delta\right)}\right)\\ 0\end{bmatrix}=f\left(\mathbf{x}(t),u(t)\right) \tag{21}$$

$$\mathbf{y}(t)=\begin{bmatrix}a_x\\ a_y\\ r\\ \boldsymbol{\omega}\end{bmatrix}=\begin{bmatrix}\frac{1}{m}\left\{F_{xf}\,\mathbf{cos}(\delta)-F_{yf}\,\mathbf{sin}(\delta)+F_{xr}-\frac{1}{2}\rho_{air}C_d A v_x^2-mg\,\mathbf{sin}(\theta)\right\}\\ \frac{1}{m}\left\{F_{xf}\,\mathbf{sin}(\delta)+F_{yf}\,\mathbf{cos}(\delta)+F_{yr}\right\}\\ r\\ \boldsymbol{\omega}\end{bmatrix}=h\left(\mathbf{x}(t),u(t)\right) \tag{22}$$

Please keep in mind that above state space form applies only when clutch is slipping. Viscous term is not appeared in the equation but its effect is not dominant for estimation. When clutch is fully lock up, estimated clutch torque value which is obtained from the above equation does not comply with the real value anymore so that clutch torque should be dealt with differently. In a rigid shaft, the amount of transferred torque at both sides of the shaft is proportional to load. Thus the wheel vertical load contributes the transferred amount of torque in AWD vehicle. Then front and rear weight shifting was considered to determine the amount of distribution torque. Shifted weight was calculated from the following equation:

$$F_{zf}=\frac{l_r}{l}mg-\frac{h_{cg}}{l}ma_x,\; F_{zr}=\frac{l_f}{l}mg+\frac{h_{cg}}{l}ma_x \tag{23}$$

Then distributed torque is determined as follows:

$$T_f=\frac{F_{zf}}{F_{zf}+F_{zr}}T_t,\; T_r=\frac{F_{zr}}{F_{zf}+F_{zr}}T_t \tag{24}$$

The algorithms of discrete time extended Kalman filter can be expressed as follows [7]:

$$\hat{\mathbf{x}}_k(-)=\hat{\mathbf{x}}_{k-1}(+)+\int_{t_{k-1}}^{t_k} f\left(\mathbf{x}(\tau),u(\tau)\right)d\tau \tag{25}$$

$$\mathbf{P}_k(-)=\mathbf{P}_{k-1}(+)+\int_{t_{k-1}}^{t_k}\left\{\mathbf{F}(\tau)\mathbf{P}(\tau)+\mathbf{P}(\tau)\mathbf{F}^T(\tau)+\mathbf{L}\mathbf{L}^T\right\}d\tau \tag{26}$$

$$\mathbf{K}_k(-)=\mathbf{P}_k(-)\mathbf{H}_k^T\left[\mathbf{H}_k\mathbf{P}_k(-)\mathbf{H}_k^T+\mathbf{R}_k\right]^{-1} \tag{27}$$

$$\hat{\mathbf{x}}_k(+)=\hat{\mathbf{x}}_k(-)+\mathbf{K}_k\left[\mathbf{y}_k-h\left(\hat{\mathbf{x}}_k(-),u(t)\right)\right] \tag{28}$$

$$\mathbf{P}_k(+) = [\mathbf{I} - \mathbf{K}_k \mathbf{H}_k] \mathbf{P}_k(-) \tag{29}$$

Where **F** and **H** are the linearization of equation (21) and (22) around $\hat{\mathbf{x}}_k(-)$ at each time step. **P** is the error covariance and **L** is the disturbance input matrix.

4 SIMULATION

In this paper, carsim software was used to validate the AWD system driving simulation. Transfer case developed by simulink was implemented in carsim. Except for the transfer case part, carsim internal model was used. To represent the hydraulic pressure response generated my motor pump, AMESIM model was applied in the clutch engaging force input. Vehicle driveline is based on Front ship Rear wheel drive (FR) type. Estimator sample time wat set to 0.001s but measurement signal was updated by 0.01s. At first, vehicle starts to move from a stop state with a steady throttle. And a step throttle is inserted between 15-25 seconds. Meanwhile, AWD is activated between 5-25 seconds to analyze the responses in gear upshifting and downshifting. AWD is also activated between 30-35 seconds for the validation for clutch lock up engagement. To consider various driving conditions, estimator was simulated at both high mu and low mu surfaces. To verify the robustness of integrated clutch torque estimation, 10% uncertainty of mass and clutch mu parameters were applied in the simulation.

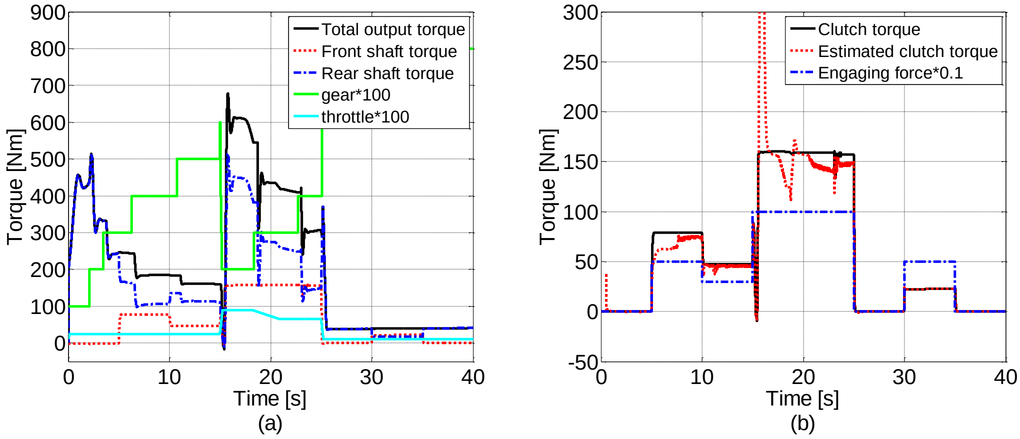

Figure 2 Clutch torque estimation in high mu (0.85) (a) Drivetrain response (b) clutch torque estimation result

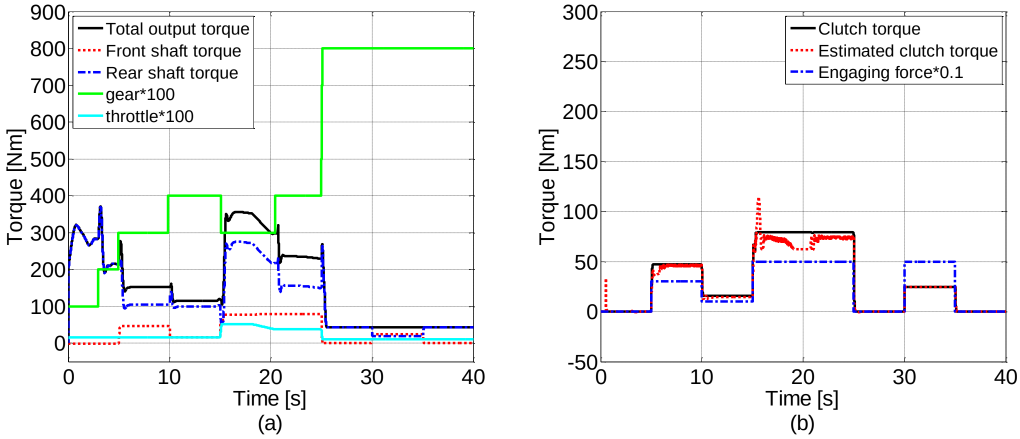

Figure 3 Clutch torque estimation in low mu (0.3) (a) Drivetrain response (b) clutch torque estimation result

Figure 2(a) and figure 3(a) show the response of driving situation which is described in the above paragraph. Figure 2(b) and figure 3(b) show the simulation result of AWD clutch torque estimation in each road surface. Total output torque means engine torque multiplied by transmission gear ratio. Front and rear shaft torque are the amount of torque which is delivered to the each shaft. To grasp the driving situation easily, throttle and gear shifting response are also added in the figure. In both conditions, clutch torque was underestimated initially at certain engaging force, then followed real clutch torque value at low steady throttle region. At severe transient slipping region of rapid acceleration, high clutch slip caused spike-shaped estimation error and it was worse with strong engaging force. However, error was reduced as slip amount becomes low. As can be seen in high steady throttle region, estimated clutch torque didn’t converge the real value since clutch dynamic model is incomplete. Obviously, no discrepancy appeared in lock up condition at the range of 30-35s.

5 CONCLUSION

In this paper, clutch torque estimation method using vehicle dynamics model was suggested. To consider the feasibility of real time application, simulation was conducted at both of high mu and low mu surfaces. With clutch mu parameter and vehicle mass uncertainties, clutch torque estimator showed robust performance by compensating clutch mu discrepancy. Proposed torque estimation algorithm can be applied for the vehicle state estimation and tyre parameter estimation, ultimately vehicle dynamics control area, for the logic development of advanced AWD driving. However, estimator experimental validation is required for future work.

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