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MODIFIED GENERALISED MAXWELL SLIP BASED ELECTRO-HYDRAULIC TRANSFER CASE MODELING FOR ALL-WHEEL DRIVE VEHICLE SIMULATION

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ABSTRACT–This paper describes the modeling of a torque-on-demand transfer case and an all-wheel drive (AWD) vehicle simulation through model integration. To develop the AWD vehicle controller, reliable and robust mathematical modeling of transfer case for AWD vehicle simulation should be proceeded. However, conventional wet clutch model cannot be applied for simulating the AWD vehicle especially when clutch is fully lock up state because of the chattering response of torque followed by the change of external factors. By replacing a slip-regulation equation of lock-up state with a degree of freedom (DOF) reduction equation in the generalized maxwell slip (GMS) model, the chattering and instability issue of original GMS model for AWD vehicle simulation was solved in this study. For parameter verification of the wet clutch, the simulation of the transfer case module and validation with the experimental data were conducted first. Then, the simulation of an AWD vehicle was conducted through the integration of the developed one in CarSim software. Through the comparison of the modified GMS model with the original GMS model, the former is verified to be superior to the latter for stable simulation.

KEY WORDS : All-wheel drive, Drag torque model, Generalized maxwell slip model, Transfer case, Wet clutch model

NOMENCLATURE

T_t : transmission output torque, N·m
T_c : transfer case clutch torque, N·m
T_f : front shaft torque, N·m
T_r : rear shaft torque, N·m
J_t : transmission output shaft inertia, kg·m^2
J_c : transfer case clutch inertia, kg·m^2
i_f : transfer case gear ratio, -
k_f : front shaft spring constants, N/rad
k_r : rear shaft spring constants, N/rad
b_f : front shaft damping constants, N·s/rad
b_r : rear shaft damping constants, N·s/rad
θ_t : transmission output shaft rotational angle, rad
θ_c : transfer case clutch rotational angle, rad
θ_f : front shaft rotational angle, rad
θ_r : rear shaft rotational angle, rad
F_c : transfer case clutch engagement force, N
M_i : friction torque of ith maxwell element, N·m
M_s : static friction torque, N·m
M_c : coulomb friction torque, N·m
M_d : hydrodynamic drag torque, N·m
k_i : integral gain of wet clutch model in sticking state, -
α_i : normalized maximum friction torque that each maxwell element can sustain, -
C : attractor gain that the rate of friction torque follows the stribeck effect, -
μ : road surface friction coefficient, -
μ_s : static friction coefficient, -
μ_c : coulomb friction coefficient, -
μ_f : dynamic viscosity of fluid, kg/(m·s)
r_i : inner radius of clutch friction plate, mm
r_o : outer radius of clutch friction plate, mm
h : height from ground to vehicle's center of gravity, m
h_d : clearance length of the clutch pack, m
Re_h : reynolds number, -
ρ : density of the fluid, kg/m^3
L : wheelbase length, m
L_r : distance from vehicle's center of gravity to front axle, m
a_x : longitudinal acceleration of vehicle, m/s^2
g : gravitational acceleration, m/s^2

1. INTRODUCTION

Unlike two-wheel drive (2WD) vehicles that transfer torque only to the front or rear wheels, an all-wheel drive (AWD) system distributes engine torque to both of the front and rear wheels. When first introduced, AWD systems were usually mounted on off-road vehicles that traveled through rough terrain. Currently, on-road vehicles are also equipped

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with AWD systems, including high-performance sedans, because some 2WD vehicles cannot maintain their maximum traction force with the increase in engine power. Among a various types of AWD vehicles, torque-on-demand AWD vehicle equipped with a transfer case of electromechanical type has become popular in the market of all types of AWD vehicles based on its ample technological advantages (Williams, 2006).

The active AWD vehicle discussed in this paper is different from the conventional passive AWD vehicle, which distributes the drive torque to the front and rear wheels deterministically based on its hardware structure; the active AWD vehicle enables the drive torque transmitted from the rear end of the transmission to be distributed intentionally, depending on the current vehicle status, through the control of commands to the clutch system by the module. Due to the advantage of arbitrary clutch engagement in torque-on-demand AWD vehicle, the task of designing AWD control algorithm is at a high priority for enhancing longitudinal driving performance and lateral stability of the AWD vehicle (Osborn and Shim, 2006; Kim *et al*., 2015; Jung *et al*., 2022). To verify the effectiveness of an AWD vehicle controller in simulation environment, an AWD vehicle simulation should provide reliable outcomes that agree well with the experimental data. Especially, it is important that simulation result should guarantee a stable torque representation at each output shaft followed by arbitrary model switching between slipping and lock-up state (Holgerson, 1997).

There have been several approaches to the AWD system simulation. Yang *et al.* (1998) included the wet clutch model in the transfer case. However, its friction model was obtained from an empirical formula. Subramanyam *et al.* (2000) and Wheals *et al*. (2004) suggested a simulation method for an AWD system. However, they did not address the characteristics of the wet clutch model in detail. Deur *et al*. (2005) suggested the dynamic wet clutch model that includes fluid dynamics. Ompusunggu *et al.* (2013) suggested another wet clutch model by applying generalized Maxwell slip (GMS) model. However, the precise model of drag torque, which is one of the principal characteristics in a wet clutch, was not considered. Alizadeh and Boulet (2014) suggested a wet clutch model for a robust control application. Although it was simpler than other wet clutch models, its modeling accuracy was not satisfactory. Above all things, most critical deficiency of previous approaches was no consideration for model switching in multiple output shafts (see Figure 1), especially when external factors that may affect the amount of torque distribution between front and rear shaft such as road friction coefficient, difference of front and rear final reduction gear ratio, and difference of tire effective radius, etc. are changed.

Through the modification of original GMS model, this study introduces a wet clutch model that can be applied to the transfer case for AWD vehicle simulation. Specifically, lock-up state model, which is expressed as a relative slip-regulation equation in the original GMS model, was replaced with a degree of freedom (DOF) reduction equation. This study has its contribution on suggesting an AWD vehicle simulation method with high fidelity in the presence of changing external factor such as road friction coefficient.

The remainder of this paper is organized as follows. In Section 2, the modeling of the transfer case is discussed in three parts: the driveline dynamics, wet clutch model, and hydraulic component model. In Section 3, the results of the simulation of the transfer case module are provided and compared with the experimental results. Finally, in Section 4, the integration of the transfer case model into the entire vehicle model is introduced. The results of simulating arbitrary clutch engagement and disengagement while the vehicle is driving are provided. Also, the enhancement of transfer case torque response of the suggested model is discussed in detail through the comparison with original GMS model. Then, AWD vehicle simulation is compared with the experimental data to verify its effectiveness.

2. TRANSFER CASE MODELING

2.1. Dynamic Equation

The vehicle dynamic model that includes AWD system can be different depending on the amount of torque that is distributed to the front and rear drive shafts. By changing the clutch engagement force in the transfer case, an AWD system can control the torque that is transferred to the sub-drive shafts. For validating developed control algorithm of AWD system through simulation, it is possible to simulate variable torque distribution in a slipping state and then smoothly transition to the lock-up state without undesirable instability. Figure 1 shows the schematic diagram of a transfer case system that represents the driveline model from the transmission output shaft to both of the front and rear shafts. In contrast with a general clutch system, it is noteworthy that transfer case have multiple output shafts. Then, the mathematical expressions are as follows:

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

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

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

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

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

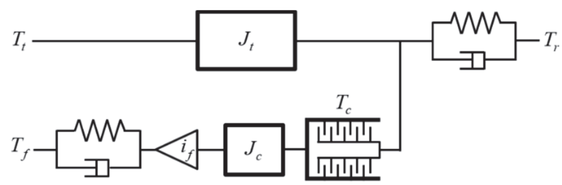


Figure 1. Transfer case model of single input and double outputs.

where ω_{slip} is the relative angular velocity defined as follows:

$$\omega_{slip} = \omega_t - \omega_c \tag{6}$$

In Equation (2), $f(F_c, \omega_{slip})$ is the clutch torque function, which will be described in detail in the following section.

2.2. Wet Clutch Model

Clutch system in the vehicle has a role of transferring the torque from the upper shaft to the lower shaft. A clutch is described as a dry or wet clutch, depending on its type of friction material. A dry clutch has a higher friction coefficient than a wet clutch. However, dry clutches have a tendency to easily wear out due to heat, if they slip for a long time. To address the weaknesses of the dry clutch, the wet clutch uses oil to prevent the friction material from heating up dramatically. However, the wet clutch has a low friction coefficient due to its different material and grooves. Therefore, to allow high torque transfer, a wet clutch generally consists of multiple friction plates.

2.2.1. GMS friction model

In the case of a system that has a contact force between two objects, such as a clutch, it is important to improve the accuracy of the friction model to assure the feasibility of the simulation. There have been many studies concerning friction models, e.g., LuGre, Dahl, Leuven, and GMS friction model (Lampaert *et al*., 2003; Piatkowski, 2014). The GMS friction model is a modification of the LuGre friction model. As in the Leuven model, the GMS model includes the hysteresis effect but it is much simpler (Lampaert *et al*., 2003). In discerning the sticking and slipping states, the GMS model is expressed as follows:

·Sticking (lock-up)

When the clutch is fully engaged, the differential equation for the torque transferred to the clutch is as follows (Ompusunggu *et al*., 2013; Lampaert *et al*., 2003):

$$\frac{dM_i}{dt} = k_i \cdot \omega_{slip} \tag{7}$$

·Slipping

When the clutch actuation force is positive but the clutch is not fully engaged, the differential equation for the torque transferred to the clutch due to the static friction and Coulomb friction is as follows:

$$\frac{dM_i}{dt} = \mathrm{sgn}\left(\omega_{slip}\right) \cdot C \cdot \left(\alpha_i - \frac{M_i}{s\left(\omega_{slip}\right)} \right) \tag{8}$$

where $\sum_{i=1}^{N} \alpha_i = 1$ and $s(\omega_{slip})$ is the Stribeck friction equation which is expressed as follows:

$$s\left(\omega_{slip}\right) = \mathrm{sgn}\left(\omega_{slip}\right) \left\{ M_s + \left(M_c - M_s\right) e^{-\left|\frac{\omega_s}{\omega_{slip}}\right|} \right\} \tag{9}$$

$$M_s = \alpha_i \mu_s r_c F_c,\ M_c = \alpha_i \mu_c r_c F_c \tag{10}$$

By assuming a uniform distribution of pressure along the entire clutch plate, the effective radius of the clutch r_c is as follows:

$$r_c = \frac{2}{3} \frac{r_o^3 - r_i^3}{r_o^2 - r_i^2} \tag{11}$$

For the values of the static and dynamic friction coefficients that are used in the Stribeck equation, the experimental values for the material that is manufactured for the vehicle were applied in the simulation.

2.2.2. Drag torque model

Drag torque refers to the torque required to overcome the rotational resistance generated by the driveline system to rotate the unloaded clutch plate. In a wet clutch, the drag torque changes depending on some hydraulic parameters and the hardware structure. The inclusion of drag torque in the model can improve the accuracy of the front and rear wheel transmission torque values (Hashimoto *et al*., 1983; Yuan *et al*., 2006).

In a wet clutch, the drag torque is generated by shear stress due to the fluid viscosity between the friction plate and the disc and can be modeled through hydrodynamic analysis. With some assumptions about the fluid, the model of hydrodynamic drag torque, M_d is as follows (Yuan *et al*., 2006):

$$M_d = 2\pi N \int_{r_i}^{r_o} \frac{\mu_f \omega_{slip} r^3}{h_d} \left(1 + 1.2 \cdot 10^{-3}\, \mathrm{Re}_h^{0.94}\right) dr \tag{12}$$

where Re_h is the Reynolds number, defined by:

$$\mathrm{Re}_h = \frac{\rho \omega_{slip} r h_d}{\mu_f} \tag{13}$$

Finally, the torque transferred to the lower shaft is described follows:

$$T_c = f\left(F_c, \omega_{slip}\right) = \begin{cases} \sum_{i=1}^{N} M_i, \ \textit{sticking state} \\ \sum_{i=1}^{N} M_i + M_d, \ \textit{slipping state} \end{cases} \tag{14}$$

2.2.3. State transition between sticking and slipping

It is important to determine the exact moment of the state transition, i.e., from slipping to sticking and vice versa, in the AWD system model. Unlike the single drive shaft in a 2WD, an AWD transfer case has two output shafts when engaged. Therefore, the torque transferred to each shaft can be different, depending on the load. Theoretically, sticking happens when the relative slip becomes zero. However, due to the issue of discretization in modeling, the information about the relative slip itself is not sufficient to determine the state transition. Therefore, an allowable torque limit condition is used simultaneously. Unlike other studies of clutch systems with a single output shaft (Ompusunggu *et al*., 2013; Oh *et al*., 2014; Bachinger *et al*., 2015), it is difficult to determine the torque limit for the dual output shafts, because the allowable torque changes depending on the load conditions at both sides. Albeit torque limit is affected by these external conditions in AWD vehicle, the transmission output torque is distributed depending on a front and rear weight shifting ratio for a homogenous road surface, which is the most common driving condition. Therefore, this study used two conditions simultaneously to prevent instability during the state transition. The state transition condition from slipping to sticking is as follows:

$$T_c \geq \left(\frac{L_r}{L} - \frac{a_x}{g}\frac{h}{L}\right)T_t \ \& \ 0 \leq \omega_{slip} \leq 0.01\,rad/s \tag{15}$$

2.3. DOF Reduction for the Sticking Condition

The clutch sticking model introduced in the previous subsection has an integral gain to regulate the relative slip. However, it was found that integral gain tuning is insufficient for representing the sticking conditions just after the state transition to deal with various driving situations, e.g., rapid full throttle in a μ-split condition. The simulation stability of the entire model should be addressed systematically through the process of switching between the two dynamic models (Duan *et al*., 2013). To solve this problem, a DOF reduction model that synchronizes the rotational angular velocity between the clutch drive plate and the driven plate was adopted in this study. The driveline dynamics model of sticking, using the DOF reduction model, is as follows:

$$\dot{\omega}_t = \dot{\omega}_c \ \& \ \omega_t = \omega_c \tag{16}$$

$$T_t - T_f - T_r = \left(J_t + J_c\right)\dot{\omega}_t \tag{17}$$

$$T_f = k_f\left(\theta_t - \theta_f\right) + b_f\left(\dot{\theta}_t - \dot{\theta}_f\right) \tag{18}$$

$$T_r = k_r\left(\theta_t - \theta_r\right) + b_r\left(\dot{\theta}_t - \dot{\theta}_r\right) \tag{19}$$

2.4. Hyraulic Parts Model

The inclusion of the hydraulic parts of the transfer case is important to represent the transient response of the system. In order to obtain simulation results that best match with the experimental data, this study used the AMESim software, which is useful for modeling minor hydraulic systems, for the hydraulic model simulation. The effects of the hydraulic parts are discussed in the simulation section. Some key parts of the hydraulic model are as follows:

- Vapor ventilation valve: This valve has a role to release the undesirable vapor that is generated in the hydraulic line. It is located in the upper part of piston and is designed to open in 2 bar and re-close in 3 bar.
- Relief valve: This valve serve to reduce the pressure of hydraulic line, when the pressure reached its limit. It is designed with 15-bar cracking pressure.
- Hydraulic pump: This pump has a role to compress the fluid. In the experiment, a gerotor pump was used. In the AMESim simulation, the pump was modeled as a nominal block. The pump displacement is 0.25 cc/rev and its typical speed is 6000 revolutions per minute (RPM).

3. SIMULATION VALIDATION OF TRANSFER CASE MODEL

The feasibility of parameter usage and usability of the transfer case model were verified through the comparison of the experimental data with the simulation results. One output shaft, i.e., the sub-shaft, was set to be fixed (no rotation) and the other output shaft, i.e., the main shaft, was set to be free-rotating. The verifications were conducted using two methods. First was the pressure-torque (PT) test. In this test, the pressure was stepped up and down repeatedly, at a clutch oil temperature of 50 °C and a relative angular velocity of 50 RPM, by gradually increasing the pressure to 1 bar. The second was the hysteresis (HS) test. In this test, the pressure was ramped up and down continuously, and the experiment was conducted three times under the same conditions. To ascertain the influence of temperature variation on the torque response, the HS tests were conducted at two clutch oil temperatures, 25 °C and 50 °C, and at a relative angular velocity of 50 RPM. The material properties that were used in the simulation are listed in Table 1.

Table 1. Parameter values of the transfer case.

Parameter	Value	Parameter	Value
J_t	0.075 kg·m^2	r_o	66.25 mm
J_c	0.08 kg·m^2	r_i	50.12 mm
μ_s	0.148	h_d	1.5 mm
μ_c	0.139	μ_f	7.2·10^{-3} kg/m·s
b_f	56 N·s/rad	ρ	900 kg/m^3
b_r	55 N·s/rad	N	20
k_f	9010 N/rad	C	1925
k_r	9020 N/rad	i_r	1
ω_s	21 rad/s	k_i	8204

3.1. Pressure-Torque Test

Figure 2 shows the simulation results and the experimental data from the PT test. The p_{cmd} is pressure command and the T_{exp} is torque of experimental data, and the T_{sim} is torque of simulated data. The simulated torque at low pressure was almost identical with the experimental result. However, some differences occurred at transient response of high pressure. This is due to the upper limit of the attractor gain C, which can lead system instability if a value above a certain upper limit is used. Also, error still existed after pressure input reached steady-state especially in 5 ~ 10 bar. The potential causes for this error are the temperature-related variation in the clutch friction coefficient and the invalid assumption of a uniform pressure distribution; the latter may trigger more. The simulation results were acceptable, considering these possible causes. Also, as the pressure increased, clutch torque also increased. However, after the certain point, it started to converge to the limit in both the experiment and simulation, due to the pressure drop at the relief value.

3.2. Hysteresis Test

Figures 3 (a) and (b) show the simulation results with the experimental data under the full-load conditions at clutch temperatures of 25 °C and 50 °C, respectively. Each graph shows the pressure versus the transfer torque. In this experiment, hysteresis appeared at low pressure region due to the dead zone of initial pressure engagement process, and then it disappeared in high pressure region. Also, its tendency was different depending on the clutch oil temperature. Table 2 shows the root mean square (RMS) error of the transfer case module test. Through the HS test, it was confirmed that the developed transfer case model could simulate the hysteresis characteristics similar to the actual phenomenon. The RMS error of HS test was lower than that of PT test because PT test includes extremely rapid transient pressure command.

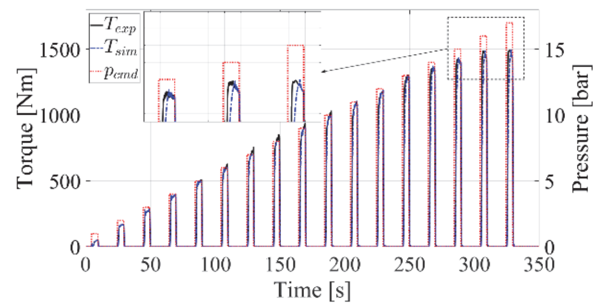
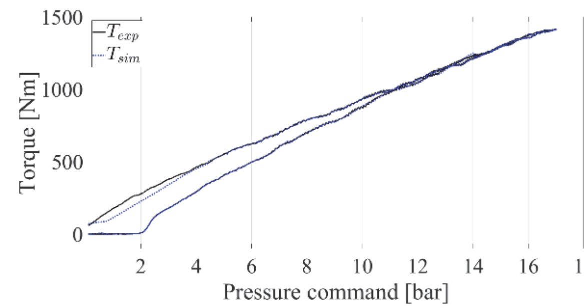
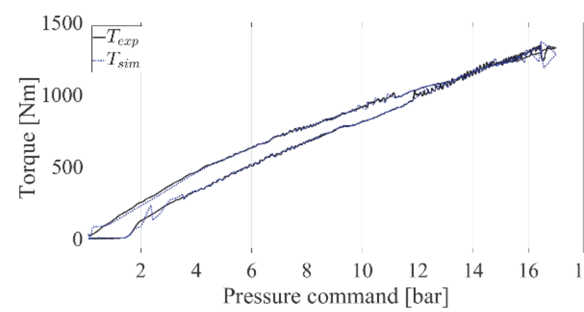


Figure 2. Pressure-torque test.



(a)



(b)

Figure 3. Hystresis test at: (a) 25 °C clutch oil temperature; (b) 50 °C clutch oil temperature.

Table 2. RMS error of transfer case module test.

PT test	HS test1(25 °C)	HS test 2(50 °C)
58.0 N·m	15.8 N·m	18.5 N·m

4. AWD VEHICLE SIMULATION

4.1. Transmission Output Torque Smoothing

CarSim has been a proven tool used for vehicle dynamic simulation results in many other studies, but its engine

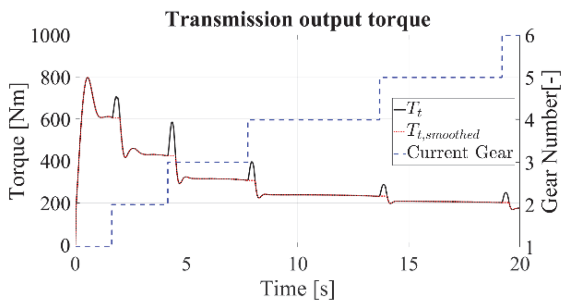


Figure 4. Effectiveness of torque smoothing during the gear shifting process.

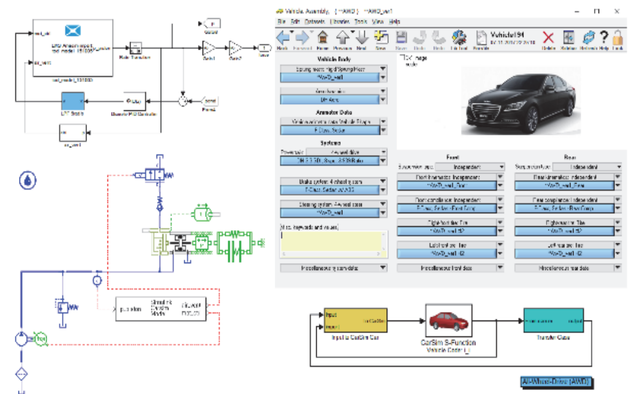


Figure 5. Model integration for AWD vehicle simulation.

model has limitations in describing the torque response during gear shifting. In the transient section, the crankshaft's rotational inertial torque is reflected in the engine output torque, resulting in a torque spike phenomenon in CarSim powertrain model. However, commercial vehicles use a transmission control unit (TCU) control command that reduces the fuel rate to provide a smooth torque response during the gear change process, which is missed in CarSim. Since the AWD system distributes the torque generated by the engine to both drive shafts, it is necessary to ensure the accuracy of the powertrain model's transmission output torque value. In CarSim, the automatic transmission's gear shifting occurs in 0.25 s, and the torque spike phenomenon occurs during this time. Here, the torque transmitted to transmission output shaft was processed to maintain the constant value during the gear-shift interval, by using a data hold block in Simulink, which helped removing the torque spike of CarSim transmission model in gear shifting.

Figure 4 shows the simulation results of torque smoothing algorithm for μ = 0.9. It can be seen that the torque spike was removed, as desired. To avoid instability, this torque smoothing was applied when the drive gear was higher than third in low-μ surface conditions.

4.2. Simulation Results

To validate the developed transfer case model for AWD vehicle simulation, CarSim, a vehicle simulation software, was used in this study for mutual model integration.

Figure 5 shows a representative block for each model that was used for the integration of the AWD system. To verify whether the suggested method could handle various driving scenarios with high reliability, several tests were conducted. A front ship, rear-wheel drive (RWD) drive train was simulated in this study. All simulation verifications were conducted at a 1 ms sampling time.

4.2.1. Slipping to sticking state transition

First, a simulation was conducted to verify the validity of the GMS model in the transient section. This verification method was used to investigate the time of the state transition from a slipping to a sticking state, according to the change in the clutch engagement force. In a situation such as acceleration in a low gear, when a large driving torque is generated, if the clutch engagement force is not strong, then most of the driving torque will be transferred to the main drive shaft, and only a portion of the driving torque will be transferred to the sub-drive shaft (slipping state). For the clutch to be fully engaged, the output torque of the transmission must be reduced, or the clutch engagement force must be strengthened. While keeping the clutch engagement force constant, a state transition point followed by a transmission torque reduction was verified in this simulation. Figure 6 shows the simulation results when the vehicle was accelerating on a high-μ surface from the stopped state with a constant throttle pedal position of 0.35. The black solid line is the transmission output torque, the red dashed line is the clutch torque, and the green dotted line is the vehicle longitudinal velocity. The blue dash-dot line just represents the timing of the state transition without unit in y-axis. The clutch engagement force occurred at 1 s. It was found that the stronger the engagement force, the stronger the clutch torque in the slipping state, and the faster the state transition occurred. That is, it was possible to transfer part of the driving torque to the rear wheels, with a certain value, without engaging the clutch completely. The longitudinal velocities, as related to the engagement force, were almost same, which means that tire grip was sufficient in the high-μ surface. Furthermore, an undesirable discontinuity or instability of the system followed by a variation of the clutch engagement force and state transition were not observed. Figure 7 shows the simulation results for a low-μ surface. All other conditions were the same as for the high-μ surface. In this case, the transmission output torque was lower than in the high-μ surface. The state transitions were also different, depending on the engagement force. Additionally, the longitudinal velocities were slightly different depending on the magnitude of the engagement force.

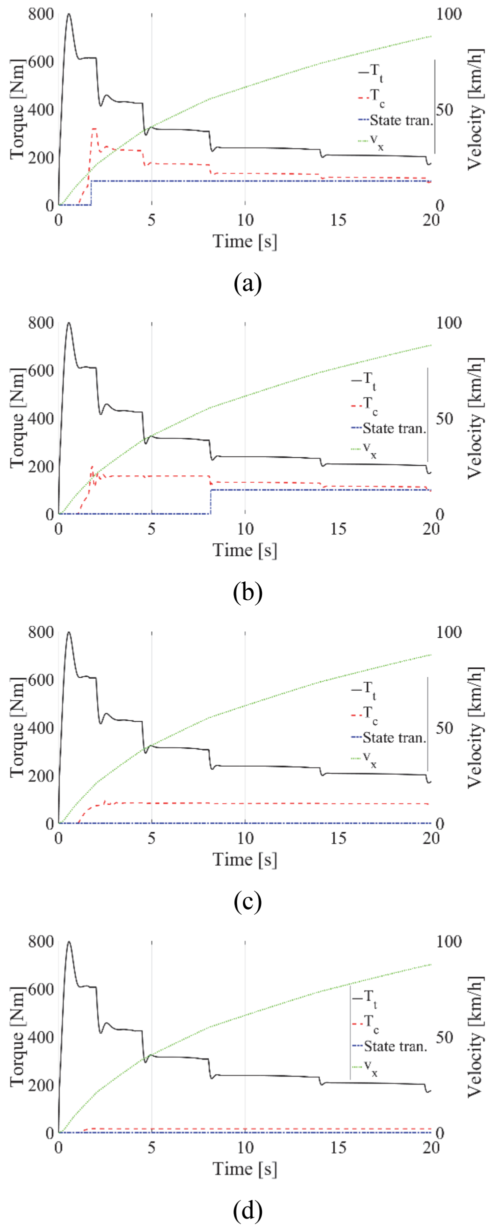


Figure 6. AWD system simulation results with varying engagement force ($\mu = 0.9$): (a) $F_c = 3000$ N; (b) $F_c = 1000$ N; (c) $F_c = 500$ N; (d) $F_c = 100$ N.

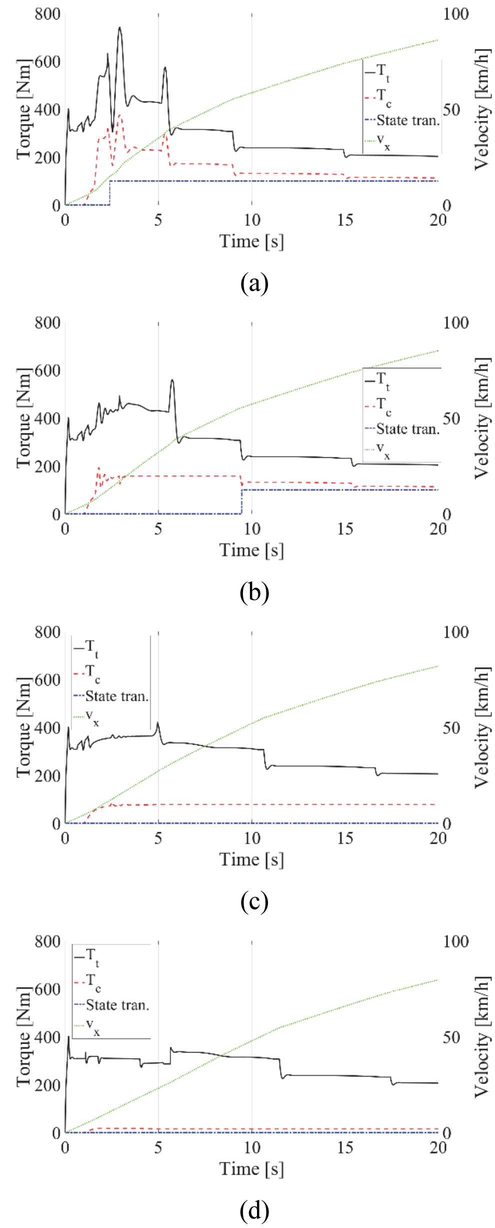
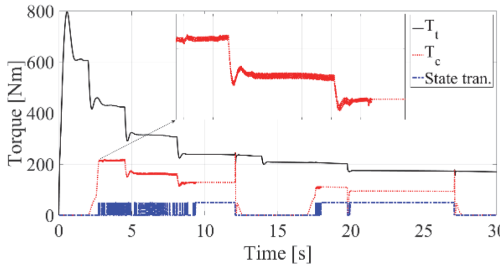


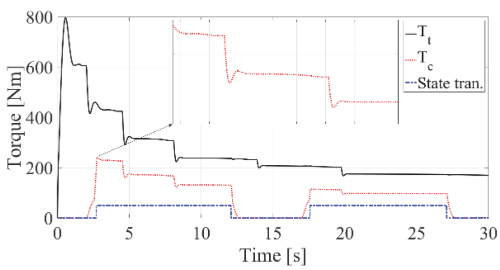
Figure 7. AWD system simulation results with varying engagement force ($\mu = 0.3$): (a) $F_c = 3000$ N; (b) $F_c = 1000$ N; (c) $F_c = 500$ N; (d) $F_c = 100$ N.

4.2.2. Clutch engagement and disengagement

To verify applicability of the suggested method to the AWD controller development, an arbitrary 2WD/AWD transition simulation that included repetitive clutch engagements and disengagements was conducted. Figure 8 shows the simulation results in which the clutch was engaged at 2 s and 17 s, and disengaged at 12 s and 27 s. Unlike the previous situation, the state transition of sticking to slipping was included. Here, the vehicle started to accelerate from a stopped state with a constant throttle position of 0.35 on a high-μ surface. The engagement force for both sections was set to 3000 N. Figure 8 (a) shows the simulation result using original GMS model. Here, chattering response occurred in the clutch torque when clutch was locked-up (t = 2 to 10 s). Although response of clutch torque was stabilized in second engagement, there was another undesirable response of undershoot (t = 20 s) and overshoot (t = 27 s) in clutch torque. Figure 8 (b) shows the simulation result using suggested model. Compared with the original GMS model, the stability of the transfer case model during the clutch

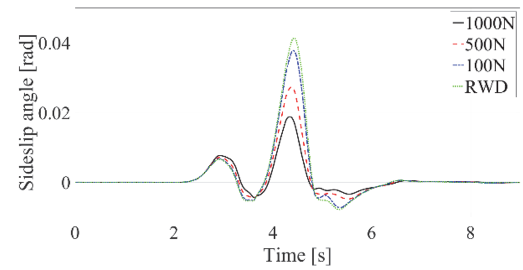


(a)

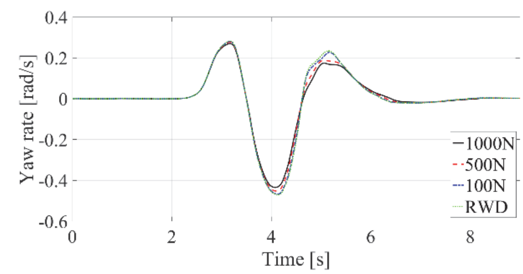


(b)

Figure 8. AWD system simulation with arbitrary clutch engagement and disengagement: (a) Original GMS model; (b) Suggested model.



(a)



(b)

Figure 9. AWD vehicle simulation during a double lane change: (a) Sideslip angle; (b) Yaw rate.

actuations was verified through the clear response in clutch torque through the entire region.

4.2.3. Clutch engagement with lateral motion

In the case of an RWD vehicle, there is a possibility that a sudden steering input accompanied by a rapid throttle input could cause oversteer. An AWD system can prevent this oversteer response through tire grip redistribution. Double lane change is a general verification method used to observe the lateral behavior of a vehicle in transient situations. Here, it was used to assess the difference in the lateral stability of 2WD and AWD vehicles with different clutch engagements while the vehicle was performing a double lane change with a stepped throttle input at an initial speed of 60 km/h on a road with a friction coefficient of 0.9, which means the engine output torque value of the vehicle was kept constant regardless of the AWD system operation. Figure 9 shows the change in the yaw rate and sideslip angle during the double lane change. The AWD vehicle demonstrated stable vehicle behavior, with a low yaw rate and low sideslip angle, compared to the RWD vehicle. More specifically, the peak values of the yaw rate and sideslip angle were reduced with increasing engagement differential, it cannot compensate for the differences in force. Since the proposed system does not include a center rotational speed between the front wheels and the rear wheels that inevitably occur during steady-state cornering. This constrains its maneuverability. Therefore, an AWD vehicle tends to behave more stably.

4.2.4. Shaft torque response in horizontal split-μ

To ensure a changeable torque distribution in the lock-up state followed by the variation of shaft external load, an AWD vehicle simulation on a horizontal split-μ was conducted. Here, vehicle starts to accelerate with a constant throttle pedal position of 0.35 while transfer case clutch is fully locked-up (F_c = 10000 N). Figure 10 shows the simulation result. Black solid line is the transmission output torque; red dotted line is the front shaft torque using suggested model; blue dash-dot line is the front shaft torque using original GMS model, and red dashed line is the one using CarSim internal model. CarSim internal model represents the fully locked-up transfer case. Therefore, simulation result of the front shaft torque should be identical with the result of CarSim internal model. Although the original GMS model showed an undesirable torque response at the end point of the horizontal split-μ, the suggested model represented torque response that better matches well with CarSim internal model through the entire region.

4.2.5. Simulation validation through experimental data

The final validation was comparing the simulation to an experiment. In the experiment, a production full-size sedan with a transfer case was used. Longitudinal acceleration from the stopped state with a constant throttle position was conducted. Then, the transmission output torque, transfer case engagement torque, and longitudinal velocity were compared to the simulation values. The experiment was

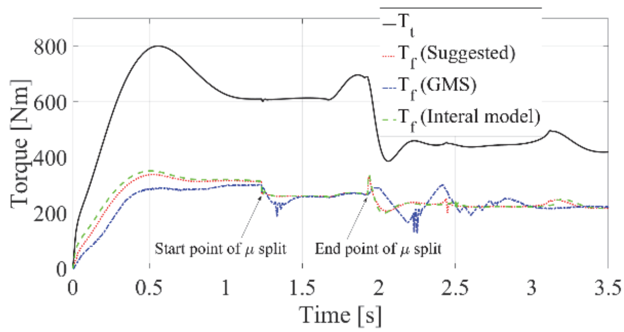
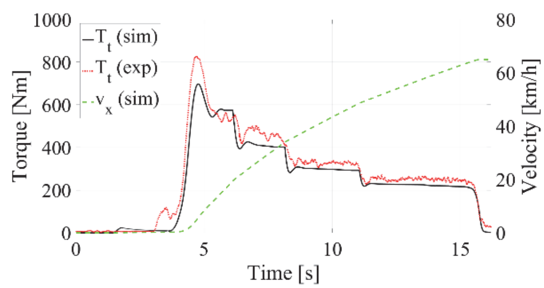
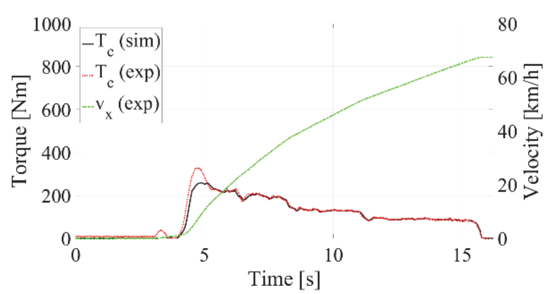


Figure 10. AWD vehicle simulation on a horizontal split-μ (transition from 0.3 to 0.9).



(a)



(b)

Figure 11. Verification of AWD vehicle simulation model in dry asphalt (μ = 0.95): (a) Transmission output torque; (b) Front shaft torque.

conducted on high-μ surface using data acquisition devices that saves the data of controller area network (CAN) signals and additional measurements. The throttle position, steering angle, and clutch engagement force data from the experiment were used as simulation inputs. In the simulation, all vehicle-related parameters, including the engine map, vehicle mass, wheel radius, and front and rear weight shifting ratio, among others, were configured to be identical to real experimental vehicle. Figures 11 (a) and (b) show the verification results of transmission output torque and front shaft torque, respectively. Black solid line is the torque response obtained from simulation and red dotted line is the torque response obtained from experiment. The initial peak transmission output torque was lower in the simulation. This error was due to the inaccuracy of engine map, especially at low RPM. At the initial region of engagement, clutch torque in the simulation was also slightly lower than that in the experiment, because the sticking condition of Equation (15) constrained the clutch torque. Except for initial region, the transmission output torque and clutch torque were almost same in both cases. Longitudinal velocity response in simulation was also quite similar with experimental results in both cases. These simulation results verify that the proposed transfer case model can sufficiently represent a real AWD vehicle.

5. CONCLUSION

In this study, a novel method of modeling and simulating a torque-on-demand AWD vehicle was suggested. As compared to the previously reported methods, the proposed one is more promising because of the following feature: by modifying the slip regulation method of the GMS model as the DOF reduction method when clutch is in the lock-up state, a stable and precise modeling method of transfer case for AWD vehicle simulation was developed. In contrast to the original GMS model that showed unstable response of clutch torque in AWD vehicle simulation, the suggested model did not cause not only undesirable overshoot and undershoot of clutch torque but also chattering responses. Moreover, the suggested method certainly showed simulation result of accurate and robust torque response even though the external factor such as horizontal split-μ changed. Two major findings of this study that simulation result of the longitudinal acceleration agreed well with the experimental data and that simulation result can deal with variation of vehicle motion states prove that the suggested model can replace an AWD vehicle driving experiment. The inclusion of additional models, e.g., a thermal model that is a principal factor in the wet clutch friction coefficient, is recommended for future studies. It is expected that the proposed simulation model can be applied to the controller development for torque-on-demand AWD vehicles, as well as to the model integration of vehicle chassis control systems.

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