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Modeling and parameter estimation of automatic transmission for heavy-duty vehicle using dual clutch scheme

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Abstract

This paper focuses on modeling of the heavy-duty vehicle drivetrain with automatic transmission by using dual clutch scheme. The planetary gear set in the automatic transmission is complicated structure and difficult to understand. The advantage of the dual clutch scheme is that it can be used to represent the complex planetary gear set intuitively, which is a great help to understand the gear shifting process. It is also suitable for being used in the controller due to its low order. Some conditions are required to convert the planetary gear set to the dual clutch model. The heavy-duty vehicle driveline can be converted to the dual clutch model due to its heavy engine and vehicle inertia.

This paper also proposes system prameter estimation methods to represent the driveline model. The main parameters are lumped inertia, lumped gear efficiency, output shaft compliance and friction coefficient of clutches. First, a method for estimating lumped inertia and lumped gear efficiency is proposed using WLSE(Weighted Least Sqaure Estimation) when gear is engaged. Second, resonance frequency of the system is obtained from the lock-up oscillation data occurring at the end of the gear shifting. The output shaft compliance is calculated by analyzing resonance frequency of the system. Third, the slip and friction coefficients of the clutch over time are calculated in the inertia phase. Using those data, the relationship between the dynamic friction coefficients and slip of the wet clutch can be obtained.

Finally, a simulation is constructed to verify the accuracy of the proposed dual clutch model and the estimated system parameters. Simulation result is compared with experimental data.

Introduction

Transmission is a mechanical device that transmits engine power to vehicle wheels. There are several types of transmission. Among them, automatic transmission is widely used in the vehicle industry due to its smooth gear shift performance. The automatic transmission is also used necessarily in heavy-duty vehicle, because torque converter amplifies torque in launch situation. But the automatic transmission has a disadvantage. The torque converter transmits power through fluid, large amount of energy is lost. Therefore, fuel efficiency of the automatic transmission is inevitably low. In the heavy-duty vehicle that uses a lot of energy, its disadvantage is even worse. To overcome low fuel efficiency, a lock-up clutch in torque converter is engaged from the low gear.

In the past decades, many researches have been done to improve gear shift control performance [1–4]. However, the first thing to do before using various control techniques is to model the driveline system. Through system modeling, the characteristics of the system can be understood and it is possible to use appropriate model-based control techniques in each situation. In addition, computer simulations can be conducted using system model. Simulation can be used to verify feasibility before applying the control techniques to a real system. It can save a lot of time and money.

Many researches about driveline modeling equipped with automatic transmission were conducted. In [5, 6], the torque converter is modeled using a steady state map and planetary gear set is modeled using its dynamics. In [7], planetary gear set is modeled using its kinematics. However the planetary gear set is a complicated structure. If planetary gear set is modeled as it is, it is difficult to understand the gear shifting process intuitively. However the planetary gear set can be converted into a dual clutch model, if a certain conditions are satisfied. The converted dual clutch model makes it easier to understand the gear shift characteristics.

This paper focuses on the gear shift modeling from 2nd to 3rd gear in heavy-duty vehicle. The lock-up clutch in the torque converter is engaged in the 1st gear to improve fuel efficiency. Therefore, the lock-up clutch is engaged during from 2nd to 3rd gear shift. The lock-up clutch model is used instead of the torque converter model. Also, the planetary gear set is modeled using dual clutch scheme. In addition, it proposes how to estimate the parameters like inertia, output shaft compliance, and clutch friction coefficients using dynamometer experimental data. Finally, the simulation using the dual clutch modeling scheme and estimated parameters is conducted. The simulation results are compared with experimental data to verify that the modeling describes the real system well.

Using the above method, it is possible to model other gear states. The typical 2-3 gear situation is modeled as an example. In addition, the parameter estimation algorithm is applicable to MT and DCT.

Driveline modeling

Figure 1 is a diagram showing the driveline modeling with automatic transmission. The torque converter is modeled using the steady state map. The planetary gear set is modeled using torque dynamics and speed kinematics of the planetary gear. Also, engine, impeller, turbine, vehicle, and each part of planetary gear set are described using lumped inertia. Also, the output shaft compliance model is used. In this paper, since the lock-up clutch is engaged during 2nd to 3rd gear shift, the lock-up clutch model is used instead of the torque converter map.

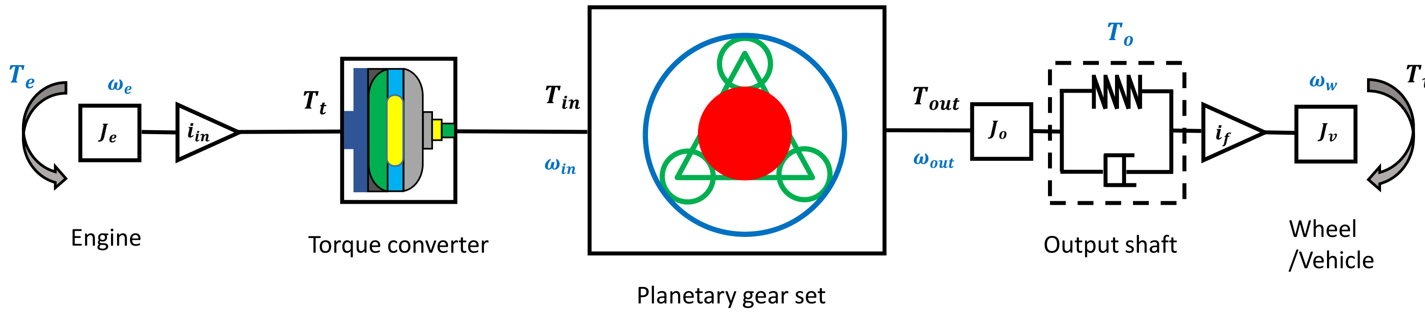

Figure 1: Driveline modeling diagram equipped with automatic transmission

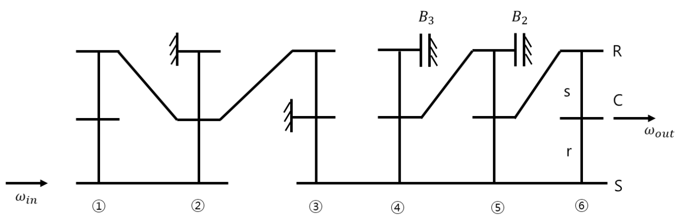

Figure 3: Detailed structure of planetary gear set

Figure 3 is detailed structure of planetary gear set. It consists of a combination of six planetary gears. A brake 2 is engaged in 2nd gear. A brake 3 is engaged in 3rd gear. The gear upshift occurs through a series of processes in which brake 2 is disengaged and brake 3 is engaged. This process is similar to gear shifting in DCT. But the dynamics of the planetary gear set and the dual clutches are different. In [5, 8], the planetary gear set in figure 3 can be represented like the DCT if the following conditions are satisfied.

$$\begin{aligned} &I_{R5} + I_{C4} = 0 \\ &I_{R4} = 0 \end{aligned} \tag{1}$$

Where S is the sun gear, subscript R is the ring gear, C is the carrier. Heavy-duty vehicle has quite heavy vehicle mass and engine inertia. The inertia of ring gear and carrier is very small value compared to the vehicle and engine inertia. Therefore it can be met above condition. Figure 2 is the dual clutch modeling scheme representing automatic transmission. The brake 2 and brake 3 of Figure 3 are expressed as the clutch 2 and clutch 3 in Figure 2.

In Figure 2, the gear connecting the engine and transmission input is expressed as an input gear. In addition, the gear ratio caused by the planetary gear set between the input gear and brake 2 is expressed by i_{c2}. The gear ratio caused by the planetary gear set between the brake 2 and transmission output is expressed by i_{out2}. Similarly, the gear ratio connected to brake 3 is represented by i_{c3} and i_{out3} respectively. The dynamics of the engine and dual clutch in Figure 2 are expressed as follows.

$$T_e = f\left(\phi, \omega_e\right) \tag{2}$$

$$\begin{aligned} T_e\eta - \frac{T_{c2}}{i_{in}i_{c2}} - \frac{T_{c3}}{i_{in}i_{c3}} = \\ \left(\left(J_e + \frac{J_i + J_t}{i_{in}^2}\right)\eta + \left(\frac{J_{c2}}{i_{c2}^2 i_{in}^2} + \frac{J_{c3}}{i_{c3}^2 i_{in}^2}\right)\right)\dot{\omega}_e \end{aligned} \tag{3}$$

Where J is the inertia, ω is the rotational speed, T is the torque, i is the gear ratio, ϕ is the engine throttle position, η is the gear efficiency. Subscript e is the engine, i is the impeller, t is the turbine, in is the input gear, $c2$ is the clutch 2, and $c3$ is the clutch 3. Function f is represented the steady state engine map. The 3-dimensional steady state map of the throttle position and the engine speed is used. Also, all gears lose energy when transmitting the torque. Although the loss rate is not large in each gear, the amount of total energy loss increases when multiple gears are gathered. Many gears are in the planetary gear set. Therefore, the total energy loss efficiency in planetary gear set is expressed as η. To simplify model, it is assumed that lumped energy loss is occurred in i_{c2} and i_{c3}.

The power transmitted through the clutch is expressed according to the state of the each clutch. The torque transmitted through the clutch 2 is expressed as follows.

$$\begin{aligned} T_{c2} = i_{in}i_{c2}\left(T_e\eta - \frac{T_{c3}}{i_{in}i_{c3}}\right) - i_{in}i_{c2} \\ \left(\left(J_e + \frac{J_i + J_t}{i_{in}^2}\right)\eta + \left(\frac{J_{c2}}{i_{c2}^2 i_{in}^2} + \frac{J_{c3}}{i_{c3}^2 i_{in}^2}\right)\right)\dot{\omega}_e \end{aligned} \tag{4}$$

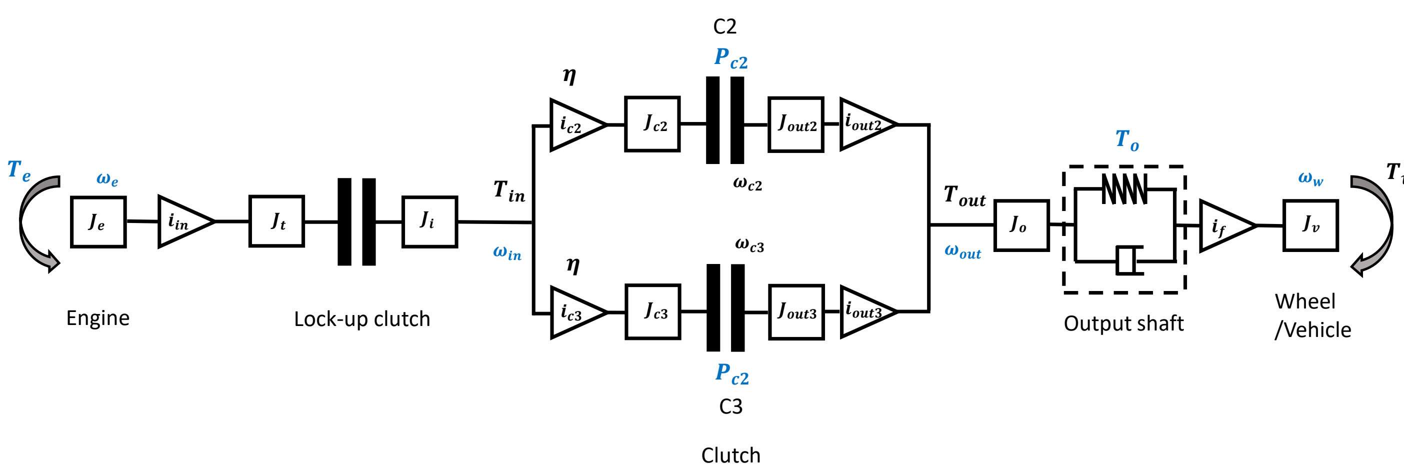

Figure 2: Equivalent dual clutch model

$$T_{c2} = \mu_{k2} N_2 D_2 A_2 r_2 \left(P_{c2} - P_0\right) \operatorname{sgn}\left(\omega_e / i_{in} i_{c2} - \omega_{c2}\right) \quad (5)$$

$$T_{c2} = 0 \quad (6)$$

Where μ_k is the friction coefficient, N is the number of clutch plates, D is the de-rating factor, A is the area of the clutch, r is the effective radius, and P is the pressure applied to the clutch. P_0 is the pressure that filling phase of the clutch ends. Equation (4) represents transmitting torque when the clutch is engaged. The clutch is slipping in (5) and disengaged in (6).

Similarly, the torque balance equation of the output shaft and the vehicle body is expressed as follows.

$$T_{c2} i_{out2} + T_{c3} i_{out3} - T_o = \left(J_{out2} i_{out2}^2 + J_{out3} i_{out3}^2 + J_o\right) \dot{\omega}_{out} \quad (7)$$

$$\dot{T}_o = c_o \left(\dot{\omega}_{out} - \dot{\omega}_w i_f\right) + k_o \left(\omega_{out} - \omega_w i_f\right) \quad (8)$$

$$T_o i_f - T_v = J_v \dot{\omega}_w \quad (9)$$

Where subscript $out2, 3$ is the gear after clutch 2, 3, o is the output shaft, f is the final gear, w is the wheel, and v is the vehicle. In (8), the output torque is expressed as 2nd order shaft compliance model. In (9), the external resistance torque contains grade, aerodynamics, and rolling resistance. During experiment, external resistance torque remains constant value. In addition, it is assumed that there is no energy loss, since there are not many gears in this part.

Parameter identification

The purpose of this chapter is to create the simulation that describes the experiment. Putting the same inputs(engine torque and brake 2, 3 pressure) with experiment in the simulation, the output results(output torque, transmission input speed, and transmission output speed) of simulation should be same as the experiment result. To improve the simulation results, parameter estimation algorithm is proposed using input and output data of experiment.

Experimental data

The gear shift experiment was carried out on the dynamometer by using gear shift algorithm embedded in the production TCU. The gear shift experiment was carried out under the condition of 60% throttle. Inputs of the system are the engine torque and pressure of brake 2, 3. The engine torque value during the experiment can be obtained from the TCU. The pressure of brake 2, 3 is measured by a pressure sensor. In addition, the output shaft torque is measured by a torque sensor. In short, the measurable data are expressed to blue letters in Figure 2.

Figure 4 is described as follows for each section. The first section corresponds to 2nd gear. Brake 2 is engaged and brake 3 is disengaged. The second section corresponds to the torque phase. In this section, brake 2 is engaged and brake 3 is slipping. The characteristic in the torque phase is that torque dip occurs in the output torque. The third section corresponds to the inertia phase. Brake 2 is disengaged and brake 3 is slipping. This section is a process in which the engine speed is synchronized with the speed of the 3rd gear. In inertia phase in Figure 4, vibration of the hydraulic actuator pressure caused vibration of the output torque. This characteristic is usually not seen in the gear shifting process. The fourth section corresponds to 3rd gear. In this section, brake 2 is disengaged and brake 3 is engaged. Characteristic of this section is oscillation of the output torque. It is called lock-up oscillation.

Inertia and lumped efficiency and external resistance estimation using WLSE

Figure 5 shows the simplified driveline model when driving in 2nd gear. The entire driveline is divided into three lumped inertia. J_1

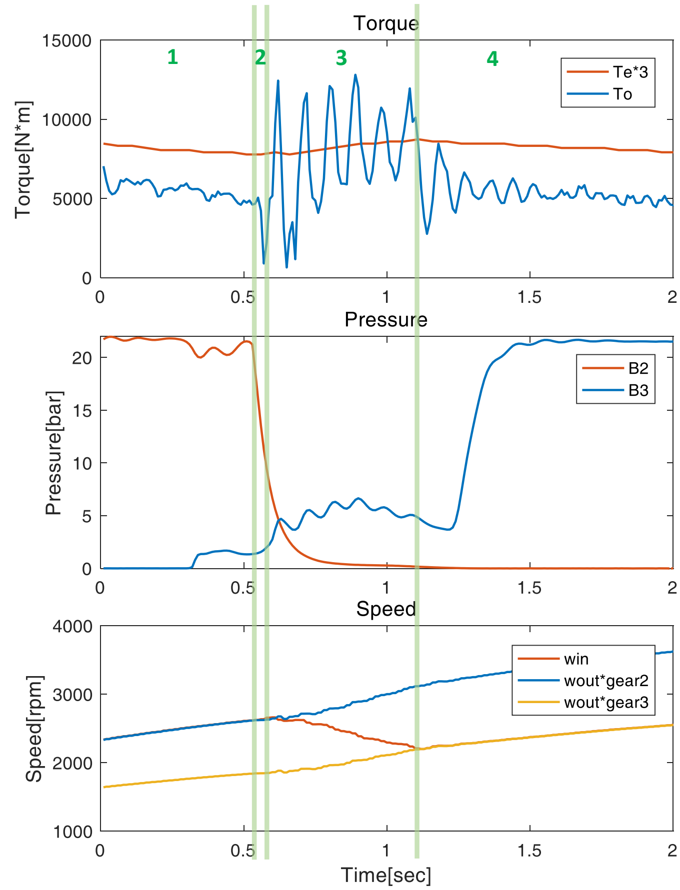

Figure 4: Gear shift experimental data from dynamometer

represents lumped inertia of all parts before the input gear. J_2 represents lumped inertia of all parts from input gear to output shaft. J_3 represents lumped inertia of all parts behind the output shaft. As mentioned above, it is assumed that the energy loss of the planetary gear set is lumped in the i_{c2} and i_{c3}. It is expressed as η. The blue letters in the Figure 5 are the values that can be obtained from the sensor. The red letters in the figure 5 are the unknown value. The simplified torque balance equation is as follows.

$$T_e \eta - \frac{T_o}{i_{in} i_{c2} i_{out2}} = \left(J_1 \eta + \frac{J_2}{i_{in}^2 i_{c2}^2 i_{out2}^2}\right) \dot{\omega}_e = J' \dot{\omega}_e \quad (10)$$

$$T_o i_f - T_v = J_3 \dot{\omega}_w \quad (11)$$

In (10), the η and J' are the values to be estimated using the data. In (11), J_3 and T_v are also the values to be estimated.

If the parameters are estimated by applying the LSE(Least Square Estimation) method to (10) and (11), the following problems occur. Differential values of rotational speed are used for parameter estimation. Due to the sensor noise and sampling rate, the accuracy of the angular acceleration value is low. This causes a problem that the accuracy of the parameters estimated using LSE becomes low.

In order to prevent the above problem, WLSE(Weighted Least Square Estimation) method is proposed in which the weight is assigned differently according to the accuracy of data. Assigning the weight of data is as follows. The exact value of inertia is unknown, but the range of values can be known from the system design specification. Calculating data using (10) and (11), the data with the inertia value within the specified range are set to high weight. Other data is set to low weight. If $5 \leq J' \leq 40$, the weight of data is 1 and other data is 0.01. If $5000 \leq J_3 \leq 9000$, the weight of data is 1 and other data is 0.01. Since the η and T_v values are firstly required to assign the

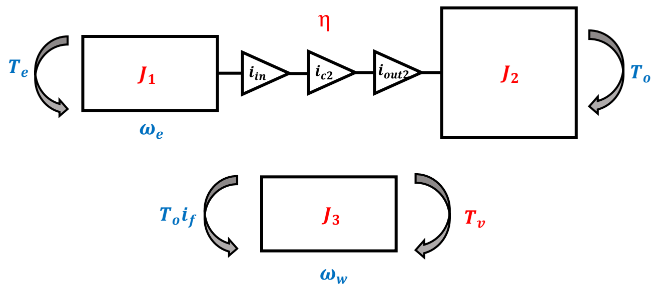

Figure 5: Simplified driveline model with lumped inertia

weight of data, two parameters in (10), (11) cannot be estimated simultaneously, but it must be estimated sequentially.

The algorithm for estimating η and J' is as follows. First, η is changed from 0.7 to 1 in 0.05 units. The left-hand side value of (10) is calculated using each η and measured data. After dividing the left-hand side value by $\dot{\omega}_e$, it is compared with $5 \leq J' \leq 40$. Then, the weight of the data is set according to the comparison result. Second, J' is estimated using LSE with weighted data for each η. Finally, RMS error is calculated using estimated J' for each η. The η with the lowest RMS error and the corresponding J' value are selected as the final estimated parameters.

Similarly, the algorithm for estimating T_v and J_3 is as follows. First, T_v/i_f is changed from 0 to 4000 in 500 units. The weight of data according to each T_v is set using (11) in the same way as the above algorithm. Second, after obtaining J_3 value for each T_v using WLSE, T_v and J_3 with the lowest RMS error is adopted. Finally, the value of the corresponding T_v/i_f is changed by 100 units. The T_v and J_3 values obtained by repeating the above process are the final estimated parameters.

At first table in Table 1, the RMS error is the smallest when $\eta = 0.85$ and $J' = 17.5$. At first graph in Figure 6, the blue data is a plot when $\eta = 1$. The green data is a plot when $\eta = 0.85$. Due to the change in the η value, the data is shifted to the left in the graph. Therefore, the data is gathered around the origin. For this reason, it is possible to estimate more accurate value of J'. T_v and J_3 can be estimated in the same way. The contents are summarized and attached to the second table in Table 1 and the second graph in Figure 6.

Table 1: Estimated inertia value and RMS error for each variables

Efficiency	**Inertia** $(kg \cdot m^2)$	**RMSE** (rad/s^2)
1.00	23.2	523.2
0.95	20.8	401.0
0.90	18.5	306.6
0.85	17.5	273.9
0.80	16.1	310.3
0.75	14.4	399.5
0.70	12.6	515.7
T_v/i_f	**Inertia** $(kg \cdot m^2)$	**RMSE** (rad/s^2)
500	7133	2349
1000	6964	2029
1500	6627	1754
2000	6234	1563
2300	6056	1520
2500	5962	1525
3000	5701	1634
3500	5723	1925
4000	5489	2644

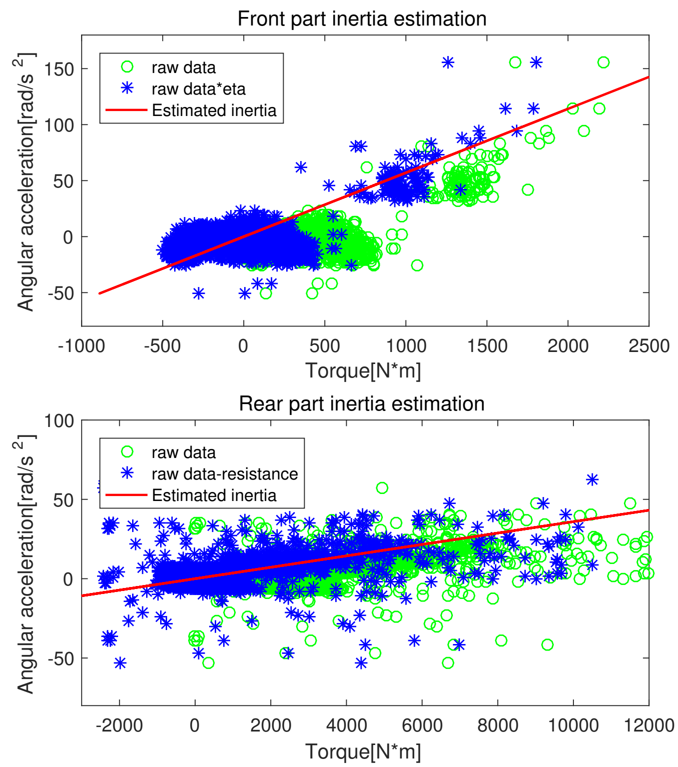

Figure 6: Experminetal data and estimated inertia using WLSE

Output shaft compliance and damping coefficient estimation

The resonance frequency of the driveline is determined by k and c values of the output shaft. The data where oscillation from resonance frequency appears is the lock-up oscillation part after gear shifting is finished. At the fourth section of output shaft torque graph in Figure 4, the resonance frequency is $12Hz$. Using this value, the output shaft compliance can be calculated inversely using model.

J_4 in Figure 7 is estimated using WLSE method and 3rd gear data. In addition, J_5 is the same value as J_3. The system with inertia attached on both sides of the spring and damper can be calculated the resonance frequency using below equations.

$$\frac{1}{J} = \frac{1}{J_4} + \frac{1}{J_5} \tag{12}$$

$$J\ddot{\theta} + c\dot{\theta} + k\theta = 0 \tag{13}$$

$$\omega_n = \sqrt{\frac{k}{J}}, \zeta = \frac{c}{2\sqrt{Jk}}, \omega_d = \omega_n\sqrt{1-\zeta^2} = 2\pi f \tag{14}$$

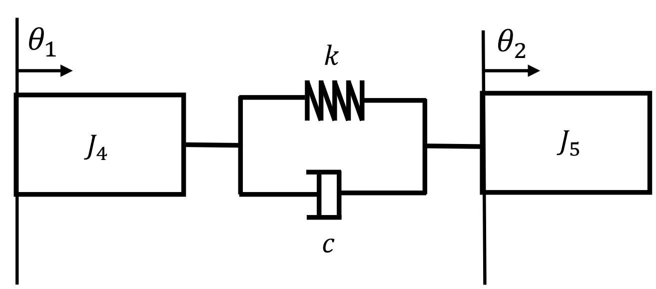

Figure 7: Simplified driveline model in 3rd gear

Where J is the equivalent inertia, ω_n is the natural frequency, ζ is the damping ratio, ω_d is the damped natural frequency. f is the resonance frequency of system. The equivalent inertia J can be calculated from (12). Governing equation of simplified model is (13). As seen in the Figure 4, resonance frequency of the system is $f = 12Hz$. In addition, the damping ratio can be calculated from decreasing amount of output torque oscillation. The output shaft compliance(k) and damping coefficient(c) is easily calculated using (14).

Clutch friction coefficient estimation

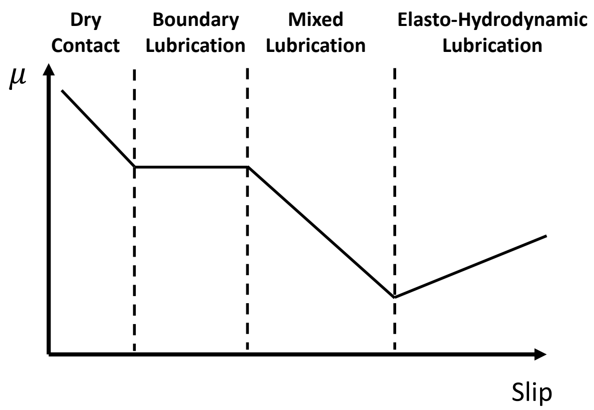

Figure 8: Stribeck effect

The friction coefficient of clutch affects the system at inertia phase in Figure 4. Inertia phase is the section where clutch 3 transmits torque by slipping. The transmitted torque follows equation (5). The clutch used in the automatic transmission is a wet type clutch, which is used to transmit large torque. To transmit large torque, there is lubricant in wet clutch. Due to lubricant, friction coefficient of wet clutch changes according to the slip. [9] explains that the friction coefficient of wet clutch changes according to the slip. This effect is called stribeck effect. Figure 8 illustrates stribeck effect well. It is expected that estimated friction coefficient of the clutch changes according to slip. To get μ-slip curve like Figure 8, the friction coefficient and slip value over time should be obtained from the experimental data. First, the clutch slip in the inertia phase is calculated by its kinematics. Kinematics is obtained from the dual clutch modeling scheme.

$$\omega_{c3,slip} = \frac{\omega_{in}}{i_{c3}} - i_{out3}\omega_{out} \tag{15}$$

Where ω_{in} is the transmission input speed and ω_{out} is the transmission output speed. The transmission input speed and transmission output speed data is measured. Therefore, the clutch 3 slip is obtained using (15). The clutch 3 slip is plotted in Figure 9.

The clutch 3 friction coefficient over time in the inertia phase is obtained by following method. In Figure 4, the output torque data in

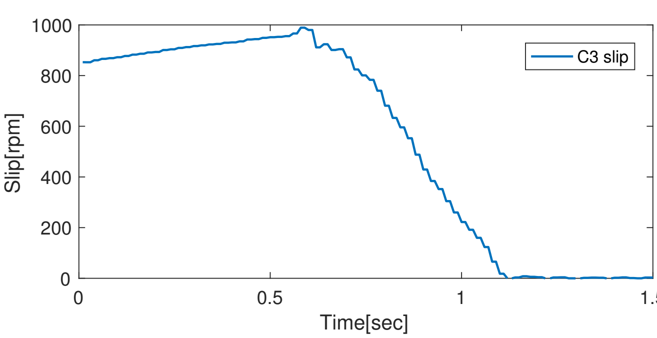

Figure 9: Clutch 3 slip plot during inertia phase

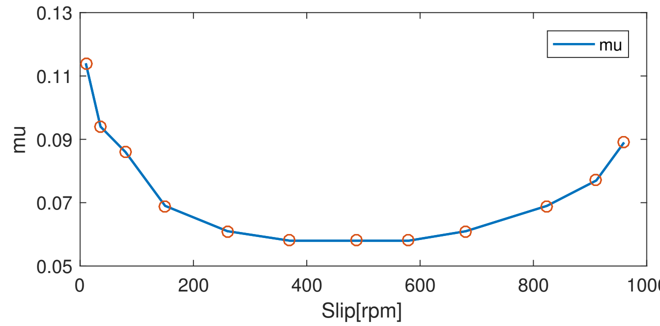

Figure 10: Clutch 3 μ-slip graph

the inertia phase has oscillating characteristic. The lowest and highest point of each oscillation are selected. In addition, the relationship between the output torque and the clutch 3 torque in the inertia phase can be described by (7). The differential value of encoder is inaccurate due to its low sampling rate. Therefore, the clutch 3 torque obtained by (7) is inaccurate. Clutch 3 torque is calculated using the following equation.

$$T_{c3} \approx \frac{T_o}{i_{out3}} \tag{16}$$

The clutch 3 friction coefficient is calculated by using (5) and (16). Combining the clutch 3 friction coefficient data with the slip in Figure 9, μ-slip graph of clutch 3 is obtained. The μ-slip graph in Figure 9 is obtained by smoothing the previous μ-slip graph.

Simulation result

This chapter examines whether the simulation composed of the estimated parameters describes the experiment well. When the inputs(P_{c2},P_{c3}, and T_e), which are the same as the experiments, are applied to the simulation, the outputs(ω_{in},ω_{out}, and T_o) from the simulation are compared with the experiment data. The simulation is constructed using Matlab's simdriveline.

In Figure 11, the above two graphs correspond to the system inputs. The bottom three graphs correspond to the system outputs. The blue line represents the experimental data and the red line represents the simulation data. Analyzing the simulation data, it has similar characteristics to the experiment. The gear shifting starts around 0.6 seconds. Then it goes through the torque phase and inertia phase. After then, the gear shifting is completed around 1.1 seconds. The characteristics of each phase and the role of the parameters required to describe them are as follows. The output torque dip occurring at the torque phase is well expressed in the 3rd graph. The slope of engine speed and the timing of synchronization at the inertia phase are well described in the 4th graph. This result means that inertia, lumped gear efficiency, and external resistance torque values obtained through WLSE are accurate. In addition, the oscillation of output torque is well described. This result means that μ-slip curve of the clutch 3 is well estimated. And the lock-up oscillation occurred after gear shifting is well expressed. It means that the calculated k and c values are accurate.

In summary, parameters estimated from experimental data were used to construct the simulation. The simulation that applied the same input as the experiment well described the experimental data. This indicated that the dual clutch modeling scheme could describe the driveline of the heavy-duty vehicle equipped with the automatic transmission. In addition, it was verified indirectly that the parameter values estimated using the proposed algorithm were accurate.

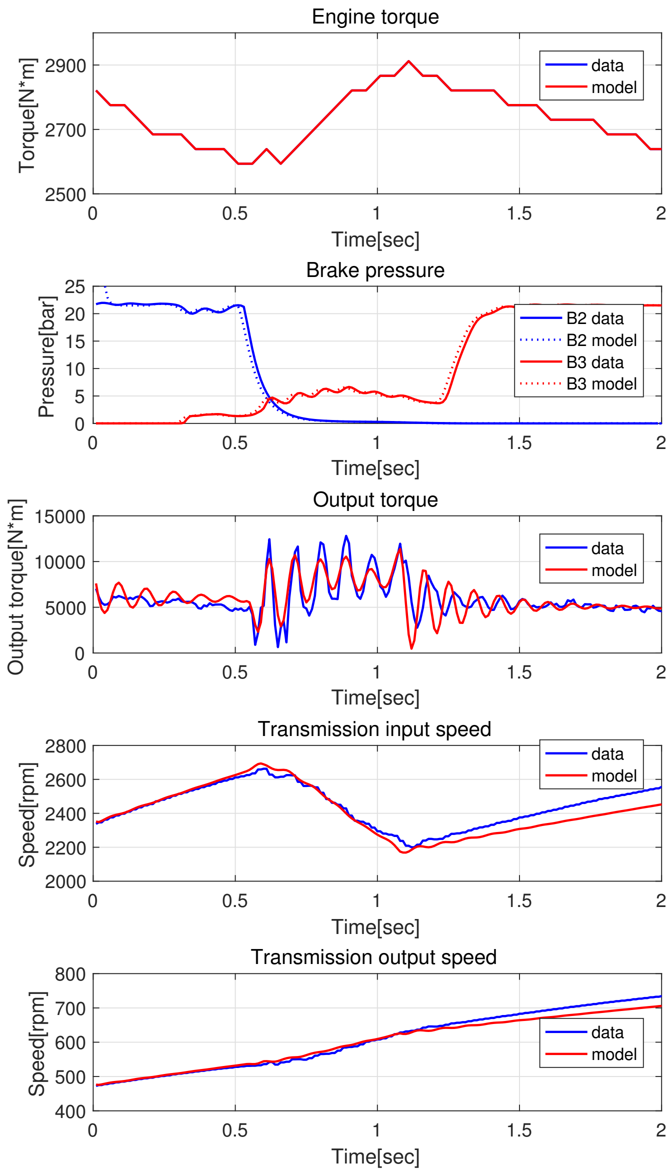

Figure 11: Comparison of simulation and experiment

Conclusion

In this paper, conditions and methods for modeling the driveline of the heavy-duty vehicle equipped with the automatic transmission with the dual clutch scheme are introduced. In order to construct the modeling, the system parameter values must be known. The algorithm which estimates the system parameters constituting the driveline model using experimental data is proposed. The WLSE is used to estimate lumped inertia, lumped gear efficiency, and the external resistance torque. The vibration characteristics of lock-up oscillation is analyzed to estimate the compliance and the damping coefficient of the output shaft. In addition, it is discussed that how to estimate the mu-slip curve of the wet clutch by analyzing the experimental data in the inertia phase. Finally, by constructing the dual clutch model simulation with the estimated parameters, it was proved that the proposed simulation model describes the experimental results well through comparison.

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