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OTIMIZATION METHOD OF REFERENCE SLIP SPEED IN CLUTCH SLIP ENGAGEMENT IN VEHICLE POWERTRAIN

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ABSTRACT−Vehicle environmental regulations are strengthening all over the world, and vehicles are getting more and more intelligent due to electronicization. Thus, there is an increasing interest in improving vehicle efficiency and ride comfort. Appropriate control of a clutch in the vehicle powertrain can play a major role in improving the vehicle efficiency ride comfort. Previously, many researches on tracking controllers of clutch slip speed and clutch transmitted torque in clutch slip engagement control have been conducted. However, little research on an optimization method of the reference clutch slip speed and clutch torque has been performed. Therefore, in this study, the optimization method of the reference clutch slip speed to improve the vehicle efficiency and ride comfort in clutch slip engagement in the vehicle powertrain is proposed. The proposed method is verified through clutch slip engagement simulations using AMESIM and MATLAB Simulink.

KEY WORDS : Clutch engagement, Slip engagement, Clutch control, Powertrain control, Reference slip speed, Reference clutch torque

NOMENCLATURE

T_e : engine torque, N·m
T_{e0} : engine torque at the initial time of the clutch slip engagement, N·m
T_{c1} : the first(odd gear) clutch torque, N·m
T_{c2} : the second(even gear) clutch torque, N·m
T_o : output shaft torque, N·m
T_r : road load torque, N·m
T_{r0} : road load torque at the initial time of the clutch slip engagement, N·m
ω_e : engine speed, rad/s
ω_{c1} : the first(odd gear) clutch speed, rad/s
ω_{c2} : the second(even gear) clutch speed, rad/s
ω_w : wheel shaft speed, rad/s
ω_{s1} : slip speed of the first(odd gear) clutch, rad/s
$\omega_{s1.0}$: slip speed of the first(odd gear) clutch at the initial time of the clutch slip engagement, rad/s
$\omega'_{s1.0}$: slip speed of the first(odd gear) clutch at the time when the optimization is conducted, rad/s
i_1 : gear ratio of the first(odd gear) clutch, -
i_2 : gear ratio of the second(even gear) clutch, -
i_f : final gear ratio, -
t : time, s
t_{end} : clutch slip end point, s
t_p : past time from the beginning of the clutch slip engagement, s
a : optimization variable, N·m
f : tuning parameter, N·m

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1. INTRODUCTION

Vehicle environmental regulations are strengthening all over the world, and vehicles are getting intelligent due to electronicization. As a result, there is an increasing interest in improving vehicle efficiency and ride comfort. Appropriate control of a clutch in the automotive powertrain can play a major role in improving the vehicle efficiency and ride comfort.

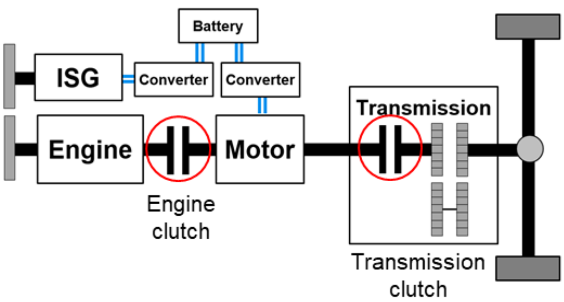

Figure 1. Schematic diagram of the powertrain of a parallelhybrid vehicle

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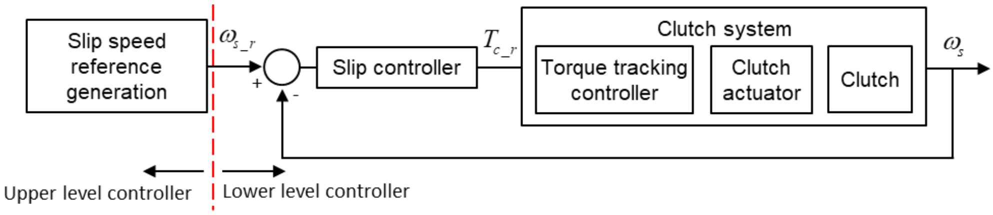

Figure 2. Structure of a clutch slip controller

Fig. 1 shows the schematic diagram of the powertrain of a parallel hybrid vehicle. As shown in Fig. 1, a parallel hybrid vehicle has two types of a clutch: an engine clutch located between the engine and driving motor, and a transmission clutch used for the transmission.

There are two methods to engage the clutch. The first is the synchronized engagement method. In this method, the speed of both sides of the clutch is synchronized and then the clutch is engaged. For the synchronization engagement method, energy loss, and clutch wear due to friction of the clutch do not occur during the clutch engagement. The second is the slip engagement method. In this method, the clutch is engaged when the speed of both sides of the clutch is different. For the slip engagement method, energy loss, and clutch wear due to friction of the clutch can occur during the clutch engagement. Also, the torque to synchronize the speed of both sides of the clutch is transmitted through the clutch, which causes vehicle jerk. Referring to these two methods, it is advisable to engage the clutch using the synchronized engagement method. However, there are some situations in which the slip engagement method should be needed.

For example, in the case of a parallel hybrid vehicle, when a parallel hybrid vehicle launches on a gradient road but the torque of the driving motor is insufficient to launch, there is not enough time to synchronize the speed of both sides of the clutch and thus the engine clutch should be engaged by the slip engagement method. Also, the transmission clutch is usually engaged by the slip engagement method unless the additional control is performed to synchronize the speed of the driving motor shaft with the transmission input shaft using the driving motor during gear shifting.

Therefore, when a clutch is utilized in the powertrain system, the slip engagement is required, and energy loss due to friction of the clutch, and vehicle jerk inevitably occur during the slip engagement. However, these two variables directly affect the vehicle efficiency and ride comfort.

The clutch friction energy loss and vehicle jerk in the clutch slip engagement are related with the clutch transmitted torque and clutch slip engagement time. The clutch torque is in turn determined by the engine torque, road load torque, and clutch engagement time. Therefore, the clutch friction energy loss and vehicle jerk are generally related with the engine torque, road load torque, clutch engagement time.

On the other hand, since the engine torque is controlled by a driver and the road load torque is determined by external environment, the controllable variable among the engine torque, road load torque, and clutch engagement time in the clutch slip engagement is only the clutch engagement time. Therefore, it is necessary to control the clutch friction energy loss and vehicle jerk appropriately by adjusting the clutch engagement time in the clutch slip engagement.

Generally, reducing the clutch engagement time in the clutch slip engagement reduces the clutch friction energy loss and increases the vehicle jerk. Conversely, increasing the clutch engagement time increases the clutch friction energy loss and reduces the vehicle jerk. Therefore, the clutch friction energy loss and vehicle jerk in the clutch slip engagement are in a trade-off relationship with respect to the clutch slip engagement time, and it is impossible to reduce both variables simultaneously. So, it is necessary to compromise the friction energy loss and vehicle jerk by adjusting the clutch slip engagement time appropriately according to the engine torque and road load torque in the clutch slip engagement, and the criterion to compromise the two variables should be needed.

Fig. 2 shows the structure of a clutch slip controller. Referring to Fig. 2, the structure of the clutch slip controller can be divided into a lower level controller and an upper level controller. The lower level controller represents the controller to track the given reference clutch slip speed or clutch torque. And, the upper level controller represents the controller that generates the reference signals of the clutch slip speed or clutch torque that the lower level controller should track.

Previously, many researches on the lower level controller have been performed. The lower level controller for the slip engagement of a dry clutch was addressed in the studies(Horn et al., 2003, Ni et al., 2009, Oh et al., 2016, Kim et al., 2018, Kim and Choi, 2018, Park et al., 2019a). Also, the lower level controller for the slip engagement of a wet clutch was addressed in the studies(Jeong and Lee, 2000a, Jeong and Lee, 2000b, Kong et al., 2016). However, little research on the upper level controller, that adjusts the clutch friction energy loss and vehicle jerk in the clutch slip engagement has been conducted. In the previous studies(Dassen and Serrarens, 2003, Gao et al., 2009, Gao et al., 2010, Lu et al., 2011, Tran et al., 2012), when controlling the clutch slip speed in the vehicle launch or shift condition, the reference slip speed was generated using a third order polynomial or an exponential function, and the clutch engagement time was fixed. In the previous study(Kim et al., 2017), an upper controller was proposed that generates reference clutch torque and engine torque using one tuning parameter. As a result, the clutch slip engagement time was adjusted by the generated reference clutch torque and engine torque, but the shape of the reference slip speed was not considered.

In this study, a method of determining the clutch slip engagement time and optimizing the reference clutch slip speed in the clutch slip engagement based on an optimization technique is proposed. In the proposed method, the weighted sum of the clutch friction energy loss and vehicle jerk is minimized according to the engine torque proportional to the vehicle acceleration desired by the driver and the road load torque that interferes with the vehicle acceleration.

Also, In the proposed method, since the engine torque and the road load torque are considered when determining the clutch slip engagement time, the influence of the driver's acceleration intention and the external environment can be naturally considered in the clutch slip engagement. And, the ratio between the clutch friction energy loss and vehicle jerk, and the clutch engagement feeling can be adjusted by varying a weighting parameter between the clutch friction energy loss and vehicle jerk in the weighted sum of the two variables.

The optimization method of the reference clutch slip speed proposed in this paper is applied to the clutch slip engagement in the launch situations of a vehicle equipped with a dual clutch transmission (DCT), and verified with simulations using AMESIM and MATLAB Simulink.

The remaining part of this paper is organized as follows. Section 2 describes the optimization method of the reference clutch slip speed. Section 3 addresses the simulation results of the proposed method. And, Section 4 concludes this paper.

2. OPTIMIZATION METHOD OF REFERENCE CLUTCH SLIP SPEED

As mentioned in the introduction section, referring to Fig. 2, the structure of the clutch slip controller can be divided into a lower level controller and an upper level controller. In this paper, as the upper level controller shown in Fig. 2, a optimization method of the reference clutch slip speed in the clutch slip engagement control is described. Also, the proposed method is applied to the clutch slip engagement control in the launch situation of a vehicle equipped with DCT.

A general PID controller is utilized as the slip controller in the lower level controller part of Fig. 2. The PID controller outputs the reference clutch torque, and it is assumed in this study that the clutch torque tracking is ideally performed. The clutch torque tracking control has been extensively studied in previous studies(Jeong and Lee, 2000a, Jeong and Lee, 2000b, Horn et al., 2003, Ni et al., 2009, Kong et al., 2016, Oh et al., 2016, Kim et al., 2018, Kim and Choi, 2018, Park et al., 2019a) but it is not discussed in detail in this paper.

Above all, it is necessary to define a cost function in order to optimize the reference clutch slip speed in the clutch slip engagement control. As mentioned in the introduction section, the clutch friction energy loss and vehicle jerk which directly affect the vehicle efficiency and ride comfort are in a trade-off relationship according to the clutch engagement time in the clutch slip engagement. Therefore, in this study, the cost function in the optimization technique consists of the clutch friction energy loss term and vehicle jerk term, and the vehicle jerk term is replaced by the derivative term of the output shaft torque since the vehicle jerk is directly related with derivative value of the output shaft torque in the vehicle driveline. More details will be covered in the later subsection.

Also, after defining the cost function, in order to optimize the reference clutch slip speed, the reference clutch slip speed that minimizes the cost function must be found. This means that the cost function should be expressed only in terms of the clutch slip speed. That is, both the clutch friction energy loss term and derivative term of the output shaft torque should be represented as the clutch slip speed in the clutch slip engagement. To address this issue, it is described in the next subsection how to use a driveline model to replace the clutch

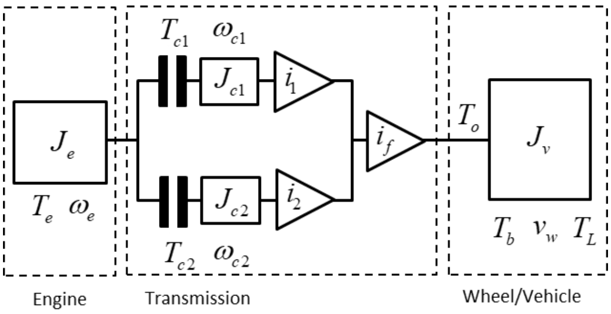

Figure 3. Lumped inertia driveline model of a DCT vehicle

friction energy loss term and derivative term of the output shaft torque with the clutch slip speed.

2.1. Driveline model

Fig. 3 shows a lumped inertia driveline model of a DCT vehicle utilized in this study. Generally, in a production vehicle, engine speed, two input shaft speeds of DCT, and wheel speed are measurable. Therefore, the lumped inertia driveline model for the variables measurable in a production vehicle can be expressed as follows(Kulkarni et al., 2007, Oh et al., 2014, Oh and Choi, 2015, Park et al., 2019b).

$$\dot{\omega}_e = \frac{1}{J_e}T_e - \frac{1}{J_e}T_{c1} - \frac{1}{J_e}T_{c2} \tag{1}$$

$$\dot{\omega}_{c1} = \frac{1}{J_{c1}}T_{c1} + \frac{1}{J_{c1}}T_{c2}\frac{i_2}{i_1} - \frac{1}{J_{c1}}\frac{T_o}{i_1 i_f} \tag{2}$$

$$\dot{\omega}_{c1} = \frac{1}{J_{c1}}T_{c1} + \frac{1}{J_{c1}}T_{c2}\frac{i_2}{i_1} - \frac{1}{J_{c1}}\frac{T_o}{i_1 i_f} \tag{3}$$

$$\dot{\omega}_w = \frac{1}{J_v}(T_o - T_r) \tag{4}$$

Where T, ω, J, and i represent torque, rotational speed, inertia moment, and gear ratio, and the subscript e, $c1$, $c2$, o, w, v, r, 1, 2, and f mean engine, first gear clutch, second gear clutch, output shaft, wheel, vehicle, road load, first gear, second gear, and final gear.

In this study, the clutch slip engagement control in the launch situation of the DCT vehicle is dealt with, so the transmitted torque of the second (even gear) clutch is zero in the launch simulations of the DCT vehicle. Therefore, by using the relation between the input shaft speed of the first (odd gear) clutch and second (even gear) clutch, considering the transmitted torque of the second clutch as zero, the above driveline model can be modified as follows.

$$\omega_{c2} = \omega_{c1}\frac{i_2}{i_1} \tag{5}$$

$$\dot{\omega}_e = \frac{1}{J_e}T_e - \frac{1}{J_e}T_{c1} \tag{6}$$

$$\dot{\omega}_{c1} = \frac{1}{J_1}T_{c1} - \frac{1}{J_1}\frac{T_o}{i_1 i_f} \tag{7}$$

$$\dot{\omega}_w = \frac{1}{J_v}(T_o - T_r) \tag{8}$$

$$J_1 = J_{c1} + \left(\frac{i_2}{i_1}\right)^2 J_{c2} \tag{9}$$

Since the input shaft and wheel shaft are mechanically connected, the following relationship exists between the input shaft speed and wheel speed, and the above driveline model can be reduced as follows.

$$\omega_{c1} = i_1\omega_w \tag{10}$$

$$\dot{\omega}_e = \frac{1}{J_e}T_e - \frac{1}{J_e}T_{c1} \tag{11}$$

$$\dot{\omega}_{c1} = \frac{1}{J}T_{c1} - \frac{1}{J}\frac{T_r}{i_1 i_f} \tag{12}$$

$$J = J_{c1} + \left(\frac{i_2}{i_1}\right)^2 J_{c2} + \left(\frac{1}{i_1 i_f}\right)^2 J_v \tag{13}$$

Here, since the slip speed of the transmission clutch means the difference between the engine speed and clutch speed, the following equation can be derived by using equations (11) and (12).

$$\dot{\omega}_{s1} = \frac{1}{J_e}T_e + \frac{1}{J}\frac{T_r}{i_1 i_f} - (\frac{1}{J_e} + \frac{1}{J})T_{c1} \tag{14}$$

$$T_{c1} = \frac{J_e J}{J_e + J}\frac{1}{J_e}T_e + \frac{J_e J}{J_e + J}\frac{1}{J}\frac{T_r}{i_1 i_f} - \frac{J_e J}{J_e + J}\dot{\omega}_{s1} \tag{15}$$

Where, the subscript $s1$ means slip speed between the engine and first gear clutch.

The relationship between the clutch torque and output shaft torque has the following relationship by the transmission gear ratio.

$$T_o = i_1 i_f T_{c1} \tag{16}$$

From equations (15) and (16), it can be seen that the clutch torque term and output shaft torque term can be replaced with the clutch slip speed when the engine torque and road load torque are known.

In a real vehicle, the engine torque can be estimated using an engine torque map, which is a function of the engine speed, intake manifold air pressure, intake manifold air temperature(Cho and Hedrick, 1989, Hendricks and Sorenson, 1990, Rajamani, 2011, Eriksson and Nielsen, 2014). And, it can be also estimated using an engine torque map, which is a function of the engine speed, throttle air flow rate, intake manifold air temperature.

Furthermore, various estimation methods of the road load torque were proposed in previous studies(Kim et al., 2006, Oh et al., 2013). In this study, the road load torque was estimated based on a Kalman filter. For the consistency of the paper, the method of estimating the road load torque is not described in detail in this paper but the equations of the estimator is represented in the appendix. Therefore, the engine torque and road load torque are considered to be known in the later description.

2.2. Optimization method of reference clutch slip speed

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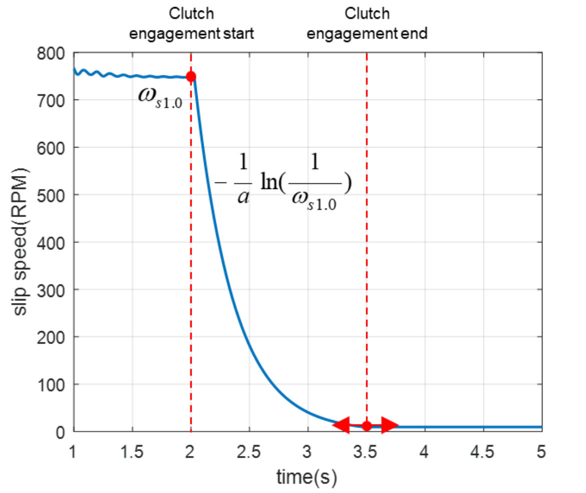

Figure 4. Basic shape of the reference clutch slip speed

As mentioned in the introduction section, the clutch friction energy loss and vehicle jerk which directly affect the vehicle efficiency and ride comfort are in a trade-off relationship in the clutch slip engagement, and the vehicle jerk is directly related with the derivative value of the output shaft torque in the vehicle driveline.

In the optimization method of the reference clutch slip speed proposed in this study, the cost function is composed of the clutch friction energy loss term and derivative term of the output shaft torque as follows. In the below equation, the integration value of the left term during the clutch slip engagement represents the amount of the clutch friction energy loss and the integration value of the right term represents the average derivative value of the output shaft torque.

$$C = \int_0^{t_{end}} T_{c1}\omega_{s1} + f\frac{|\dot{T}_o|}{t_{end}}dt \tag{17}$$

Where, t_{end} represents clutch slip engagement time, and f is a tuning parameter.

The tuning parameter f serves to adjust the ratio between the clutch friction energy loss and average derivative value of the output shaft torque in the cost function. It will be discussed later but the tuning parameter f can be used to adjust the clutch engagement feeling during the clutch slip engagement.

In the previous studies(Dassen and Serrarens, 2003, Gao et al., 2009, Gao et al., 2010, Lu et al., 2011, Tran et al., 2012) , a third order polynomial and an exponential functions were utilized as a function of the reference slip speed. In this study, the reference slip speed is expressed as an exponential function as follows. Fig. 4 shows the basic shape of the reference clutch slip speed used in this study.

$$\omega_{s1} = \omega_{s1.0}e^{-at} \tag{18}$$

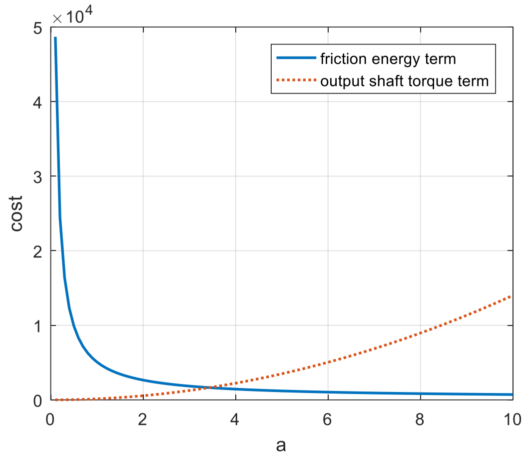

Figure 5. Clutch friction energy loss and average derivative value of the output shaft torque during the clutch slip engagement

Where, $\omega_{s1.0}$ represents initial clutch slip speed between the engine and first gear clutch at the initial time of the clutch slip engagement.

As shown in Fig. 4, in this study, the clutch slip slip engagement time is determined to minimize the cost function composed of the clutch friction energy loss term and derivative term of the output shaft torque, so that the reference clutch slip speed can be generated in the clutch slip engagement.

In addition, the clutch slip engagement time is regarded as the time when the clutch slip speed becomes 1 rad/s as shown below using equation (18).

$$t_{end} = -\frac{1}{a}\ln(\frac{1}{\omega_{s1.0}}) \tag{19}$$

Where, the subscript 0 means initial value at the initial time of the clutch slip engagement.

Then, using equations (15), (16), (18), and (19), equation (17) can be rearranged as follows.

$$C = \int_0^{t_{end}} \left[\begin{array}{l} \left(\begin{array}{l} \frac{J_e J}{J_e + J}\frac{1}{J_e}T_{e.0} \\ + \frac{J_e J}{J_e + J}\frac{1}{J}\frac{T_{r.0}}{i_1 i_f} \\ + \frac{J_e J}{J_e + J} a\omega_{s1.0}e^{-at} \end{array} \right) \cdot \left(\omega_{s1.0}e^{-at}\right) \\ + f\frac{i_1 i_f \frac{J_e J}{J_e + J} a^2 \omega_{s1.0} e^{-at}}{-\frac{1}{a}\ln(\frac{1}{\omega_{s1.0}})} \end{array} \right] dt \tag{20}$$

Here, since the engine torque, road load torque, and slip speed at the initial time of the clutch slip engagement are all known, the unknown variable in the

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Table 1. Vehicle parameters

Parameter	Value
J_e, $kg \cdot m^2$	0.05
J_{c1}, $kg \cdot m^2$	0.02
J_{c2}, $kg \cdot m^2$	0.02
J_o, $kg \cdot m^2$	0.3
J_v, $kg \cdot m^2$	120.6
i_1, -	3.46
i_2, -	2.05
i_f, -	4.06

cost function is only a, which is related with the clutch slip engagement time. Hereinafter, a is referred as the optimization variable.

Now, in order to minimize the cost function for the optimization variable a, equation (20) is differentiated with respect to the optimization variable a, and the following equation can be derived. Here, it is assumed that the engine torque and road load torque are maintained constant at initial value during the clutch slip engagement.

$$a = \sqrt[3]{\frac{\frac{1}{J_e}T_{e.0}\log(\omega_{s1.0}) + \frac{1}{J}\frac{T_{r.0}}{i_1 i_f}\log(\omega_{s1.0})}{2\,f i_1 i_f}} \tag{21}$$

Fig. 5 shows the clutch friction energy loss and the average derivative value of the output shaft torque during the clutch slip engagement. The principle of the optimization method described above can be visually explained with reference to Fig. 5. The solid line in Fig. 5 represents the friction energy loss ($\int_0^{t_{end}} T_{c1}\omega_{s1}dt$)according to the optimization variable a which is related with the clutch slip engagement time, also the dotted line represents the average derivative value ($\int_0^{t_{end}} f\frac{|\dot{T}_o|}{t_{end}}dt$) of the output shaft torque according to the optimization variable a. The value of 50 Nm, 500 Nm, 10 as the engine torque, road load torque, and tuning parameter was utilized to calculate the clutch friction energy loss and the average derivative value of the output shaft torque, and other parameters are shown in Table. 1. In Fig. 5, there would exist the optimization variable a which minimizes the cost function of equation (20) near the point where the solid line and dotted line meet, and the exact point can be expressed like equation (21).

As the optimization variable a of the reference clutch slip speed function increases, the slip engagement time decreases. On the other hand, As the optimization variable a decreases, the clutch slip engagement time increases. Therefore, As the initial engine torque and road load torque at the initial time of the clutch slip engagement increase, the clutch slip engagement time decreases. On the other hand, As the initial engine torque and road load torque decrease, the clutch slip engagement time increases.

Referring to Fig. 5, the larger the engine torque and road load torque at the initial time of the clutch slip engagement, the more the solid line moves to the upper right. Then, the point where the sold line and dotted line meet moves to the right, and the optimization variable a becomes larger. Conversely, the smaller the engine torque and road load torque, the solid line moves to the lower left. Then, the point where the solid line and dotted line meet moves to the left and the optimization variable a becomes smaller.

In this way, it is possible to determine the optimization variable a of the reference clutch slip speed which minimizes the cost function, consists of the clutch friction energy loss and average derivative value of the output shaft torque, according to the engine torque and road load torque at the initial time of the clutch slip engagement, and finally, the reference clutch slip speed can be generated using equations (18) and (21).

2.3. Real-time optimization method of the reference clutch slip speed

In subsection 2.1, the optimization of the reference clutch slip speed is performed only once at the initial time of the clutch slip engagement using the initial engine torque and road load torque. That is, the clutch slip speed is updated only once at the initial time. In this case, the optimization of the reference clutch slip speed cannot cope with the change of the engine torque and road load torque during the clutch slip engagement.

Fig. 6 shows an example of the engine torque and

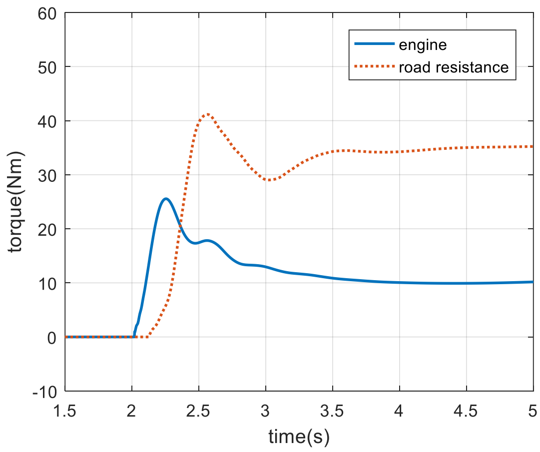

Figure 6. Examples of the engine torque and road load torque in vehicle launch

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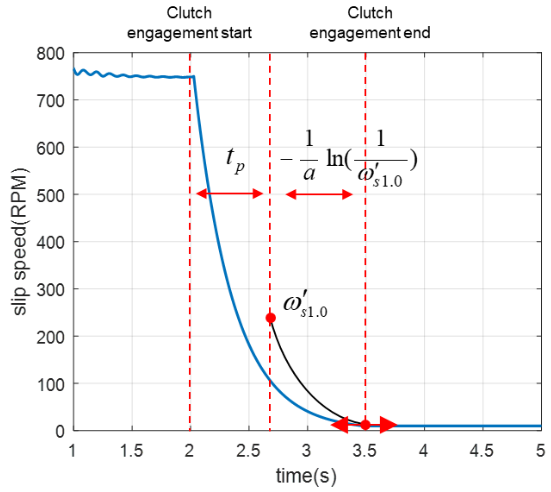

Figure 7. Concept of the reference slip speed function in the real-time optimization method

road load torque in vehicle launch. In this example, the vehicle launch and clutch slip engagement are started at around 2 seconds.

As shown in Fig. 6, the engine torque increases from 0 when the vehicle launches. Also, the actual road load torque always has a constant value, but when the road load torque is estimated, the road load torque can be estimated only when the engine torque and the speeds of the driveline are not zero. Thus, the road load torque also increases from 0 when the vehicle launches.

Therefore, when the optimization of the reference clutch slip speed is performed only once at the initial time of the vehicle launch, the optimization variable *a* will be calculated as zero since the zero value of the engine torque and road load torque are utilized in equation (21).

To solve this issue, the optimization method of the reference clutch slip speed should be modified so that the real-time engine torque and road load torque are utilized. In the real-time optimization method of the reference clutch slip speed, the function of the reference clutch slip speed expressed like equation (18) is modified as follows so that it can be expressed using the real-time clutch slip speed.

$$\omega_{s1} = \omega'_{s1.0} e^{-a(t-t_p)} \tag{22}$$

Where, $\omega'_{s1.0}$ represents clutch slip speed at the time when the optimization is conducted, and t_p means past time from the beginning of the clutch slip engagement.

Fig. 7 shows the concept of the reference clutch slip speed function in the real-time optimization method. The clutch slip engagement time is defined as the point when the clutch slip speed becomes 1, which can be expressed as follows.

$$t_{end} = t_p - \frac{1}{a}\ln(\frac{1}{\omega'_{s1.0}}) \tag{23}$$

The cost function expressed like equation (17) can be rearranged as follows using equations (22) and (23).

$$C = \int_{t_p}^{t_{end}} \left[\begin{array}{l} \left(\begin{array}{l} \frac{J_e J}{J_e + J}\frac{1}{J_e}T_e \\ +\frac{J_e J}{J_e + J}\frac{1}{J}\frac{T_r}{i_1 i_f} \\ +\frac{J_e J}{J_e + J} a\omega'_{s1.0}e^{-a(t-t_p)} \end{array} \right) \\ \cdot\left(\omega'_{s1.0}e^{-a(t-t_p)}\right) + f\dfrac{i_1 i_f a^2 \omega'_{s1.0}e^{-a(t-t_p)}}{-\frac{1}{a}\ln(\frac{1}{\omega'_{s1.0}})} \end{array} \right] dt \tag{24}$$

Here, it is assumed that the engine torque and road load torque are maintained constant at initial value during the clutch slip engagement. Then, the cost function is differentiated with respect to the optimization variable *a*, and the following equation which is the same with the equation (21) can be derived. The detailed derivation process is omitted since it is a simple mathematical operation.

$$a = \sqrt[3]{\frac{\frac{1}{J_e}T_e \log(\omega'_{s1.0}) + \frac{1}{J}\frac{T_r}{i_1 i_f}\log(\omega'_{s1.0})}{2 f i_1 i_f}} \tag{25}$$

Through the above-mentioned process, the method of obtaining the optimization variable *a* of the reference clutch slip speed was derived which minimizes the cost function, consists of the clutch friction energy loss and average derivative value of the output shaft torque, according to the real-time engine torque, road load torque, and clutch slip speed during the clutch slip engagement. Now, It is possible to generate the reference clutch slip speed in real time during the clutch slip engagement using equations (22) and (25).

3. SIMULATION RESULTS

In this section, the optimization method of the reference clutch slip speed in the clutch slip engagement, proposed in this study, is verified with simulations using AMESIM and MATLAB Simulink, and is applied to vehicle launch situations.

The real-time optimization method which is described in subsection 2.2 and the one-time optimization method which is described in subsection 2.1 are compared with each other.

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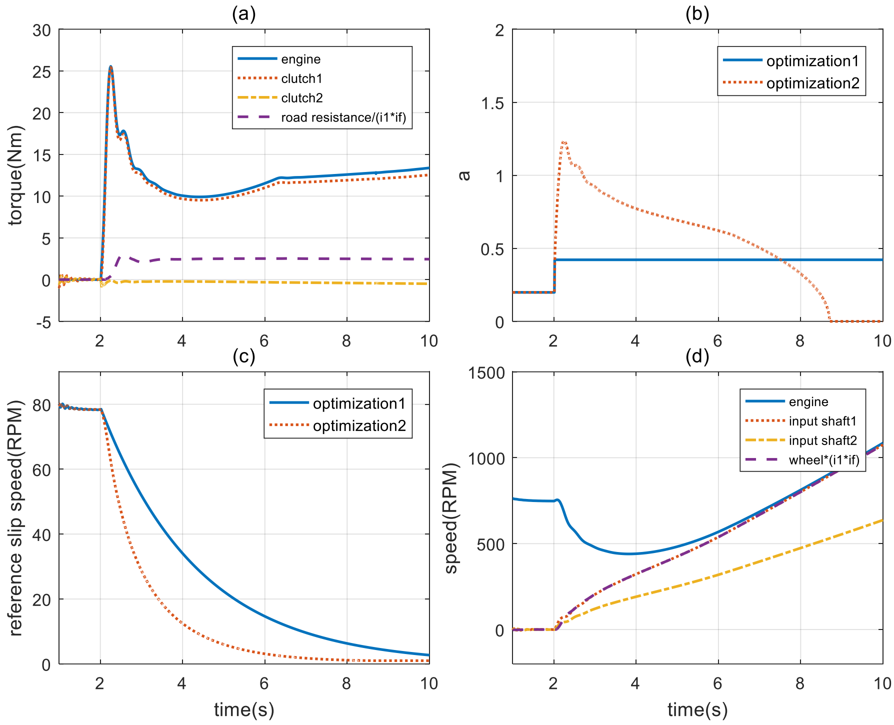

Figure 8. Behavior of the vehicle in the vehicle launch: (a) driveline torque, (b) optimization variable *a*, (c) reference clutch slip speed, (d) driveline speed

Here, the one-time optimization method is the method of performing optimization once using the initial engine torque, road load torque, and slip speed at the initial time of the clutch slip engagement, obtaining the optimization variable *a*, and generating the reference clutch slip speed. Whereas, the real-time optimization method is the method of performing optimization in real time using the real-time engine torque, road load torque, and slip speed during the clutch slip engagement, obtaining the optimization variable *a*, and generating the reference clutch slip speed.

Furthermore, the real-time optimization method is verified by applying it to vehicle launch situations in which the engine torque, road load torque, and tuning parameter *f* are different.

3.1. One-time and real-time optimization method

Fig. 8 shows the behavior of a vehicle in vehicle launch and the results of applying the optimization methods. Fig. 8(a) shows the driveline torque in the vehicle launch: the engine torque, odd gear clutch torque of DCT, even gear clutch torque of DCT, and road load torque divided by the first gear ratio and final gear ratio. Fig. 8(b), and (c) show the optimization variable *a* which is calculated by the one-time optimization method and the real-time optimization method, and the reference clutch slip speed which is generated by the one-time optimization method and the real-time optimization method. In the legends, optimization1 represents the one-time optimization method, and optimization2 represents the real-time optimization method. Fig. 8(d) show the driveline speed in the vehicle launch: the engine speed, first input shaft(odd gear) speed of DCT, second input shaft(even gear) speed of DCT, and wheel speed divided by the first gear ratio and final gear ratio. In Fig. 8, the slip engagement was started from 2 seconds.

In this simulation, the parameters in Table 1 were utilized in Equation (24), and the tuning parameter *f* was set to 40.

Referring to Fig. 8(b) and (c), in the case of the one-time optimization method, the optimization of the reference clutch slip speed is performed only once using the initial engine torque, road load torque, and the clutch slip speed at the initial time of the clutch slip engagement. Thus, the optimization variable *a* was maintained constant during the slip engagement and the value was almost zero since the initial engine torque and road load torque were almost zero at the initial time of the slip engagement. The reason why the optimization variable *a* was not exactly zero although the initial engine torque and road load at the initial time were zero is that the minimum value of the engine torque and road load was limited to a threshold. Also, the shape of the reference slip speed did not change during the slip

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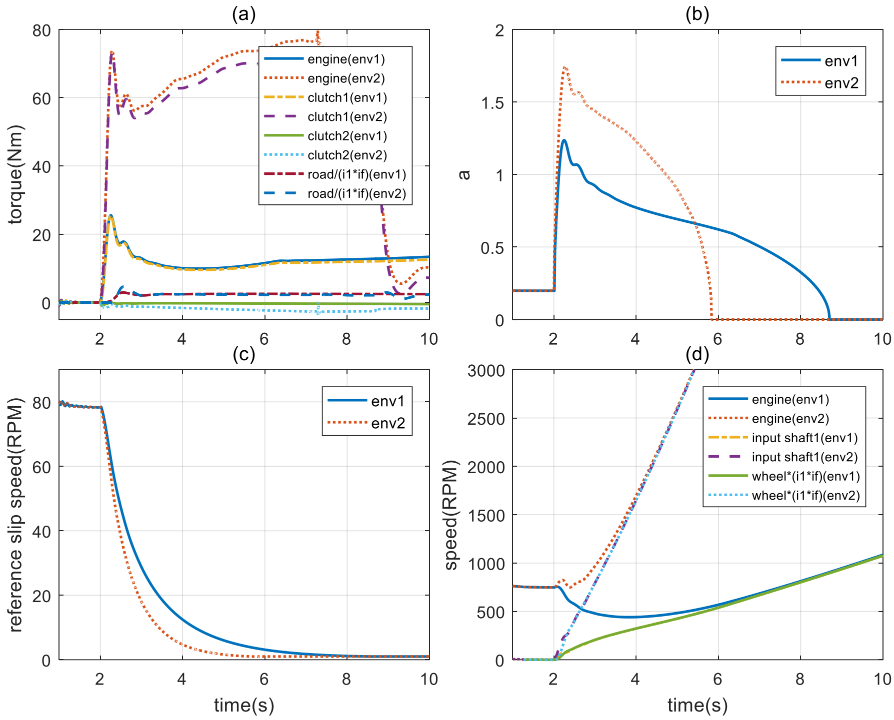

Figure 9. Behavior of the vehicle in the vehicle launch situations when the engine torque is different: (a) driveline torque, (b)optimization variable *a*, (c) reference clutch slip speed, (d) driveline speed

engagement after generated at the initial time. Therefore, it can be seen that the optimization of the reference clutch slip speed is not performed appropriately when the one-time optimization method is applied to the vehicle launch simulation.

Whereas, in the case of the real-time optimization method, the optimization of the reference clutch slip speed is performed in real time using the real-time engine torque, road load torque, and clutch slip speed during the clutch slip engagement. Thus, the optimization variable *a* and reference clutch slip speed changed in real time.

In Fig. 8(d), it can be seen that the vehicle launched well when the reference slip speed generated using the real-time optimization method was used.

3.2. Situations when the engine torque is different

In this subsection, it is addressed that how the reference clutch slip speed is generated using the real-time optimization method when a vehicle launches in two different situations when the engine torque is different. In each situation, the amount of throttle opening in the vehicle launch was set differently.

In the legends of the later figures, each situation when the engine torque is different is represented as env1 and env2.

Fig. 9(a) shows the driveline torque in two situations when the road load torque is almost the same but the engine torque is different. The driveline torque shown is the engine torque, odd gear clutch torque of DCT, even gear clutch torque of DCT, and road load torque divided by the first gear ratio and final gear ratio. Fig. 9(b), and (c) show the optimization variable *a* in two situations which is calculated by the real-time optimization method, and the reference clutch slip speed which is generated by the real-time optimization method. Fig. 9(d) show the driveline speed in two situations: the engine speed, first input shaft(odd gear) speed of DCT, second input shaft(even gear) speed of DCT, and wheel speed multiplied by the first gear ratio and final gear ratio. In Fig. 9, the slip engagement is started from 2 seconds.

In this simulation, the parameters in Table 1 were utilized in Equation (24), and the tuning parameter *f* was set to 40.

Referring to Fig. 9(b) and (c), it can be seen that, in the real-time optimization method, as the engine torque increases, the optimization variable *a* which minimizes the cost function, consists of the clutch friction energy loss and vehicle jerk, increases, and then the clutch slip engagement time becomes shorter.

In Fig. 9(d), it can be seen that the vehicle launched well in two situations when the reference slip speed generated using the real-time optimization method.

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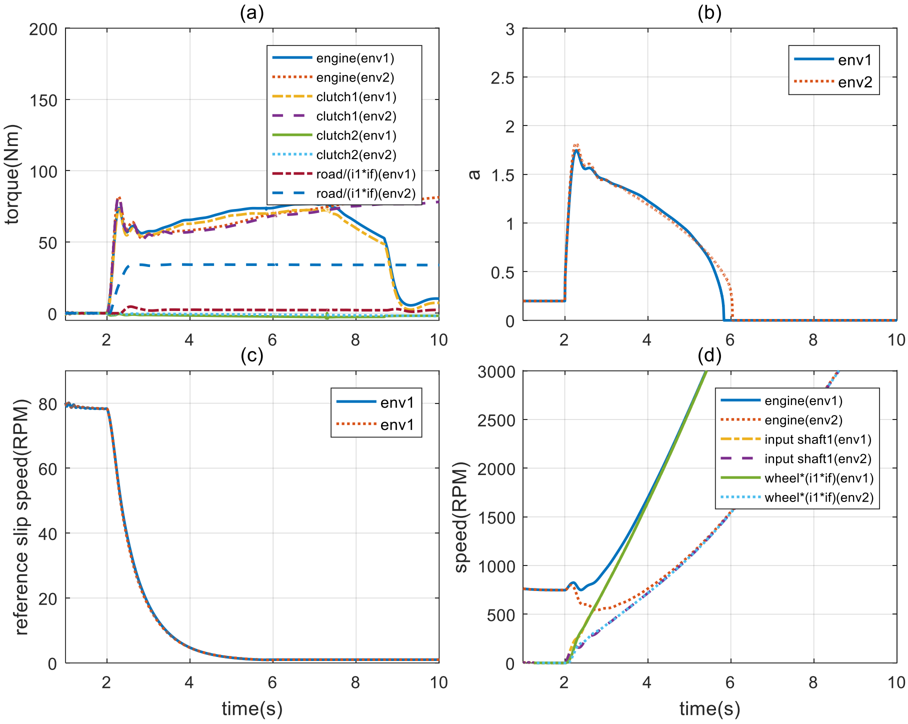

Figure 10. Behavior of the vehicle in the vehicle launch situations when the road load torque is different: (a) driveline torque, (b)optimization variable *a*, (c) reference clutch slip speed, (d) driveline speed

3.3. Situations when the road load torque is different

In this subsection, it is addressed that how the reference clutch slip speed is generated using the real-time optimization method when a vehicle launches in two different situations when the road load torque is different. In each situation, the road slope was set differently.

In the legends of the later figures, each situation when the road load torque is different is represented as env1 and env2.

Fig. 10(a) shows the driveline torque in two situations when the engine torque is almost the same but the road load torque is different. The driveline torque shown is the engine torque, odd gear clutch torque of DCT, even gear clutch torque of DCT, and road load torque divided by the first gear ratio and final gear ratio. Fig. 10(b), and (c) show the optimization variable *a* in two situations which is calculated by the real-time optimizaiton method, and the reference clutch slip speed which is generated by the real-time optimization method. Fig. 10(d) show the driveline speed in two situations: the engine speed, first input shaft(odd gear) speed of DCT, second input shaft(even gear) speed of DCT, and wheel speed multiplied by the first gear ratio and final gear ratio. In Fig. 10, the slip engagement is started from 2 seconds.

In this simulation, the parameters in Table 1 were utilized in Equation (24), and the tuning parameter *f* was set to 40.

Referring to Fig. 10(c) and (d), it can be seen that the optimization variable *a* calculated by the real-time optimization method and the generated reference clutch slip speed were almost the same for the two situations when the road load torque was different.

This was analyzed because the inertia J in front of the road load torque in equation (25) is considerably large. The inertia J in front of the road load torque was around ten times larger than the inertia J_e in front of the engine torque, which means that the effect of the road load torque on the optimization variable *a* is much smaller than that of the engine torque.

From the above simulations, it was seen that, in the real-time optimization method, as the engine torque increases, the optimization variable *a* which minimizes the cost function, consists of the clutch friction energy loss and vehicle jerk, increases, and then the clutch slip engagement time becomes shorter. However, it was seen that, as the road load torque changes, the optimization variable *a* and clutch slip engagement time do not change so much. This was analyzed because the inertia J in front of the road load torque is considerably large.

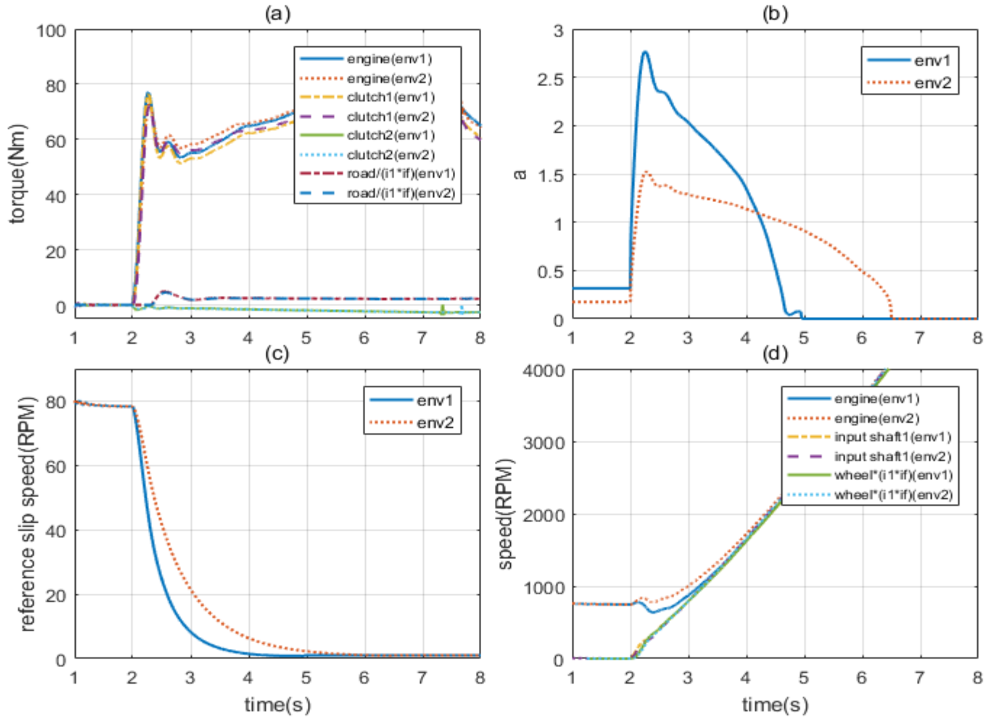

Figure 11. Behavior of the vehicle in the vehicle launch situations when the tuning parameter is different: (a) driveline torque, (b)optimization variable *a*, (c) reference clutch slip speed, (d) driveline speed

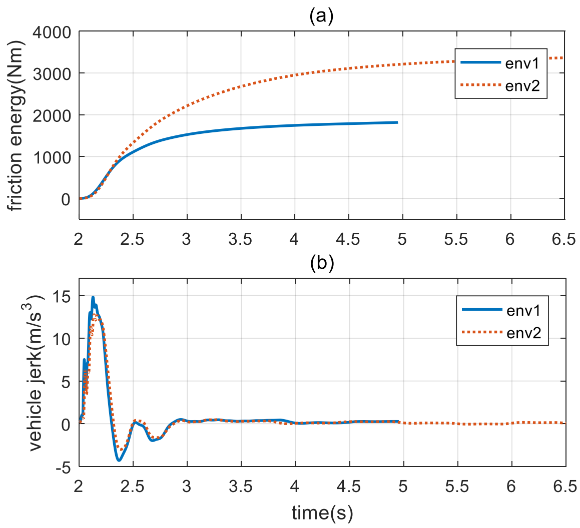

Figure 12. Friction energy loss and vehicle jerk

3.4. Situations when the tuning parameter *f* is different

In this subsection, it is addressed how the reference clutch slip speed is generated when the tuning parameter *f* is different. In each simulation, the tuning parameter *f* were set to 10 and 60, and in legend of the following figures, each situation is represented as env1 and env2.

Fig. 11(a) shows the driveline torque in two situations when the tuning parameter is different. The driveline torque shown is the engine torque, odd gear clutch torque of DCT, even gear clutch torque of DCT, and road load torque divided by the first gear ratio and final gear ratio. Fig. 11(b), and (c) show the optimization variable *a* in two situations which is calculated by the real-time optimizaiton method, and the reference clutch slip speed which is generated by the real-time optimization method. Fig. 11(d) show the driveline speed in two situations: the engine speed, first input shaft(odd gear) speed of DCT, second input shaft(even gear) speed of DCT, and wheel speed multiplied by the first gear ratio and final gear ratio. In Fig. 11, the slip engagement is started from 2 seconds.

Fig. 12(a) shows the integral value of the clutch friction energy loss over time in the two situations. Fig. 12(b) shows the vehicle jerk in the two situations. Table 2 shows the friction energy loss and the peak value of the vehicle jerk in the two situations.

Table 2. Friction energy loss and maximum vehicle jerk whenthe tuning parameter *f* is different.

Situation	Friction energy loss(Nm)	Maximum vehicle jerk(m/s^3)
env1	0.05	14.8230
env2	0.02	12.7843

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As shown in Fig. 11(a), the engine torque and road load torque were almost the same in the two situations, but the tuning parameter *f* was different in the two situations.

If the tuning parameter *f* is small in the cost function in equation (16), it means that the clutch friction energy loss in the cost function is relatively more important. Whereas, if the tuning parameter *f* is large, it means that the vehicle jerk is relatively more important.

Therefore, as shown in Fig. 11(c), in the env1, the clutch slip engagement time was shorter because the focus is on the clutch friction energy loss in the cost function, whearas, in the env2, the clutch slip engagement time was longer because the focus is on the vehicle jerk.

As can be seen in Fig. 12 and Table 2, in env1 where the tuning parameter *f* was relatively small, the clutch friction energy loss was small and the peak of the vehicle jerk was large. Also, In env2 where the tuning parameter *f* was relatively large, the clutch friction energy loss was large and the peak of the vehicle jerk was small.

From this simulation, it was seen that, in the proposed optimization method of the reference clutch slip speed, the tuning parameter *f* can be used to adjust the ratio between the clutch friction energy loss and the average derivative value of the output shaft torque in the cost function. Thus, the clutch engagement feeling in the slip engagement can be adjusted depending on the tuning parameter *f*.

4. CONCLUSION

In this paper, a new optimization method of the reference clutch slip speed to improve the vehicle efficiency and ride comfort in the clutch slip engagement was proposed. In the optimization method proposed in this paper, the cost function consists of the clutch friction energy loss and vehicle jerk which directly affect the vehicle efficiency and ride comfort in the clutch slip engagement, and the reference clutch slip speed is generated to rind the reference clutch slip speed which minimized the cost function. Also, the clutch engagement feeling in the clutch slip engagement can be adjusted by adjusting the ratio between the clutch friction energy loss and vehicle jerk in the cost function. It was verified by simulations that the proposed method works well in vehicle launch. In the later study, the ride comfort in the clutch slip engagement according to the shape of the reference slip speed will be studied.

APPENDIX

$$\dot{\omega}_e = \frac{1}{J_e}(T_e - T_{c1} - T_{c2})$$

$$\dot{\omega}_{c1} = \frac{1}{J_1}(T_{c1} + T_{c2}\frac{i_2}{i_1} - \frac{T_o}{i_1 i_f})$$

$$\dot{\omega}_{c2} = \frac{1}{J_2}(T_{c1}\frac{i_1}{i_2} + T_{c2} - \frac{T_o}{i_2 i_f})$$

$$\dot{\omega}_w = \frac{1}{J_v}(T_o - T_r)$$

$$\dot{\omega}_{c1} = \frac{(1+\frac{i_1}{i_2})}{(J_1 + \frac{i_2}{i_1}J_2)}T_{c1} + \frac{(1+\frac{i_2}{i_1})}{(J_1 + \frac{i_2}{i_1}J_2)}T_{c2} - \frac{(\frac{1}{i_1 i_f} + \frac{1}{i_2 i_f})}{(J_1 + \frac{i_2}{i_1}J_2)}T_o$$

$$\dot{\omega}_{c2} = \frac{(1+\frac{i_1}{i_2})}{(J_1\frac{i_1}{i_2} + J_2)}T_{c1} + \frac{(1+\frac{i_2}{i_1})}{(J_1\frac{i_1}{i_2} + J_2)}T_{c2} - \frac{(\frac{1}{i_1 i_f} + \frac{1}{i_2 i_f})}{(J_1\frac{i_1}{i_2} + J_2)}T_o$$

$$T_{c1} = f(d_1)\ (|\omega_e - \omega_{c1}| > 1)$$

$$T_{c2} = f(d_2)\ (|\omega_e - \omega_{c2}| > 1)$$

Where, d_1, and d_2 represents the actuator position measurement of the first gear clutch, and second gear clutch

$$\dot{X}_1 = A_1 X_1 + B_1 U_1,\ Y_1 = C_1 X_1$$

$$X_1 = \begin{pmatrix} \dot{\omega}_e \\ \dot{\omega}_{c1} \\ \dot{T}_{c1} \\ \dot{T}_o \end{pmatrix},\ U_1 = \begin{pmatrix} T_e \\ T_{c2} \end{pmatrix}$$

$$A_1 = \begin{pmatrix} 0 & 0 & -\frac{1}{J_e} & 0 \\ 0 & 0 & \frac{(1+\frac{i_1}{i_2})}{(J_1 + \frac{i_2}{i_1}J_2)} & -\frac{(\frac{1}{i_1 i_f} + \frac{1}{i_2 i_f})}{(J_1 + \frac{i_2}{i_1}J_2)} \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{pmatrix}$$

$$B_1 = \begin{pmatrix} \frac{1}{J_e} & -\frac{1}{J_e} \\ 0 & \frac{(1+\frac{i_2}{i_1})}{(J_1+\frac{i_2}{i_1}J_2)} \\ 0 & 0 \\ 0 & 0 \end{pmatrix},\ C_1 = \begin{pmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \end{pmatrix}$$

$$\dot{X}_2 = A_2 X_2 + B_2 U_2,\ Y_2 = C_2 X_2$$

$$X_2 = \begin{pmatrix} \dot{\omega}_e \\ \dot{\omega}_{c2} \\ \dot{T}_{c2} \\ \dot{T}_o \end{pmatrix},\ U_2 = \begin{pmatrix} T_e \\ T_{c1} \end{pmatrix}$$

$$A_2 = \begin{pmatrix} 0 & 0 & -\frac{1}{J_e} & 0 \\ 0 & 0 & \frac{(1+\frac{i_2}{i_1})}{(J_1+\frac{i_2}{i_1}J_2)} & -\frac{(\frac{1}{i_1 i_f}+\frac{1}{i_2 i_f})}{(J_1+\frac{i_2}{i_1}J_2)} \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{pmatrix}$$

$$B_2 = \begin{pmatrix} \frac{1}{J_e} & -\frac{1}{J_e} \\ 0 & \frac{(1+\frac{i_1}{i_2})}{(J_1+\frac{i_2}{i_1}J_2)} \\ 0 & 0 \\ 0 & 0 \end{pmatrix},\ C_2 = \begin{pmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \end{pmatrix}$$

$$\mathbf{F}_n = e^{\mathbf{A}_n T_s},\ \mathbf{G}_n = \left(\int_0^{T_s} e^{\mathbf{A}_n \tau} d\tau\right)\mathbf{B}_n,\ \mathbf{H}_n = \mathbf{C}_n$$

$$\mathbf{X}_{n.k} = \mathbf{F}_n \mathbf{X}_{n.k-1} + \mathbf{G}_n \mathbf{U}_{n.k-1}$$

$$\mathbf{Y}_{n.k} = \mathbf{H}_n \mathbf{X}_{n.k}$$

$$\hat{\mathbf{X}}^-_{n.k} = \mathbf{F}_n \hat{\mathbf{X}}^+_{n.k-1} + \mathbf{G}_n \mathbf{U}_{n.k-1}$$

$$\mathbf{P}^-_{n.k} = \mathbf{F}_n \mathbf{P}^+_{n.k-1} \mathbf{F}_n^T + \mathbf{Q}_n$$

$$\mathbf{K}_{n.k} = \mathbf{P}^-_{n.k} \mathbf{H}^T_{n.k} \left[\mathbf{H}_{n.k} \mathbf{P}^-_{n.k} \mathbf{H}^T_{n.k} + \mathbf{R}_n\right]^{-1}$$

$$\hat{\mathbf{X}}^+_{n.k} = \hat{\mathbf{X}}^-_{n.k} + \mathbf{K}_{n.k}\left[\mathbf{Y}_{n.k} - \mathbf{H}_n \hat{\mathbf{X}}^-_{n.k}\right]$$

$$\mathbf{P}^+_{n.k} = \left(\mathbf{I} - \mathbf{K}_{n.k}\mathbf{H}_n\right)\mathbf{P}^-_{n.k}$$

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