Friction Coefficient, Torque Estimation, Smooth Shift Control Law for an Automatic Power Transmission

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For shift quality improvement, torque sensors are currently too expensive to be used on production vehicles. To achieve smooth acceleration shift, the reference trajectory of the clutch slip speed for accomplishing the shift process within a designated shift completion time and its relationship with the clutch actuating torque were suggested by Jeong and Lee (1999). In order to facilitate the proposed algorithm, nonlinear estimators for necessary information such as the axle shaft torque, clutch friction and turbine torque were designed using only speed sensors. Accounting for the modeling error, a control law for this indirect smooth shift was proposed based on the above mentioned suggestions. Simulation results of the proposed estimators and shift controller were presented and further considerations for practical applications are discussed.

Key Words : Clutch-to-Clutch Shift, Dynamic Shift Condition, Nonlinear Reduced Order Observer, Indirect Smooth Acceleration Shift

1. Introduction

Notwithstanding low efficiency and high cost, automatic power transmission systems are widely used in passenger vehicles due to easy drive-ability, torque amplifying and torsional damping characteristics of torque converter. A conventional automatic transmission commonly uses planetary gear sets and over-running sprags to effect smooth transition between speed ranges. This expensive mechanical component may be eliminated and the kinematic arrangement can be significantly simplified if a smooth clutch-to-clutch shift is made possible. This featare offers economic advantages but presents a challenging control problem. Meanwhile, sensors currently used for measuring the shaft torque are too expensive to be used on production vehicles for control purpose, while magnetic pick-up speed sensors can be widely applied for speed checking, shift decision, engine control and anti-skid braking, among others.

Cho and Hedrick (1989a) developed a power-train model for control. The modeling effort was directed to achieving a reasonable trade-off between simplicity and comprehensiveness of the model comprised of three major components of the engine, transmission and drive-train. The torque estimation of the vehicle axle shaft was studied by using inexpensive RPM sensors to facilitate a nonlinear control algorithm (Masmoudi and Hedrick, 1992). The sliding mode theory developed for state observation (Slotine, et al., 1987) was applied. A sliding mode controller was designed for clutch-to-clutch shift and an alternative formulation was presented to deal with uncertainties in actuator or control dynamics (Cho and Hedrick, 1989b). The anthors set the speed gaps between the desired and actual value

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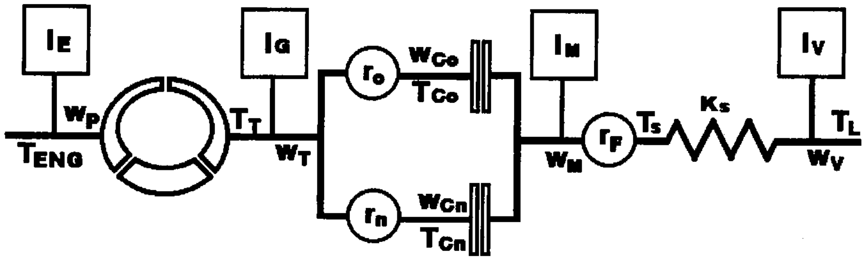

Fig. 1 Shift model of automatic transmission

of the reaction carrier and torque converter turbine as two sliding surfaces. But there was no mention about the desired speed trajectories, which may be more important in some sense. Jeong sought indirect shift control using only speed sensors and proposed a desired speed trajectory (Jeong and Lee, 1994a; 1994b). He further refined the idea, studied shift characteristics in depth. As a result, desired slip speed for smooth acceleration shift, aimed at accelerating the vehicle smoothly from the initial state to the predetermined final state while satisfying the *dynamic shift condition*, was designed (Jeong and Lee, 1999).

For shift control, two important issues are shift point decision and shift quality control (Jeong, 1993). Two primary objectives in shift quality control are smooth transient and short shift completion time. The smoothness of shift is related to the change in the vehicle acceleration which is directly proportional to the axle shaft torque. The second is related to the clutch life, and the clutch energy dissipation may be reduced for components' longevity.

This paper proposes a smooth acceleration shift control law that tracks the proposed slip speed trajectory (Jeong and Lee, 1999) for shift quality while accomplishing gear ratio change within a designated shift completion time for prolonged system life. For design simplicity and practical applicability, we seek the control law without regard for detailed model of the engine and torque converter. To this end, necessary information such as the turbine torque, axle shaft torque and friction coefficient will be obtained by nonlinear estimators using only speed information. The remainder of this paper is organized as follows: Section 2 describes our system model. Estimators for the shaft torque, friction coefficient and turbine torque are designed in Sec. 3. Section 4 proposes an indirect shift control law and conclusions are followed.

2. Clutch-to-Clutch Shift Model

A typical clutch-to-clutch shift model of an automatic transmission can be depicted as shown in Fig. 1. It consists of the engine, torque converter, gear sets, final drive gear, drive shaft and load torque. The system in the figure may be described mathematically as follows:

$$I_{EP}\dot{\omega}_P = T_{ENG} - T_P \tag{1}$$

$$I_{TG}\dot{\omega}_T = T_T - \left(\frac{1}{r_o}T_{Co} + \frac{1}{r_n}T_{Cn}\right) \tag{2}$$

$$I_M\dot{\omega}_M = (T_{Co} + T_{Cn}) - \frac{1}{r_F}T_s \tag{3}$$

$$I_V\dot{\omega}_V = T_s - T_L \tag{4}$$

$$\dot{T}_s = K_s\left(\frac{1}{r_F}\omega_M - \omega_V\right), \tag{5}$$

where subscript o means the old (off-going) clutch, n the new (up-coming) clutch, F the final drive and others are as denoted in the figure. The speed relations before and after engaging the new clutch and the conesponding governing equations are grien by Eqs. (6) and (7), respectively.

$$\begin{aligned} &\textit{before} : \omega_T = r_o\omega_{Co} = r_o\omega_M \\ &\textit{after} : \omega_T = r_n\omega_{Cn} = r_n\omega_M \end{aligned} \tag{6}$$

$$\begin{aligned} &\textit{before} : (r_o^2 I_{TG} + I_M)\,\dot{\omega}_M = r_o T_T - T_s/r_F \\ &\textit{after} : (r_n^2 I_{TG} + I_M)\,\dot{\omega}_M = r_n T_T - T_s/r_F \end{aligned} \tag{7}$$

The drive-train is simply modeled as a lumped parameter torsional spring and vehicle inertia (Cho and hedrick, 1989a). The load torque mainly consists of the tire rolling resistance, aerodynamic drag force and grade resistance. Note that the system characteristics such as the parameter values, input torque and system order vary with the progression of the speed ratio change process. During shift, the clutch torque generated by hydraulic system can be given as follows

$$T_{Ci}=\mu nARP_{Ci},\ i=o,\ n \quad (8)$$

where μ is the friction coefficient, n the number of frictional disk, A the disk area, R the effective radius and P_C the actuating pressure. However, when the clutch is engaged the torque is determined by a entirely different principle. The so-called *engaged clutch torque* is just the torque required to prevent slipping of the two clutch faces as given by Eq. (9), which is independent of the actuating pressure (Jeong and Lee, 1998; 1999).

$$T_{Ei}=\frac{r_i^2 I_{TG} I_M}{r_i^2 I_{TG}+I_M}\left(\frac{1}{r_i I_{TG}}T_T+\frac{1}{I_M}T_s\right),\ i=o,\ n \quad (9)$$

Throughout this paper, the load information is assumed known and the old clutch torque, T_{Co}, is assumed to be cut off instantaneously as shift starts. Furthermore, the clutch actuators are assumed to have unlimited bandwidth with no delay. Let us denote the desired shift completion time τ and occasionally denote the new clutch torque T_{Cn}, as $u(t)$ to emphasize that it is a control input variable.

3. Estimator Design

In order to accomplish indirect smooth acceleration shift using only RPM sensors, one unfortunately needs torque information on the turbine shaft of the torque converter and drive axle shaft (Jeong and Lee, 1999). To avoid using expensive torque sensors, torque estimators were designed. For design simplicity and practical applicability, we seek the estimator design without regard to detailed model of the engine and torque converter. Three important criteria were selected for the estimator design. One is the prevention of oscillation of the estimated torques since it is directly related to control input and therefore affects the shift quality. The second is the rate of convergence considering short shift time, which is generally set at less than 1.5 sec. for components' longevity. As mentioned in the previous section, the clutch torques behave differently depending on the clutch states. Hence, the third is the ability to estimate both the shifting and engaged states. The accuracy and robustness were also considered.

3.1 Shaft torque estimator

Shaft torque is one of the state variables of system Eqs. (3)-(5). Comparative studies of existing nonlinear state observer techniques are given by Walcott and Zak (1987), Misawa and Hedrick (1989), and Thein and Misawa (1995). For the pressent study, we apply the sliding mode observer theory since it satisfies our design criteria and offers several advantages such as the order reduction, easiness of design and implementation, decoupling property of the design procedure and the robustness against modelling error. As for as the technique is concerned, our design is almost the same as that of Masmoudi and Hedrick (1992) except for minor modifications and different interpretations.

First, the sliding mode observer theory (Slotine, et al., 1997) is briefly reviewed. Consider a nonlinear system

$$\dot{x}=f(x,\ t)+d(t),\ x\in R^n \quad (10)$$
$$y=Cx+n(t),\ y\in R^p$$

where the function $f(x,\ t)$ is not exactly known, but the level of imprecision $|\Delta f|$ of $f(x,\ t)$ is upper bound by a known continuous function. The disturbance $d(t)$ is also unknown but its magnitule is bound. If we define the i-th element of sliding surfaces, s_i, as the error $\tilde{y}_j$ between the measurement value y_i and estimated value $\hat{y}_i$

$$s_i=\tilde{y}_i=C_i\tilde{x}=C_i(\hat{x}-x),\ i=1,\ \cdots,\ p \quad (11)$$

where C_i is ith row of $p\times n$ C matrix. Then, the existence condition for the sliding surface $s_i=0$ to be attractive is

$$s_i\dot{s}_i=\tilde{y}_i\dot{\tilde{y}}_i<0 \quad (12)$$

If this condition holds, the states approach the sliding surface and then slides along the surface $s_i=0$, i. e., the error $\tilde{y}_i=0$. To guarantee the condition (12), a nonlinear switching term, Kv_s, is added to the conventional linear observer

$$\dot{\hat{x}}=\hat{f}(\hat{x}, t)-K_f\tilde{y}-Kv_s, \tag{13}$$

where $\hat{f}$ is a model of f, K_f and K are gain matrices to be determined. The error dynamics can be given from Eqs. (10) and (13)

$$\dot{\tilde{x}}=\hat{f}(\hat{x}, t)-f(x, t)-K_fC\tilde{x}-Kv_s \tag{14}$$

The sliding condition may be given from (12) and the switching gain K should be large enough to satisfy the condition. On the sliding surface $s=C\tilde{x}=0$, the equivalent value of the switching term v_s can be obtained from Eq. (14)

$$v_s\approx(CK)^{-1}C(\hat{f}-f) \tag{15}$$

Thus, the reduced order $(n-p)$ dynamics on the sliding manifold yield

$$\dot{\tilde{x}}\approx[I-K(CK)^{-1}C](\hat{f}-f),\quad C\tilde{x}\approx 0 \tag{16}$$

In Masmoudi and Hedrick (1992), the gain K_f was selected based on the Kalman filter theory for a linearized model. However, note that the gain does not affect the reduced dynamics. It has an influence only on the so-called reaching phase. Therefore, in contrast to Masmoudi and hedrick (1992), the diagonal terms $K_{f,ii}$, $i=1, \cdots, p$ can be selected to adjust the rate of convergence and the off-diagonal elements can be set zero. If the disturbance channel is available, i. e., structured uncertainty, the concept of unknown input observer may also be applied for the selection of the gain K_f which was developed in fault diagnostic systems (Chang et al., 1994). The idea was further enhanced by Takahashi and Peres (1996) by unifying the existing disturbance decoupling observers based on projector theory. In Masmoudi and Hedrick (1992), discontinuous switching term v_s was selected as follows

$$v_s=1_s=[sign(s_1)\cdots sign(s_p)]^T$$

While the invariance properties may be preserved against the matched disturbances, undesirable chattering problem may be induced. To avoid this harmful feature, especially for our shifting case, many continuous approximations are proposed, e. g., a saturation function using the boundary layer concept, the sigmoid function, $tanh(s)$, which is commonly used as a transfer function of the neuron (Jeong and Utkin, 1998). Another choice is $v_s=s/(|s|+\varepsilon)$ where ε is a small positive constant (Burton and Zinober, 1996). These continuous functions are means of eliminating chatter while retaining, or yielding a good approximation to, the inherent properties of ideal sliding motion.

The proposed estimator for the shaft torque observer is

$$\begin{aligned}\dot{\hat{\omega}}_M&=\frac{1}{I_M}\hat{T}_C-\frac{1}{\gamma_F I_M}\hat{T}_s+k_M\tilde{\omega}_M\\&\quad+M_Mv_s(\tilde{\omega}_M)\\\dot{\hat{\omega}}_V&=\frac{1}{I_V}\hat{T}_s-\frac{1}{I_V}\hat{T}_L+k_V\tilde{\omega}_V+M_Vv_s(\tilde{\omega}_V)\\\dot{\hat{T}}_s&=K_s(\frac{1}{\gamma_F}\omega_M-\omega_V)+k_{s1}\tilde{\omega}_M\\&\quad+k_{s2}\tilde{\omega}_V+M_{s1}v_s(\tilde{\omega}_M)+M_{s2}v_s(\tilde{\omega}_V),\end{aligned} \tag{17}$$

where $\hat{T}_C$ denotes $\hat{\mu}nARP_C$ and $\hat{\mu}$ is the estimated value of the friction coefficient that will be estimated in the following subsections. Define the estimation errors, $\tilde{\omega}_M$ and $\tilde{\omega}_V$, as two sliding surfaces. Selecting the gains M_M and M_V to guarantee the condition (12), one can enforce $s_i\approx 0$. Thus, on the sliding manifold, the equivalent values for the switching terms can be derived from their error dynamics as given by Eq. (14) as follows

$$\begin{aligned}M_Mv_s(\tilde{\omega}_M)&\approx\frac{1}{I_M}\tilde{T}_C-\frac{1}{\gamma_F I_M}\tilde{T}_s,\\\tilde{T}_C&=\tilde{\mu}nARP_C\\M_Vv_s(\tilde{\omega}_V)&\approx\frac{1}{I_V}\tilde{T}_s-\frac{1}{I_V}\tilde{T}_L\end{aligned} \tag{18}$$

Substituting Eq. (18) into the corresponding error dynamics of the shaft torque yields

$$\begin{aligned}\dot{\tilde{T}}_s&\approx-M_{s1}v_s(\tilde{\omega}_M)-M_{s2}v_s(\tilde{\omega}_V)\\&\approx-\left(\frac{M_{s2}}{M_V}\frac{1}{I_V}-\frac{M_{s1}}{M_M}\frac{1}{\gamma_F I_M}\right)\tilde{T}_s\\&\quad-\frac{M_{s1}}{M_M}\frac{1}{I_M}\tilde{T}_C+\frac{M_{s2}}{M_V}\frac{1}{I_V}\tilde{T}_L\end{aligned} \tag{19}$$

This is a first-order differential equation. Thus, no overshoot that can degrade the shift quality may be expected. To be stable, $\{M_{s2}/(M_VI_V)-M_{s1}/(M_M\gamma_FI_M)\}$ should be positive. And for fast convergence it should be as large as possible.

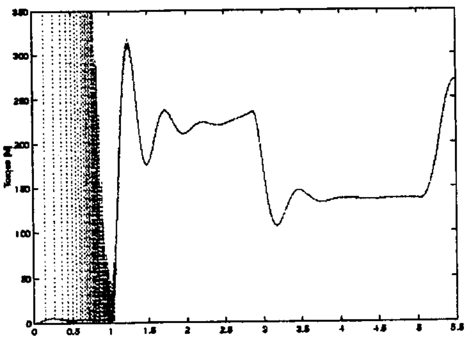

Fig. 2 Estimation of axle shaft torque

Note that it also affects the steady state error which will be analyzed in the later subsection. The gain k_{s1} is set at zero because it has no special role on the performance as can be seen from (19). The gain k_{s2} can be set K_s to expedite the convergence rate.

The shaft torque estimation results are shown in Fig. 2. In all figures in this section, all solid lines indicate the actual values and dotted lines indicate the corresponding estimated values. After 1 sec. of idle running with a constant engine and load torque, the old clutch, i. e. the upper part in Fig. 1, the engagement in initiated with a step -like clutch actuating pressure. These facts may be noticed from the step-like shaft torque of certain level in the figure. At 5 sec., the old clutch was disengaged and at the same time the actuating pressure applied to the new clutch, the lower part in Fig. 1. For the first 1 sec., convergence does not occur since the clutch pressure is not introduced yet. After that period, it is hard to discriminate the two lines. Figure 2 shows fast convergence and good estimation accuracy. In accordance with our design criteria, the figure also shows no oscillation of the estimator. The slow oscillations shown in the figure are due to the elastic effect of the shaft and it dies out gradually. Steadily increasing torque between 2.5 sec. and 2.8 sec. is due to the effect of increasing the static friction coefficient. Note also that the shaft torque of both the slip and engaged case are well estimated in accordance with our design purpose. A sudden drop at around 3 sec. indicates that two surfaces of the old clutch are finally engaged. At this moment, the clutch torque switches from frictional torque (8) into engaged torque (9), which in turn affects the shaft torque.

3.2 Friction coefficient estimator

Unlikely the other data of mechanical components, the friction coefficient of the clutch is highly suspectible to variation. It is generally known to vary with the relative speed. It may also depend on the fluid used and the temperature of the system. Moreover, it heavily influences the shift quality since it is a multiplying parameter of the control pressure input as shown in Eq. (8). There are several techniques for parameter identification and adaptive observer. Some of the recent results are given in references (Bastin and Gevers, 1988; Friedland, 1997; Friedland and Park, 1992; Kreisselmeir, 1977; Marino, 1990; Xu and Hashimoto, 1993). Friedland developed a new algorithm for estimating parameters in dynamic systems, which involves two nonlinear functions to be determined (Friedland, 1997). It was a further generalization of the earlier work for estimating friction coefficient in mechanical systems to improve the accuracy of position control by compensating the friction force (Friedland and Park, 1992). In order to fulfill our design criteria, we applied and slightly modified the technique.

The proposed estimator for friction coefficient are

$$\hat{\mu} = z - k\omega_M{}^m \qquad (20)$$
$$\dot{z} = km\omega_M{}^{(m-1)}\left(\frac{1}{I_M}\hat{\mu} nARP_C - \frac{1}{r_F I_M}\hat{T}_s\right),$$

where the gain k and exponent m are design parameters. It can be considered to be a nonlinear reduced order observer. To study the performance of the proposed estimator, define the error between the actual parameter μ and its estimate $\hat{\mu}$ as $\tilde{\mu}$. Assuming that the true value is constant, we have

$$\dot{\tilde{\mu}} = -\dot{\hat{\mu}} = -\dot{z} + km\omega_M{}^{(m-1)}\dot{\omega}_M$$
$$= km\omega_M{}^{(m-1)}\left(\dot{\omega}_M - \frac{1}{I_M}\hat{\mu} nARP_C + \frac{1}{r_F I_M}\hat{T}_s\right)$$

Substituting Eq. (3) to the above equation,

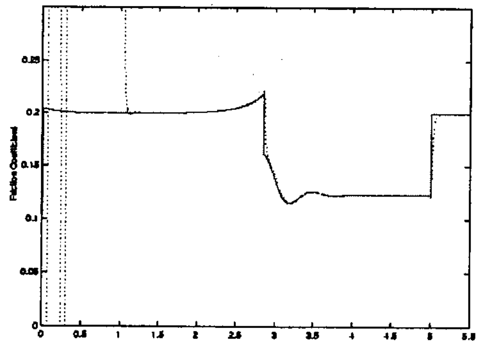

Fig. 3 Estimation of friction coefficient

$$\dot{\tilde{\mu}} = km\omega_M{}^{(m-1)}\left(\frac{1}{I_M}\tilde{\mu}nARP_C - \frac{1}{r_F I_M}\tilde{T}_s\right) \tag{21}$$

For $k<0$, $m>0$, the error converges asymptotically to zero since ω_M is positive. From (21), it can be seen that there is no oscillation due to a negative real pole. The convergence rate can be made arbitrarily fast by adjusting the design parameters k, m. Note also that the error depends on the estimation error of the shaft torque which will be analyzed in the later subsection.

The estimation results are simulated in Fig. 3. For time varying friction, $\mu = \mu_d + (\mu_s - \mu_d)$ exp $(-0.5|\Delta\omega|)$ was used in the simulation where μ_d and μ_s denote the dynamic and static coefficients, respectively. The solid line in the figure is the calculated friction coefficient based on Eqs. (8) and (9). The figure shows that our three design goals are successfully achieved. It tracks the time-varying friction very well as can be seen for up to 2.8 sec.

Around 2.8 sec., note that the coefficient suddenly drops down. As shown in Fig. 2, the shaft torque also drops at the same time. As discussed in the previous subsection, the clutch torque switches from frictional torque into engaged torque when the clutch engagement is completed. Recall that even for the estimation of the engaged case we used Eq. (3) instead of Eq. (7). In other words, the clutch torque of the engaged case was regarded as the torque given by Eq. (8) of the shifting case. Therefore, the estimated coefficient in the figure between 2.8 sec. and 5 sec. may be determined by $\hat{T}_{Eo}/(nARP_C)$. Thus, the proposed estimator also provides a method for estimating the engaged clutch torque. If the data nAR are suspectible, μnAR instead of μ may be estimated by a procedure similar to the one proposed above.

3.3 Turbine torque estimator

A simple static model of the torque converter is that of Kotwicki (1982) which may be suitable for control or estimation purpose

$$T_P = a_1\omega_P^2 + a_2\omega_P\omega_T + a_3\omega_T^2 \tag{22}$$
$$T_T = b_1\omega_P^2 + b_2\omega_P\omega_T + b_3\omega_T^2$$

The parameters a_i, b_i, $i=1, 2, 3$, vary according to the operating modes i. e. the fluid coupling mode and torque converter mode. When the vehicle cruises, it is usually in the coupling mode where the torque ratio T_T/T_P is almost unity. While shifting, it generally changes into the converter mode in which the output torque of which is larger than the input torque. The estimation of the turbine torque based on Eq. (22) requires the identification of parameters b_i. This approach may face an issue of convergence, in some cases it may fall into local minima. The identifiability condition should be met which is related to the richness of the frequency components of the system input. It means that there must be at least $(n+1)$ different frequencies in the system input for the linear time invariant systems, which may be somewhat difficult to satisfy in many practical situations. Discussions about the identifiability condition may be found in Xu and Hashimoto (1993) and the references mentioned therein.

In order to avoid these drawbacks, we design a turbine torque estimator without detailed model of the torque converter. Friedland's parameter identification methods are applied to the turbine torque estimation. The proposed structures of the reduced order estimator are

$$\hat{T}_T = z - k\omega_T{}^m \tag{23}$$
$$\dot{z} = km\omega_T{}^{(m-1)}\left(\frac{1}{I_{TG}}\hat{T}_T - \frac{1}{r_n I_{TG}}\hat{T}_C\right)$$

It is a type of input estimator since the estimated Value is one of the input terms of dynamic Eq. (2). Defining the estimation error $\tilde{T}_T = T_T - \hat{T}_T$, the corresponding error dynamics may yield

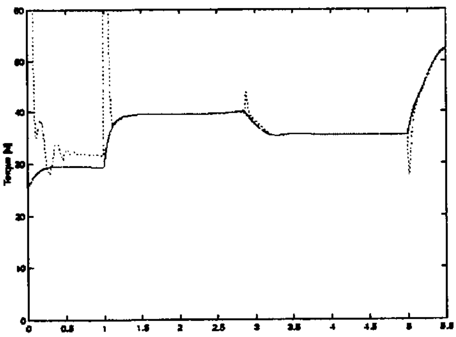

Fig. 4 Estimation of turbine torque

$$\dot{\tilde{T}}_T = \dot{T}_T - \dot{\hat{T}}_T = \dot{T}_T - \dot{z} + km\omega_T^{(m-1)}\dot{\omega}_T$$
$$= \dot{T}_T + km\omega_T^{(m-1)}\left(\dot{\omega}_T - \frac{1}{I_{TG}}\hat{T}_T + \frac{1}{r_n I_{TG}}\hat{T}_C\right)$$

Substituting into Eq. (2) yields

$$\dot{\tilde{T}}_T = km\omega_T^{(m-1)}\left(\frac{1}{I_{TG}}\tilde{T}_T - \frac{1}{r_n I_{TG}}\tilde{T}_C\right) + \dot{T}_T \quad (24)$$

This is a first-order dynamics with respect to the estimation error. Hence, no oscillation is expected. For $k<0$, $m>0$, the error may converge to zero asymptotically. The convergence rate can be made arbitrarily fast by adjusting the design parameters k, m. Note that the steady state error depends on the estimation error of the clutch torque.

Figure 4 shows the simulation result. In the simulation, a constant engine torque was applied. No oscillation, good estimation accuracy and fast convergence may be seen from the figure. The rate of convergence can be made even faster. Short peaks at 1 sec., 2.8 sec. and 5 sec. are due to $\dot{T}_T$, the last term of (24). These times correspond to the moments when the clutch starts the actuation and when the clutch settles into the engaged state. However, note that this peak can be made short enough to be negligible by adjusting the rate of convergence of the estimator.

3.4 Error dynamics analysis

Assuming rather slow variation of the turbine torque, $\dot{T}_T \approx 0$, each error dynamics of the three estimators are repeated here

$$\dot{\tilde{T}}_s = -A\tilde{T}_s - B\tilde{T}_C + D\tilde{T}_L$$
$$\dot{\tilde{T}}_C = \frac{C(t)}{r_F}\tilde{T}_s - C(t)\tilde{T}_C \quad (25)$$
$$\dot{\tilde{T}}_T = -E(t)\tilde{T}_T + \frac{E(t)}{r_n}\tilde{T}_C,$$

where $\tilde{\mu}$ is replaced with $\tilde{T}_C$ by applying Eq. (8). Several system parameters in Eqs. (19), (21) and (24) are integrated into respective capital letters. Note that the third eigenvalue corresponding to $\tilde{T}_T$ dynamics is fixed as $-E(t)$. Thus, let us check that the characteristic equation of the first two equations of (25) is

$$\Delta(s) = s^2 + (A + C(t))s + AC(t) + BC(t)/r_F$$

To avoid the oscillation of the estimator, its discriminant D should be positive

$$D = (A - C(t))^2 - 4BC(t)/r_F > 0$$

This inequality may be easily achieved, e. g. by setting $B = M_{s1}/(M_M r I_M)$ small. The steady state estimation errors can be obtained from (25) as follows

$$\tilde{T}_C = \frac{1}{r_F}\tilde{T}_s = \frac{D}{r_F A + B}\tilde{T}_L \quad (26)$$
$$\tilde{T}_T = \frac{D}{r_n(r_F A + B)}\tilde{T}_L$$

It shows that the accuracy of the load torque information, which is assumed to be known in this paper, is crucial for the accuracy of the estimators in this section. Note also that constants A, B, and D are adjustable by suitable choice of estimator design parameters.

4. Indirect Smooth Acceleration Control

Smooth acceleration shift was defined through the shift analysis by Jeong and Lee. They suggested a desired clutch slip speed trajectory to achieve the so-called indirect acceleration shift. The relation between the slip speed and clutch torque was derived, and it may be used as a controller (Jeong and Lee, 1999). In the previous section, the necessary torque information such as the turbine and shaft torque are provided. Therefore, combining the designed estimators and the suggested slip speed trajectory, the smooth acceleration shift may be accomplished by using only speed sensors. Let us repeat the desired slip speed and clutch torque relation in reference Jeong and Lee (1999) here

$$\dot{\omega}_s^* = \dot{\omega}_s^o + \frac{1}{r_n I_{TG}}(T_T - T_{To}) - \frac{1}{r_F r_n^2 I_{TG}}(T_s - T_{so}) \tag{27}$$

$$\dot{\omega}_s = \frac{1}{r_F I_M} T_s + \frac{1}{r_n I_{TG}} T_T - \beta_M T_{Cn},$$

$$\beta_M = \frac{1}{I_M} + \frac{1}{r_n^2 I_{TG}}, \tag{28}$$

where $\dot{\omega}_s^o$ is a pre-determined smooth curve satisfying the static *shift condition* (Jeong and Lee, 1999). The last two terms of Eq. (27) are inserted to compensate for dynamically varying turbine and axle shaft torques, while T_{To}, T_{so} are the initial torques.

Replacing the torque and friction information in Eqs. (27)-(28) with the estimated ones, the following control law for indirect acceleration shift is proposed

$$u(t) = P_C(t) \tag{29}$$

$$= \frac{\mu_o}{\hat{\mu}} \frac{1}{\beta_M \mu_o n A R}\left(-\dot{\omega}_s^* + \frac{1}{r_F I_M} \hat{T}_s + \frac{1}{r_m I_{TG}} \hat{T}_T\right) + M_P v_s(\tilde{\omega}_s)$$

The first term of (29) is given by substituting the desired trajectory (27) into (28). It represents a feed-forward control for good shift quality. It is based on the perfect information about the system parameters except for friction. These may possibly include measurement errors even though most parameters are mechanical quantities. Although the adoption of a self-zero-approaching curve, instead of w_s^*, below some critical slip speed may complete the shift eventually, the data imperfection may degrade the shift performance. Therefore, the second term of a nonlinear switching function $v_s(\cdot)$ is added to account for the model or parameter error. Its a role is to complete the shift despite the modeling error because the sliding surface is taken as the slip speed error, $\tilde{\omega}_s = \omega_s - \omega_s^o$, based on the sliding mode theory. Note that since our purpose is not the perfect tracking but smooth transients the gain M_P should be chosen relatively small. Generally, the static friction coefficient is larger than the dynamic friction coefficient. This tendency may increase jerk at the end of the shift (Jeong, 1993), which may be inferred from the response at 2.8 sec. of Fig. 2. However, note that the multiplying term $\mu_o/\hat{\mu}$ in the proposed control (29) where μ_o represents a nominal dynamic friction coefficient can cope with this phenomenon.

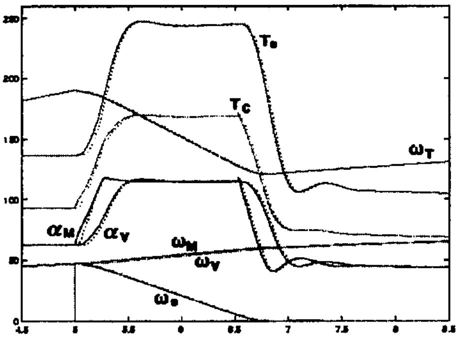

Fig. 5 Indirect smooth acceleration control

Simulation results of the indirect shift are plotted in Fig. 5. Here, the shift completion time of 2 sec. was taken for easy visualization. It is incorporated in the desired smooth curve $\dot{\omega}_s^o$. But, a shorter time period will not change the basic nature of the shift. All solid lines in the figure are responses of the indirect control using Eq. (29) while dotted lines are results using the actual torque information based on Eqs. (27)-(28). The solid line shows smooth vehicle acceleration as well as smooth transmission output acceleration in accordance with our expectation. The actuating clutch torque and turbine speed shows also quite smooth transient. That is, on behalf of the performance of the proposed estimators, indirect smooth acceleration shift using only speed sensors is realized quite well. However, the indirect responses are a little faster than the dotted lines. It is due to an added nonlinear switching term. From about 6.5 sec., a self-zero-approaching curve is adopted and it forces the shift to be completed while satisfying the unpredictable dynamic shift condition (Jeong and Lee, 1999). After 6.8 sec., oscillations occur. It corresponds to the moment when the two clutch faces are stuck together. These are mainly induced by the torsional spring effect of the axle shaft. From that moment on, the system is uncontrollable because the transmitted torque through the clutch are changed into the engaged clutch torque, Eq. (9). In this sense, we can say that a stiffer shaft will show better response after the shift is completed. The presence of the modeling error, similar shift quality may be inferred from the shift analysis results (Jeong and Lee, 1999). Such robustness can be seen from the fact that the relative degree of the system, Eq. (28), is 1 and that the control input is the clutch torque T_{Cn}. However, the effects of the modeling error are not explicitly stated here to save space.

5. Conclusions

An indirect smooth acceleration shift controller that can accomplish the shift process within a designated shift completion time by using speed sensors only is developed. In order to utilize the proposed slip speed trajectory by Jeong and Lee (1999) and to show the applicability of the slip speed-clutch torque relationship, estimators for some necessary information are designed. Three design criteria selected for the estimator design are the prevention of oscillation, rate of convergence and ability to estimate both shifting and engaged states.

Shaft torque was estimated based on the sliding mode theory. Relevant topics such as the linear gain selection, choice of nonlinear switching terms and error dynamics are discussed. A parameter identification method of Friedland is applied for estimating the friction coefficient and turbine torque of the torque converter. These are reduced order nonlinear observers. Additionally, the friction coefficient estimator provides a method for estimating the engaged clutch torque. Steady state error analysis shows that the load torque information is crucial to the accuracy of the proposed estimators. Finally, an indirect control law was proposed. The proposed law can adapt to the time varying friction of the clutch. It consists of two terms. One is a feed-forward control for good shift quality. And the other conducts an auxiliary role to complete the shift in the presence of the modeling error.

Authors acknowledge that the above results are a starting point rather than a final solution. Above all, the load information is assumed to be known. But, it depends on the various operating conditions in actual systems. In practical implementations, signal filtering of noisy and vibrating torque data should also be considered. More specifically, the shaft torque vibrates as it is continually excited by the engine firing pulse. Fortunately, it vibrates at almost fixed frequency corresponding to the axle shaft first torsional mode. It can be easily identified and excluded, e. g. by using a notch filter. Furthermore, the clutch actuator dynamics are not considered in this paper; it can be very important for achieving good shift quality. But, the cost is also a critical factor in mass production. Thus, the actuator delay, so-called the fill time, is inevitable. Also the time gap or overlap between the off-going clutch and on-coming clutch affects the shift quality. This is tightly connected to the presence of inertia phase at the beginning of the shift. To cope with these facts, introduction of the off-going clutch should be deliberately considered. That is, until the on-coming clutch pressure is fully developed the off-going clutch takes part in the elimination of the inertia phase. Applying these methods, one may achieve satisfactory performance, perhaps equal to the results obtained in this paper. All of these considerations could be topics for further research.

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