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Estimation of Turbine Torque Using an Adaptive Nonlinear Observer for Closed-Loop Control of Vehicle Automatic Transmissions

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This paper presents an adaptive sliding observer-based method for the estimation of turbine torque. The estimations of turbine torque can be used in the closed-loop control of vehicle power train for better shift quality. A sliding observer structure including saturation functions and boundary layers is applied to adapt torque converter parameters. The proposed method has been investigated via simulation and laboratory experimental studies. It has been shown via simulation and experiments that the proposed methodology is promising for the estimations of turbine torque since it uses only inexpensive angular velocity measurements and is not sensitive to parametric uncertainties.

Key Words: Sliding observer, Adaptive, Estimation, Automatic Transmission, Turbine Torque

Nomenclature

I_P, I_T, I_S : Moment inertias of pump, turbine, and stator, including fluid inertia
T_P, T_T, T_S : Torque of pump, turbine, and stator,
w_p, w_t, w_s : Angular velocities of pump, turbine, and stator,
R_{po}, R_{to}, R_{so} : Fluid exit radii of pump, turbine, and stator, when $Q>0$
A_{po}, A_{to}, A : Fluid exit areas of pump, turbine and stator, when $Q>0$
a_{po}, a_{to} : Fluid exit blade angles of pump, turbine and stator, when $Q>0$
Q : Flow rate of torque converter,
ρ : Density of fluid,
S_P : $\int_{CV,pump} R\tan(a)\,dL$
S_T : $\int_{CV,turbine} R\tan(a)\,dL$
S_S : $\int_{CV,stator} R\tan(a)\,dL$
L_f : $\int_{CV,total} \frac{1}{A\cos^2 a}(a)\,dL$
P_{Loss} : Power loss,
A, R : Arbitrary area and radius perpendicular to average flow path L,
dL : Infinitesimal of average flow path L,
CV : Control volume.
A_c : Effective clutch area,
R_c : Effective clutch radius,
P : Hydraulic pressure applied to clutch,
μ_s : Static friction coefficient,
μ_d : Dynamic friction coefficient,
ν_{couple} : Coupling speed ratio of torque converter

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1. Introduction

The shift quality of automatic transmissions is very closely concerned with shift torque, and it has been well recognized that nonlinear closed-loop control algorithm can be developed which yields better shift quality provided an accurate information on the shaft torque is available (Hedrick, 1991; Chung, 1993). Torque sensors for rotating shafts can not be used on production vehicles because they are expensive and not swable in the environment of the automotive system. Because the speed sensors of automatic transmission are inexpensive, they are already applied in vehicle control such as the electronic control of automatic transmission, brake control, eg., Anti-skid Brake Systems (ABS), and traction control etc. The operating torque of torque converter depends both on the impeller speed and on the turbine speed.

The steady-state pump and turbine torques of torque converters can be represented as nonlinear functions of pump and turbine angular velocities (Hedrick, 1991; Chung, 1993). Although the nonlinear functions may be obtained from the experimental characteristic curves of a torque converter, they are not useful in the implementation of feedback control of vehicle powertrain since they do provide only the steady-state information and not the dynamic torque information. The estimation of turbine torque is feasible by the use of nonlinear observers which needs only the impeller and turbine speeds of the torque converter.

The theory of adaptive observers is well documented in the literature (Kudva, 1973; Bastin, et al. 1988;Marino, 1990;Teel, 1993). In this paper, we use the adaptive sliding observer technique. It offers easy application to a large class of linearly parametrized nonlinear systems. An adaptive sliding observer-based method for the estimation of the turbine torques of an automatic transmission will be proposed. The goal of the proposed observer-based method is to obtain good estimations of the turbine torques using only angular velocity measurements which can be obtained by inexpensive pickup sensors and a timer.

2. Vehicle/Transmission Model

The vehicle/transmission model has been developed for simulation, and a simplified model for observer design has been obtained from the complete model. Figure 1 shows a schematic of the nonlinear vehicle/transmission model whose states are:

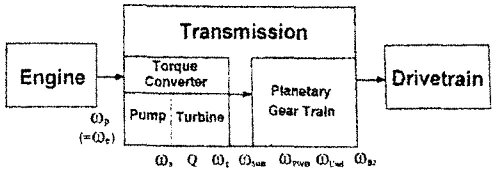

Fig. 1 Two-mass-spring-damping system with disturbance.

- ω_p : angular velocity of torque converter pump shaft
- ω_t : angular velocity of torque converter turbine shaft
- ω_s : angular velocity of torque converter staor shaft
- Q : flow rate of torque converter inside
- ω_{Sun} : angular velocity of Rear & FWD sun gear
- ω_{FWD} : angular velocity of Rear & FWD carrier
- ω_{Und} : angular velocity of Under Drive carrier
- ω_{B2} : angular velocity of brake shaft

The equations of motion for torque converter are written as follows (Chung, 1993):

$$I_P\dot{\omega}_p+\rho S_p\dot{Q}=-\rho Q[(R_{p0}^2\omega_p+R_{p0}\left(\frac{Q}{A_{p0}}\right)\tan a_{p0})-(R_{s0}^2\omega_s+R_{p0}\left(\frac{Q}{A_{p0}}\right)\tan a_{p0})]+T_P$$

$$I_T\dot{\omega}_t+\rho S_t\dot{Q}=-\rho Q[(R_{t0}^2\omega_t+R_{t0}\left(\frac{Q}{A_{t0}}\right)\tan a_{t0})-(R_{p0}^2\omega_s+R_{p0}\left(\frac{Q}{A_{p0}}\right)\tan a_{p0})]+T_T$$

$$I_S\dot{\omega}_s+\rho S_s\dot{Q}=-\rho Q[(R_{s0}^2\omega_s+R_{s0}\left(\frac{Q}{A_{s0}}\right)\tan a_{s0})-(R_{t0}^2\omega_t+R_{t0}\left(\frac{Q}{A_{t0}}\right)\tan a_{t0})]+T_S$$

$$\begin{aligned}L_f\dot{Q}+S_p\dot{\omega}_p+S_T\dot{\omega}_t+S_s\dot{\omega}_s=&R_{p0}^2\omega_p^2+R_{t0}^2\omega_t^2+R_{s0}^2\omega_s^2\\&+R_{p0}^2\omega_p\omega_t+R_{t0}^2\omega_t\omega_s+R_{s0}^2\omega_p\omega_s\\&+\left[\left(\frac{R_{po}}{A_{po}}\right)\tan a_{po}-\left(\frac{R_{so}}{A_{so}}\right)\tan a_{so}\right]Q\omega_p\\&+\left[\left(\frac{R_{to}}{A_{to}}\right)\tan a_{to}-\left(\frac{R_{po}}{A_{po}}\right)\tan a_{so}\right]Q\omega_t\\&+\left[\left(\frac{R_{so}}{A_{so}}\right)\tan a_{so}-\left(\frac{R_{to}}{A_{to}}\right)\tan a_{to}\right]Q\omega_s-\frac{P_{Loss}}{\rho|Q|}\end{aligned}\tag{1}$$

The Simplified torque converter model can be written as follows[Chung, 1993]:

$$I_{Peq}\dot{\omega}_p = T_E - T_{LU} - T_P$$
$$I_{Teq}\dot{\omega}_t = -T_T + T_{LU} - R_1(T_{c1} + T_{ce}) \quad (2)$$

where I_{Peq}, I_{Teq} are equivalent inertias of pump and turbine, respectively, T_E is engine torque, T_{LU} is Lock-up clutch torque, R_1 is gear ratio, T T_{C1}, T_{C2} are the torques on the off-going and on-coming clutches, and T_P, T_T are pump and turbine torques. Depending on the turbine and pump angular velocity ratio, the pump and turbine torques are represented by[Hedrick, 1991; Chung, 1993]:

A high torque transfer phase (converter mode $\omega_t/\omega_p < \nu_{couple}$)

$$T_P = c_1\omega_p^2 + c_2\omega_p\omega_t + c_3\omega_t^2$$
$$T_T = c_4\omega_p^2 + c_5\omega_p\omega_t + c_6\omega_t^2 \quad (3)$$

Fluid coupling mode $(\omega_t/\omega_p \geq \nu_{couple})$

$$T_T = T_P = c_7\omega_p^2 + c_8\omega_p\omega_t + c_9\omega_t^2 \quad (4)$$

The clutch torques T_{C1}, T_{C2} are related to the applied hydraulic pressures by:

$$T_{ci} = A_{ci}R_{ci}P_i\mu_i \quad (5)$$

where the friction coefficient includes both static and dynamic friction as follows:

$$\mu_i = (\mu_s + \mu_d|\omega_{slip}|)\, sign(\omega_{slip})$$
$$\omega_{slip} = \omega_t - \frac{\omega_{cr}}{R_i}$$

where w_{cr} is carrier speed and R_i is gear ratio.

3. An Adaptive Sliding Observer for Nonlinear Systems

The theory of adaptive observers is well documented in the literature (Kudva, 1973, Kreisselmeier, 1977; Bastin, et al. 1988; Teel, 1993). Adaptive observers for linear systems have been developed by Kudva et al.(1973) The first adaptive observer for a nonlinear system was presented by Bastin and Gevers.(1988) It is based on certain coordinate transformations and an auxiliary filter. A simple but restricted observer based on the satisfaction of a certain SPR condition was proposed by Marino(1990). An observer based identifier for nonlinear systems has been discussed by Teel, et al. (1993).

In this paper, we use the adaptive sliding observer technique. It offers easy of application to a large class of linearly parametrized nonlinear systems.

Consider a nonlinear system of the following form:

$$\dot{x}(t) = f(x, \theta, t) + g(x, t)u(t)$$
$$y = Cx$$

where $x \in R^n$, $u \in R^m$, $u \in R$ are state, measurements and input, respectively, $\theta \in R^p$ is a vector of unknown parameters, f and g are vectors of nonlinear functions. It is assumed that $f(x, \theta, t)$ can be parametrized linearly in the unknown parameters as follows:

$$f(x, \theta, t) = F(x, t)\theta$$

where $F(x, t) \in R^{n\times p}$.

Define a Lyapunov function candidate V by:

$$V = \frac{1}{2}\Delta S^T\Delta S + \frac{1}{2}\tilde{\theta}^T R\tilde{\theta}$$

where R is $p\times p$ positive definite matrix and,

$$\Delta S = \left[s_1 - \phi_1 sat\left(\frac{s_1}{\phi_1}\right), \cdots, s_m - \phi_m sat\left(\frac{s_m}{\phi_m}\right)\right]^T$$
$$s_i = \tilde{y}_i = y_i - \hat{y}_i; \quad i = 1, 2, \cdots, m$$
$$\tilde{\theta} = \theta - \hat{\theta}.$$

$\hat{y}_i$ and $\hat{\theta}$ are the estimated values of y_i and θ, respectively. ϕ_i is thickness of boundary layer and $sat(\cdot)$ represents the saturation functions.

$$sat(x) = \begin{cases} 1 & \text{if } x \geq 1 \\ x & \text{if } |x| < 1 \\ -1 & \text{if } x \leq -1 \end{cases}$$

ΔS will approach zero if:

$$\dot{V} = \Delta S^T\dot{\Delta S} + \tilde{\theta}^T R\dot{\tilde{\theta}} < 0 \text{ for } all\ nonzero\ \Delta S. \quad (6)$$

If the inequality (6) holds, then $|s_i| = |\tilde{y}_i| < \phi_i$ is guaranteed.

To guarantee that the inequality (6) holds, the observer of the following structure is used.

$$\dot{\hat{x}} = \hat{F}(\hat{x}, t)\hat{\theta} + \hat{g}(\hat{x}, t)u + K_f(y - \hat{y}) + K\, sat_1 \quad (7)$$

where hat x is the estimated value of x hat F , ~ hat g are our model of F and g Kf , K are n times m observer gain matrices to be determined and saT1 represents the vector of saturation functions.

$$sat_1 = \left[sat\left(\frac{\tilde{y}_i}{\phi_1}\right) \cdots sat\left(\frac{\tilde{y}_m}{\phi_m}\right)\right]^T$$

The resulting error dynamics can be written:

$$\dot{\tilde{x}} = \Delta F\theta + \hat{F}(\tilde{x})\tilde{\theta} + \Delta g\, u - K_f\tilde{y} - K\, sat_1 \quad (8)$$

where

$$\Delta F = F(x, t) - \hat{F}(\tilde{x}, t)$$
$$\Delta g = g(x, t) - \hat{g}(x, t).$$

The values of ΔF and Δg depend both on the modeling complexity and on the observation error. In this paper, we assume that dynamic uncertainties ΔF, Δg are explicitly bounded.

Since

$$\Delta \dot{s}_i = \dot{s}_i - \frac{d}{dt}\left[\phi_i\, sat\left(\frac{s_i}{\phi_i}\right)\right]$$
$$= \begin{cases} C_i\dot{\tilde{x}} & \text{if } |s_i| > \phi_i \\ 0 & \text{if } |s_i| > \phi_i \end{cases}$$

and, under the assumption that θ is slowly varying, i.e., $\dot{\theta} \approx 0$,

$$\dot{\tilde{\theta}} = -\dot{\hat{\theta}},$$

$\dot{V}$ along the trajectory of (8) is written as follows:

$$\dot{V}_{(8)} = \Delta s^T C\{\Delta F\theta + \Delta g\, u - K_f\tilde{y} - Ksat_1\} + \tilde{\theta}^T\{\hat{F}(\tilde{x})^T C^T \Delta S - R\dot{\hat{\theta}}\} \quad (9)$$

If we choose the observer gains as follows;

$$\sigma_i(CK_f) \geq \alpha \cdot \beta$$
$$\sigma_j(K) \geq \gamma u_{\max} + \frac{\eta}{\sigma_{\max}(C)} \quad ; i, j = 1, \cdots, m \quad (10)$$

where $\sigma_j(M)$ is *i-th singular value* of matrix M, η is a positive constants, α, β and γ are constants satisfying:

$$|\Delta F| < \alpha|\tilde{x}|$$
$$|\theta| < \beta$$
$$|\Delta g| < \gamma$$

and prescribe the following adaptation law:

$$\dot{\hat{\theta}} = R^{-1}\hat{F}(\tilde{x})^T C^T \Delta S, \quad (11)$$

then $\dot{V}_{(8)}$ is expressed as follows:

$$\dot{V}_{(8)} = \begin{cases} < -\eta\sum_{i=1}^{m}|\Delta S_i| < 0 & \text{if } |s_i| \geq \phi_i \text{ for } \textit{some } i \\ 0 & \text{if } |s_i| \geq \phi_i \text{ for } \textit{all } i \end{cases} \quad (12)$$

and we can achieve attractiveness of the boundary layers.

It should be noted that convergence of the parameter estimates is not guaranteed, but the parametric error magnitude, $|\tilde{\theta}|$, is bounded and the error bound can be represented, when $|s_i| \leq \phi_i$ for all i, as follows:

$$|\tilde{\theta}| < \frac{\delta_K + \delta_\Delta}{\delta_F} \quad (13)$$

where

$$\delta_K = \max|\Phi^T C(K_f\Phi + K[1])|$$
$$\delta_\Delta = \max|\Phi^T C(\Delta F\ \theta + \Delta g\ u)|$$
$$\delta_F = \min|\Phi^T C\hat{F}(\tilde{x})|$$

$$\Phi = \begin{bmatrix} \phi_1 \text{ or } -\phi_1 \\ \phi_1 \text{ or } -\phi_1 \\ \vdots \\ \phi_m \text{ or } -\phi_m \end{bmatrix}, \quad [1] = \begin{bmatrix} 1 \text{ or } -1 \\ 1 \text{ or } -1 \\ \vdots \\ 1 \text{ or } -1 \end{bmatrix}$$

As can be seen in Eq. (13) the parametric error bound is proportional to ΔF, Δg, Φ and the observer gains, K_f, K. The values of $\Delta F = F(x, t) - \hat{F}(\tilde{x}, t)$, $\Delta g = g(x, t) - \hat{g}(\tilde{x}, t)$ depend on the modelling effort and the computational complexity allowable to the observer. The thickness of the boundary layer, Φ, and the observer gains, K_f, K, are the design parameters. Therefore the observer gains should be chosen to be as small as possible while satisfying the condition (10). In the following simulations and experimental study, it was found that the adaptive observer was able to provide a good approximation of $f(x, \theta, t)$ by adapting the parameter estimates even if the parameter estimates did not converge to the true values. The thickness of the boundary layer, Φ, and the observer gains, K_f, K, were tuned to obtain the acceptable results in the simulations, and the tuned values were successfully used in the experimental studies.

4. Estimation of the Turbine Torque Using an Adaptive Sliding Observer

4.1 Turbine torque observer design

Using the simplified model described in section 2, a nonlinear observer based on adaptive sliding observer theory is designed to estimate the turbine torque T_T. In order to estimate the turbine torque with the engine and turbine speed measurements, $\omega_e(=\omega_p)$, ω_t, firstly, the turbine speed is first estimated using an observer of the following form:

$$\dot{\hat{\omega}}_t = \frac{1}{I_{Teq}}[\hat{T}_T - R_1(\hat{T}_{c1} + \hat{T}_{c2})] + K_f(\omega_t - \hat{\omega}_t)$$

$$+k_1 SAT\left(\frac{\omega_t-\hat{\omega}_t}{\phi}\right) \tag{14}$$

where $\hat{T}_T$, $\hat{T}_{ci}$ are the estimated turbine torque and clutch torque, K_f, k_1 are the gains to be determined, Φ is the boundary layer thickness and $SAT(\cdot)$ represents the saturation functions.

$$SAT(x)=\begin{cases} x & \text{if } |x|<1 \\ 1 & \text{if } |x|\geq 1 \end{cases}.$$

The clutch torques are calculated from the applied pressure command using Eq.(5).

The turbine torque T_T is estimated by the following algebraic equation.

$$\hat{T}_T=\hat{\theta}^T\bar{\omega} \tag{15}$$

where $\hat{\theta}$ and $\bar{\omega}$ omega are the vectors of parameter estimates and measurements. They are represented as follows:

$$\hat{\theta}^T=\begin{cases}[\hat{c}_4\ \hat{c}_5\ \hat{c}_6] & \text{if, } \omega_t/\omega_p<0.9 \\ [\hat{c}_7\ \hat{c}_8\ \hat{c}_9] & \text{if, } \omega_t/\omega_p<0.9\end{cases} \tag{16}$$

$$\bar{\omega}^T=[\omega_p^2\ \omega_p\omega_t\ \omega_t^2] \tag{17}$$

The parameter adaptation law is studied first. The observer error dynamics is given by,

$$\dot{\tilde{\omega}}=\frac{1}{I_{Teq}}[\tilde{T}_T-R_1(\tilde{T}_{c1}+\tilde{T}_{c2})]-K_f\,\tilde{\omega}_t$$
$$-k_1 SAT\left(\frac{\tilde{\omega}_t}{\phi}\right) \tag{18}$$

where $\tilde{x}$ is the difference between the true and estimated value of x.

Define a sliding surface S as the error $\tilde{\omega}_t$ between the measurement ω_t and estimated value $\hat{\omega}_t$,

$$S=\omega_t-\hat{\omega}_t,$$

and ΔS by:

$$\Delta S=S-\phi\cdot SAT\left(\frac{S}{\phi}\right).$$

Consider the Lyapunov function candidate, V,

$$V=\frac{1}{2}\Delta S^2+\frac{1}{2}\tilde{\theta}^T R\tilde{\theta}$$

where R is a 3×3 positive definite matrix. Differentiation of V yields:

$$\dot{V}_{(8)}=\Delta S\,\Delta\dot{S}+\tilde{\theta}^T R\dot{\tilde{\theta}}. \tag{19}$$

Since

$$\Delta\dot{S}=\dot{S} \qquad \text{for } |S|>\phi,$$
$$\dot{\tilde{\theta}}=-\dot{\hat{\theta}} \tag{20}$$

$\dot{V}$ along the trajectory of (18) can be rewritten as follows:

$$\dot{V}_{(18)}=\Delta S\left[-\frac{R_1}{I_{Teq}}(\tilde{T}_{c1}+\tilde{T}_{c2})-K_f S-k_1 SAT\left(\frac{S}{\phi}\right)\right]$$
$$+\tilde{\theta}^T\left(\Delta S\frac{1}{I_{Teq}}\bar{\omega}-R\dot{\hat{\theta}}\right) \tag{21}$$

In order to achieve attractiveness of the boundary layer we choose the sliding observer gain k_1 as follows:

$$k_1=\frac{R_1}{I_{Teq}}\gamma+\eta \tag{22}$$

where

$$\gamma=|T_c|_{\max}=|\tilde{T}_{c1}+\tilde{T}_{c2}|_{\max},$$
$$\eta>0,$$

and prescribe the following adaptation law.

$$\dot{\hat{\theta}}=R^{-1}\Delta S\frac{1}{I_{Teq}}\bar{\omega} \tag{23}$$

Then $\dot{V}_{(18)}$ is expressed as follows:

$$\dot{V}_{(18)}=\begin{cases}<-\eta|\Delta S|-\Delta S\cdot SK_f<0, & |S|>\phi \\ 0, & |S|<\phi\end{cases} \tag{24}$$

It can be shown that the turbine torque estimation error is bounded when $|s|<\phi$ as follows:

$$|\tilde{T}_T|_{\max}\leq R_1|\tilde{T}_c|_{\max}+I_{Teq}(K_f\Phi+k_1)$$
$$=2R_1|\tilde{T}_c|_{\max}+I_{Teq}(K_f\Phi+\eta).$$

The adaptive sliding observer for the estimation of the turbine torque can be summarized as follows:

$$\dot{\hat{\omega}}_t=\frac{1}{I_{Teq}}[\hat{T}_T-R_1(\hat{T}_{c1}+\hat{T}_{c2})]+K_f(\omega_t-\hat{\omega}_t)$$
$$+k_1 SAT\left(\frac{\omega_t-\hat{\omega}_t}{\phi}\right)$$
$$\dot{\hat{\theta}}=R^{-1}\Delta S\frac{1}{I_{Teq}}\bar{\omega}$$
$$\hat{T}_T=\hat{\theta}^T\bar{\omega}.$$

5. Simulations

The adaptive sliding observer is tested with the complete eight state vehicle/transmission model described in Sec. 2. Figure 2 shows the observer simulation results. The "actual" turbine torque is the value calculated by the complete model. The "estimated" one is the value calculated by the adaptive sliding observer based on the simplified model. Simulation has been performed during the

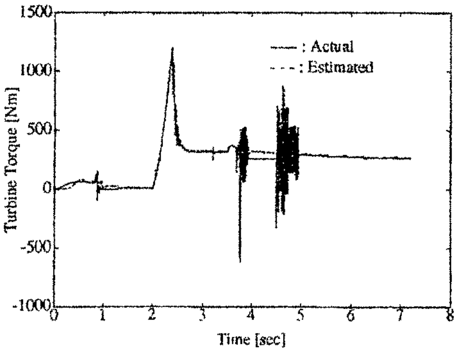

Fig. 2 Comparison of actual and estimated turbine torque.

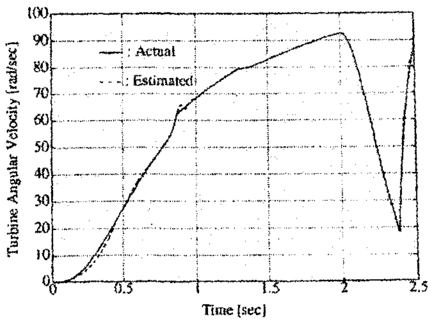

Fig. 3 Comparison of actual and estimated turbine angular velocities.

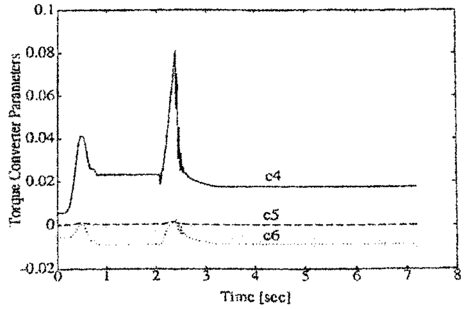

Fig. 4 Adaptation of torque converter parameters c_4, c_5, c_6.

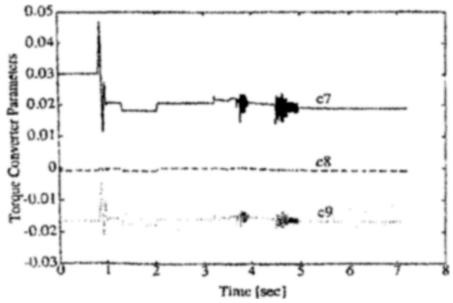

Fig. 5 Adaptation of torque converter parameters c_7, c_8, c_9.

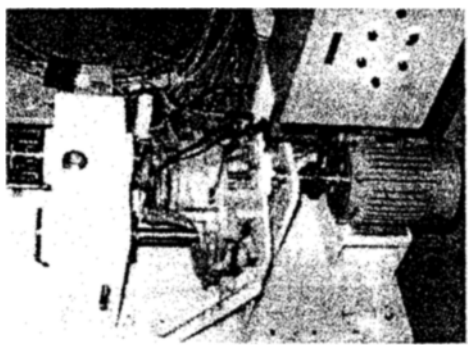

Fig. 6 Experimental test setup.

shift from first gear to second gear. As can be seen, the turbine torque estimation is accurate after 1.5 seconds and during the shift (2.0-2.5 seconds). Large estimation error between 3. 75-4. 5 seconds is due to the lockup of the torque converter. In this simulation, lockup is not considered.

Figure 3 compares the actual and the estimated turbine angular velocity. The estimation error is quite small after 1 second.

Figures 4 and 5 show time histories of adapted torque converter parameters. The parameter adaptation is executed when the turbine velocity estimation error is greater than the boundary layer. The parameters c_4, c_5, c_6 and c_7, c_8, c_9 are adapted only when the turbine and pump velocity ratio ω_t/ω_p is less than or greater than $\nu_{couple}(0.9)$, respectively. Both parameters are constants after 5 second. It should be noted that the fact that both parameters are constants does not imply that the parameter values have converged to correct values.

6. Experimental Studies

Experimental studies using a transmission test setup were conducted to test the proposed turbine torque estimation method. A photograph of the experimental test setup is shown in Fig. 6. Also, a schematic diagram depicting the test setup is shown in Fig. 7. The experimental setup consists of an AC motor, a motor drive, an automatic

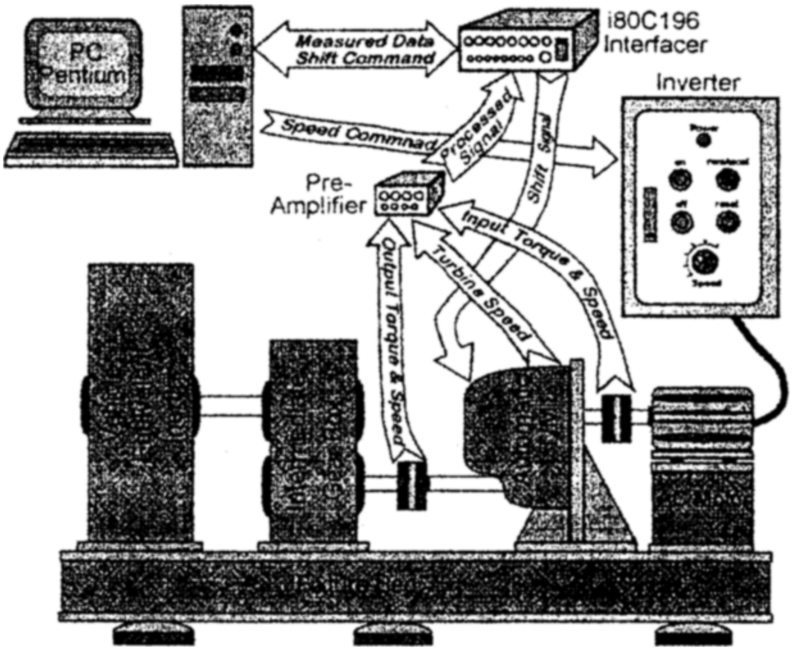

Fig. 7 Schematic diagram of experimental test setup.

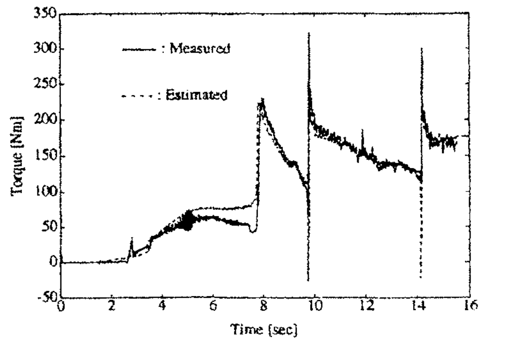

Fig. 8 Comparison of measured and estimated turbine torque.

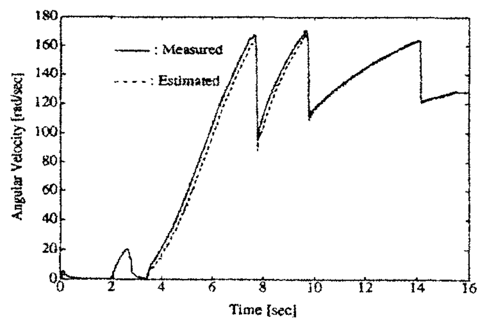

Fig. 9 Comparison of measured and estimated turbine angular velocity.

transmission, a gear box, an inertia equivalent to a automotive vehicle mass, sensors, and an electronic control system. The test setup is equipped with various sensors for measuring the important states of automatic transmission such as shaft torques, pump angular speeds, turbine angular speeds and vehicle speeds.

Figure 8 shows a test result comparing the "measured" and "estimated" turbine torques. The "measured" turbine torque is obtained using strain gauges and a slip ring. The "estimated" turbine torque is the value calculated by the adaptive sliding observer based on the simplified model. Tests have been performed during the shift from first gear to fourth. As can be seen, the turbine torque estimation is close to the measured values. Actual turbine inertia in the test setup is not exactly known and an assumed values are used. The turbine torque estimation is accurate despite the fact that the turbine inertia was not exactly known.

A comparison of the measured and estimated turbine angular velocities is shown in Fig. 9. The measurement is obtained using *a pickup sensor* and *timer* which indicate the *number of teeth* that passed by during a given length of time and the *absolute time*, respectively. It is possible to obtain quite accurate angular velocities using both the number of pulses and the time period between the first and last pulses. As can be seen in Fig. 9, estimation error decreases as time increases.

7. Conclusions

An adaptive sliding observer-based method for the turbine torque estimations of an automatic transmission was presented. It has been recognized that an accurate estimate of the shaft torques can be very useful for closed-loop vehicle powertrain control. The goal of the proposed observer-based method is to obtain good estimates of the turbine torques using only angular velocity measurements which can be obtained by inexpensive pickup sensors and timer.

The performance of the proposed turbine torque observer methodology was investigated via simulations and laboratory experimental studies. The observer gains have been tuned to obtain good observer performance in simulations and the tuned gains have been used successfully in the experimental studies. The simulation and experimental studies have shown that the turbine torque estimation is accurate despite the uncertainty of the turbine inertia. The closed-loop automatic transmission control using the estimated torques is the topic of our current research.

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