2005 American Control Conference
June 8-10, 2005. Portland, OR, USA

Implementation of On-Line Clutch Pressure Estimation for Stepped Automatic Transmissions

Sarawoot Watechagit and Krishnaswamy Srinivasan

Abstract— **On-line estimation of clutch pressures in hydraulically powered stepped automatic transmissions can be useful for on-line diagnostics or development of closed-loop transmission control to improve shift quality. This paper presents the development and implementation of a model-based sliding mode observer to provide on-line estimates of clutch pressures for the clutches involved in gear shifts. The estimation of transmission input torque (torque converter turbine torque), which is needed by the observer for accurate clutch pressure estimation but is not available, is also addressed here and implemented using an adaptive sliding mode observer. The resulting observers are validated via off-line simulation tests, as well as on-line tests at different sampling frequencies on a test vehicle, in order to demonstrate observer performance and establish the feasibility of the approach.**

Index Terms—**Automatic transmission, Clutch pressure, Turbine torque, On-line estimation, Sliding mode observer, Adaptive observer**

I. Introduction

It is known that if clutch pressures or clutch/shaft torques are measured or otherwise estimated in real time in automatic transmissions, closed-loop control algorithms can be designed to improve shift quality [1]-[3]. However, such pressure or torque sensors are usually not used in production transmissions due to sensor cost and reliability, as well as difficulty in sensor installation and maintenance. The development of a nonlinear model-based observer to estimate clutch pressure using speed measurements is presented here, along with results from experimental implementation.

A brief description of the dynamic model for the transmission of interest is presented in section II. The development of a clutch pressure observer using the sliding mode approach is presented in section III, followed by the development of a turbine torque observer to enhance the accuracy of the clutch pressure estimation, in section IV. The performance of the observer is evaluated by simulation and experimental tests, with the experimental results being shown in section V. Conclusions and recommendations for future work are given in section VI.

Manuscript received September 27, 2004. This work was supported by Delphi Automotive Systems. Sarawoot Watechagit is currently a lecturer in the Department of Mechanical Engineering, Mahidol University, Thailand (e-mail: watechagit.2@osu.edu). Krishnaswamy Srinivasan is Chairperson of the Department of Mechanical Engineering, The Ohio State University, Columbus, OH 43210, USA (e-mail: srinivasan.3@osu.edu).

II. Transmission Model

The transmission of interest has four speeds and five clutches involved in the gear shifts. The power-on 2^{nd}–3^{rd} up shift is considered here as an example. A dynamic model of the transmission with a stiff drive shaft is considered here since the proposed observer uses filtered speed signals as inputs, and the effect of a high frequency oscillation from the driveshaft is suppressed. The reader is referred to [5] for more modeling details.

The transmission components modeled here include the torque converter, the transmission mechanical system, which includes the planetary gear train, the transmission shift hydraulic system, and the driveline. The engine dynamic model is not included, experimental engine data being used instead as input to the transmission model. We note also that the torque converter model used in this research is the widely-accepted static model developed by Kotwicki [4], the coefficients in the model being chosen depending on measured torque converter characteristics.

A. Combined Mechanical System and Driveline

The dynamic model of the combined mechanical system and driveline for the transmission for operation in the 2^{nd} and 3^{rd} gears and the 2^{nd}-3^{rd} power-on up shift is presented here. In the 2^{nd} gear, two clutches, the underdrive clutch (UD) and the second clutch (2ND), are fully engaged. In the 3^{rd} gear, the two clutches that are engaged are the UD clutch and the overdrive clutch (OD). Therefore, the 2^{nd}-3^{rd} up shift involves releasing the 2ND clutch and simultaneously applying the OD clutch. Figure 1 shows a simplified free-body diagram for 2^{nd}-3^{rd} up shift. It is known that two phases are involved during any gearshift, namely the torque phase and the inertia phase. Due to space limitations, only the dynamics of the torque phase are considered here.

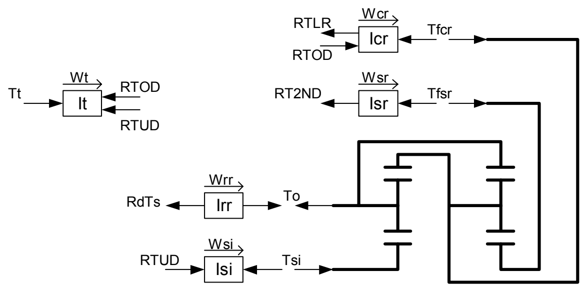

Fig. 1 Free body - stick diagram for the 2nd-3rd up shift

1) 2nd-Gear Dynamic Model

When both the UD clutch and the 2ND clutch are fully engaged, the following equation describes the dynamic behavior.

$$\dot{\omega}_t = \frac{1}{I_{23S}}\left(-R_d d_{11} r(c_1 + c_2(R_d d_{11} r)^2 \omega_t^2) \quad + \quad T_t\right) \tag{1}$$

Here, ω_t is the turbine speed, T_t is the turbine torque, R_d is the final drive ratio, r is the tire radius, c_1 is the rolling friction coefficient, and c_2 is the aerodynamics friction coefficient. $I_{23S} = I_{23} + (R_d d_{11})^2 I_v$, where I_{23} is the lumped inertia of all the moving elements in the planetary gear set in 2nd gear, I_v is the equivalent vehicle inertia which includes the vehicle inertia and the wheel inertia, and d_{11} is the gear ratio in second gear.

2) 2nd - 3rd Up Shift – Torque phase

The torque phase starts when the OD clutch cavity is filled and the clutch starts to exert torque on the transmission. When the torque phase starts, the governing equation is

$$\dot{\omega}_t = \frac{1}{I_{23S}}\left(-R_d d_{11} r(c_1 + c_2(R_d d_{11} r)^2 \omega_t^2) - (d_{21} - 1)T_{OD} + T_t\right)$$

(2)

Here, T_{OD} is the torque exerted by the OD clutch, and d_{21} is a coefficient which depends on the planetary gear train kinematics. During this phase, the torque exerted by the 2ND clutch decreases.

3) 3rd-Gear Dynamic Model

After the 2ND clutch is fully released and the OD clutch locks up, the transmission is in the 3rd gear. The dynamic behavior of the transmission is governed by

$$\dot{\omega}_t = \frac{1}{I_{3S}}\left(-R_d r(c_1 + c_2(R_d r)^2 \omega_t^2) \quad + \quad T_t\right) \tag{3}$$

In this case, $I_{3S} = I_3 + (R_d)^2 I_v$, and I_3 is the lumped inertia of all rotating elements in the planetary gear train in the 3rd gear.

4) Shift hydraulic model

In this work, a simplified model of the hydraulic system, valid for low frequency response and described in [5], is used. For the OD clutch, the clutch pressure is described by,

$$\dot{P}_{c,OD} = C_{xOD}\sqrt{(0.01 \times (DutyCycle) \times P_l - P_{c,OD})} \tag{4}$$

where DutyCycle is the duty cycle command input, P_l is the main line pressure, $P_{C,OD}$ is the OD clutch pressure, and C_{xOD} is a constant calculated from the clutch geometry. The relationship between the clutch pressure and the clutch torque is given by

$$T_{OD} = \begin{cases} \mu(\Delta\omega) P_{C,OD} R_{OD} A_{OD}\,\mathrm{sgn}(\Delta\omega) & \text{if} \quad P_{C,OD} A_{OD} > F_{c\_preload} \\ 0 & \text{if} \quad P_{C,OD} A_{OD} \leq F_{c\_preload} \end{cases} \tag{5}$$

where $\Delta\omega$ represents the slip speed at the OD clutch, $\mu(\Delta\omega)$ is the slip-speed dependent friction coefficient of the clutch plate, R_{OD} is the effective radius of the clutch plate, A_{OD} is the pressurized area of the clutch plate, and $F_{c\_preload}$ is the clutch return spring preload. The model of the 2ND clutch can be described by similar equations. Note that equation (5) neglects the transmitted torque due to viscous shear force, which occurs at the beginning of the torque phase and can be significant at low transmission oil temperature.

III. Clutch Pressure Observer

In this section, a nonlinear sliding mode observer is developed to estimate the clutch pressure. Clutch pressures cannot be estimated in-gear since the clutches are locked up, and clutch torques are below clutch capacities. When the gear shift starts, the estimation of clutch pressure is performed separately for the torque and inertia phases. Due to similarities in the observer design process for the two phases, only the design of the observer for the torque phase is presented here. The turbine torque is assumed to be computed using the static torque converter model and measured turbine and pump speeds, error from torque converter model uncertainty being compensated by the use of the turbine torque observer described in section IV.

A. Observer formulation

Using the model from the previous section, the dynamic behavior of the transmission with a rigid drive line, in second gear, can be described by

$$\begin{bmatrix} \dot{\omega}_t \\ \dot{P}_{c,OD} \end{bmatrix} = \begin{bmatrix} -\frac{1}{I_{23S}} R_d d_{11} r(c_1 + c_2(R_d d_{11} r)^2 \omega_t^2) - \quad d'_{21} P_{c,OD} \\ C_{xOD}\sqrt{0.01 \times DutyCycle \times P_l - P_{c,OD}} \end{bmatrix} + \begin{bmatrix} \frac{1}{I_{23S}} \\ 0 \end{bmatrix} T_t \tag{6}$$

where $\quad d'_{21} = (d_{21} - 1)\mu(\omega_t - \omega_{Cr}) R_{OD} A_{COD}\,\mathrm{sgn}(\omega_t - \omega_{Cr}) \qquad (7)$

We follow directly the sliding mode observer designprocedure suggested by [6]. From equation (6), letting x_1 and x_2 represent ω_t and $P_{c,OD}$ respectively, we can write

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} = \begin{bmatrix} f_1'(\mathbf{x}) \\ f_2(\mathbf{x}) \end{bmatrix} \tag{8}$$

where $\mathbf{x} = [x_1 \quad x_2]^T = [\omega_t \quad P_{c,OD}]^T$ (9)

$$f_1'(\mathbf{x}) = -\frac{1}{I_{23S}} R_d d_{11} r (c_1 + c_2 (R_d d_{11} r)^2 \omega_t^2) - \quad d_{21}' P_{c,OD} + \quad c_t T_t \tag{10}$$

$$f_2(\mathbf{x}) = C_{xOD} \sqrt{0.01 \times DutyCycle \times P_l - P_{c,OD}} \tag{11}$$

$$c_t = 1 / I_{23S} \tag{12}$$

The measured signal in this case is x_1 (ω_t). We assume the observer equations are in the following form.

$$\begin{aligned} \dot{\hat{x}}_1 &= \hat{f}_1'(\hat{\mathbf{x}}) - k_{1tS}\,\mathrm{sgn}(\tilde{x}_1) \\ \dot{\hat{x}}_2 &= \hat{f}_2(\hat{\mathbf{x}}) - k_{2tS}\,\mathrm{sgn}(\tilde{x}_1) \end{aligned} \tag{13}$$

where T_t is assumed to computed from the static torque converter model. Also, $\tilde{x}_i = \hat{x}_i - x_i$, and k_{itS}, i=1,2 are switching gains. The sliding surface is given by

$$S_1 = \tilde{x}_1 = \hat{x}_1 - x_1 = 0 \tag{14}$$

and the sliding condition by

$$S_1 \dot{S}_1 < 0 \tag{15}$$

or

$$S_1 \dot{S}_1 = \tilde{x}_1 [\Delta f_1' - k_{1tS}\,\mathrm{sgn}(\tilde{x}_1)] < 0 \tag{16}$$

Here, the term $\Delta f_1' = \hat{f}_1'(\hat{\mathbf{x}}) - f_1'(\mathbf{x})$ represents the modeling error, which is assumed to be bounded, i.e.,

$$|\Delta f_1'| < \alpha_1(\mathbf{x}, t) > 0 \tag{17}$$

$\Delta f_1'$ in equation (17) includes the uncertainty due to the turbine torque input and the friction coefficient μ embedded in the parameter d_{21}'. The switching gain, k_{1tS} must be chosen to be large enough to satisfy (17). In particular, based on equation (16), the sliding surface is attractive if

$$k_{1tS} > \alpha_1(\mathbf{x}, t) > |\Delta f_1'| \tag{18}$$

For this particular system, when the sliding surface is reached, the observer becomes a reduced order observer of order 1. In particular, the estimation error dynamics computed by using equations (8) and (13) are given by

$$\begin{aligned} \dot{\tilde{x}}_1 &= \Delta f_1' - k_{1tS}\,\mathrm{sgn}(\tilde{x}_1) \\ \dot{\tilde{x}}_2 &= \Delta f_2 - k_{2tS}\,\mathrm{sgn}(\tilde{x}_1) \end{aligned} \tag{19}$$

Using the equivalent control concept, when the sliding surface is reached, we have

$$\hat{x}_1 - x_1 \approx 0 \quad \to \quad \dot{\tilde{x}}_1 \approx 0, \quad \tilde{x}_1 \approx 0 \tag{20}$$

Therefore, by substituting equation (20) into equation (19), the switching terms can be approximated by

$$\mathrm{sgn}(\tilde{x}_1) = \frac{\Delta f_1'}{k_{1tS}} \tag{21}$$

The equivalent dynamics on the reduced order manifold are described by the error dynamics of $\tilde{x}_2$, where the switching terms are now substituted using equation (21) . Specifically, we have

$$\dot{\tilde{x}}_2 = \Delta f_2 - k_{2tS} \frac{\Delta f_1'}{k_{1tS}} \tag{22}$$

When the transmission is in 2nd gear, the dynamic behavior of the transmission can be described by equation (6) with the OD clutch pressure set equal to zero, $P_{c,OD} = 0$. The observer in this case is of same form as equation (13), with the dynamics of $\hat{x}_2$ being ignored. In this case, the gain k_{1tS} selected in the next section is used to maintain the system on the sliding surface.

B. Observer gain selection

The value of the gain k_{1tS} depends on estimated upper bound for the uncertainty as described by equation (18). Its value will guarantee the attraction of the sliding surface, S_1. The equivalent dynamics on the reduced order manifold described by equation (22) will then be used to find the value of the gain k_{2tS}.

<u>Gain k_{1tS}</u>

We compute $\Delta f_1'(\mathbf{x})$ by considering error in the assumed clutch friction coefficient, measurement error for turbine speed, estimation error for turbine torque, and clutch pressure in the expression for $f_1'(\mathbf{x})$ given by equation (10). We then have,

$$\Delta f_1'(\mathbf{x}) = -\frac{1}{I_{23S}} (2 c_2 (R_d d_{11} r)^2 \bar{\omega}_t) \tilde{\omega}_t - \frac{d_{21}'}{I_{23S}} \tilde{P}_{c,OD} - \frac{\bar{P}_{c,OD}}{I_{23S}} \frac{\partial d_{21}'}{\partial \mu} + c_t \tilde{T}_t \tag{23}$$

At this point, we ignore the effect of error in the friction coefficient μ, and in the computed turbine torque input. Therefore, equation (23) becomes

$$\Delta f_1'(x) = -\frac{1}{I_{23S}} (2 c_2 (R_d d_{11} r)^2 \bar{\omega}_t) \tilde{\omega}_t - \frac{d_{21}'}{I_{23S}} \tilde{P}_{c,OD} \tag{24}$$

We assume that the maximum error bound for the OD clutch pressure is 85 psi or 0.58 MPa., and for the turbine speed is 30 rad/sec. Following conventional methods for evaluating uncertainty bounds as presented in [1], we get

$$|\Delta f_1| \approx 20.61 \quad rad/\sec^2 \quad \to \quad k_1 \approx 30 \quad rad/\sec^2 \tag{25}$$

<u>Gain k_{2tS}</u>

Selection of the gain k_{2tS} affects the estimation error dynamics of the clutch pressure, $P_{C,OD}$. Based on equation (22) the error dynamics of $P_{C,OD}$ can be written as

$$\dot{\tilde{P}}_{c,OD} = \Delta f_2 - k_{2tS} \frac{\Delta f_1'}{k_{1tS}} \tag{26}$$

In this case, Δf_2 represents the error in the shift hydraulic system model. We consider two limiting cases in designing the gain k_{2tS}, viz. when the clutch pressure is near zero, and when the clutch pressure is near the line pressure input. Since both cases give similar results, only the former is shown here.

The case when the clutch pressure is near zero represents the moment when the OD clutch cavity is almost full but its pressure is still low. Linearizing $f_2(\mathbf{x})$ around this point, substituting for $\Delta f_1'$ from equation (24), and assuming that $\tilde{\omega}_t \approx 0$, equation (26) is reduced to

$$\dot{\tilde{P}}_{c,OD} = \left[\frac{-0.5 \times C_{xOD}}{\sqrt{\left(0.01 \times \overline{DutyCycle} \times \overline{P}_l\right)}} + \frac{k_{2tS}}{k_{1tS}} \frac{d_{21}'}{I_{23S}} \right] \tilde{P}_{c,OD} \quad (27)$$

Note that terms with bars represent nominal values of corresponding variables and parameters, i.e. $\overline{P}_{c,OD} = 0$, $\overline{P}_l$ = line pressure for 2nd-3rd gearshift, and $\overline{DutyCycle}$ is at 100%. In can be seen that the dynamics of the estimation error for $P_{c,OD}$ are stable if the coefficient of $\tilde{P}_{c,OD}$ is less than zero. The gain k_{2tS} can be selected to ensure satisfactory response of $\tilde{P}_{c,OD}$. In our application, we need the estimation error to go to zero relatively fast as compared to the shift duration, especially the duration of the torque phase which is in the range of 100-150 millisecond. Therefore, a 10 millisecond decay time for this estimation error is acceptable. Therefore, we place the eigenvalue of equation (27) at –100 rad/sec. After substituting numerical values, this choice of eigenvalue gives

$$\left[\frac{-0.5 \times C_x}{\sqrt{\left(0.01 \times \overline{DutyCycle} \times \overline{P}_l\right)}} + \frac{k_{2tS} d_{21}'}{k_{1tS} I_{23S}} \right] = -100 \rightarrow \quad k_{2tS} = -8.37\ Pa/\sec \quad (28)$$

The gain k_{2tS} chosen above is for the case when the clutch pressure is nearly zero. As the clutch pressure gets higher, this choice of gain is still valid, as shown in [5].

IV. Adaptive Torque Converter Model for Turbine Torque Estimation

In the design of the clutch pressure estimator thus far, the turbine torque is assumed to be computed using the fixed static torque converter model. However, it is well known that torque converter characteristics do change with operating conditions, for instance, with temperature change [5]. Therefore, the Kotwicki model with constant coefficients cannot be used to estimate turbine torque accurately. Error in estimating the turbine torque induces error in the clutch pressure estimation.

In this section, we use an adaptive sliding mode observer to improve accuracy of the turbine torque estimation, by adjusting the parameters in the Kotwicki model. The adaptation scheme presented here follows the work presented in [7]. The parameter adaptation is implemented only during in-gear operation as we seek to compensate for adjusting slowly varying parameters, such as change of transmission fluid temperature. Moreover, use of the adaptation scheme during the gear shift will conflict with the clutch pressure estimation scheme presented previously since both schemes rely on error between speed signals estimated by the models and measurements to drive the estimation. The torque converter model parameters are held constant at their last adapted values once the gear shift starts, and clutch pressure estimation is then activated.

The following derivation shows the development of the torque converter parameter adaptation mechanism for operation in the 2nd gear. The same approach can be easily applied to other gears.

Consider the dynamic behavior of the transmission with a rigid shaft in second gear, described by the following equation.

$$\dot{\omega}_t = -\frac{1}{I_{23S}} R_d d_{11} r (c_1 + c_2 (R_d d_{11} r)^2 \omega_t^2) + \frac{1}{I_{23S}} T_t \quad (29)$$

With some modification of the 2-3 up shift torque phase estimation, the observer for the second gear is of the form,

$$\dot{\hat{\omega}}_t = -\frac{1}{I_{23S}} R_d d_{11} r (c_1 + c_2 (R_d d_{11} r)^2 \hat{\omega}_t^2) + \frac{1}{I_{23S}} \hat{T}_t - k_{1tS} \operatorname{sgn}(\tilde{\omega}_t) \quad (30)$$

Here, based on the torque converter model, the turbine torque can be parameterized as follows

$$\hat{T}_t = \hat{\theta}^T \Omega \quad (31)$$

where

$$\hat{\theta}^T = \begin{cases} [\hat{c}_{1tc} \quad \hat{c}_{2tc} \quad \hat{c}_{3tc}]; & \text{if } \textit{speed ratio} \ \left(\omega_t / \omega_p \right) < 0.9 \\ [\hat{c}_{4tc} \quad \hat{c}_{5tc} \quad \hat{c}_{6tc}]; & \text{if } \textit{speed ratio} \ \left(\omega_t / \omega_p \right) \geq 0.9 \end{cases} \quad (32)$$

$$\Omega^T = \left[\omega_p^2 \quad \omega_p \omega_t \quad \omega_t^2 \right] \quad (33)$$

The use of the adaptation scheme developed here eliminates switching between two operating modes at different speed ratios. Therefore, equation (32) is reduced to

$$\hat{\theta}^T = [\hat{c}_{1ta} \quad \hat{c}_{2ta} \quad \hat{c}_{3ta}] \quad \text{for all } \omega_t / \omega_p \quad (34)$$

Following the same derivation as presented for the 2-3 up shift torque phase observer development, the following error dynamics can be derived.

$$\dot{\tilde{\omega}}_t = -\frac{1}{I_{23S}}(2c_2(R_d d_{11} r)^2 \bar{\omega}_t)\tilde{\omega}_t + \frac{1}{I_{23S}}\tilde{T}_t - k_{1tS}\,\mathrm{sgn}(\tilde{\omega}_t) \quad (35)$$

The sliding surface in the case is defined by,

$$S_1 = \tilde{\omega}_t = \hat{\omega}_t - \omega_t = 0 \quad (36)$$

The following Lyapunov function candidate is assumed,

$$V = \frac{1}{2}S^T S + \frac{1}{2}\tilde{\theta}^T R_a \tilde{\theta} \quad (37)$$

In this case, R is a 3×3 positive definite matrix to be selected. We then have

$$\begin{aligned}\dot{V} &= \tilde{\omega}_t\dot{\tilde{\omega}}_t + \tilde{\theta}^T R_a \dot{\tilde{\theta}} \\ &= -\frac{1}{I_{23S}}(2c_2(R_d d_{11} r)^2 \bar{\omega}_t)\tilde{\omega}_t^2 + \ldots \\ &\quad \frac{1}{I_{23S}}\hat{\theta}^T \Delta\Omega\tilde{\omega}_t - k_{1tS}\,\mathrm{sgn}(\tilde{\omega}_t) + \tilde{\theta}^T\left(\frac{1}{I_{23S}}\tilde{\omega}_t\Omega + R_a\dot{\tilde{\theta}}\right)\end{aligned} \quad (38)$$

Here, we use

$$\tilde{T}_t = \hat{\theta}^T \Delta\Omega + \tilde{\theta}^T\Omega \quad (39)$$

and also assume that the parameter θ is slowly varying, i.e. $\dot{\tilde{\theta}} = \dot{\hat{\theta}} - \dot{\theta} = \dot{\hat{\theta}}$. The switching gain k_{1tS} is already selected from the 2-3 up shift torque phase estimation development, and will not be repeated here. From equation (38), the stability requirement of the estimation scheme gives the following adaptive law.

$$\dot{\hat{\theta}} = R_a^{-1}\frac{1}{I_{23S}}\tilde{\omega}_t\Omega \quad (40)$$

V. Implementation Results

The test vehicle used in this research is equipped with all the standard sensors, viz. transmission input/output speed sensors, engine speed sensor etc, as well as pressure transducers to measure the clutch pressures. The pressure transducers are installed only for validation purposes, and are not installed on production vehicles. The personal computer used here is equipped with an Intel Pentium III 700 MHz processor and 256 MB of RAM. The estimator is programmed using Simulink®. Data needed to implement the observer on-line are clutch pressures, engine speed, transmission input speed, transmission output speed, and clutch command duty cycle.

The choice of the sampling frequency generally affects the performance of the sliding mode observer [5]. The results shown in this section are based on implementation using two sampling frequencies. A high sampling frequency of 1 kHz, which is the highest sampling frequency for real time implementation achievable for the designated computer, is used to show the performance of the resulting observer. A low sampling frequency of 64 Hz, which is the frequency used by the transmission control unit on the production vehicle, is also used in order to test the feasibility of implementing the resulting observer on current production hardware. All data input into the observer are captured by the transmission control unit at a sampling frequency of 128 Hz.

We note also that the observer gains used in most of the results shown here are fine-tuned during the implementation. The switching gain, k_{1tS}, is at the value noted in section III. However, the switching gain for the clutch pressure error dynamics, k_{2tS}, had to be reduced by a factor of 0.01.

Finally, open-loop estimation for the off-going clutch is also used in order to help set up proper initial conditions for the start of the torque phase and the accuracy of estimation of the 2ND clutch pressure. During the clutch fill phase that precedes the torque phase, there is no speed change and hence no feedback to improve the pressure estimation accuracy.

Figures 2 and 3 show estimation results for the case of wide throttle acceleration, throttle setting $\approx 85-90\%$, with the use of the adaptive torque converter model as well as the open-loop off-going clutch pressure estimation in addition to the sliding mode observer. A sampling frequency of 1 kHz is used in this case. In each figure, a so-called "flag" is used to indicate the gear shift phase, viz. torque phase, inertia phase, or in-gear. The observer is able to predict the clutch pressures reasonably well for both the OD clutch pressure, shown in Figure 2, and the 2ND clutch pressure, shown in Figure 3. The observer seems to have more difficulty estimating the 2ND clutch pressure at low pressure levels.

Results for clutch pressure estimation using the adapted and static torque converter models show some improvement in accuracy of pressure estimation with the adaptation of the torque converter model. See [5] for results.

Figure 4 shows estimation results for low throttle acceleration with a throttle opening $\approx 35-45\%$. A sampling frequency of 64 Hz is used for this case. It can be seen that the observer is able to predict clutch pressure reasonably well even at the low sampling frequency used here. The interested reader may consult [5] for results from other tests involving torque converter model adaptation.

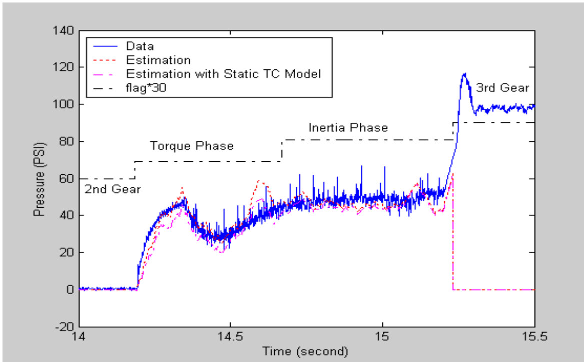

Fig. 2 OD clutch pressure estimation for wide throttle acceleration - High sampling frequency

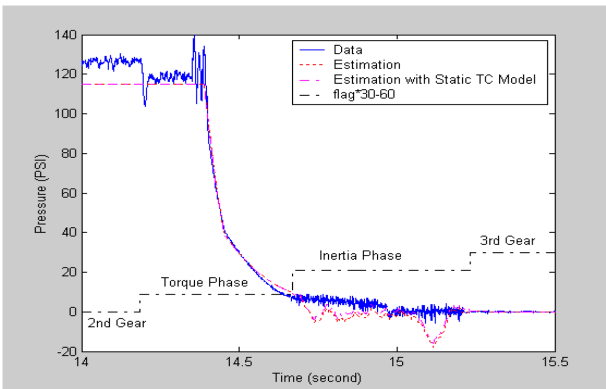

Fig. 3 2ND clutch pressure estimation for wide throttle acceleration – High sampling frequency

VI. Conclusions

A model-based adaptive sliding mode observer for the simultaneous estimation of clutch pressure and transmission input torque, or turbine torque, for an automatic transmission is presented here. On-line implementation results using the developed observers are presented for two sampling frequencies – a high sampling frequency of 1 kHz, to demonstrate the performance of the observer when there is less of a computational limitation imposed on the implementation, and a low sampling frequency of 64 Hz, to demonstrate the feasibility of using the observer on current production vehicles.

Results are presented here for the 2^{nd} – 3^{rd} gear up shift for a four-speed transmission. An open loop estimator is used to determine better initial conditions for closed loop estimation in the torque phase of the up shift. The implementation results show that the observer is able to predict clutch pressures as well as the turbine torque with a reasonable degree of accuracy. Evaluation and refinement of the observer design for a greater variety of gear shifts and for a wider range of transmission operating conditions is warranted. In

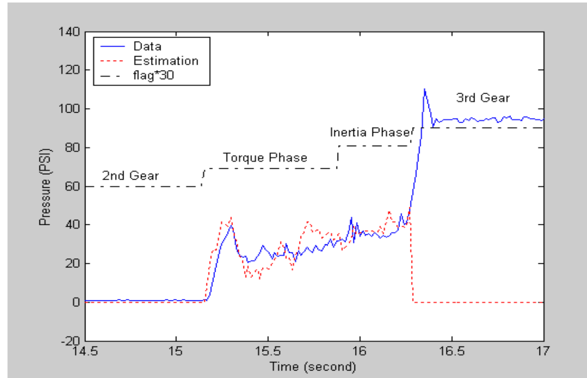

Fig. 4 OD clutch pressure estimation for low throttle acceleration – Low sampling frequency

particular, the models used need to better accommodate variable road load, torque converter clutch application, driveline compliance, and clutch drag. More flexibility in observer pole placement is also essential for the resulting clutch pressure estimates to be useful for shift control.

References

[1] Ibamoto, M., Kuroiwa, H., Minowa, T., Sata, K., and Tsuchiya, T., "Development of Smooth Shift Control System with Output Torque Estimation," *SAE Technical Paper*, No. 950900, 1995

[2] Lin, W. C., "Shaft-Torque Estimation for Closed-Loop Transmission Clutch Control," Proceedings of the ASME Winter Annual Meeting, *ASME Dynamic Systems and Control Division, DSC* - Vol. 44, pp. 293-298, 1992

[3] Masmoudi, R. A., and Hedrick, J. K., "Estimation of Vehicle Shaft Torque Using Nonlinear Observers," *ASME Journal of Dynamic Systems, Measurement, and Control*, Vol. 114, pp. 394-400, September, 1992

[4] Kotwicki, A. J., "Dynamic Models for Torque Converter Equipped Vehicles," *SAE Technical Paper*, No. 820393, 1982

[5] Watechagit, S., "Modeling and Estimation for Stepped Automatic Transmission with Cluth-to-Clutch Shift Technology," *Ph.D. Dissertation*, Department of Mechanical Engineering, The Ohio State University, 2003

[6] Slotine, J. -J., Hedrick, J. K., and Misawa, E. A., "On Sliding Observers for Nonlinear Systems," *ASME Journal of Dynamic Systems, Measurement, and Control*, Vol. 109, pp. 245-252, 1987

[7] Yi, K., Shin, B.-K., and Lee, K.-I., "Estimation of Turbine Torque of Automatic Transmissions Using Nonlinear Observers," *ASME Journal of Dynamic Systems, Measurement, and Control*, Vol. 122, pp. 276-283, June, 2000