Proceedings of the
47th IEEE Conference on Decision and Control
Cancun, Mexico, Dec. 9-11, 2008

A Reduced-Order Nonlinear Clutch Pressure Observer for Automatic Transmission Using ISS

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***Abstract*— For a new kind of automatic transmissions using proportional pressure valves to control the clutches directly, a clutch pressure observer design method is suggested in the concept of input-to-state stability (ISS). Model uncertainties including steady state errors and unmodelled dynamics are considered as additive disturbance inputs and the observer is designed such that the error dynamics is input-to-state stable. Lookup tables, which are widely used to represent complex nonlinear characteristics of engine systems, appear in their original form in the designed reduced-order observer. Finally, the designed pressure observer is tested on an AMESim powertrain simulation model.**

***Index Terms*— Automatic transmission, clutch pressure, pressure estimation, nonlinear observer, input-to-state stable (ISS)**

I. INTRODUCTION

Automotive transmissions transfer the engine torque to the vehicle with desired ratios. To improve fuel economy, reduce emission and enhance driving performance, many new technologies have been introduced in the transmission area in recent years, such as Dual Clutch Transmission (DCT) and new Automatic Transmissions (AT) controlling clutches independently [1]. Furthermore, smart proportional valves with large flow rate are developed for direct clutch pressure control, without using the pilot duty solenoid valve [2].

In both Dual Clutch Transmissions and new Automatic Transmissions, the change of the speed ratio can be regarded as a process of one clutch to be engaged while another being disengaged, namely, clutch-to-clutch shifts. Although in current production automatic transmissions, the clutch pressure is usually controlled by feed-forward control with gain-scheduling [3], some studies, such as [4], [5], [6], [7], [8], have tried to construct the model-based control law for clutch control. If state feedback is involved in clutch control systems [7], [8], the clutch cylinder pressure is necessary to be known for feedback control law. The sensors for measuring the clutch pressure, however, are seldom used because of the cost and durability. Hence, as already pointed out in [9], it is necessary to estimate the clutch pressure/torque by observers for the improvement of clutch control systems.

There have been some studies for the estimation of transmission shaft torque and clutch pressure, mostly using sliding mode observer design techniques [10]. For example, a sliding mode observer is designed to estimate the torque of automotive drive shaft [11]. In [12], an adaptive sliding mode algorithm is proposed to estimate the turbine torque of torque converter. Furthermore, [9] uses the sliding mode method to estimate clutch pressures in hydraulic powered stepped automatic transmissions and the same algorithm is extended in [13] to estimate the clutch pressures and transmission output shaft torque simultaneously.

Because of the complex nonlinearities in automotive powertrain, such as the engine's speed-torque relationship and the characteristics of torque converter, it is very hard to model the whole dynamics with physical principles. Lookup tables, which are obtained from large numbers of experiments in the steady state, are widely used to describe those strongly nonlinear characteristics. Hence, there inherently exist model uncertainties including steady state errors and unmodelled dynamics. In this paper, by using rotational speeds as measured output, a reduced-order clutch pressure estimation method is proposed in the concept of input-to-state stability (ISS) [14], [15], [16]. Model uncertainties are considered as additive disturbance inputs and the observer is designed such that the error dynamics is input-to-state stable. Moreover, lookup tables for complex nonlinear characteristics of engine systems appear in the designed reduced-order observer in their original form.

The rest of the paper is organized as follows. In Section II, a dynamic model for the transmission of interest is derived for shift inertia phase. The development of a clutch pressure observer is designed based on ISS in Section III. In Section IV, the proposed observer is tested on a complete powertrain simulation model.

II. CLUTCH SYSTEM MODELING AND PROBLEM STATEMENT

For convenience, the powertrain to be considered here is a passenger vehicle with a two-speed automatic transmission. It is represented schematically in Fig. 1. As shown in the Fig. 1, a planetary gear set is adopted as the shift gear, and two clutches are used as the actuators. In this study, two proportional pressure control valves are adopted to control

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The work is supported by the National Science Fund of China for Distinguished Young Scholar (No. 60725311) and the "985 Project" of Jilin University.

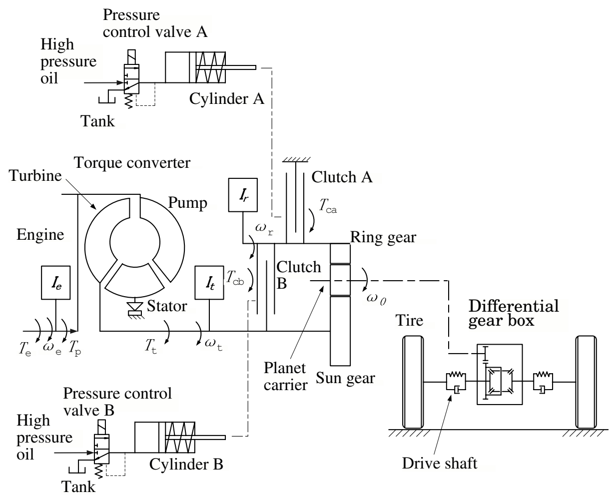

Fig. 1. Schematic graph of Automatic Transmission

the two clutches respectively. When clutch A is engaged and clutch B disengaged, the powertrain operate in 1^{st} gear, the speed ratio is

$$i_1 = 1 + \frac{1}{\gamma}\,, \tag{1}$$

where γ is the ratio of the sun gear's teeth number Z_s to that of the ring gear Z_r,

$$\gamma = \frac{Z_s}{Z_r}\,. \tag{2}$$

While clutch A is disengaged and clutch B engaged, the vehicle is driven in 2^{nd} gear with the speed ratio

$$i_2 = 1. \tag{3}$$

Normally, a typical gear shift process includes the torque phase and the inertia phase [17]. For example, at the beginning of power-on 1^{st} to 2^{nd} up shift, clutch B starts to exert torque on the transmission, and at the same time, the torque transmitted through clutch A begins to decrease, namely the off-going gear torque phase. When most torque is taken over by clutch B, the inertia phase begins, during which the pressure of clutch B is controlled so that the speed difference of clutch B can be reduced to zero, *i.e.*, the clutch engagement.

During shift process of this kind of automatic transmissions, the oncoming and offgoing clutches are controller by the proportional pressure control valves independently, thus the shift timing and coordination of the clutches are guaranteed. Because the sensors for measuring the pressure of clutch cylinder are not used, the clutch pressure estimation is necessary for optimizing the transmission operation [9].

A. *Clutch system modeling*

The 1^{st} to 2^{nd} up shift is considered here as an example, and the pressure observer is designed to estimate clutch pressure during the shift inertia phase. The driveline dynamic equations of the 1^{st} to 2^{nd} up shift inertia phase can be described by the following equation [8]

$$\dot{\omega}_t = C_{11}T_t + C_{13}T_{cb} + C_{14}T_{ve}, \tag{4a}$$

$$\dot{\Delta\omega} = (C_{11} - C_{21})T_t + (C_{13} - C_{23})T_{cb} + (C_{14} - C_{24})T_{ve}, \tag{4b}$$

where ω_t is the turbine speed, $\Delta\omega$ is the speed difference between turbine and ring gear, *i.e.*, speed difference of clutch B, T_t is the turbine torque of the torque converter, T_{cb} is the torque transferred by cylinder B, T_{ve} is the equivalent resistant torque delivered from tire to drive shaft and C_{ij} is the constant coefficients decided by inertia moments of vehicle and transmission shafts.

Turbine torque T_t can be estimated by the steady-state characteristics of torque converter,

$$T_t = \, t(\lambda)\, C(\lambda)\omega_e^2 \tag{5}$$

where $C(\lambda)$ denotes the capacity factor of the torque converter, $t(\lambda)$ is the torque ratio, ω_e is engine speed and λ is the speed ratio defined as

$$\lambda = \frac{\omega_t}{\omega_e} \tag{6}$$

On the other hand, the transferred torque T_{cb} during clutch slipping is determined by the cylinder's pressure. If the force of return spring is treated as a constant, the relationship between clutch torque and cylinder pressure can be described as

$$T_{cb} = \mu(\Delta\omega)\, RN \cdot (A\, p_{cb} - F_s) \tag{7}$$

where μ is friction coefficient of clutch plates, R is effective radius of push force acted on the plates of clutch B, N is plates number of clutch B, A is piston area of clutch B, p_{cb} is pressure of cylinder B, F_s is return spring force of clutch B.

The cylinder's pressure is determined by the input current of the proportional pressure control valve. The dynamics of the proportional valve can be simplified as a first-order system [5],

$$\tau_{cv}\dot{p}_{cb} = -p_{cb} + K_{cv}\, i_b \tag{8}$$

where τ_{cv} is the time constant of the valve B, K_{cv} is the gain of the valve B, and i_b is the electric current of the valve B.

Moreover, if the torsion dynamics of drive shaft, tire slip and road grade are ignored, the resistant torque delivered from tire to drive shaft T_{ve} can be calculated as

$$T_{ve} = \frac{T_w}{i_{df}} + \frac{C_A R_w^3}{i_{df}^3}\omega_0^2 \tag{9}$$

where T_w denotes the rolling resistant moment of tires, R_w is the radius of tire, i_{df} is the gear ratio of the differential gear box, ω_0 is output speed of transmission, C_A is a constant coefficient.

B. Estimation problem statement

Given the above equations and select turbine speed ω_t, speed difference $\Delta\omega$, and pressure of cylinder B p_{cb} as the state variables x_1, x_2, x_3 respectively, the inertia phase of 1^{st} to 2^{nd} gear up shift process can be described in state space form:

$$\dot{x}_1 = C_{13}\mu(x_2)RNA\, x_3 + f_1(x_1, x_2), \tag{10a}$$
$$\dot{x}_2 = (C_{13} - C_{23})\mu(x_2)\, RNA\, x_3 + f_2(x_1, x_2), \tag{10b}$$
$$\dot{x}_3 = -\frac{1}{\tau_{cv}} x_3 + \frac{K_{cv}}{\tau_{cv}} u, \tag{10c}$$

where $u = i_b$ is the control input and

$$f_1(x_1, x_2) = C_{11}T_t + C_{14}T_{ve} - C_{13}\mu(x_2)RNF_s, \tag{11a}$$
$$\begin{aligned} f_2(x_1, x_2) = (C_{11} - C_{21})T_t + (C_{14} - C_{24})T_{ve} \\ - (C_{13} - C_{23})\mu(x_2)RNF_s. \end{aligned} \tag{11b}$$

The problem considered in this paper is to estimate the pressure of clutch B from the measured output and control input, *i.e.*, the measured rotational speeds of transmission and valve electric current.

III. Reduced-order nonlinear state observer

A. Reduced-order nonlinear observer with ISS property

In this section we will make use of the special structure of the clutch pressure system to derive a reduced-order pressure observer. To do this, we denote the measured variables and the variable to be estimated as $\boldsymbol{y}$ and z respectively, *i.e.*,

$$\boldsymbol{y} = [x_1\ x_2]^T, \tag{12a}$$
$$z = x_3. \tag{12b}$$

Then, we can rewrite the state equations as:

$$\dot{\boldsymbol{y}} = \boldsymbol{F}(\boldsymbol{y}, u) + \boldsymbol{G}(\boldsymbol{y}, u)z + \boldsymbol{w}(\boldsymbol{y}, u, z), \tag{13a}$$
$$\dot{z} = A_{22}(u)z + B_2(u), \tag{13b}$$

where $\boldsymbol{w}(\boldsymbol{y}, u, z)$ summarizes model errors and

$$\boldsymbol{F}(\boldsymbol{y}, u) = \begin{pmatrix} f_1(x_1, x_2) \\ f_2(x_1, x_2) \end{pmatrix},$$
$$\boldsymbol{G}(\boldsymbol{y}, u) = \begin{pmatrix} C_{13}\mu(x_2)RNA \\ (C_{13} - C_{23})\mu(x_2)RNA \end{pmatrix},$$
$$A_{22}(u) = -\frac{1}{\tau_{cv}},$$
$$B_2(u) = \frac{K_{cv}}{\tau_{cv}} u.$$

The powertrain system has a large extent of model uncertainties, including the uncertain parameters such as the vehicle mass and the damping coefficient of shafts, and the uncertain dynamics such as the characteristics of the torque converter. The model errors shown in (13) may cause the pressure observer, designed on the nominal model, to be unstable or have a much degraded performance if model errors are not taken into account during the design process.

The ISS theory gives the stability analysis of nonlinear systems with perturbations. In the following, we show how to use this to design a nonlinear pressure observer with ISS property, considering the modeling errors.

Let the observer be designed in the form of

$$\dot{\hat{z}} = A_{22}(u)\hat{z} + B_2(u) + \boldsymbol{L}\left(\dot{\boldsymbol{y}} - \boldsymbol{F}(\boldsymbol{y}, u) - \boldsymbol{G}(\boldsymbol{y}, u)\hat{z}\right), \tag{14}$$

where $\boldsymbol{L} \in R^{1\times 2}$ is the to be determined observer gain. By defining the observer error as

$$e = z - \hat{z}, \tag{15}$$

the error dynamics can then be described by

$$\dot{e} = A_{22}(u)e - \boldsymbol{L}\left(\boldsymbol{G}(\boldsymbol{y}, u)e + \boldsymbol{w}\right). \tag{16}$$

We define $V = \frac{1}{2}e^T e$ and differentiate it along the solution of (16) to obtain

$$\dot{V} = e^T\left(A_{22}(u) - \boldsymbol{L}\boldsymbol{G}(\boldsymbol{y}, u)\right)e - e^T\boldsymbol{L}\boldsymbol{w}. \tag{17}$$

By the use of Young's Inequality [16], the above equality becomes

$$\begin{aligned} \dot{V} &\le e^T\left(A_{22}(u) - \boldsymbol{L}\boldsymbol{G}(\boldsymbol{y}, u)\right)e + \kappa_1 e^T e + \frac{1}{4\kappa_1}\boldsymbol{w}^T\boldsymbol{L}^T\boldsymbol{L}\boldsymbol{w} \\ &= e^T\left(A_{22}(u) - \boldsymbol{L}\boldsymbol{G}(\boldsymbol{y}, u) + \kappa_1\right)e + \frac{1}{4\kappa_1}\boldsymbol{w}^T\boldsymbol{L}^T\boldsymbol{L}\boldsymbol{w}, \end{aligned} \tag{18}$$

where $\kappa_1 > 0$. We now choose $\boldsymbol{L}$ such that

$$A_{22}(u) - \boldsymbol{L}\boldsymbol{G}(\boldsymbol{y}, u) + \kappa_1 \le -\kappa_2 \tag{19}$$

with $\kappa_2 > 0$, then we arrive at

$$\dot{V} \le -e^T\kappa_2 e + \frac{1}{4\kappa_1}\boldsymbol{w}^T\boldsymbol{L}^T\boldsymbol{L}\boldsymbol{w}, \tag{20}$$

which implies that the error dynamics admits the input-to-state stability property if the model error $\boldsymbol{w}$ is supposed to be bounded in amplitude. Moreover, it follows from (20) that

$$\dot{V} \le -2\kappa_2 V + \frac{1}{4\kappa_1}\boldsymbol{w}^T\boldsymbol{L}^T\boldsymbol{L}\boldsymbol{w}. \tag{21}$$

Upon multiplication of (21) by $e^{2\kappa_2 t}$, it becomes

$$\frac{d}{dt}\left(Ve^{2\kappa_2 t}\right) \le \frac{1}{4\kappa_1}\boldsymbol{w}^T\boldsymbol{L}^T\boldsymbol{L}\boldsymbol{w}e^{2\kappa_2 t}. \tag{22}$$

Integrating it over $[0, t]$ leads to

$$\begin{aligned} V(t) &\le V(0)e^{-2\kappa_2 t} \\ &\quad + \frac{1}{4\kappa_1}\int_0^t e^{-2\kappa_2(t-\tau)}\boldsymbol{w}(\tau)^T\boldsymbol{L}^T\boldsymbol{L}\boldsymbol{w}(\tau)d\tau \end{aligned} \tag{23}$$

Hence, we conclude the following results for the property of the error dynamics of the designed observer (14).

Theorem 1: Suppose that

- $\kappa_1 > 0, \kappa_2 > 0$;
- the observer gain $\boldsymbol{L}$ is chosen to satisfy (19).

Then, the error dynamics of the observer (14) is

(i) input-to-state stable, if $\boldsymbol{w}$ is bounded in amplitude, *i.e.*, $\boldsymbol{w} \in \mathcal{L}_\infty$;

(ii) exponentially stable with $2\kappa_2$ for $\boldsymbol{w} = 0$.

Remark 3.1: Now we give some discussions for choosing parameters κ_1 and κ_2. It follows from (23) that κ_2 can be chosen according to the required decay rate of the error. If $\boldsymbol{w}$ is bounded in amplitude, *i.e.*, $\boldsymbol{w} \in \mathcal{L}_\infty$, then (23) becomes

$$\|e(t)\|^2 \leq \|e(0)\|^2 e^{-2\kappa_2 t} + \frac{\|\boldsymbol{w}\|_\infty^2 \sup_{[0,t]} \lambda_{\max}(\boldsymbol{L}^T\boldsymbol{L})}{2\kappa_1} \int_0^t e^{-2\kappa_2(t-\tau)} d\tau, \quad (24)$$

which implies that

$$\|e(t)\|^2 \to \frac{\|\boldsymbol{w}\|_\infty^2 \sup\left(\lambda_{\max}(\boldsymbol{L}^T\boldsymbol{L})\right)}{4\kappa_1\kappa_2} \quad \text{as } t \to \infty. \quad (25)$$

Hence, one may choose larger κ_1 to reduce the offset. From (19), however, one should notice that the larger the κ_1, the higher the observer gain. That is, the choice of κ_1 requires the trade-off between the offset and the observer gain.

Remark 3.2: We stress that (25) gives just an upper bound of the estimation offset, if the bound of the modeling error is given. The real estimation offset could be much smaller, due to the multiple use of inequalities in the derivation. Moreover, if the modeling error is much underestimated, the real estimation offset may be larger than computed. However, because the observer is designed to be input-to-state stable, the estimation error will be not diverge so long as the modeling error is bounded.

Remark 3.3: We do not consider modeling errors in the state equation for the variable to be estimated. If there are modeling errors that should be considered, (13b) may be rewritten as

$$\dot{z} = A_{22}(u)z + B_2(u) + \boldsymbol{w}_2(u, z) \quad (26)$$

and then the error dynamics (16) becomes

$$\dot{e} = (A_{22}(u) - \boldsymbol{L}\boldsymbol{G}(\boldsymbol{y}, u))\, e - (\boldsymbol{L}\boldsymbol{w} - \boldsymbol{w}_2). \quad (27)$$

It can be seen that the impact of $\boldsymbol{w}_2$ can be somehow combined into the summarized modeling error $\boldsymbol{w}$.

B. Implementation issues

In order to avoid taking derivatives of the measurements, the following transformation is made, which is usually used when designing reduced-order observer for linear systems. Let

$$\eta = \hat{z} - \boldsymbol{L}\boldsymbol{y}, \quad (28)$$

then, we can obtain for constant observer gains that

$$\dot{\eta} = (A_{22}(\boldsymbol{y}, u) - \boldsymbol{L}\boldsymbol{G}(\boldsymbol{y}, u))(\eta + \boldsymbol{L}\boldsymbol{y}) + B_2(\boldsymbol{y}, u) + \boldsymbol{L}\boldsymbol{F}(\boldsymbol{y}, u). \quad (29)$$

Equations (28) and (29) constitute then the reduced-order state observer for the nonlinear clutch slip control system. It can be seen that the nonlinearities of the powertrain system appear in the observer in their original form. Therefore, a merit arises: the characteristics of powertrain mechanical systems, such as characteristics of engine and torque converter, can be represented in the form of lookup tables, which is easy to be processed in computer control.

According to Theorem 1 and Remark 3.1, we now give in the following procedure to design the reduced-order nonlinear clutch pressure observer in the form of (28) and (29):

- S1 choose the parameter κ_2 according to the required decay rate of the estimation error;
- S2 choose the parameter κ_1, where the larger the value of κ_1, the smaller the offset of the estimation error;
- S3 determine the observer gain $\boldsymbol{L}$ such that (19) is satisfied;
- S4 for given the model error, use (25) to compute the estimation offset and check if the offset is acceptable.
- S5 If the offset is acceptable, end the design procedure. If not acceptable, go to S2.

C. Observer design for clutch pressure

Now we apply the proposed method to design a clutch pressure observer, where the parameters (τ_{cv}, K_{cv}) are regarded as constant for simplicity. Together with other parameters, they are listed in Table I. The nonlinear functions f_1, f_2

TABLE I

PARAMETERS FOR OBSERVER DESIGN

C_{13}	-25.85	$\frac{1}{\text{kg}\cdot\text{m}^2}$	C_{23}	17.38	$\frac{1}{\text{kg}\cdot\text{m}^2}$
R	0.13	m	N	3	
A	0.01	m^2	τ_{cv}	0.04	s
K_{cv}	0.85	MPa/A			

and μ are given as lookup tables in the observer. Following the procedure given in the above section, we first choose $\kappa_2 = 40$ to meet the requirement for desired decay rate of the estimation error.

Then, we choose $\kappa_1 = 40$ with the purpose of achieving a smaller offset of the estimation error. Following (19), $\boldsymbol{L}$ should satisfy

$$A_{22}(u) - \boldsymbol{L}\boldsymbol{G}(\boldsymbol{y}, u) \leq -(\kappa_1 + \kappa_2) = -80. \quad (30)$$

Although $\boldsymbol{L}$ is allowed to depend on $(\boldsymbol{y}, u)$, we choose $\boldsymbol{L}$ as constant for easy implementation and use the robust poles displacement algorithm of [18], and the command "*place*" of MATLAB/Control Toolbox to calculate $\boldsymbol{L}$. For

$$A_{22}(u) = -\frac{1}{\tau_{cv}} = -25 \quad (31a)$$

$$\boldsymbol{G}_{\min}(\boldsymbol{y}, u) = \begin{pmatrix} C_{13}\mu_{\min}RNA \\ (C_{13} - C_{23})\mu_{\min}RNA \end{pmatrix} = \begin{pmatrix} -0.0101 \\ -0.0169 \end{pmatrix}, \quad (31b)$$

where $\mu_{\min} = 0.1$, $\boldsymbol{L}$ can be calculated to be

$$\boldsymbol{L} = (-1500, -2500). \quad (32)$$

In order to check if the estimation offset acceptable, we now estimate roughly the bound of modeling errors. It is very difficult to obtain a good estimation of the modeling error bound. Hence, we only consider some major uncertainties and estimate the value of $\boldsymbol{w}$ by simulations for simplicity. The major uncertainties considered here are the calculation error of $\boldsymbol{F}(\boldsymbol{y}, u)$ in (13). The simulations are implemented on a powertrain simulation model constructed by the commercial simulation software AMESim, which will be discussed in the next section. Given different driving conditions, the values of $\boldsymbol{F}(\boldsymbol{y}, u)$ can be calculated, and the range of $\boldsymbol{F}(\boldsymbol{y}, u)$ is regarded as the bound of $\boldsymbol{w}$.

In this way, we determine a bound of $\boldsymbol{w}$ as $\|\boldsymbol{w}\|_\infty = 1600\,\mathrm{rad/s^2}$. According to (25), we can then calculate an upper bound of the offset of the designed observer. The value is of $e(\infty) = 0.06$ MPa, which we regard as satisfying for the considered uncertainties.

IV. Simulation results

A. Powertrain simulation model

In this section, the proposed clutch pressure observer is evaluated on a powertrain simulation model. The model is established by commercial simulation software AMESim, which supports the Simulink environment by S-Function and provides a library named “POWERTRAIN” for automotive powertrain modeling. As shown in Fig. 1, the following submodels are contained in the powertrain simulation model,

- 2000cc injection gasoline engine, with torque map shown in Fig. 2;
- torque converter, with capacity factor $C(\lambda)$ and torque ratio $t(\lambda)$ given in Fig. 3;
- transmission box, which contains a planetary gear set and two clutches, which have time-variant friction coefficient μ;
- differential gear box and drive shaft;
- tires in consideration of longitudinal slip;
- car body of 1500kg and road grade setting;
- clutch actuators with parameters (τ_{cv}, K_{cv}) no longer constant but vary for different inputs of current i_b and valve's output port pressure p_{cb}.

Thus, the constructed model can capture the important transient dynamics during vehicle shift process, such as the drive shaft oscillation and tire slip.

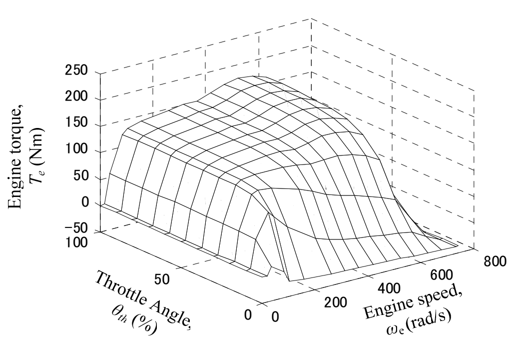

Fig. 2. Engine torque map with speed ω_e and throttle opening θ_{th}

B. Simulation results

The proposed clutch pressure observer is programmed using MATLAB/Simulink and combined with the above complete powertrain simulation model through S-function.

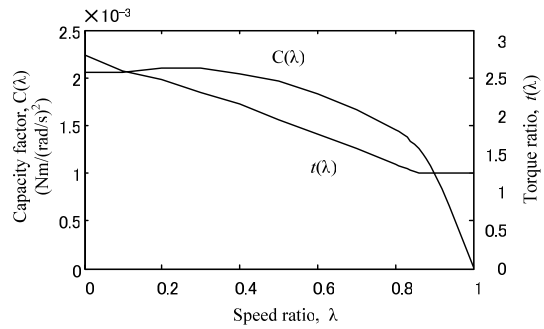

Fig. 3. Capacity factor and torque ratio of torque converter

In this study, the major concern is put on the inertia phase of a typical power-on 1^{st} to 2^{nd} gear up shift process, during which, the torque is mostly transmitted by clutch B, and the pressure of clutch B is controlled to make it lock up smoothly in required duration. This phase is crucial for good shift quality, such as little shock and short shift time.

Fig. 4 gives the simulation results of the shift process with the nominal driving condition, *i.e.*, the condition for observer design. In the inertia phase (from 8s to 8.4s), the pressure of cylinder B, *i.e.*, x_3 is estimated by the proposed observer. For comparison, the estimation result by the simplified control valve dynamics, *i.e.*, (10c), without feedback correction term, is also given as well. From Fig. 4, it can be seen that when the observer works, the estimation error decay rapidly and the estimated pressure can then track the true value without large error. On the other hand, the estimated value by open loop estimation can not track real pressure very well because of the model errors of control valve. When inertia phase ends at about 8.4s, the proposed observer can not work any longer because of the lock up of the clutch. The estimated pressure value calculated by the proposed observer at 8.4s, however, can be used as the initial value for the integration of the open loop method. Therefore, there are two dashed lines during a short duration after 8.4s.

Fig.5 shows the estimation results of a driving condition greatly deviate from the nominal setting. As shown in (5), the turbine torque T_t (torque of transmission input shaft) is determined by the steady-state characteristics of the torque converter. This turbine torque calculation method, however, can not produce accurate values when the torque converter works in transient state. Furthermore, the change of temperature will result in the variation of torque converter characteristics [9]. As to the output side of the transmission, the change of road grade angle will affect T_{ve} greatly. Therefore, we change the engine from a 2000cc injection gasoline engine to a 3000cc injection gasoline engine, and subsequently the capacity of the torque converter is also enlarged. These changes are achieved by modifying the steady-state characteristics of the engine and the torque converter. Moreover, the vehicle mass is increased from 1500kg to 2000kg, and the road grade angle is changed from 0 to 5 deg.

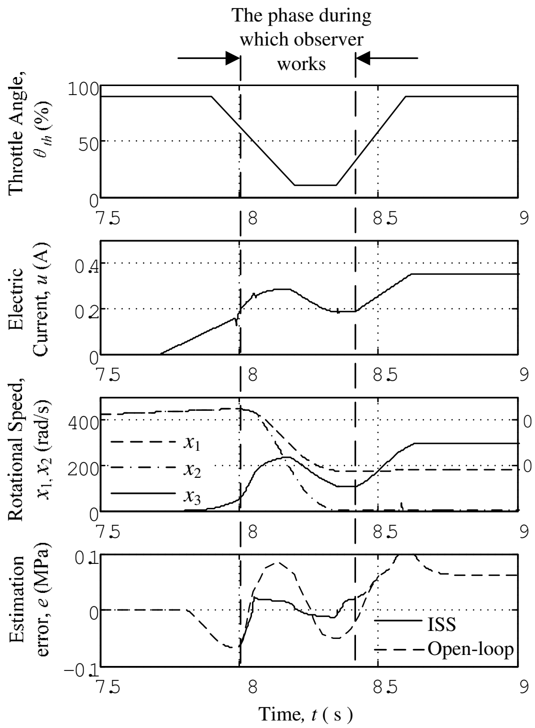

Fig. 4. Simulation results of observer tracking performance

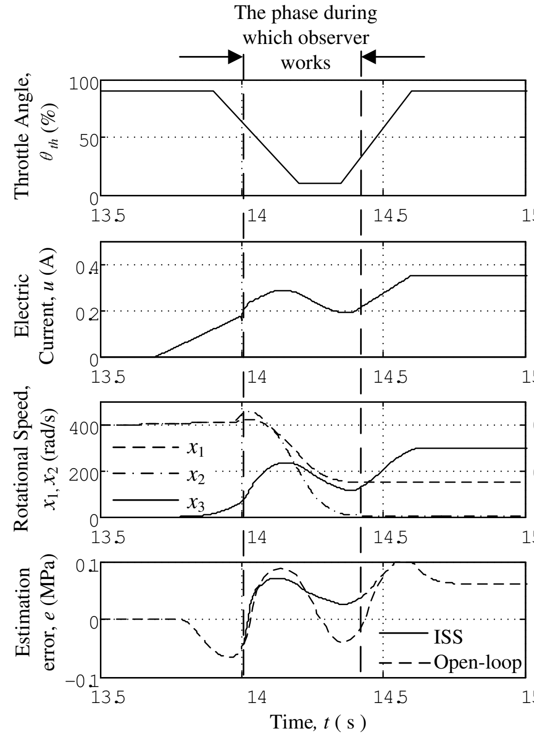

Fig. 5. Simulation results of the different driving condition

From Fig. 5, we can see that because there are large model errors, the pressure estimation error is greater than that of Fig. 4. The estimation error with peak value of $0.07\mathrm{MPa}$, however, is still less than that of the open loop estimation.

V. Conclusions

For the new kind of automatic transmission using proportional pressure control valves to control the clutches, a clutch pressure observer is designed based on ISS. Simulation results show that the estimation error can be restricted in the required bound even a large extent of model errors exist, which shows the potential of the pressure observer's application to real transmission systems.

We also stress that the proposed pressure observer design method can not only be applied to the Automatic Transmission, but also to other types of clutch slip control systems, such as Dual Clutch Transmission, which adopt clutch-to-clutch shifts to change speed ratio.

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