Preprints of the 18th IFAC World Congress
Milano (Italy) August 28 - September 2, 2011

Design of Observer-based Output Feedback Clutch Slip Controller for Automatic Transmission ⋆⋆

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Abstract: In this paper, an observer-based output feedback controller is designed for clutch slip control of automatic transmission where not all states are measurable. The control system consists of a reduced-order observer and a nonlinear state feedback controller. The main contribution of this paper is that an integral action of the tracking error is introduced to reducing the steady-state error of the system. Robustness of the tracking error system is discussed in the framework of Input-to-State Stability (ISS) theory, where the disturbance inputs are considered as estimation error from the observer and unmodelled dynamics respectively. Moveover, lookup tables, which are widely used to represent complex nonlinear characteristics of engine systems, appear in their original form in the designed controller. Finally, the designed controller is tested on an AMESim powertrain simulation model to demonstrate the effectiveness of the proposed control scheme.

Keywords: Automatic transmission, clutch slip control, nonlinear control, lookup tables, output feedback, observer-based.

1. INTRODUCTION

To improve fuel economy, reduce emission and enhance driving performance, many new technologies have been introduced in the transmission area in recent years Sun and Hebbale [2005], such as Dual Clutch Transmission (DCT) Matthes and Guenter [2005], Kulkarni et al. [2007] and new Automatic Transmissions (AT) Minowa et al. [1999] controlling the clutches independently. Furthermore, smart proportional valves with large flow rate are developed for direct clutch pressure control, without using the pilot duty solenoid valve Shioiri [2007].

In both DCT and new AT transmissions, the change of speed ratio can be regarded as a process of one clutch to be engaged while another being disengaged, namely, clutch-to-clutch shifts Cho [1987]. During the shift inertia phase Goetz et al. [2005], the applying (oncoming) clutch slips, and the rotational speeds change intensively. The clutch slip control during inertia phase effects shift quality (smoothness and efficiency) greatly.

The clutch slip control of stepped ratio transmission has received a considerable amount of attention from several researches Glielmo et al. [2006]. Because the clutch engagement is expected to satisfy different and sometimes conflicting objectives: minimizing clutch lock-up time; minimizing the friction losses during the slipping phase; ensuring a smooth acceleration of the vehicle, optimization based algorithm is a potential solution for this problem Dolcini and Béchart [2005], Dolcini et al. [2008], Glielmo and Vasca [2000], Garofalo et al. [2002]. But it's always used for standing start-up scenario, where the duration time is relatively long and there are large amount of friction losses.

Different from optimal algorithms which use penalty functions to formulate multiple control objectives simultaneously, there is another kind of controller design method widely used for clutch slip control, in which the only control objective is to make clutch speed track the reference trajectory Cho [1987], Yokoyama [2008], Quan Zheng and Rizzoni [1999], Sanada and Kitagawa [1998], Gao et al. [2008]. Although during the shift operations, speed tracking control does not consider the multiple control objectives directly, the required shift time and shift comfort can be reached by selecting proper reference trajectory; the friction losses can also be reduced by suitable engine torque coordination during shift process Gao et al. [2008].

Previous work was focused on the later method, *i.e.*, to carry out clutch speed tracking control during the shift inertia phase of a stepped ratio automatic transmission. State feedback is involved in clutch control systems widely, 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 Watechagit and Srinivasan [2005], it is necessary to estimate the clutch

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⋆⋆The work is supported by the National Science Fund of China for Distinguished Young Scholars under Grant No. 60725311, and the National Nature Science Foundation of China under Grant No.90820302 and No.51005193

pressure by observers for the improvement of clutch control systems.

The objective of this paper is to address the problem of observer-based output feedback controlling for clutch slip only depending on measured states. Here, basing on present engineering practice, we consider the turbine speed and speed difference of clutch as two outputs and the pressure of cylinder is not available. Most of clutch slip control methods need state feedback, however, the clutch cylinder pressure is unmeasurable. Following our previous results Gao et al. [2010b,a], we design an observer-based output feedback clutch slip controller. The novelty of this method is giving a control law, which only uses feedback variables from measured outputs of transmission system. Furthermore, model uncertainties are considered as additive disturbances, and Input-to-State Stability (ISS) is used to analyze the robustness of closed-loop tracking error system Sontag [2005]. Simultaneously the integral term of tracking error is introduced in our control system, in order to minimize the offset. As is demonstrated in simulations, the output feedback tracking controller proposed in this paper yield performance as good as their counterparts with state feedback.

The rest of the paper is organized as follows. Section 2 contains a dynamic model of the considered driveline and the formulation of our problems. The clutch slip controller is designed only using measurable variables and an integral term is included to reduce the offset in Section 3. In Section 4, the proposed controller is tested on a complete powertrain simulation model, and the comparison with the state-feedback algorithm Gao et al. [2010a] is given as well.

2. CLUTCH SYSTEM MODELING AND PROBLEM STATEMENT

We consider the powertrain in passenger vehicles with a two-speed AT, as schematically shown in Fig. 1. A planetary gear set is adopted as the shift gear. Two clutches are used as the actuators. Two proportional pressure valves are used to control the two clutches respectively. When clutch A is engaged and clutch B disengaged, the powertrain operates on the 1st gear and the speed ratio is given out by $i_1 = 1 + \frac{1}{\gamma}$, where γ is the ratio of the teeth number of the sun gear to that of the ring gear. While clutch A is disengaged and clutch B engaged, the vehicle is driven on the 2nd gear with a speed ratio of $i_2 = 1$.

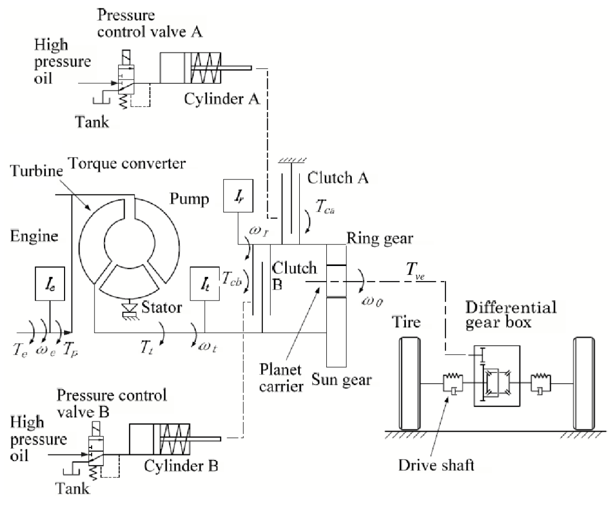

Fig. 1. Schematic graph of automatic transmission

Normally, a typical gear shift process includes the torque phase and the inertia phase Ishihara [1980]. For example, at the beginning of power-on 1st to 2nd 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 controlled by the proportional pressure control valves independently, thus the shift timing and coordination of the clutches are guaranteed.

The driveline dynamic equations of the 1st to 2nd up shift inertia phase can be described by the following equation Gao et al. [2010b]:

$$\dot{x}_1 = C_{13}\mu(x_2)RNAx_3 + f_1(x_1, x_2), \tag{1a}$$

$$\dot{x}_2 = (C_{13} - C_{23})\mu(x_2)RNAx_3 + f_2(x_1, x_2), \tag{1b}$$

$$\dot{x}_3 = -\frac{1}{\tau_{cv}}x_3 + \frac{K_{cv}}{\tau_{cv}}u, \tag{1c}$$

with

$$f_1(x_1, x_2) = C_{11}T_t + C_{14}T_{ve} - C_{13}\mu(x_2)RNF_s, \tag{2a}$$

$$\begin{aligned} f_2(x_1, x_2) = {} & (C_{11} - C_{21})T_t + (C_{14} - C_{24})T_{ve} \\ & - (C_{13} - C_{23})\mu(x_2)RNF_s. \end{aligned} \tag{2b}$$

where $u = i_b$ is the current of valve B, and it is also the control input; turbine speed ω_t, speed difference $\Delta\omega$, and pressure of cylinder B p_{cb} are selected as the state variables x_1, x_2, x_3 respectively; C_{ij} is the constant coefficients decided by inertia moments of vehicle and transmission shafts; 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; τ_{cv} is the time constant of the valve B; K_{cv} is the gain of the valve B; F_s is return spring force of clutch B; T_t is the turbine torque of the torque converter; T_{ve} is the equivalent resistant torque delivered from differential to transmission.

As mentioned above, the problem considered in this paper is to make the speed difference of clutch B $\Delta\omega$ track a reference trajectory. It's shown in Dolcini and Béchart [2005], Bemporad et al. [2001], Haj-Fraj and Pfeiffer [2001] that the clutch engagement should satisfy the so-called nolurch condition. Therefore, the desired reference trajectory is shown in Gao et al. [2010a] as Fig 2.

There are many methods to define the reference trajectory shown in Fig. 2, such as transfer functions or polynomials. Here, we use a third-order polynomial to describe the reference trajectory which can be seen in Gao et al. [2010a].

3. CLUTCH SLIP CONTROLLER

The purpose of this section is to introduce an observer-based output feedback design method to solve the tracking problem formulated in Section 2. As mentioned in Introduction. The diagram of the suggested control system is

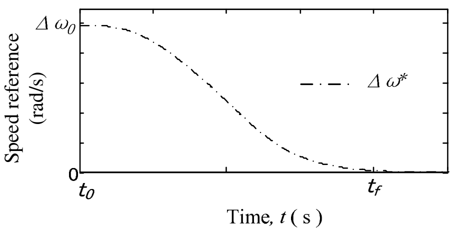

Fig. 2. Reference trajectory of clutch speed difference.

shown in Fig 3. Here the controller is combining state feedback control law with a pressure observer. Moreover, an integral action of tracking error is introduced to reduce the offset. This way, we obtain an observer-based output feedback controller with integral action.

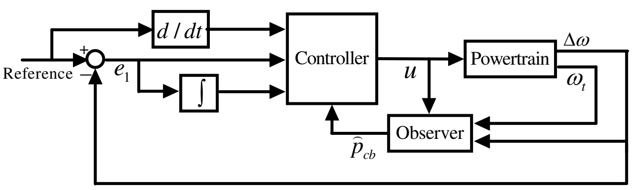

Fig. 3. The diagram of controller for clutch slip control.

3.1 Observer design

Because the sensors for measuring the pressure of clutch cylinder are not used, the clutch pressure estimation is necessary for optimizing the transmission operation Watechagit and Srinivasan [2005]. In this section, we briefly introduce a reduced-order pressure observer, which is proposed in Gao et al. [2010b]. We denote the measured variables and the variable to be estimated as $\boldsymbol{y}$ and z respectively, *i.e.* ,

$$\boldsymbol{y} = [x_1 \;\; x_2]^T, \tag{3a}$$
$$z = x_3. \tag{3b}$$

Then, we can rewrite the state equations as

$$\dot{\boldsymbol{y}} = F(\boldsymbol{y}) + G(\boldsymbol{y})z + w(\boldsymbol{y}, u, z), \tag{4a}$$
$$\dot{z} = A_{22}z + B_2 u, \tag{4b}$$

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

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

Nonlinear functions $f_1(x_1, x_2)$, $f_2(x_1, x_2)$ and $\mu(x_2)$ are given as lookup tables. Then the reduced-order observer is given in the form of

$$\dot{\hat{\eta}} = (A_{22} - \boldsymbol{LG}(\boldsymbol{y}))(\hat{\eta} + \boldsymbol{Ly}) + B_2 u - \boldsymbol{LF}(\boldsymbol{y}). \tag{5}$$

It is shown in Gao et al. [2010b] that the error dynamics system is input to state stable (ISS) with respect to the additive disturbance w, if $A_{22} - \boldsymbol{LG}(\boldsymbol{y}) < 0$. Moreover, an upper-bound of the estimation error offset is given by

$$\|e_0(t)\|^2 \le \frac{\|w\|_\infty^2 \sup(\lambda_{\max}(L^{\mathrm{T}}L))}{4k_1k_2} \quad \text{as } t \to \infty\ , \tag{6}$$

where $e_0 = \eta - \hat{\eta}$ is the observation error.

3.2 Nonlinear controller design

This paper focuses on the clutch slip control during shift inertia phase. During the inertia phase, the pressure of clutch A has already been reduced to a very low level, and the dynamics of the clutch system can be described by the following equation:

$$\dot{x}_1 = a_1\mu(x_1)x_2 + f_2(x_1, \omega_e, \omega_t) + b_{11}w_1, \tag{7a}$$
$$\dot{x}_2 = a_2x_2 + b_{22}u + b_{21}w_2, \tag{7b}$$

where, $x_1 = \Delta\omega$, $x_2 = p_{cb}/1000$, so that x_1 and x_2 are at the same order of magnitude; w_1 and w_2 summarize model uncertainties and b_{11} and b_{21} are known scaling factors. Moreover, $a_1 = (C_{13} - C_{23})RNA \times 1000$, $a_2 = -\frac{1}{\tau_{cv}}$, $b_{22} = \frac{K_{cv}}{\tau_{cv} \times 1000}$. The electric current of valve B is chosen as system input $u = i_b$.

First, we design a state feedback control law using backstepping to the nominal system without considering disturbances. The control objective is to track a given smooth trajectory of x_1, denoted as x_{1d}. To do this, we define the tracking error as $e_1 = x_1 - x_{1d}$. Then we consider x_2 as control input and determine a control law of x_{2d}, such that the tracking error is asymptotically tended to zero.

Let $V_1 = \frac{1}{2}e_1^2$ and differentiating it along (7a), we infer

$$\dot{V}_1 = e_1\dot{e}_1 = e_1\left[a_1\mu(x_1)x_2 + f_2(x_1, \omega_e, \omega_t) - \dot{x}_{1d}\right]. \tag{8}$$

Note that $\mu(x_1) \neq 0$ in our case. Hence, we can choose

$$x_{2d} = \frac{-\kappa_1 e_1 - f_2(x_1, \omega_e, \omega_t) + \dot{x}_{1d}}{a_1\mu(x_1)}, \tag{9}$$

with $\kappa_1 > 0$ to guarantee

$$\dot{V}_1 = -\kappa_1 e_1^2 < 0. \tag{10}$$

Then, we conclude that the control law (9) renders the subsystem (7a) asymptotically exact tracking of x_{1d}. We define $e_2 = x_2 - x_{2d}$. Let $V_2 = \frac{1}{2}e_1^2 + \frac{1}{2}e_2^2$ and in view of (9) and (7b), we infer

$$\dot{V}_2 = -\kappa_1 e_1^2 + e_2[a_1\mu(x_1)e_1 + a_2\mu(x_2, u)x_2 + b_{22}u - \dot{x}_{2d}]. \tag{11}$$

Because K_{cv} and τ_{cv} are simplified as positive constants for the controller design, $b_{22} \neq 0$ is satisfied in our case. Hence, the control law is chosen as

$$u = \frac{-\kappa_2 e_2 - a_1\mu(x_1)e_1 - a_2x_2 + \dot{x}_{2d}}{b_{22}}, \tag{12}$$

where $\kappa_2 > 0$. Hence, we have

$$\dot{V}_2 = -\kappa_1 e_1^2 - \kappa_2 e_2^2 < 0. \tag{13}$$

Note that x_2 is not measured. By replacing x_2 with the estimated one $\hat{x}_2$, we can get an observer-based output feedback clutch slip controller as follows:

$$u = \frac{-\kappa_2(\hat{x}_2 - x_{2d}) - a_1\mu(x_1)(x_1 - x_{1d}) - a_2\hat{x}_2 + \dot{x}_{2d}}{b_{22}}. \tag{14}$$

Moreover, $\dot{x}_{2d}$ is needed for the implementation of (14). Since $\mu(x_1)$ and $f(x_1, \omega_e, \omega_t)$ are given as lookup tables, it is impossible to obtain the explicit form of $\dot{x}_{2d}$ by differentiating (9). Hence, we apply the input shaping technique Chang and Park [2005], which is on occasion named as Dynamic Surface Control (DSC) in the backstepping literature Swaroop et al. [2000]. The result of (9) is labeled as $\bar{x}_2$, and passed through a first order filter,

$$\tau_2\dot{x}_{2d} + x_{2d} = \bar{x}_2, x_{2d}(0) = \bar{x}_2(0), \tag{15}$$

which yields $\dot{x}_{2d} = \frac{\bar{x}_2 - x_{2d}}{\tau_2}$.

3.3 Properties of the closed-loop system

As shown in (7), model uncertainties w_1, w_2, e_0 including the observer error and unmodelled dynamics are considered as additive disturbance inputs, which are not included during the controller design. In this section, robustness of the tracking error system is discussed in the framework of Input-to-State Stability (ISS) theory.

In view of (7), (9) and (14), we have a closed-loop error system including the observer error as follows:

$$\dot{e}_1 = a_1\mu(x_1)e_2 - k_1e_1 + b_{11}w_1, \tag{16a}$$

$$\dot{e}_2 = -a_1\mu(x_1)e_1 - k_2e_2 + (a_2 + k_2)e_0 + b_{21}w_2. \tag{16b}$$

Let $e = [e_1, e_2]^T$, $w = [w_1, w_2, e_0]^T$ then

$$\dot{e} = \begin{bmatrix} -\kappa_1 & a_1\mu(x_1) \\ -a_1\mu(x_1) & -\kappa_2 \end{bmatrix} e + \begin{bmatrix} b_{11} & 0 & 0 \\ 0 & b_{21} & a_2 + \kappa_2 \end{bmatrix} w \tag{17}$$

We define $V = \frac{1}{2}e_1^2 + \frac{1}{2}e_2^2$ and differentiate it along the solution of (17) to obtain

$$\dot{V} = -\kappa_1 e_1^2 - \kappa_2 e_2^2 + b_{11}e_1w_1 + b_{21}e_2w_2 + (a_2+\kappa_2)e_2e_0. \tag{18}$$

By the use of Young's Inequality H.K.Khalil [2002], the above equality becomes

$$\dot{V} \le -(\kappa_1 - \kappa_3)e_1^2 - (\kappa_2 - \kappa_4 - \kappa_5)e_2^2 + \frac{|b_{11}|}{4\kappa_3}w_1^2 + \frac{|b_{21}|}{4\kappa_4}w_2^2 + \frac{(a_2+\kappa_2)^2}{4\kappa_5}e_0^2. \tag{19}$$

As shown in Gao et al. [2010a], we choose $\kappa_i > 0, i = 1 \sim 5$, $\kappa_1 - \kappa_3 > 0$, $\kappa_2 - \kappa_4 - \kappa_5 > 0$. Then, the error system (16) under controller (14) is ISS, if w_1, w_2 and observer error e_0 are bounded in amplitude, *i.e.*, w_1, w_2 and $e_0 \in \mathcal{L}_\infty$.

Then, we first analyze the effect of the observer error upon the closed-loop system. Therefore, $w_1 = 0, w_2 = 0$. In view of (18), by using Young's Inequality, we have

$$\dot{\mathrm{V}} \le -\kappa_1 e_1^2 - (\kappa_2 - \kappa_5)e_2^2 + \frac{(a_2+\kappa_2)^2}{4\kappa_5}e_0^2. \tag{20}$$

Suppose that $\kappa_i > 0,\ i = 1, 2, 5$, $\kappa_2 - \kappa_5 > 0$. We define $\kappa = 2\min\{\kappa_1, \kappa_2 - \kappa_5\}$, the inequality (20) above can be rewrite as follows

$$\dot{\mathrm{V}} \le -\kappa\mathrm{V} + \frac{(a_2+\kappa_2)^2}{4\kappa_5}e_0^2. \tag{21}$$

The adjustable parameter $\kappa_1, \kappa_2, \kappa_5$ and κ should satisfy the suppose above, implying $\kappa_1 \ge \frac{1}{2}\kappa$ or $\kappa_2 \ge \kappa_5 + \frac{1}{2}\kappa$. Then, we choose $\kappa_1 = \frac{1}{2}\kappa, \kappa_2 = \kappa_5 + \frac{1}{2}\kappa$, in order to minimize the controller gain, namely: $\kappa = 2\kappa_1, \kappa_5 = \kappa_2 - \kappa_1$.

Therefore, the procedure of parameter selection can be simplified for only κ_1 and κ_2. Hence, (21) can be rewrite as

$$\dot{\mathrm{V}} \le -2\kappa\mathrm{V} + \frac{(a_2+\kappa_2)^2}{4(\kappa_2 - \kappa_1)}e_0^2. \tag{22}$$

Upon multiplication of (22) by $e^{2\kappa t}$ and integrating it over $[0, t]$ leads to

$$V(t) \le V(0)e^{-2\kappa t} + \frac{(a_2+k_2)^2}{4(\kappa_2 - \kappa_1)}\int_0^t e_0^2 e^{-2\kappa(t-\tau)}d\tau, \tag{23}$$

and hence

$$\|e(t)\|^2 \le \|e(0)\|^2 e^{-2\kappa t} + \frac{(a_2+k_2)^2}{2(\kappa_2 - \kappa_1)}\int_0^t e_0^2 e^{-2\kappa(t-\tau)}d\tau. \tag{24}$$

If e_0 is bounded in amplitude, *i.e.*, $e_0 \in \mathcal{L}_\infty$, (24) becomes

$$\|e(t)\|^2 \le \|e(0)\|^2 e^{-2\kappa t} + \frac{(a_2+k_2)^2}{2(\kappa_2 - \kappa_1)}\|e_0\|_\infty^2 \int_0^t e^{-2\kappa(t-\tau)}d\tau. \tag{25}$$

Thus, we have the following conclusion: If the observation error is bounded in amplitude, then

- initial value of error dynamics decaying with index of κ;
- $\|e(t)\|^2 \le \frac{(a_2+\kappa_2)^2}{4\kappa_1(\kappa_2-\kappa_1)}\|e_0\|_\infty^2$ as $t \to \infty$.

We stress that the above inequality gives just an upper bound of the tracking offsets, if the bound of the observation error is given.

In the same way, we analyze the influence of disturbances w_1 and w_2 respectively. Then we have the following conclusions:

- initial value of error dynamics decaying with index of κ, $\kappa = \min\{\kappa_1 - \kappa_3, \kappa_2\}$;
- $\|e(t)\|^2 \le \frac{|b_{11}|\,\|w_1\|_\infty^2}{8\kappa\kappa_3}$ as $t \to \infty$,

where $w_2, e_0 = 0$, $\|w_1\|_\infty$ can be estimated approximately as 543, which is introduced in Gao et al. [2010a].

Also we have

- initial value of error dynamics decaying with index of κ, $\kappa = \min\{\kappa_2 - \kappa_4, \kappa_1\}$;
- $\|e(t)\|^2 \le \frac{|b_{21}|\,\|w_2\|_\infty^2}{8\kappa\kappa_4}$ as $t \to \infty$,

where $w_1 = 0, e_0 = 0$.

Based on the procedure of designing the clutch slip controller and the pressure observer parameters, which are given above, $\kappa_i, i = 1, 2$ and L can be determined as: $\kappa_1 = 82$; $\kappa_2 = 15$; $L = (-1607, -2689)$.

3.4 Integral term modification for the controller

From ISS analysis in last section, the system existence static tracking error caused by model uncertainties. Thus an integral action of tracking error is introduced in our controller (14) for reducing the offset. In this section, a derivation of controller with integral term modification is given briefly.

An integral term of the tracking error is added to the virtual control input (9), which then is modified as

$$x_{2d} = \frac{-\kappa_1 e_1 - f(x_1, \omega_e, \omega_t) + \dot{x}_{1d} - h\chi}{a_1\mu(x_1)}, \tag{26}$$

where $h > 0, \chi = \int_0^t e_1(\tau)d\tau$. Choosing a modified Lyapunov function as $V_1 = h\chi^2/2 + e_1^2/2$, then we have $\dot{V}_1 = -\kappa_1 e_1^2 < 0$.

Repeating the same derivation to (10)-(14), we can get the observer-based output feedback controller after modifying.

$$u = \frac{-\kappa_2}{b_{22}}(\hat{x}_2 - \frac{-\kappa_1 e_1 - f(x_1, \omega_e, \omega_t) + \dot{x}_{1d} - h\chi}{a_1\mu(x_1)}) - \frac{a_1\mu(x_1)(x_1 - x_{1d}) - a_2\hat{x}_2 + \dot{x}_{2d}}{b_{22}}. \quad (27)$$

4. SIMULATION RESULTS

The powertrain simulation model is established by commercial simulation software AMESim. AMESim allows to model vehicle driveline and complete transmission, including vibrations and shift dynamics. Except for the simplified 2-speed transmission, the parameters used in this paper represent a typical front-wheel-drive mid-size passenger car equipped with a 2000cc injection gasoline engine. The constructed model captures the important transient dynamics during the vehicle shift process, such as

- the drive shaft oscillation and the tire slip;
- time-delay in control and time-varying parameters of the proportional valves Sanada and Kitagawa [1998].

The proposed clutch controller is programmed using MATLAB/Simulink and combined with the aforementioned complete powertrain simulation model through co-simulations.

In this study, the major concern is put on the power-on first to second gear up-shift process during the inertia phase (between 15.94 and 16.34 s). Fig.4 gives the simulation results of the tracking error of the clutch slip speed during the inertia phase with the nominal driving condition (vehicle mass: 1500 kg; road slope : 0°; inertia moment of turbine = 0.06 kg·m^2). In Fig.4, the maximum errors are 12.2, 14.3 and 7.2 rad/s respectively for state feedback, output feedback without integral and output feedback with integral, all of which are small for the vehicle clutch system. However, output feedback with integral has smaller offset and the transmission torque is as good as that of state feedback controller. It should be pointed out that the shift process operates in the nominal driving condition, but the stiffness of the drive shaft and a tire model considering longitudinal slip are considered in the simulation model, while these are ignored in the model for designing the controller. Moreover, the time delay in control and time-varying parameters are also considered in the simulation model of the proportional valve.

By the procedures of simulation debug, we choose the gain of integral term as follows: $h = 1000$.

In order to verify the effectiveness of the algorithm, the proposed controller is tested under the driving conditions that deviate from the nominal driving setting. The result are shown in Figs. 5 and 6, where the driving condition settings are as follows: Fig. 5 shows when the torque characteristics of the engine is enlarged by 15%, and subsequently, the capacity of the torque converter is also enlarged, the vehicle mass is increased from 1500 to 2000 kg, the road grade angle is changed from 0° to 5°, and the inertia of the turbine shaft is changed from 0.06 to 0.1kg · m^2; Fig. 6 shows when the torque characteristics of the engine is reduced by 15%, and subsequently, the capacity of the torque converter is also reduced, the vehicle mass is reduced from 1500 to 1200 kg, the road grade angle is changed from 0° to 5°, and the inertia of the turbine shaft is changed from 0.06 to 0.03kg·m^2 . Note that

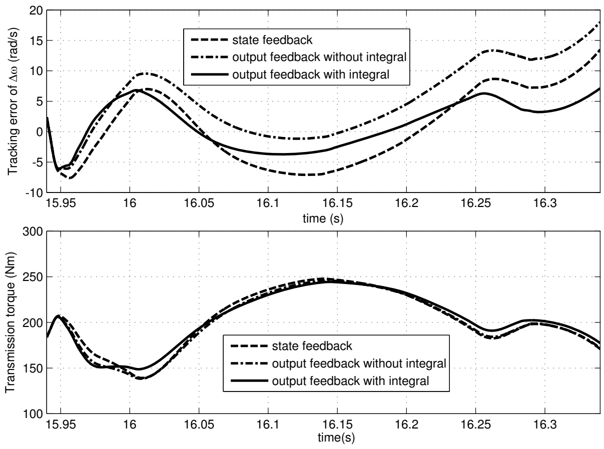

Fig. 4. Simulation results of the nominal driving condition (torque characteristics of engine and torque converter: standard, vehicle mass: 1500 kg; road slope : 0°; inertia moment of turbine = 0.06 kg · m^2 .)

although the driving conditions are changed, because the shift maneuvers are all power-on upshift when the engine load is 90%, the engine throttle control pattern of Figs. 5 and 6 are the same as Fig. 4.

It can be seen from the simulation results, the observation error leads to large offset, therefore the output feedback controller without integral term has a larger offset, whereas the one with integral term is able to reduce the offset of tracking system and has good performance in the transmission torque.

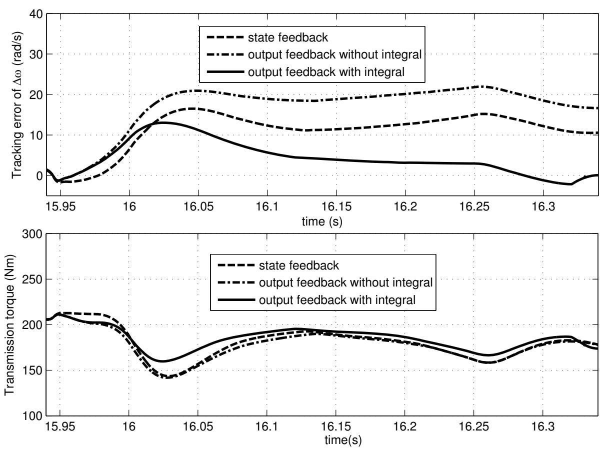

Fig. 5. Simulation results of different driving condition2 (torque characteristics of engine and torque converter:standard × 115%, vehicle mass: 2000 kg; road slope : 5°; inertia moment of turbine = 0.1 kg · m^2 .)

5. CONCLUSIONS

An observer-based output feedback controller with integral term modification is designed for the clutch slip control during shift inertia phase of a stepped ratio automatic transmission. The control system consists of a reduced-order observer to estimate the unmeasurable state and a nonlinear state feedback controller. The complex nonlinear characteristics of engine systems appear in their original

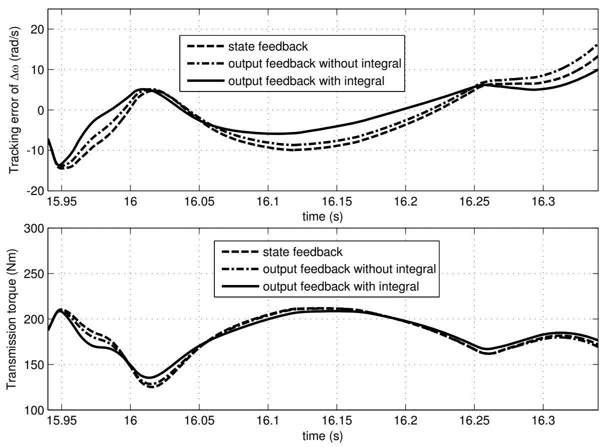

Fig. 6. Simulation results of different driving condition1 (torque characteristics of engine and torque converter:standard $\times$ 85%, vehicle mass: 1200 kg; road slope : 5°; inertia moment of turbine = 0.03 $\mathrm{kg \cdot m^2}$.)

form of lookup tables. Model uncertainties including the observer error and unmodelled dynamics are considered as additive disturbance inputs and the controller is designed such that the tracking error system is ISS.

The designed controller is tested on an AMESim powertrain simulation model, which contains complete drive train and clutch actuators. Comparison results with the state-feedback controller and the observer-based controller without integral term modification verify the potential benefits of the proposed nonlinear controller in achieving less tracking error. It is also demonstrated that the controller is robust to driving condition variations, such as change of vehicle mass and road grade. Moreover, testing the controller in HIL deserves the further attention, which is our work to be done later.

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