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CLUTCH-TO-CLUTCH SHIFT CONTROL OF AN AUTOMATIC TRANSMISSION WITH PROPORTIONAL PRESSURE CONTROL VALVES

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ABSTRACT

For a new kind of automatic transmissions using proportional pressure valves to control the clutches directly, a two-degree-of-freedom controller is designed for clutch slip control during the inertia phase of the shift process. The controller is designed based on a low order linear model which is derived from dynamics of the proportional pressure control valve and the vehicle drive line. The feedback gain is calculated by robust pole assignment methods while the feed-forward compensator aims to improve system response. Finally, the designed controller is tested on an AMESim powertrain simulation model. Simulation results show that the rotational speed difference of clutch can track the desired trajectory well, and shift shock can be reduced by designing suitable feed-forward compensator.

KEY WORDS

Automatic Transmission, Proportional Pressure Control Valves, Clutch Control, Clutch-to-Clutch Shift

NOMENCLATURE

A : piston area of clutch B
C_A : a constant coefficient depending on air density, aerodynamic drag coefficient and the front area of the vehicle.
$C(\lambda)$: capacity factor of torque converter
dS_x : longitudinal slip threshold
F_s : force of return spring
F_x : tire longitudinal force
$F_{x\max}$: maximum longitudinal force of the tire
i_b : electric current of pressure control valve B
i_{df} : gear ratio of the differential gear box
$\widetilde{K}_{cv}$: valve gain
$\widetilde{L}_{cv}$: time-lag of valve
N : number of friction plates
p_{cb} : pressure of cylinder B
p_s : input port pressure of the valve
R : effective radius of push force acted on the friction plates
R_w : tire radius
S_x : longitude slip ratio of tires
T_p : torque converter pump torque
T_t : torque converter turbine torque
T_v : resistant torque delivered from tire to drive shaft
T_w : rolling resistant moment of tires

$t(\lambda)$: torque ratio of torque converter
γ : gear ratio of sun gear to ring gear
λ : speed ratio of torque converter
ω_0 : output speed of transmission
ω_e : engine speed
ω_r : rotational speed of ring gear
ω_t : turbine speed
$\Delta\omega$: speed difference of clutch B
μ : friction coefficient of clutch plates
$\tilde{\tau}_{cv}$: time constant

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]. These valves can be used in new Automatic Transmissions to improve the ability of adapting to different driving conditions, as well as to reduce the cost and to improve packaging.

Clutch-to-clutch shift of this kind of Automatic Transmission involves electronic control of both the oncoming and offgoing clutches, thus guarantee the timing and coordination between them, which are assured by the hydraulic logic circuits in the case of traditional AT. The elimination of some shift valves and accumulators, etc. greatly simplifies the transmission mechanical content, but makes the robust control of clutch-to-clutch shifts a challenge [1, 3]. Furthermore, the sensors for measuring the pressure of clutch cylinder are seldom used because of the cost and durability. If system state feedback is to be used to enhance system control quality, the cylinder pressure needs to be estimated [4, 5].

Two-degree-of-freedom controller design is suitable to many automotive control systems for it can fulfill good tracking performance and robustness simultaneously [6]. In [7], a two-degree-of-freedom controller was designed for the speed control of a braking mechanism, in which the feedback gain was calculated by μ synthesis.

This paper, therefore, use the two-degree-of-freedom controller design method to carry out the clutch engagement control of an Automatic Transmission with proportional pressure control valves. The clutch cylinder pressure, which is necessary for state feedback control is estimated by a reduced-order state observer. The feedback gain is calculated by robust pole assignment methods while the feedforward compensator aims to improve the system response. Not only the rotational speed can track the desired trajectory well, but also can shift shock be reduced by choosing suitable feedforward compensator. Finally, the designed controller including the pressure estimator is tested on an AMESim powertrain simulation model.

CLUTCH SYSTEM MODELING

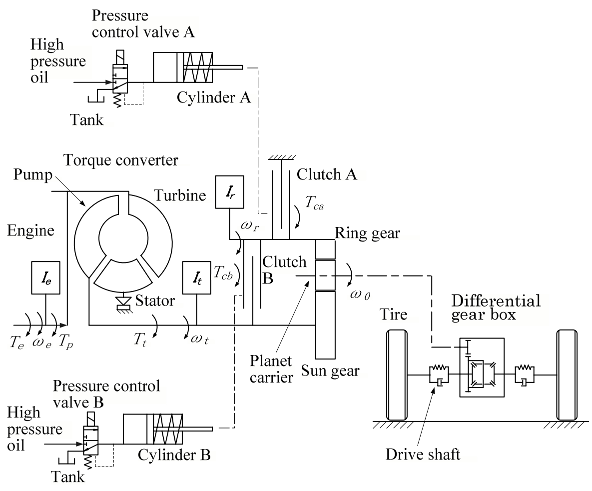

Figure 1 Schematic graph of Automatic Transmission

We consider the powertrain in passenger vehicles with a two-speed automatic transmission, as schematically shown in Figure 1. A planetary gear set is adopted as the shift gear, two clutches are used as the actuators, and two proportional pressure valves are used to control the two clutches respectively. When clutch A is engaged and clutch B disengaged, the powertrain is operating in 1st gear, the speed ratio is

$$i_1 = (1 + 1/\gamma) \tag{1}$$

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

$$i_2 = 1 \tag{2}$$

During the shift process, the oncoming and offgoing clutches are controlled by the proportional pressure control valves independently, thus the shift timing and cooperation of the clutches are guaranteed.

Proportional Pressure Control Valves

The cylinder's pressure is decided by the input electric current of proportional pressure control valve. The dynamics of the proportional valve can be simplified as a first-order system [7]:

$$\tilde{\tau}_{cv}\dot{p}_{cb} = -p_{cb} + \tilde{K}_{cv}\, i_b(t - \tilde{L}_{cv}) \tag{3}$$

The values of $\tilde{\tau}_{cv}$, $\tilde{K}_{cv}$ and $\tilde{L}_{cv}$ are not constant but vary according to different operating point.

In order to represent real valve dynamics at different operating points, the parameters $\tilde{\tau}_{cv}$, $\tilde{L}_{cv}$, $\tilde{K}_{cv}$ are given by lookup tables according to different operating points, i.e. valve current i_b and the input port pressure of the valve p_s [8].

The above time-variant model of the valves can be used for system simulation. During controller design stage, however, the parameter variations are ignored, and the dynamics of the valve can be rewritten as

$$\tau_{cv} \dot{p}_{cb} = -p_{cb} + K_{cv} i_b \tag{4}$$

where τ_{cv} and K_{cv} are constants.

Clutch System

The 1st to 2nd up shift is considered here as an example. During the inertia phase, the pressure of cylinder A is already approximately zero, and the pressure of cylinder B is controlled so that clutch B can be engaged smoothly in required time. If the variation of friction coefficient μ is ignored, the speed difference of clutch B can be described by the following equation[9],

$$\Delta\dot{\omega} = (C_{13} - C_{23})\mu RNA\, p_{cb} + (C_{11} - C_{21}) T_t + (C_{14} - C_{24}) T_v - (C_{13} - C_{23})\mu RNF_s$$

where $\Delta\omega = \omega_t - \omega_r$ and C_{ij} are the constant coefficients decided by inertia moments of vehicle and transmission shafts.

Without considering the transient dynamics of the torque converter, T_t can be calculated by the steady-state characteristics of the torque converter as follows

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

with $\lambda = \dfrac{\omega_t}{\omega_e}$.

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_v can be calculated as

$$T_v = \frac{T_w}{i_{df}} + \frac{C_A R_w^3}{i_{df}^3} \omega_0^2 \tag{6}$$

Model for Controller Design

By selecting speed difference $\Delta\omega$, and pressure p_{cb} of cylinder B as the state variables $x1$; $x2$ respectively, the inertia phase of 1st to 2nd gear up shift process can be described in the following state space form

$$\dot{x}_1 = (C_{13} - C_{23})\mu RNA\, x_2 + T_d \tag{7a}$$

$$\dot{x}_2 = -\frac{1}{\tau_{cv}} x_2 + \frac{K_{cv}}{\tau_{cv}} u \tag{7b}$$

where

$u = i_b$, the control input;

$T_d = (C_{11} - C_{21}) T_t + (C_{14} - C_{24}) T_v - (C_{13} - C_{23})\mu RNF_s$,

regarded as disturbance torque here.

It should be noted that, although T_d is regarded as disturbance torque, it has large effect on the dynamics of speed difference. Our previous study has shown that, when only linear feedback is used to carry out the control problem, it is difficult to get satisfactory control performance. Thus a two-degree-of-freedom controller will be designed and used to improve the control performance.

CONTROLLER DESIGN

The clutch pressure observer [9, 10] design is omitted here, and the clutch pressure is assumed available in the following controller design procedure.

Two-Degree-of-Freedom Controller

Two-degree-of-freedom controller is a control system with a forward compensator besides the feedback controller [11]. Its block diagram is shown in Figure 2.

If the controlled object $P(s)$ is modeled accurately enough, the transfer function from input to output turns to be,

$$\frac{Y(s)}{R(s)} = M(s) \tag{8}$$

which means that the system transfer function only depends on the dynamics of $M(s)$. Therefore, the quality of output response can be improved by giving suitable $M(s)$. While, on the other hand, the feedback controller $K_b(s)$ can be designed for high stability and robustness.

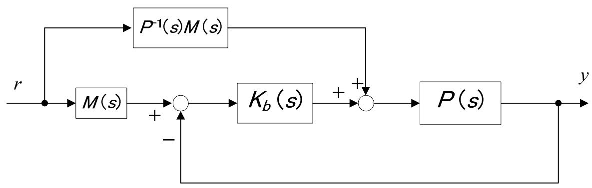

Figure 2 Block diagram of two-degree-of-freedom controller

Two-Degree-of-Freedom Controller Design for Clutch Slip Control

Clutch slip controller will be designed in this section based on equation (7). Rewrite the equation (7) in matrix form:

$$\dot{\boldsymbol{x}} = \boldsymbol{A}\boldsymbol{x} + \boldsymbol{B}u + \boldsymbol{E}d \tag{9a}$$

and the output equation is:

$$y = \boldsymbol{C}\boldsymbol{x} \tag{9b}$$

where,

$\boldsymbol{x} = [\Delta\omega \;\; p_{cb}]^T$, $y = \Delta\omega$

$$A = \begin{bmatrix} 0 & (C_{13} - C_{23})\mu RNA \\ 0 & -\dfrac{1}{\tau_{cv}} \end{bmatrix}, \quad B = \begin{bmatrix} 0 & \dfrac{K_{cv}}{\tau_{cv}} \end{bmatrix}^T$$

$E = [1 \quad 0]^T$, $C = [1 \quad 0]$, $u = i_b$, $d = T_d$

Based on the above linear state equations, the two-degree-of-freedom clutch slip controller is designed. The block diagram is given as Figure 3.

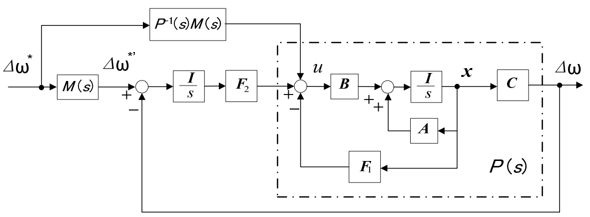

Figure 3 Two-degree-of-freedom clutch slip controller

If no feedforward compensator is included, the gain F_1 and F_2 turns to be the commonly used linear servo system for output tracking control. The robust pole assignment method proposed by [12] is used here to calculate F_1 and F_2, which is also the algorithms of command "*place*" in control toolbox of MATLAB.

After determining the feedback gain F_1 and F_2, the forward compensator can be derived. First the part circled in the dashed line is labeled as $P(s)$. Being different from the $P(s)$ of Figure 2, $P(s)$ defined here includes the state feedback besides the controlled object. This treatment allows for convenient design of the feed forward compensator and the later simulation results show its validity. Thus, $P(s)$ can be calculated by,

$$P(s) = C(sI - A')^{-1} B = \frac{P_n(s)}{P_d(s)} \quad (10)$$

where

$$A' = A - B F_1 \quad (11)$$

$P_n(s)$ is a constant and $P_d(s)$ is a second-order polynomial of the Laplace variable s.

Because $P^{-1}(s)M(s)$ must be a proper transfer function, $M(s)$ is set as a third-order transfer function with the following form,

$$M(s) = \frac{p_0^{\,3}}{(s + p_0)^3} \quad (12)$$

The desired speed difference is given first by the request of shift time, which is shown as $\Delta\omega^*$ in Figure 3. Then it is modified by reference model $M(s)$ and yields $\Delta\omega^{*'}$, the real desired trajectory. The smaller value of p_0 results in smoother shape of $\Delta\omega^{*'}$, which can be seen from Figure 7 in the section "Simulation results ".

After getting $P(s)$ and $M(s)$, the feed forward compensator $P^{-1}(s)M(s)$ can be calculated by,

$$P^{-1}(s)M(s) = \frac{P_d(s)\,M(s)}{P_n(s)} \quad (13)$$

SIMULATION RESULTS

Powertrain Simulation Model

The simulation model of the powertrain is established by commercial simulation software AMESim, which supports the Simulink environment by S-Function. The powertrain simulation model can be constructed by combining the different submodels provided by the POWERTRAIN library of AMESim.

a) Engine and torque converter

The function of engine output torque is always given as the map of engine rotational speed and engine throttle angle, which can be fitted as polynomial expression. Figure 4 shows the torque characteristic of the engine used in this study, i.e. a 2000cc injection gasoline engine.

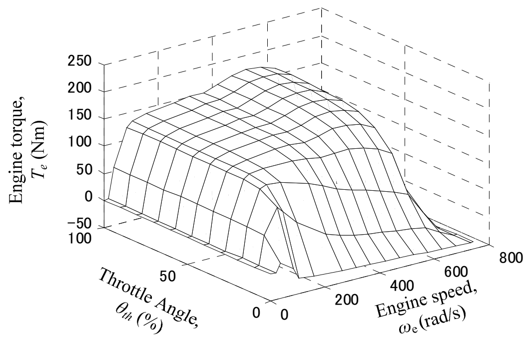

Figure 4 Engine torque map with speed and throttle opening

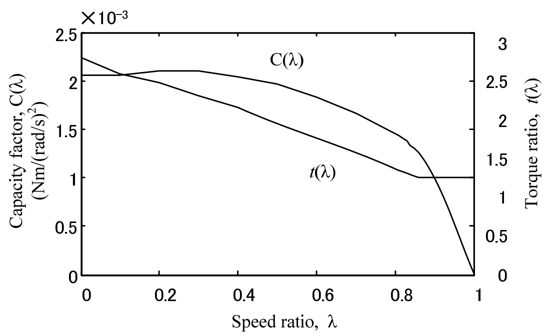

Figure 5 Capacity factor and torque ratio of torque converter

The dynamic properties of torque converter when turbine is driven forward are often characterized as follows:

$$T_p = C(\lambda)\omega_e^{\ 2} \qquad (14a)$$

$$T_t = t(\lambda)T_p \qquad (14b)$$

The capacity factor $C(\lambda)$ and torque ratio $t(\lambda)$ used here are given in Figure 5.

b) Clutches and valves

Different from model for controller design, the friction coefficient of clutch plates μ is not constant but a function of $\Delta\omega$ with the relationship shown in Figure 6.

The dynamic parameters of the proportional valve are also time-variant according to different operating points.

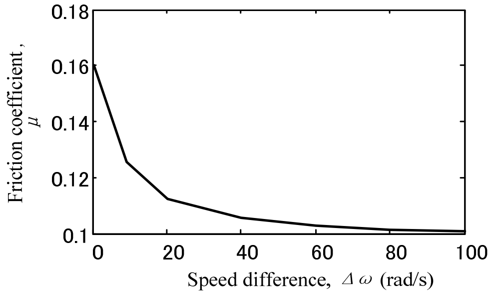

Figure 6 Friction characteristics of clutch plates

c) Driveshaft and tires

The two drive shafts between the differential gear and front wheels are represented as torsion spring with stiffness coefficient K_l and a torsion damping with damping coefficient C_l.

Only the longitudinal force of the tire is considered here. The longitudinal force of the tire F_x is represented as a tanh function of the longitude slip ratio S_x:

$$F_x = F_{x\max} \tanh\left(\frac{2S_x}{dS_x}\right) \qquad (15)$$

Simulation Results

1st to 2nd gear up shift is simulated. During the inertia phase deduced controller works to make speed difference of clutch B tracking desired trajectory.

The feedback gain $\boldsymbol{F}_1$ and $\boldsymbol{F}_2$ used here are

$$\boldsymbol{F}_1 = \left[-7.8\times10^{-3} \quad 1.9\times10^{-6}\right]$$

$$\boldsymbol{F}_2 = [-0.081]$$

and the value of p_0 is set to be

$p_0 = 30$.

The 1st to 2nd gear up shift simulation results are given by Figure 7. The gear shift process consists of three parts: before 7.94s, the 1st gear torque phase; after 8.24s, the 2nd gear torque phase and between 7.94 s and 8.24s, the inertia phase. During the toque phases the rotational speeds of shafts do not change greatly, while during the inertia phase, the rotational speeds change intensively because of the clutch slip.

The desired time of the inertia phase is set to be 0.3s. The simulation result of the speed difference of clutch B is shown in Figure 7(b), and $\Delta\omega$ and $\Delta\omega^{*'}$ are also given as well. It can be seen that the speed difference between turbine and ring gear $\Delta\omega$ can track reference value $\Delta\omega^{*'}$ without large error.

The angle of driveshaft θ_l is shown to examine the shift shock. It can be seen that at the time the inertia phase begins and ends, there is no sharp change in the electric current of valve B, which results in smooth change of driveshaft angle. Especially before 8.24s, the time clutch B locked up, the electric current of valve B decreases for a while, which makes the lock up of clutch smooth.

During the shift process, the engine is controlled to cooperate with the transmission shift. The throttle angle θ_{th} decrease and increase respectively when shift start and end, with angle and rotational speed decided in advance.

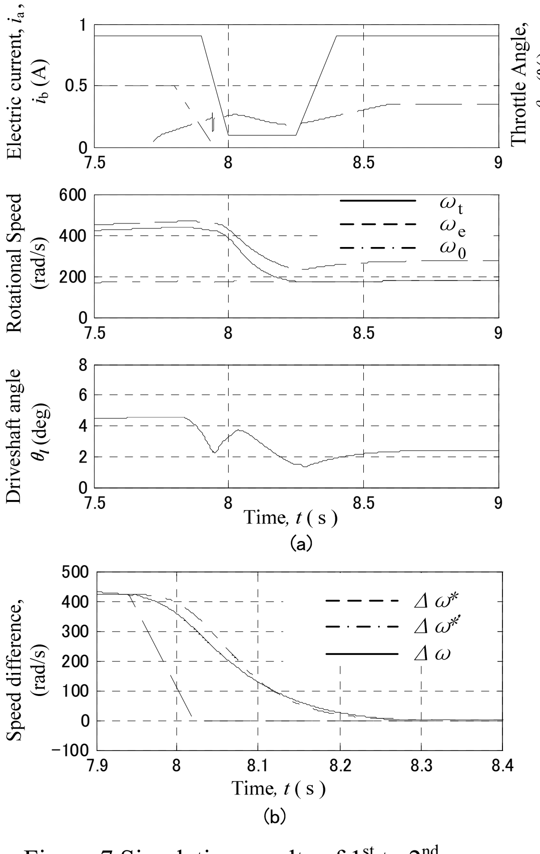

Figure 7 Simulation results of 1st to 2nd gear up shift

CONCLUSIONS

For the automatic transmission using proportional pressure control valves to control the clutches directly, a two-degree-of-freedom clutch slip controller is designed for the inertia phase of gear shift.

Given required shift time, the controller can be designed and simulation results show that the speed difference can track the desired trajectory well. The feed forward compensator can reduce shift shock as well as improve tracking performance.

ACKNOWLEDGMENTS

The work is partially supported by the National Science Fund of China for Distinguished Young Scholars under Grant No. 60725311 and by "985 Project" of Jilin University.

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