

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

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ABSTRACT

To improve shift quality of vehicle with clutch-to-clutch gear shifts, a two-degree-of-freedom controller, including a clutch pressure estimator, 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. Then the calculated controller is evaluated on the constructed nonlinear 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 feedforward compensator.

INTRODUCTION

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 the clutches independently [1]. In both 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. How to make the process smooth and efficient relies on the clutch slip control, accordingly, the control of cylinder pressure becomes very important.

In recent years, the small size but large flow rate proportional valve was developed for direct clutch pressure control without using the pilot duty solenoid valve [2]. New Automatic Transmission can use it to improve the ability of the transmission to adapt to different driving conditions, as well as reduce the cost and 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].

Two-degree-of-freedom controller design is suitable to many automotive control systems for it can fulfill good tracking performance and robustness simultaneously [5]. In [6], 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 slip 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. The designed controller, including the pressure estimator is evaluated on a whole powertrain simulation model. In particular, the simulation of gear shift process with engine control shows the robustness of the controller.

POWERTRAIN MODEL

POWERTRAIN DESCRIPTION

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. A planetary gear set is adopted as the shift gear, and two clutches are used as the actuators. Two proportional pressure control valves are adopted to control the two clutches respectively.

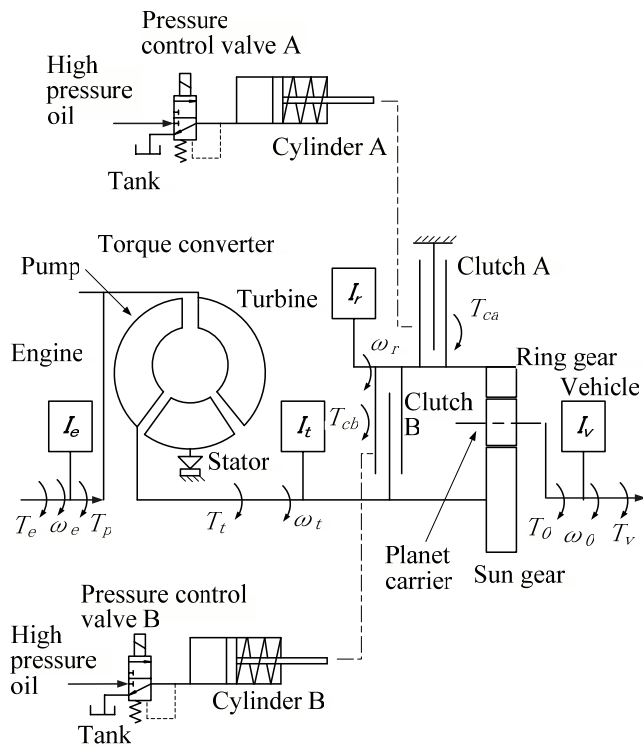


Fig. 1 Schematic graph of Automatic Transmission

When clutch A is engaged and clutch B disengaged, the powertrain is operating in 1st gear, the speed ratio is

$$i_1 = (1 + 1/\gamma), \quad (1)$$

where γ is the ratio of sun gear's teeth number to that of ring gear.

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

$$i_2 = 1 \quad (2)$$

POWERTRAIN SIMULATION MODEL

a) Engine

The dynamic equation of engine speed is described by

$$I_e \dot{\omega}_e + C_e \omega_e = T_e - T_p \quad (3)$$

where, I_e : inertia moment of engine crane and torque converter pump, C_e : damping coefficient of engine crane and torque converter pump, ω_e : engine rotational speed, T_e : engine output torque, T_p : torque converter pump torque.

Engine output torque can be simplified as the function of engine rotational speed and engine throttle angle if not considering the explicit representation of engine:

$$T_e = T_e(\omega_e, \theta_{th}) \quad (4)$$

where, θ_{th} : engine throttle angle.

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. Fig. 2 shows the torque characteristic of the engine used in this study, a 3000cc injection gasoline engine.

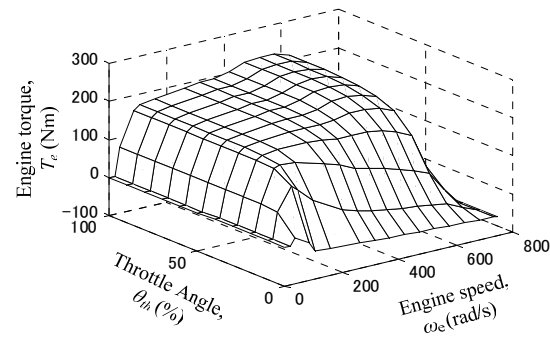


Fig. 2 Engine torque map with speed and throttle opening

b) Torque converter

The dynamic properties of torque converter when turbine is driven forward are often characterized as follows [3]:

$$T_p = C(\lambda) \omega_e^2 \quad (5a)$$

$$T_t = t(\lambda) T_p \quad (5b)$$

where, T_t : torque converter turbine torque, $C(\lambda)$: capacity factor of torque converter, $t(\lambda)$: torque ratio of torque converter, λ : speed ratio of torque converter, defined as:

$$\lambda = \frac{\omega_t}{\omega_e} \quad (6)$$

where, ω_t : torque converter turbine rotational speed.

The capacity factor $C(\lambda)$ and torque ratio $t(\lambda)$ used here are given in Fig. 3.

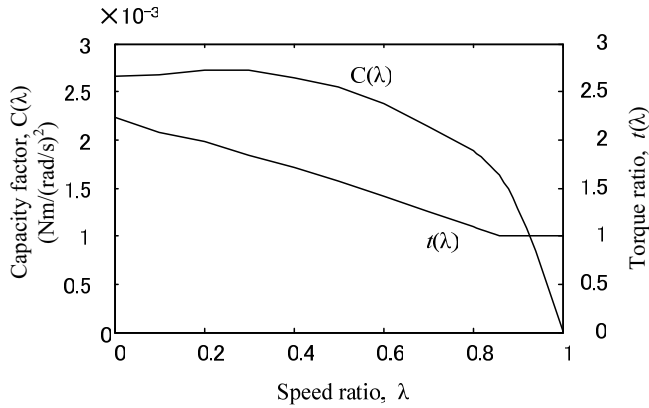


Fig. 3 Capacity factor and torque ratio of torque converter

c) Planetary gear set

Because the planetary gear set is a two-degree-of-freedom system, two states variables, such as ω_t and ω_r , can be used to describe its movement.

$$\dot{\omega}_t = C_{11}T_t + C_{12}T_{ca} + C_{13}T_{cb} + C_{14}T_v \quad (7a)$$

$$\dot{\omega}_r = C_{21}T_t + C_{22}T_{ca} + C_{23}T_{cb} + C_{24}T_v \quad (7b)$$

with constant coefficients:

$$\begin{aligned} C_{11} &= \frac{R_g^2 I_r + (R_g - 1)^2 I_v}{R_g^2 I_t I_r + (R_g - 1)^2 I_t I_v + I_r I_v}, \\ C_{12} &= -C_{11} \left[\frac{(R_g - 1) I_v}{R_g^2 I_r + (R_g - 1)^2 I_v} \right], \\ C_{13} &= -C_{11} \left[\frac{(R_g - 1) I_v}{R_g^2 I_r + (R_g - 1)^2 I_v} + 1 \right], \\ C_{14} &= C_{11} \left[\frac{-R_g I_r}{R_g^2 I_r + (R_g - 1)^2 I_v} \right], \\ C_{21} &= -\frac{(R_g - 1) I_v}{R_g^2 I_t I_r + (R_g - 1)^2 I_t I_v + I_r I_v}, \\ C_{22} &= -C_{21} \left[\frac{R_g^2 I_t + I_v}{(R_g - 1) I_v} \right], \\ C_{23} &= -C_{21} \left[\frac{R_g^2 I_t + I_v}{(R_g - 1) I_v} + 1 \right], \quad C_{24} = C_{21} \frac{R_g I_t}{I_v} \end{aligned}$$

where, ω_r : rotational speed of ring gear, T_{ca} : torque delivered by clutch A, T_{cb} : torque delivered by clutch B, T_v : resistant torque delivered from tire to drive shaft, which will be explained later, I_t : inertia moment of torque converter turbine. I_v : equivalent inertia moment

of vehicle body, which will be explained later, I_r : inertia moment of ring gear and parts connected, and

$$R_g = (1 + 1/\gamma) \quad (8)$$

d) Vehicle

The grade force is ignored here, and the resistance forces acted on the vehicle consists of two parts: the rolling resistant moment of tires T_w and aerodynamics drag F_A , of which, the aerodynamics drag is described as [8]:

$$D_A = \frac{1}{2} \rho C_D A_A V^2 = C_A V^2 \quad (9)$$

where, ρ : air density, C_D : aerodynamic drag coefficient, A_A : front area of the vehicle, V : vehicle speed, C_A : a constant coefficient,

$$C_A = \frac{1}{2} \rho C_D A_A \quad (10)$$

If vehicle has a differential gear with gear ratio of i_{df} , I_v and T_v shown in equation (7) can be obtained by:

$$I_v = \frac{m R_w^2}{i_{df}^2} \quad (11a)$$

$$T_v = \frac{T_w}{i_{df}} + \frac{C_A R_w^3}{i_{df}^3} \omega_0^2 \quad (11b)$$

where, m : vehicle mass, R_w : radius of tire, T_w : rolling resistant moment of tires, ω_0 : output speed of transmission.

e) Clutches and valves

Taking clutch B as an example, describes the dynamics of wet clutch and its actuator: cylinder and proportional pressure control valve. The transferred torque during clutch slipping is decided 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:

$$T_{cb} = \mu(\Delta\omega)RN \cdot (A p_{cb} - F_s) \quad (12)$$

where, T_{cb} : torque transferred by cylinder B, μ : friction coefficient of clutch plates, R : effective radius of push force acted on the friction plates, N : number of friction plates, A : piston area, p_{cb} : pressure of cylinder B, F_s : force of return spring, $\Delta\omega$: the speed difference between turbine and ring gear.