

Development of Ratio Control System for Toyota's New Continuously Variable Transmission

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ABSTRACT

Toyota has developed a new belt-type continuously variable transmission (CVT) for 1.5-liter compact vehicles. To improve both driveability and fuel economy over previous CVTs, pressure management was adopted as the shift control method. The new shift control system was designed using a model-based control method which uses a two-degree-of-freedom system composed of feedback and feedforward controls. Smooth shifting in all the target shift speed regions was realized by combining a feedback loop that considers the output limit of the pulley thrust into the feedforward controller. Furthermore, shift response was improved while maintaining or even improving stability. This paper describes the details of this shift control system.

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1. INTRODUCTION

Recent years have seen increasing demand for improved vehicle fuel economy to help resolve global environmental problems. In an effort to address this demand, vehicles equipped with continuously variable transmissions (CVTs) have grown in popularity throughout the world. Use of a CVT enables superior fuel economy performance because the engine can operate at the optimum fuel consumption point more often and the fine ratio control allows fuel injection to be stopped for extended periods when the vehicle is coasting. Furthermore, CVTs can have driveability superior to automatic transmissions (ATs) due to better shift response as well as smooth, shock-less shifting.

Conventionally, flow control-type CVTs have been used, which have excellent control of the target ratio in a steady state. This is because the ratio is in a state of equilibrium when the flow rate (i.e., the controlled variable) is at 0 (zero). However, one issue with these CVTs is tracking performance when the target ratio is in a transient state. Since the opening of the flow control valve is governed by the estimated pressure before and after the valve, it is easily affected by numerous variables. Consequently, it is difficult to construct effective feedforward (FF) controls so this type of CVT is largely dependent on feedback (FB) control.

Improving the precision of ratio control leads to improved driveability and fuel economy. Therefore, a pressure control-type CVT was developed (Fig. 1). A pressure control-type

CVT has excellent control when the target ratio is in a transient state. This is because a FF control that uses shift characteristics is easy to apply [1, 2, 3]. However, controllability with this type of CVT is more difficult to achieve when the target ratio is in a steady state, because the hydraulic pressure (i.e., the controlled variable) must be changed appropriately in relation to the change in the input torque, even when the target ratio is constant.

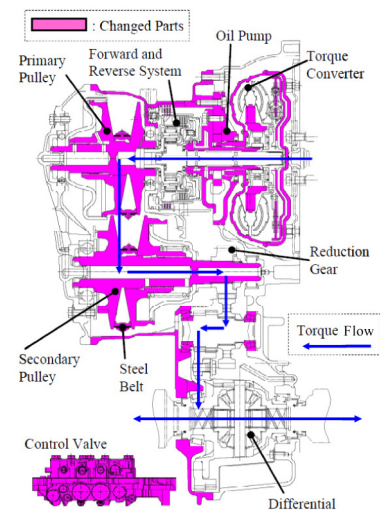


Figure 1. Cross section of new CVT

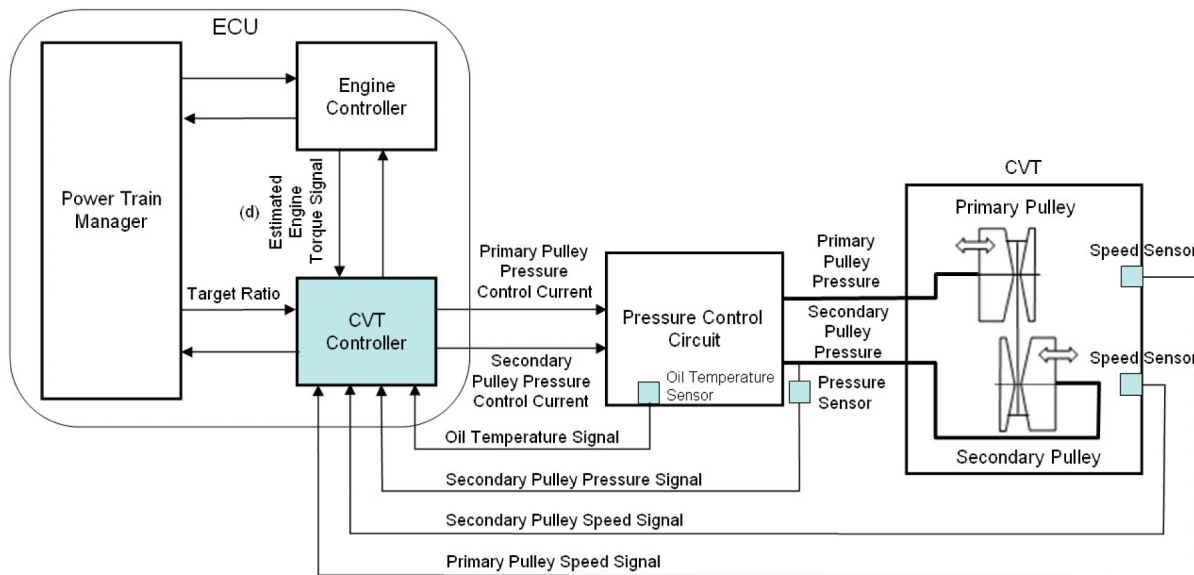


Figure 2. Configuration of CVT shift control system

Based on the characteristics described above, the shift management of the pressure control-type CVT was designed to maximize the effects of FF control over all the target shift speed regions. This control achieved both fast shift response in accordance with the driver's operations and smooth shifting. This paper describes the details of this shift control system.

2. SYSTEM CONFIGURATION

2.1. Configuration of CVT Shift Control System

Figure 2 shows the configuration of the newly developed CVT shift control system. The CVT controller observes the speed and ratio via the speed sensors equipped on each pulley. It also receives the estimated engine torque from the engine controller and the target ratio from the powertrain manager [4]. The necessary pressures for the primary and secondary pulleys are then calculated using these signals to track the target ratio. The results of these calculations are used to develop instructions for the primary and secondary pulley pressure control currents which are sent to the linear solenoids of the pressure control circuit.

2.2. Pressure Control Circuit

Figure 3 shows the pressure control circuit of the new CVT. The circuit is equipped with two separate linear solenoids and two separate pressure control valves so the primary and secondary pulley pressures can be controlled independently. In addition, a pressure sensor is equipped on the secondary pulley side. The secondary pressure can be maintained with high precision with the help of the sensing value as feedback.

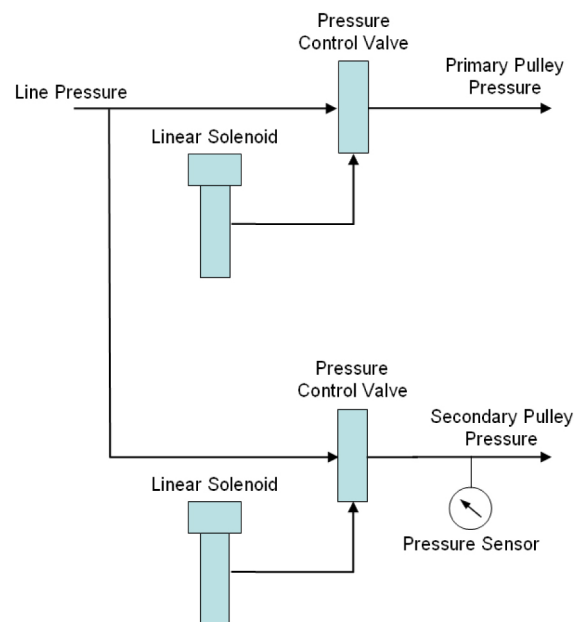


Figure 3. Pressure control circuit of new CVT

3. MODELING OF BELT-TYPE CVT

This section describes details on the modeling of the CVT, which was necessary to achieve model-based control. The shift characteristics of the CVT can be modeled by combining the steady-state and transient-state characteristics. The following list defines the physical quantities necessary for the modeling.

- N_p : Primary pulley speed
- N_s : Secondary pulley speed

- γ : Ratio (N_p/N_s)
- W_p : Primary pulley thrust
- W_s : Secondary pulley thrust
- τ : Thrust ratio in steady state (W_s/W_p)
- μ : Coefficient of friction between belt and pulley
- α : Half wedge angle of the pulley V-surface
- R_p : Pitch radius of primary pulley belt
- R_s : Pitch radius of secondary pulley belt
- T_p : Primary pulley input torque
- T_s : Secondary pulley input torque (γT_p)
- T_{plmt} : Primary pulley maximum transmittable torque without sliding slip ($2\mu R_p W_p / \cos\alpha$)
- T_{slmt} : Secondary pulley maximum transmittable torque without sliding slip ($2\mu R_s W_s / \cos\alpha$)
- κ_p : Primary pulley torque ratio (T_p/T_{plmt})
- κ_s : Secondary pulley torque ratio (T_s/T_{slmt})

3.1. Steady-State Shift Characteristics

Constant CVT ratios can be uniquely determined from the thrust ratio and torque ratio [1]. This relationship can be theoretically derived by solving a force-balance equation that considers the coefficient of friction of the belt, the tensile force of the belt, and the element compression force. In this paper the steady-state characteristics of the shift are modeled with the following equations using the ratio (γ), thrust ratio (τ), and torque ratio (κ).

$$\gamma = f(\tau, \kappa_p) \quad (1)$$

$$\gamma = g(\tau, \kappa_s) \quad (2)$$

Figure 4 and Figure 5 show the steady-state shift characteristics expressed by Equation (1) and Equation (2), respectively. Equation (1) is used when calculating the secondary pulley thrust to balance a certain ratio from the primary pulley thrust. In contrast, Equation (2) is used when calculating the primary pulley thrust to balance a certain ratio from the secondary pulley thrust.

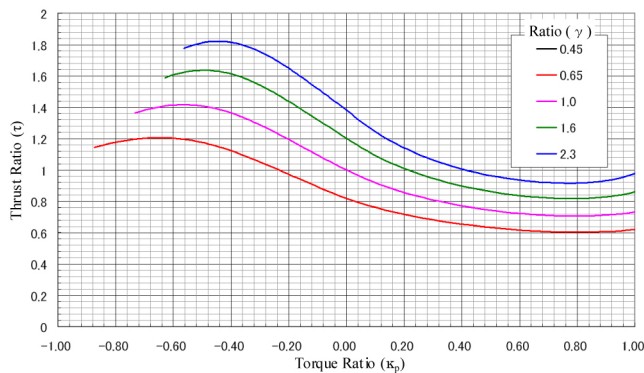


Figure 4. Steady-state shift characteristics of CVT (relationship between κ_p , γ , and τ)

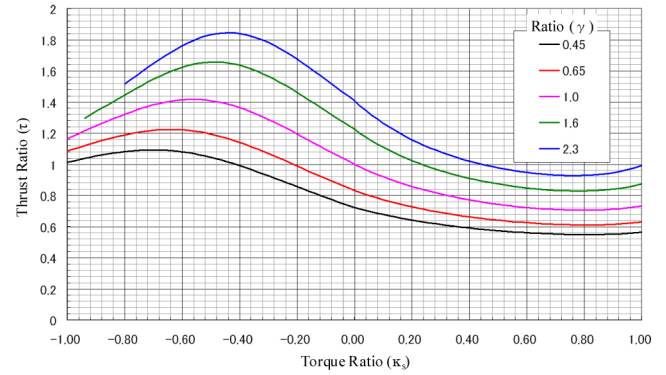


Figure 5. Steady-state shift characteristics of CVT (relationship between κ_s , γ , and τ)

3.2. Transient-State Shift Characteristics

When the ratio of the CVT is constant and thrust is added or subtracted from one of the pulleys, the steady state will deteriorate and the ratio will change. This addition or subtraction of thrust is referred to as shifting thrust. Shift speed is determined by the magnitude of this shifting thrust. It is understood from experimentation that the shift speed is principally dependent upon the shifting thrust, pulley speed, and ratio. The shift speed becomes faster as the shifting thrust or pulley speed increases [2-3]. In this paper, the transient-state characteristics of the shift are modeled with the following equations using the ratio (γ), pulley speed (N_p), and shifting thrust (ΔW).

$$\frac{d\gamma}{dt} = m(\gamma, N_p) \times \Delta W_p \quad (3)$$

$$\frac{d\gamma}{dt} = n(\gamma, N_p) \times \Delta W_s \quad (4)$$

Here, the terms ΔW_p and ΔW_s refer to the shifting thrust applied to the primary and secondary pulleys, respectively. These terms can be expressed by the following equations.

$$\Delta W_p = \frac{W_s}{\tau(\gamma, \kappa_s)} - W_p \quad (5)$$

$$\Delta W_s = W_s - W_p \tau(\gamma, \kappa_p) \quad (6)$$

The function $m(\gamma, N_p)$ is nearly equal to the function $n(\gamma, N_p)$, and has different transient-shift characteristics during upshift and downshift. Figure 6 and Figure 7 show the function $m(\gamma, N_p)$ during upshift and downshift, respectively.