Dynamic Analysis of Shift Quality for Clutch to Clutch Controlled Automatic Transmission

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Dynamic characteristics of a shift quality are investigated for a clutch to clutch controlled automatic transmission. Dynamic models of the hydraulic control system and the geartrains are obtained using Bondgraph. Based on the dynamic models, a simulation tool is developed to evaluate the shift quality. Using the simulation tool, shift performances for a 1-2 upshift and a 4-2 downshift are investigated and are compared with test results. It is found that the simulation tool is able to predict the pressure profiles of the clutch and brake, output torque, and turbine speed for a given input duty signal. It is expected that the simulation tool developed in this work can be used to improve the shift quality for the clutch to clutch controlled automatic transmission by saving time and cost in the tests.

Key Words : Clutch to Clutch Controlled Automatic Transmission, Shift Quality, PCV, PCSV

1. Introduction

Shift quality is an integral part of an automatic transmission design. In a clutch to clutch shifting, where release clutch is changed over to an apply clutch during shifting, oil pressure profile in the clutch is directly related to the shift quality. In Fig. 1, a schematic diagram of the clutch control system for the clutch to clutch controlled automatic transmission is shown. All the clutches are independently controlled by the separate hydraulic module. Each of the hydraulic module consists of a pressure control solenoid valve (PCSV) and a pressure control valve (PCV). The PCSV generates the first control pressure according to the input signal from a transmission electronic control unit. The PCV determines the second control pressure by the first control pressure from the PCSV. The second control pressure from the PCV is applied to the friction element such as a clutch or a brake, and the behavior of the friction element is determined by the pressure profile of the second control pressure. Since the shift quality in the automatic transmission depends on the torque change caused by engagement or disengagement characteristics between the friction elements, it is very important to predict the pressure profile on the friction element and to investigate the effect of the pressure profile on the torque change of a geartrain in order to obtain a desired shift quality.

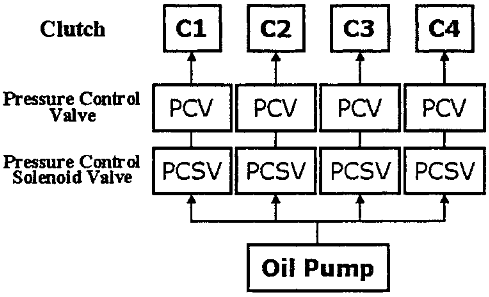

Fig. 1 Schematic diagram of a clutch to clutch system

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Since the elaborate shift control was introduced by employing the clutch to clutch control technology (Usuki et al., 1996), many researches on the subject have been reported. A linearized model for the turbine angular acceleration was presented for a clutch to clutch controlled automatic transmission to estimate the output torque which is directly related to shift quality (Jung, Cho and Lee, 1999). A control strategy was suggested to improve the shift quality by using a speed sensor including the effect of heat dissipation (Jeong and Lee, 2000). However, they used pre-determined clutch pressure instead of obtaining the pressure from dynamic models of the hydraulic control valves. If the clutch pressure can be calculated from accurate dynamic models of the hydraulic control valves for a given duty input, it will be more convenient to estimate the shift quality when the input duty changes. Since the hydraulic control valves have nonlinear characteristics, it is inevitable to depend on experimental results in describing dynamic characteristics of the valves.

In this study, dynamic models of a powertrain for a clutch to clutch controlled automatic transmission is obtained including both hydraulic control valves and geartrain. Based on the dynamic models, a simulation tool is developed to investigate the shift quality of the clutch to clutch controlled automatic transmission. Simulation results are compared with test results to validate the dynamic models obtained. Using the simulation tool, the effects of some design parameters of the hydraulic system on shift quality are investigated.

2. Clutch to Clutch Controlled Automatic Transmission

Figure 2 shows a schematic diagram of a 4-speed clutch to clutch controlled automatic transmission(AT) investigated in this study. The basic configuration of the AT includes a torque converter, two simple planetary gear sets, three sets of multi-plate clutches and two sets of multi-plate brakes on the primary shaft.

In Fig. 3, the hydraulic circuit of a release and an apply clutch is shown. Line pressure which is regulated through a regulator valve is supplied to the PCSV and the PCV of each clutch. A 3-way pulse width modulated solenoid valve is used as the PCSV. The PCSV generates the first control pressure according to the digital signal from an electronic control unit. As the first control pressure increases, the PCV spool moves to the left side and the supply port is connected to the control port, which supplies the apply pressure to the clutch. On the other hand, as the first control pressure of the PCSV decreases, the PCV spool moves to the right side. Then, the line from the clutch is connected to the exhaust port, which results in the release of the clutch.

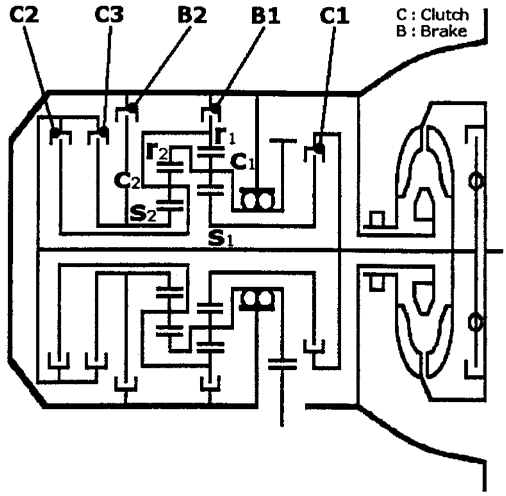


Fig. 2 Schematic diagram of a 4-speed clutch to clutch controlled AT

In the clutch to clutch shifting, a shift is carried out by changing the clutch. In Fig. 3, let us assume that the clutch-A is a release clutch and clutch-B is an apply clutch. The shift is performed as follows: the PCSV-A supplies the low first control pressure, which causes the PCV-A spool to move to the right side. Then, the exhaust port of the PCV-A opens and the clutch C1 begins to be released. Meanwhile, the PCSV-B supplies the high control pressure to the PCV-B so that the PCV-B spool can move to the left side. This causes the supply port to open and the second control pressure is generated in the clutch chamber, which results in the engagement of the clutch C2.

3. Dynamic Model of Hydraulic Control Valve

Operation of the PCSV in Fig. 3 depends on the force balance among the hydraulic pressure force, spring force and magnetic force which is applied according to the duty ratio. In this study, the PCSV is modeled as a second order system (Cho et al., 1999) based on the experimental results as follows,

$$P_{c1}=\frac{K_1\omega_n^2}{s^2+2\zeta\omega_n s+\omega_n^2}D+K_2 \tag{1}$$

where P_{cl} is the first control pressure, D is the input duty, K_1 and K_2 are constants, ζ is the damping coefficient, ω_n is the natural frequency. ζ and ω_n are determined by comparing the simulation results with the experiments.

Figure 4 shows simulation results of the PCSV characteristics compared with the experimental results. In Fig. 4(a), steady state characteristics of the PCSV are shown. It is noted from Fig. 4(a) that the PCSV control pressure shows linear

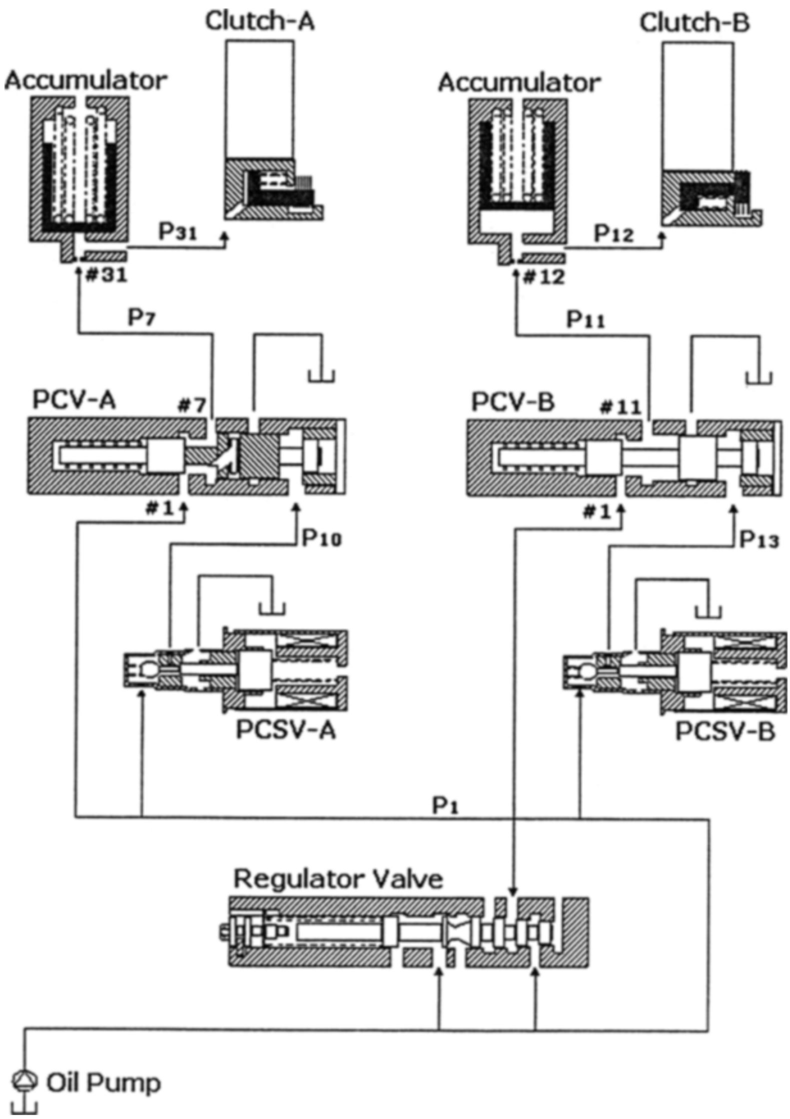
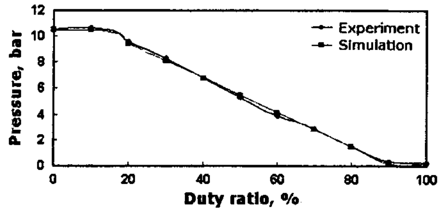
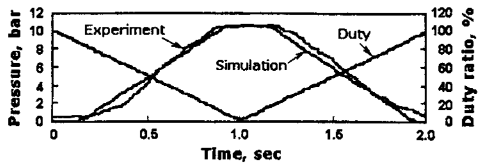


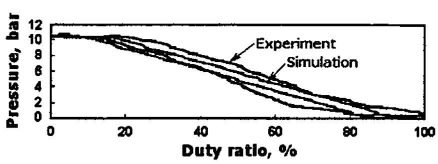
Fig. 3 Hydraulic circuit of release clutch and apply clutch



(a) Steady State Characteristics of PCV



(b) Dynamic Characteristics of PCSV



(c) Dynamic Characteristics of PCSV

Fig. 4 Characteristics of PCSV

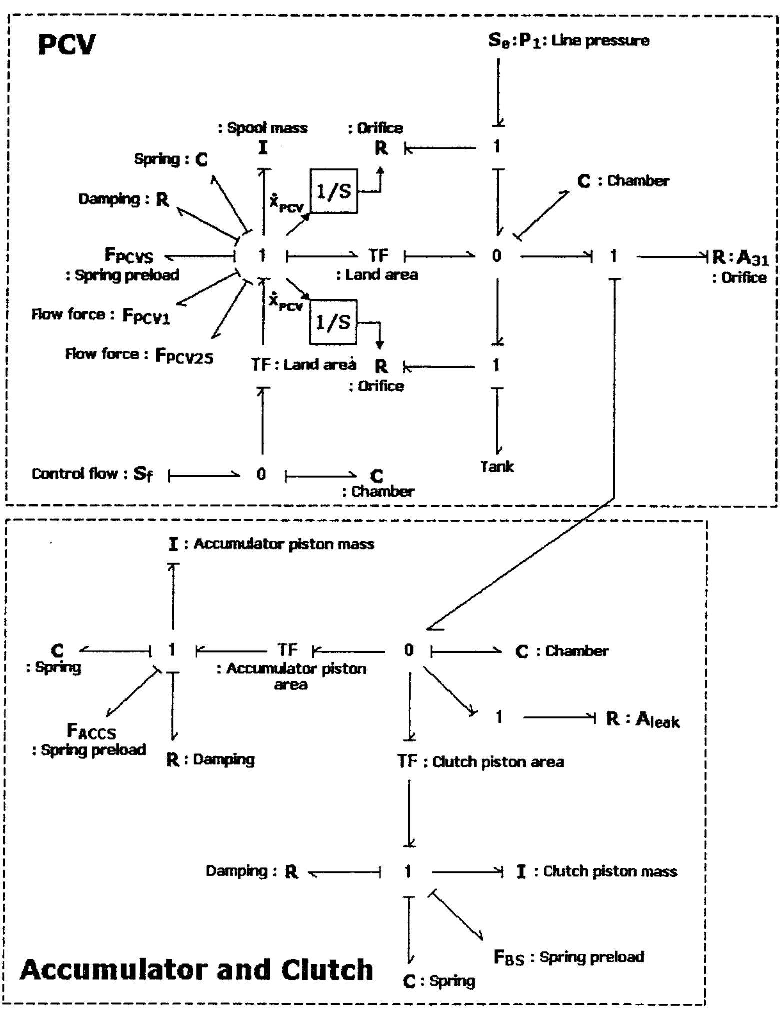


Fig. 5 Bondgraph model of PCV with accumulator and clutch

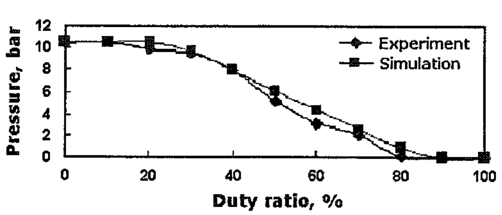
characteristics for the duty ratio of 20~90%. The simulation results agree well with the experimental results. In Fig. 4(b) and (c), dynamic responses of the PCSV are shown for a ramp input with 0.5Hz sweep frequency. A pressure hysteresis is observed in the transient response. The simulation results of the dynamic characteristics are in accordance with the experimental results, which demonstrates the validity of the dynamic model of the PCSV obtained.

Figure 5 shows a Bondgraph model of the PCV, accumulator and clutch. From the Bondgraph model, state space equation for the second control pressure by the PCV, P is obtained as,

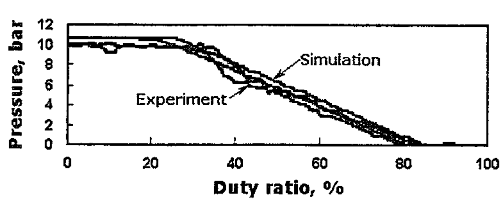
$$\frac{V+A_{ACC}x_{ACC}+A_{CLUTCH}x_{CLUTCH}}{\beta}\dot{P}=Q_{31}-Q_{leak}+A_{ACC}\left[\frac{P_{ACC}}{M_{ACC}}\right]-A_{CLUTCH}\left[\frac{P_{CLUTCH}}{M_{CLUTCH}}\right] \quad (2)$$

where V is the control volume consisting of the accumulator and the clutch, A_{ACC} is the pressurized area of accumulator piston, A_{CLUTCH} is the pressurized area of clutch piston, x_{ACC} is the displacement of accumulator piston, x_{CLUTCH} is the displacement of clutch piston, β is the effective bulk modulus of oil, Q_{31} is the flow rate through orifice #31, Q_{leak} is the leakage flow rate, P_{ACC} is the momentum of accumulator piston, M_{ACC} is the mass of accumulator piston, P_{CLUTCH} is the momentum of clutch piston, M_{CLUTCH} is the mass of clutch piston.

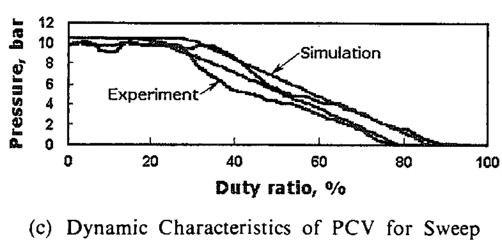
In Fig. 6, simulation results of the PCV control pressure are compared with the experimental results. As shown in Fig. 6(a), almost linear relationship is observed for the duty range of 30~80% at a steady state. In Fig. 6(b) and 6(c), dynamic responses of the PCV control pressure by the simulation are compared with those of the experiments for the ramp input with various sweep frequencies. It is noted that magnitude of the pressure hysteresis increases as the sweep frequency increases. From Fig. 6, it is found that the simulation results are in accordance with the experiments.



(a) Steady State Characteristics of PCV



(b) Dynamic Characteristics of PCV for Sweep Frequency 0.25Hz



(c) Dynamic Characteristics of PCV for Sweep Frequency 0.5Hz

Fig. 6 Characteristics of PCV

4. Dynamic Model of Geartrain

As described earlier, the clutch to clutch controlled AT used in this study consists of 2 simple type planetary gears. Before deriving dynamic equation of the geartrain, it is required to define the geartrain structure, i. e., to identify

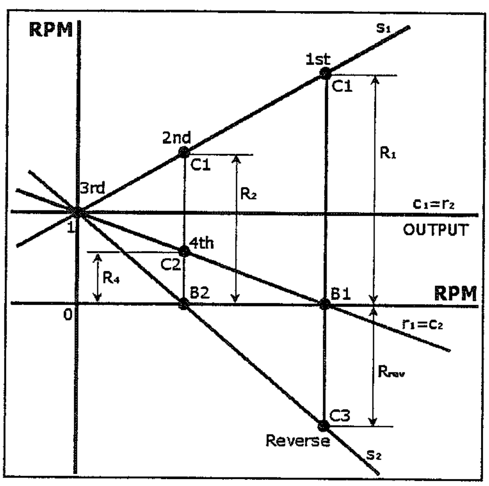


Fig. 7 R-R chart

which gear is connected to the input power during the shift or which gear is constrained. In defining the geartrain structure, R-R chart (Park, 2000) is useful. In Fig. 7, an R-R chart for a geartrain with 2 simple type planetary gears is shown. In the R-R chart, the units of vertical and horizontal axe are rpm (revolution per minute). From Fig. 2, the output of the geartrain is a carrier c_1 which is connected with a ring gear r_2. In addition, a ring gear r_1 is connected with a carrier c_2, In the R-R chart, the output speed is assumed to be 1 (one) rpm. Since the magnitude of the speed for a gear element in a simple planetary gear should be in the order of sun, carrier and ring gear or vice versa, the speed lines of s_1, c_1 and r_1 can be plotted as shown in Fig. 7. The speed lines of s_2, c_2 and r_2 can be plotted in a similar manner. All the lines cross 1 (one) rpm of the vertical axis, where the speed of every element is the same with the output speed. In the R-R chart, we can determine which element should be connected to the engine input by the clutch or should be constrained by the brake.

When s_1 is connected to the input and $r_1(c_2)$ is stopped by the brake, the output speed $c_1(r_2)$ is reduced by the ratio, $1/R_1$ which is defined as the 1st speed range. In the 1st speed, connection of s_1 to the engine input is achieved by applying the clutch C1. At the same time, $r_1(c_2)$ should be constrained by the brake B1. In a similar way, the 2nd speed can be obtained by taking the clutch C1 and the brake B2. The reduction ratio of the 2nd speed is obtained as $1/R_2$. In the 3rd speed, the input speed is the same with the output speed by taking the clutch C1 and C2, in other words, by engaging s_1 and $r_1(c_2)$ with the clutches. In the 4th speed, the output speed is increased by the ratio, $1/R_4$ by taking the clutch C2 and the brake B2. As for the reverse, a speed ratio R_{rev} is achieved by taking the clutch C3 and the brake B1. Now, we can see that 3 clutches and 2 brakes are required in order to provide 4-speed with 1 reverse range. In Table 1, application chart of the clutches and brakes are listed for each speed range of the clutch to clutch controlled AT used in this study.

Now, dynamic models of the geartrain are derived for a power-on upshift and a power-on downshift. As for the power-on upshift, 1-2 upshift is selected since the largest torque change is expected during the shifting.

Table 1 Friction element application chart

Selection lever position \ Friction element		Clutch			Brake	
		C1	C2	C3	B1	B2
Reverse				○	○	
D	1st	○			○	
	2nd	○				○
	3rd	○	○			
	4th		○			○

As shown in Table 1 and Fig. 2, the 1-2 upshift can be achieved by releasing the brake B1 and applying the brake B2. The clutch C1 is engaged during the 1-2 upshift. In Fig. 8, dynamic model for the 1-2 upshift is obtained by Bondgraph (Cho and Hedrick, 1989). From the Bondgraph model, state space equations are derived as follows,

$$\left[I_{t12}-\frac{I_{12}^2}{I_{c12}}\right]\dot{\omega}_t=T_t+\left[\frac{R_{s1}}{R_{r1}}-\frac{I_{12}}{I_{c12}}\frac{1}{R_{r1}}\right]T_{B1}+\left[\frac{I_{12}}{I_{c12}}\frac{1-R_{r1}R_{r2}}{R_{r1}R_{s2}}-\frac{R_{s1}}{R_{r1}R_{s2}}\right]T_{B2}-\frac{I_{12}}{I_{c12}}T_O \tag{3}$$

$$\left[I_{c12}-\frac{I_{12}^2}{I_{t12}}\right]\dot{\omega}_{c1}=\frac{I_{12}}{I_{t12}}T_t+\left[\frac{I_{12}}{I_{t12}}\frac{R_{s1}}{R_{r1}}-\frac{1}{R_{r1}}\right]T_{B1}+\left[\frac{1-R_{r1}R_{r2}}{R_{r1}R_{s2}}-\frac{I_{12}}{I_{t12}}\frac{R_{s1}}{R_{r1}R_{s2}}\right]T_{B2}-T_O \tag{4}$$

$$I_{t12}=I_t+I_{s1}+\left[\frac{R_{s1}}{R_{r1}}\right]^2 I_{c2}+\left[\frac{R_{s1}}{R_{r1}R_{s2}}\right]^2 I_{s2},$$

$$I_{c12}=I_{c1}+\frac{1}{R_{r1}^2}I_{c2}+\left[\frac{1-R_{r1}R_{r2}}{R_{r1}R_{s2}}\right]^2 I_{s2},$$

$$I_{12}=\frac{R_{s1}}{R_{r1}^2}I_{c2}+\frac{R_{s1}(1-R_{r1}R_{r2})}{(R_{r1}R_{s2})^2}I_{s2} \tag{5}$$

$$R_{r1}=\frac{Z_{r1}}{Z_{s1}+Z_{r1}},\ R_{r2}=\frac{Z_{r2}}{Z_{s2}+Z_{r2}},$$

$$R_{s1}=\frac{Z_{s1}}{Z_{s1}+Z_{r1}},\ R_{s2}=\frac{Z_{s2}}{Z_{s2}+Z_{r2}} \tag{6}$$

where ω_t is the turbine speed, ω_{c1} is the output shaft speed, T_t is the turbine torque, T_{B1} is the B1 brake torque, T_{B2} is the B2 brake torque, T_O is the output shaft torque, I is the inertia of the rotating

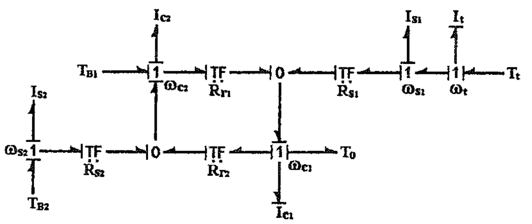


Fig. 8 Bondgraph model for 1-2 upshift

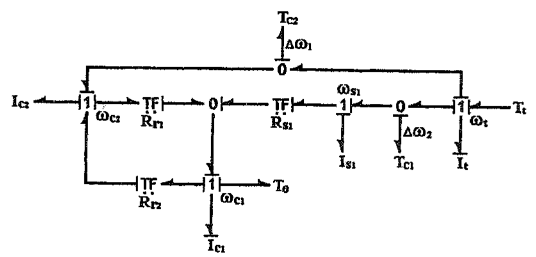


Fig. 9 Bondgraph model for 4-2 downshift

element, Z is the number of gear teeth. Subscript t means the turbine, s, r, c mean sun gear, ring gear and carrier, respectively. 1 and 2 mean the first and the second planetary gears shown in Fig. 2.

As for the power-on downshift, 4-2 downshift is investigated. In the clutch to clutch controlled AT, it is possible to perform the 4-2 downshift without passing through the 3rd speed range. In the 4-2 downshift, the clutch C2 is released and the clutch C1 is applied while the brake B2 is engaged. Bondgraph model for the 4-2 downshift is shown in Fig. 9. From the Bondgraph model, state space equations are derived as follows,

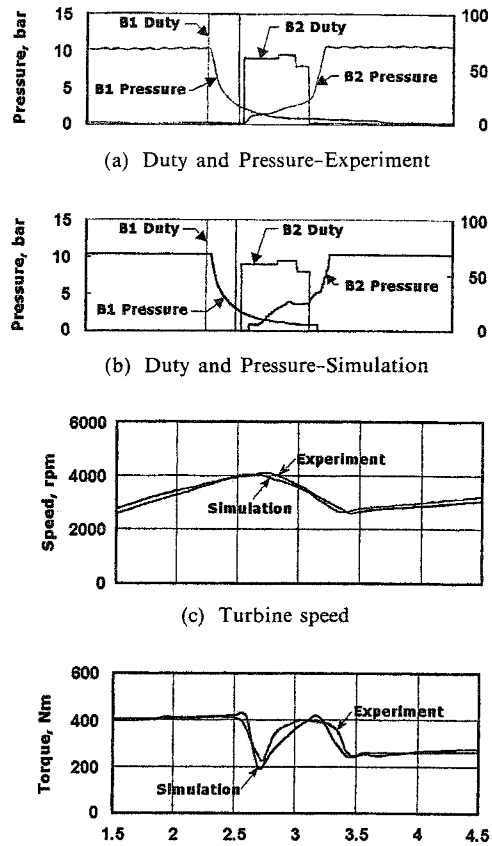
$$I_t \dot{\omega}_t = T_t - T_{C1} - T_{C2} \tag{7}$$

$$I_{C42} \dot{\omega}_{C1} = \frac{1 - R_{r1} R_{r2}}{R_{s1}} T_{C1} + R_{r2} T_{C2} - T_O \tag{8}$$

where T_{C1} is the C1 clutch torque, T_{C2} is the C2 clutch torque,

$$I_{C42} = I_{C1} + R_{r2}^2 I_{C2} + \left[\frac{1 - R_{r1} R_{r2}}{R_{s1}}\right]^2 I_{s1}$$

In solving Eqs. (3) ~ (8), the output torque T_O is determined from the lateral vehicle dynamics. In addition, engine and torque converter are modeled based on characteristic curves at a steady state. Using the dynamic models of the powertrain, a simulation tool is developed to investigate the shift performance of the clutch to clutch controlled AT.



(a) Duty and Pressure-Experiment

(b) Duty and Pressure-Simulation

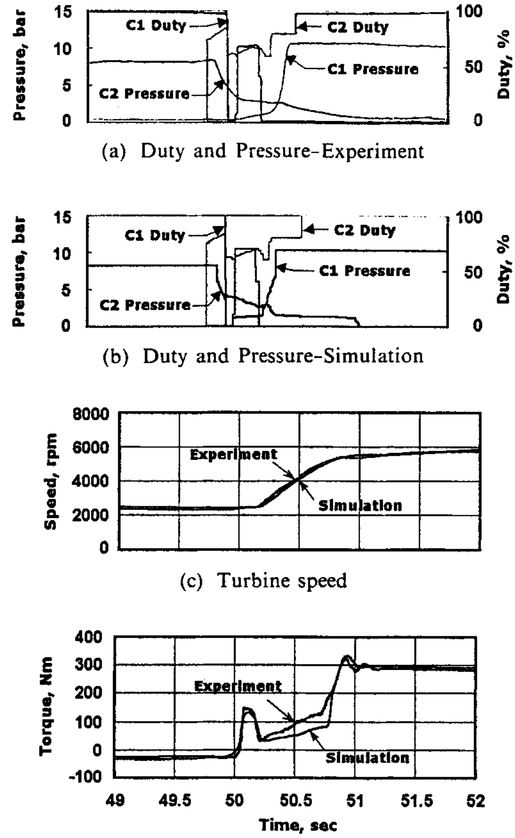
(c) Turbine speed

(d) Output torque

Fig. 10 Reponses for 1-2 upshift

In order to validate the dynamic models of the powertrain obtained, simulations are carried out for the power-on 1-2 upshift and the power-on 4-2 downshift and are compared with the experimental results. In Fig. 10, transient response of the brake pressure, engine speed and the output torque are shown for the power-on 1-2 upshift. In the 1-2 upshift, the B1 brake plays as the release element while the B2 brake becomes the applying element. For the B1 brake, a duty signal is applied in stepwise manner from 0% to 100% in order to obtain a fast release. The B1 brake pressure decreases corresponding to the duty signal. The input duty of the B2 brake is supplied from 100% to 0% and remains for 0.04sec to supply the apply pressure in the initial stage. Then, the duty signal increases and remains around 60% to adjust the increasing rate of the apply pressure as shown in Fig. 10(a). The B2 brake pressure changes depending on the duty signal. Simulation results for the 1-2 upshift are shown in Fig. 10(b). In the simulations, the duty signals used in the experiments are applied. The B1 brake pressure drops rapidly to 5 bars for 0.2 seconds and decreases slowly. The pressure of the apply brake B2 is increased in response to the duty signal. The B2 brake pressure increases slowly from 1 bar to 3 bars where the slippage between the friction elements is allowed. Then, the B2 brake pressure increases rapidly up to the maximum pressure corresponding to the stepwise decrease of the duty signal from 60% to 0%. The output torque(d) decreases in the torque phase. In the inertia phase, the output torque increases and decreases to a steady state value of the 2nd speed. It is seen from the experiments and the simulations that it takes about 1 second to complete the 1-2 upshift. The simulation results of the brake pressure, output torque are in accordance with the experimental results in overall trends, which demonstrates the validity of the dynamic models of the powertrain including the hydraulic system.

In Fig. 11, transient response for the power-on 4-2 downshift are shown. As shown in Fig. 11 (a), the duty signal of the C2 clutch is applied from 0% to 80% in stepwise manner and is increased up to 100% showing some variation. This duty signal is supplied to provide a pressure profile of the release clutch C2. The C2 pressure decreases more slowly compared with the release pressure in Fig. 10(a) due to the different duty signal. The duty signal of the apply clutch C1 is supplied from 100% to 0%. The C1 duty is increased again and remains around 70% for 0. 2 seconds. This duty increase is required to adjust the rate of the pressure increase of the apply clutch. Comparing the duty signal of the C1 clutch with those of the B2 brake in Fig. 10(a), duration time of the C1 duty around 70% is relatively short, which means that the time period for the slow increase of the apply pressure becomes shorter. This can be seen from Fig. 11 (a) and 11(b). After the duty increase, the C1 duty drops to 0%, which results in the rapid



(a) Duty and Pressure-Experiment

(b) Duty and Pressure-Simulation

(c) Turbine speed

(d) Output torque

Fig. 11 Reponses for 4-2 downshift

increase of the C1 clutch pressure. The crossover pressure where the C2 pressure curve crosses the C1 pressure curve is about 3 bar, that is higher than the crossover pressure of the 1-2 upshift(Fig. 10(a)). This high crossover pressure is provided in order to minimize the slippage between the friction elements, which results in a swift shift. The output torque(Fig. 11(d)) increases from -20Nm to 300Nm. The negative torque in the initial stage is due to the engine brake torque caused by a zero throttle valve opening. It is found from Fig. 11 that the simulation results of the clutch pressure, engine speed and output torque agree with the experimental results.

From the comparison of the simulation results with the experimental ones in Figs. 10 and 11, it is proved that the dynamic models of the clutch to clutch controlled AT powertrain are valid and it is expected that the simulation tool developed in this study can be used to estimate the effect of the design parameter on the shift quality of the clutch to clutch controlled AT.

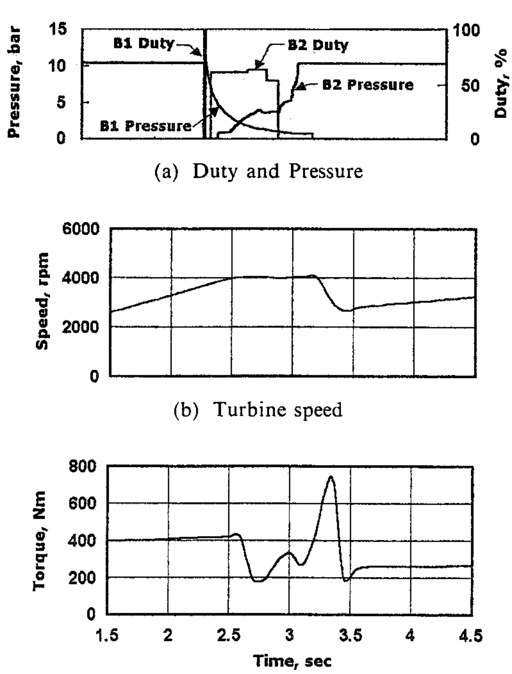
5. Prediction of Shift Quality

The clutch to clutch shifting requires not only apply clutch control but also release clutch control simultaneously. Accordingly, the prime importances in the shift control are the timing to switch from the release clutch to the apply clutch and the decreasing(increasing) rate of the release (apply) pressure. The timing of the pressure supply and release between the two clutches is adjusted so that the beginning of release clutch slippage and the beginning of engagement of the apply clutch may be accurately matched. In the followings, the effects of these design parameters on the shift performance are investigated using the simulation tool developed in this study.

In Fig. 12, simulation results of the 1-2 upshift are shown for a different timing of the pressure supply to the apply clutch. In order to investigate the effect of the apply pressure timing on the shift performance, the same input duty signal as shown in Fig. 10(a) is applied by changing the timing 0.25 seconds earlier than the duty in Fig. 10(a). As shown in Fig. 12(a), the pressure of the apply clutch shows the same profile with the one in Fig. 10(a) since the same duty signal is used. However, the pressure crossover point is shifted by 0.25 seconds due to the earlier apply pressure timing. It is seen from Fig. 12(c) that magnitude of the torque increase in the inertia phase is larger than those in Fig. 10(d).

From Fig. 12, it is found that the earlier timing of the supply pressure to the apply clutch results in a poor shift performance compared with the shift quality in Fig. 10(d).

Now, let us investigate the effect of decreasing rate of the release pressure on the shift performance. In Fig. 13, simulation results of the 4-2 downshift are shown for a different decreasing rate of the release pressure. As shown in Fig. 13(a), the input duty signal of the release clutch C2 is changed so that the decreasing rate of the release pressure can be increased. Consequently, the pressure crossover point is decreased from 3 bar

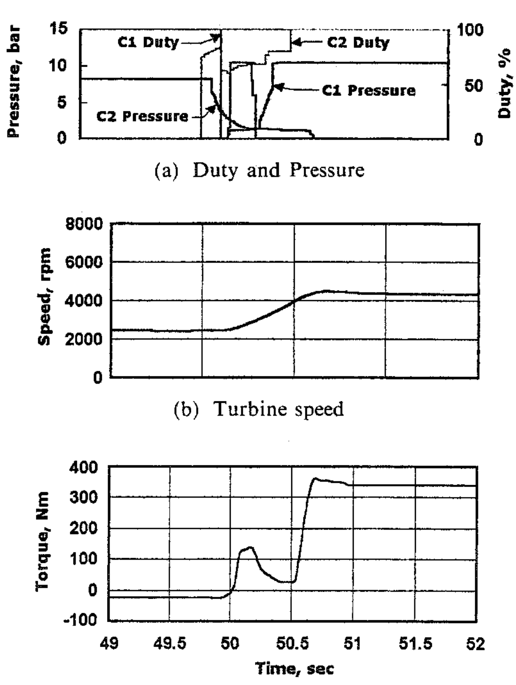


(a) Duty and Pressure

(b) Turbine speed

(c) Output torque

Fig. 12 Simulation results for 1-2 upshift



(a) Duty and Pressure

(b) Turbine speed

(c) Output torque

Fig. 13 Simulation results for 4-2 downshift

(Fig. 10) to 1.5 bar. This results in the lower turbine speed (Fig. 13(b)) at the 2nd speed compared with the turbine speed in Fig. 11(c). The change of the output torque in the inertia phase becomes larger than that of Fig. 11. However, the shift time is reduced by 0.2 seconds. Therefore, it is expected that better acceleration performance can be achieved due to the increased output torque and the reduced shift time even though increased torque change gives poor shift quality.

It is expected that the simulation tool developed in this study can be used in the improvement of the shift quality for the clutch to clutch controlled automatic transmission by saving the time and cost required in the tests.

6. Conclusion

Dynamic characteristics of a shift quality have been investigated for a clutch to clutch controlled automatic transmission. Dynamic models of a pressure control solenoid valve and a pressure control valve were obtained considering the ineffective duty range and hysteresis characteristics. In addition, dynamic models of the geartrain were derived by Bondgraph modeling method. Based on the dynamic models of the powertrain, a simulation tool has been developed to evaluate the effect of design parameters on the shift quality. It is seen that the simulation results of the release and apply pressure, output torque are in accordance with the experimental results, which demonstrates the validity of the dynamic models obtained. Using the developed simulation tool, the effects of some design parameters on the shift performance were investigated. It was found that both the timing of the pressure supply and release between the clutches and the decreasing (increasing) rate of the pressure influence the shift performance. It is expected that the simulation tool developed in this study can be used in the improvement of the shift quality for the clutch to clutch controlled automatic transmission by saving the time and cost required in the tests.

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