Smooth shift control of automatic transmissions using a robust adaptive scheme with intelligent supervision

Deok-Ho Kim

Microsystem Research Center, Korea Institute of Science and Technology, P.O. Box 131, Cheongryang, Seoul, 130-650, Korea
Fax: 82 2 958 6910
E-mail: kim-dh@kist.re.kr

Kyung-Jinn Yang and Keum-Shik Hong*

School of Mechanical Engineering, Pusan National University, 30 Changjeon-dong, Kumjeong-ku, Busan, 609-735, Korea
Fax: 82 51 514 0685
E-mail: {jinnky, kshong}@pusan.ac.kr
*Corresponding author

Jin-Oh Hahn

Department of Mechanical Engineering, Faculty Board, Korea Air Force Academy, P.O. BOX 335-2, Ssangsu, Namil-Myeon, Cheongwon-Gun, Choongbook, Korea
E-mail: stardust@afa.ac.kr

Kyo-Il Lee

School of Mechanical and Aerospace Engineering, Seoul National University, Shillim-dong, Kwanak-gu, Seoul, 151-742, Korea
Fax: 82 2 883 1513
E-mail: lki@snu.ac.kr

Abstract: In this paper, a robust adaptive control scheme with an intelligent supervisor for vehicle powertrain systems is investigated. The control objectives are to provide smooth shift transients for passenger comfort and to improve components durability. The reaction carrier speed and the turbine speed during the inertia phase are controlled to track their desired speeds, respectively. The boundedness of all signals in the closed loop system and the convergence of the reaction carrier speed error near to zero are guaranteed by applying the Lyapunov stability analysis. The adaptive compensation controller with an intelligent supervisor is implemented to keep the turbine speed error near to zero and the deviation of the shift duration within an allowable range. The proposed control algorithm is implemented and evaluated on an experimental test setup.

Keywords: vehicle powertrain system; shift quality; robust adaptive control; stability; intelligent supervisor; adaptive neuro-fuzzy inference system.

Reference to this paper should be made as follows: Kim, D.-H., Yang, K.-J., Hong, K.-S., Hahn, J.-O. and Lee, K.-I. (2003) 'Smooth shift control of automatic transmissions using a robust adaptive scheme with intelligent supervision', *Int. J. of Vehicle Design*, Vol. 32, Nos. 3/4, pp.250–272.

Biographical notes: Deok-Ho Kim received the B.S. degree in mechanical engineering from Pohang University of Science and Technology (POSTECH), Korea, in 1998 and the M.S. degree in mechanical design and production engineering from Seoul National University, Korea, in 2000. In 1996, he was an exchange student in the department of mechanical and manufacturing engineering at University of Birmingham, UK. He was a Research Assistant in Institute of Advanced Machinery and Design, Seoul National University, during 1998–2000. He is currently a Research Scientist in Microsystem Research Center, Korea Institute of Science and Technology (KIST), Korea.

Mr. Kim was a recipient of Hogil-Kim Memorial Scholarship during his study in University of Birmingham, UK. He is a Member of KSME and ICASE. His research interests include micro/nano-mechatronics, microassembly, tele-robotics, Powertrain/ITS Control, and MEMS.

Kyung-Jinn Yang received the B.S. degree in mechanical engineering, the M.S. degree in mechanical and precision engineering, and the Ph.D. degree in mechanical and intelligent systems engineering from Pusan National University, Korea, in 1994, 1998, and 2002, respectively. He is now a postdoctoral researcher of the Department of Computational Intelligence and Systems Science, Tokyo Institute of Technology. During 1994–1995, he was with Ssangyong Oil Refining Company, Korea, where he worked on control problems of steam generation systems. In 2000, he studied as a visiting research engineer at the Guidance and Control Technology Laboratory of the National Space Development Agency (NASDA) of Japan. He has been a scholarship holder of SBS (Seoul Broadcasting System) Seoam Research Foundation during his Ph.D. studies. Dr. Yang's current research interests include nonlinear systems theory, robust adaptive control, distributed parameter system control, and vehicle powertrain control.

Keum-Shik Hong received the B.S. degree in mechanical design and production engineering from Seoul National University in 1979, the M.S. degree in mechanical engineering from Columbia University, New York, in 1987, and both the M.S. degree in applied mathematics and the Ph.D. degree in mechanical engineering from the University of Illinois at Urbana-Champaign in 1991. Since Dr. Hong joined the school of mechanical engineering at Pusan National University, Korea, in 1993, he is now Associate Professor. During 1982–1985, he was with Daewoo Heavy Industries, Incheon, Korea, where he worked on vibration, noise, and emission problems of vehicles and engines.

Dr. Hong serves as Associate Editor for Automatica (2000-date) and served as Associate Editor for the Journal of Control, Automation, and Systems Engineering (1997–2000). He is a Member of ASME, IEEE, KSME, KSPE, and ICASE. Dr. Hong's current research interests include nonlinear systems theory, adaptive control, distributed parameter system control, input shaping, vehicle control, and innovative control applications to engineering problems.

Jin-Oh Hahn received the B.S. and M.S. degrees from the department of mechanical design and production engineering, Seoul National University, Korea, in 1997 and 1999, respectively. He had been an Assistant Researcher of the Institute for Advanced Machinery and Design, Seoul National University, Korea, during 1997–1999, and a researcher in the department of mechanical engineering, University of Minnesota, U.S.A., during 1999–2000. He is currently an Instructor in the department of mechanical engineering, Korea Air Force Academy, Korea. His research interests include control design, state estimation, failure identification and re-configurable control for advanced vehicle control systems. He is a member of KSME and ICASE.

Kyo-Il Lee was born in July 6, 1942 at Seoul, Korea. He received the B.S. degree in mechanical engineering from Seoul National University in 1966, and both the Dipl.-Ing. in production engineering and the Dr.-Ing. in fluid power control from Technische Hochschule Aachen, Germany, in 1971 and 1978, respectively. Since he began his academic career as an Assistant Professor in the department of mechanical design and production engineering at Seoul National University in 1978, he became Professor in 1988, and had been the Head of school of mechanical and aerospace engineering from 1996 to 1998. Before he pursued his graduate work in Germany, he worked as a Planning Engineer at Korea Bearing Company from 1966 to 1967. He also serves in the Board of Directors of Korea Electric Power Corporation since 1996. His research interests include production engineering, fluid power control, and automotive engineering.

1 Introduction

Nowadays, an automatic transmission is more often used than a manual one in passenger cars. Thus, the smooth and precise gear shifting of an automatic transmission becomes more and more essential to improve the ride quality and mechanical efficiency. The purposes of powertrain control are primarily to reduce the shock occurring during the gear shifts for passenger comfort and to prevent an excessive slip, which may cause unnecessary wear of the prismatic parts. In the conventional method, testing and calibration have been done to tune the gains of a controller. But such an approach does not assure the optimal shift quality at all times due to the variations in the hydraulic system and external disturbances.

During the shift of gears, there always exists a shock, and this shock may degrade the passenger feeling. An active pressure control of hydraulic actuators is essential to improve the shift feeling by achieving the smooth torque transition during the gear shifts. However, the hydraulic part of an automatic transmission is one of the most complicated components in the entire vehicle, which involves many uncertain parameters and model complexity. It consists of a large number of power and control elements, for example, a torque converter, clutch actuators, orifices, planetary gear sets, etc. The shift quality is also greatly affected by the variation of engine torque. If the engine speed is not controlled during a gear shift, the transmission controller will try to overcome the excessive torque transmitted from the engine during the inertia phase. This causes a decrease in fuel efficiency and results in a mismatch between clutch speeds [1].

In addition to the difficulties existing in the analysis and control of the shift mechanism, from the viewpoint of an integrated engine and transmission control, the vehicle powertrain system involves several uncertain variables that affect the shift mechanism. Those uncertainties include the change in clutch characteristics, the change in oil temperature, and disturbances due to input torque loss and external loads. In particular, the shift process consists of many operating modes and is highly affected by the nonlinear dynamics of the vehicle powertrain system. The assurance of smooth shift transitions for all possible operating modes is a challenging issue.

A number of applications of modern control theory to the powertrain system have been investigated [2–10], and various control algorithms for shifting process have been extensively studied [11–22]. A torque estimation method of vehicle axle shafts was studied by using inexpensive speed sensors to facilitate the sliding mode control algorithm [23].

However, an inaccuracy estimation of the torque, caused by the degraded performance of engine and a change in torque converter characteristics, may deteriorate shift quality.

A robust control algorithm for the turbine speed in the inertia phase during the 1–2 upshift was developed based on a linearized model and its performance was evaluated by using a nonlinear powertrain model [22]. A robust integrated engine–transmission control scheme was investigated for the clutch-to-clutch shift to deal with uncertainties in the vehicle powertrain dynamics [21]. In robust controls, a fixed controller to guarantee its robust behaviour within a realistic plant variation can be designed. However, the range of potential modelling error in vehicle powertrain systems is quite large, due to the fluctuating operation environment, wear of components, deterioration of oil, etc. Therefore, the application of a nonlinear control strategy to the power transmission control with an additional adaptation law estimating uncertainties of the system is necessary.

However, it is difficult to use a model-based engine control scheme for smooth power transmission since the vehicle powertrain system involves complex, uncertain, and highly nonlinear dynamics. One effective approach in dealing with such complexities and uncertainties is an intelligent control scheme such as fuzzy and neural network controls. For the realization of smooth shift control in this paper, an adaptive neuro-fuzzy inference system (ANFIS) [24] is utilized as a supervisory control and then an adaptive compensation scheme based upon the shift characteristics is designed for the engine control.

The present paper makes the following contributions: A robust integrated engine–transmission control of a vehicle powertrain system is investigated for enhancing shift quality. To analyse the shift transient phenomena, the dynamic models for various powertrain components, such as engine, torque converter, transmission, hydraulic line, and drivetrain, are developed. Using the Lyapunov stability analysis, a robust adaptive control law is derived for the transmission control that reduces the output torque during the gear shifts. The convergence of the tracking error near to zero is assured. To prevent an excessive clutch slip, an adaptive learning shift controller is developed for the engine control by synthesizing several basic ideas from neuro-fuzzy and conventional adaptive controls. In deriving the control laws, both the engine and automatic transmission dynamics have been included. The integrated controller proposed uses only angular velocity signals, which are inexpensive to measure, such as engine speed, turbine speed, and output speed, instead of torque signals.

The paper is structured as follows: In Section 2, the dynamic models for powertrain components are developed. In Section 3, by analysing the characteristics of a shift process, two control objectives are stated. In Section 4, a robust adaptive control scheme in order to reduce the shift shock is proposed. Using an appropriate Lyapunov function candidate, the tracking error convergence near to zero is established. Also, an adaptive compensation scheme to achieve an improved shift quality in the presence of system variations based upon an intelligent supervisory control using the neuro-fuzzy inference system is described. In Section 5, the performance of the proposed controller is investigated via experiments. It is shown via experiments that the controller designed provides much enhanced shift smoothness and an improved clutch longevity. Conclusions are given in Section 6.

2 Vehicle powertrain model

Figure 1 shows the vehicle powertrain system considered in this paper. It consists of an engine, a torque converter, a power transmission, a driveline, and a hydraulic control system. The power produced by the engine is delivered to the driving wheels through the power transmission and the automatic transmission changes gears by engaging and disengaging the hydraulically driven clutches.

2.1 Engine

The engine is a complex and highly nonlinear sub-system to model. The equation used to describe the engine dynamics is

$$I_e \dot{\omega}_e = T_e(\omega_e, \alpha) - T_p, \tag{2.1}$$

where I_e is the engine inertia (kg m^2), ω_e is the engine speed (rpm), α is the throttle angle (%), T_p is the torque converter pump torque (N m), and T_e is the engine torque that is a nonlinear function of the engine speed and the throttle angle. Figure 2 shows a typical engine map, which characterizes the steady state characteristics of the engine torque.

Figure 1 Block diagram of the vehicle powertrain system

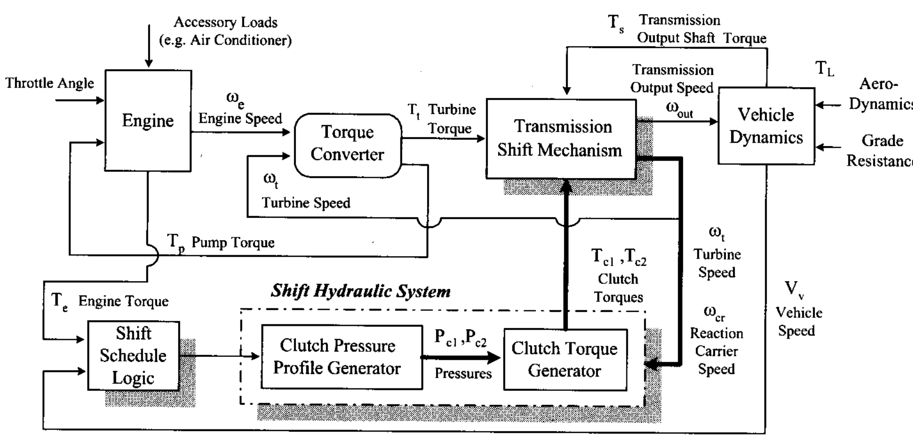
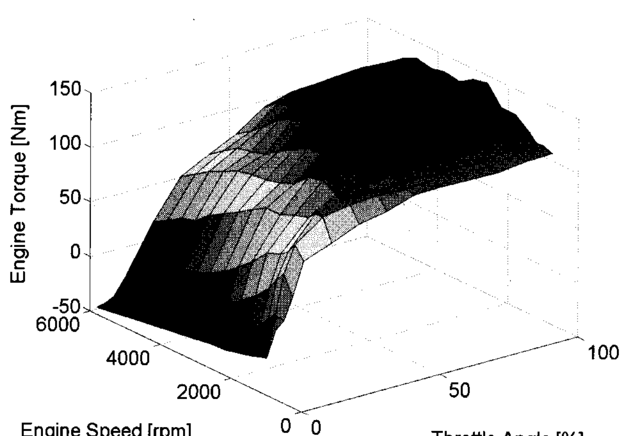


Figure 2 A typical nonlinear engine map



2.2 Torque converter

The torque converter consists of a pump (input), a turbine (output), and a stator (reaction member). The pump is attached directly to the engine and therefore turns at engine speed. Torque is transferred to the turbine as a result of the induced oil flow from the pump. The steady-state pump torque T_p and turbine torque T_t of a torque converter can be presented as follows:

$$T_p = C_p(\omega_e, \omega_t)\omega_e^2, \tag{2.2}$$

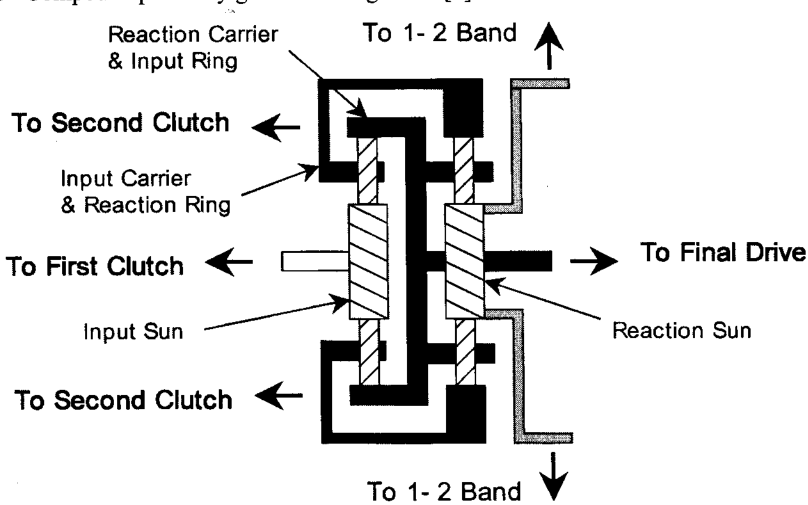
$$T_t = T_r(\omega_e, \omega_t)T_p, \tag{2.3}$$

where C_p is the capacity factor and T_r is the torque ratio. C_p and T_r are nonlinear functions of the engine and turbine rotational speeds, respectively, which are provided by the manufacturer in the form of a look-up table.

2.3 Transmission model

The automatic transmission consists of several planetary gears and the associated clutches and bands. In this paper, a two-state transmission model is considered for power-on upshift. As shown in Figure 3, the core of the transmission combines two planetary gears. The detailed operational principles can be found in [25].

Figure 3 Compound planetary gear set arrangement [3]



The turbine dynamics in first gear is given by

$$I_{t1}\dot{\omega}_t = T_t - R_1 R_d T_s, \tag{2.4}$$

where $I_{t_1} = I_t + I_{si} + R_1^2 I_{cr} + R_1^2/R_2^2\, I_{ci}$, ω_t is the turbine speed, R_1 is the first gear speed reduction ratio, R_2 is the second gear speed reduction ratio, R_d is the final drive speed reduction ratio, T_s is the axle torque (both sides combined), I_t is the converter turbine and chain inertia, I_{si} is the input sun inertia, I_{cr} is the reaction carrier inertia, and I_{ci} is the input carrier inertia.

When the vehicle reaches a shift speed, the hydraulic control circuit applies pressure on the second clutch. This initiates the starting of 1–2 upshift. The turbine dynamics during the torque phase is modelled as

$$I_t\dot{\omega}_t = T_t - R_1 R_d T_s - \left(1 + \frac{R_2}{R_1}\right) T_{c2}, \tag{2.5}$$

where T_{c2} is the torque on the second clutch, i.e., oncoming clutch. During the inertia phase, the dynamics of the converter turbine and the carrier become

$$I_t\dot{\omega}_t = T_t - T_{c2}, \tag{2.6}$$

$$I_{cr12}\dot{\omega}_{cr} = \frac{T_{c2}}{R_2} - R_d T_s, \tag{2.7}$$

where $I_{cr12} = I_{cr} + \frac{I_{si}}{R_1^2} + \frac{I_{ci}}{R_2^2}$.

When the slip speed of the second clutch ($\Delta\omega_{c2}$) reaches zero, the second clutch locks up. Hence, the state equation in the second gear is modelled as

$$I_{t2}\dot{\omega}_t = T_t - R_2 R_d T_s, \tag{2.8}$$

where $I_{t_2} = I_t + I_{ci} + R_2^2 I_{cr} + R_2^2/R_1^2\, I_{si}$.

2.4 Clutch torque model

The clutch torque T_{c2} is given as a static function of the clutch hydraulic pressure, clutch geometry, plate friction characteristics, and the clutch slip speed. For clutch torque calculation, the following equation is used:

$$T_{c2} = A_{c2} \cdot \delta \cdot P_{c2} \cdot \operatorname{sgn}(\Delta\omega_{c2}), \tag{2.9}$$

where

A_{c2} = total clutch area of gear × effective radius,

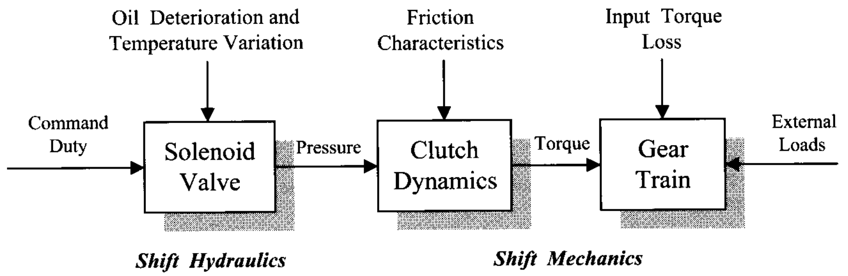
$\delta = \delta_s + \delta_d\ |\Delta\omega_{c2}|,\quad \delta_s, \delta_d > 0$,

P_{c2} = hydraulic pressure applied to the clutch,

$\Delta\omega_{c2} = \omega_t - \frac{\omega_{cr}}{R_2}$.

A hydraulic actuating system using electromagnetic valves controls the pressure of clutches and brakes. Since the hydraulic system has a large system order and high nonlinearities, it is not easy to model the hydraulic system for the analysis and control of shift transients. Thus, the model based on the steady-state characteristics of the shift hydraulic system from a look-up table is widely used for shift control. The power transmission control system with several uncertainties is illustrated in Figure 4.

Figure 4 Uncertainties in power transmission control system



2.5 *Driveline model*

The angular velocity of the final drive output shaft is the input to the axle shafts. The axle shaft is modelled as a lumped parameter torsional spring:

$$\dot{T}_s = K_s(R_d\omega_{cr} - \omega_w), \tag{2.10}$$

where K_s is the torsional stiffness and ω_w is the wheel speed. The rotational dynamics of the driving wheel is given by

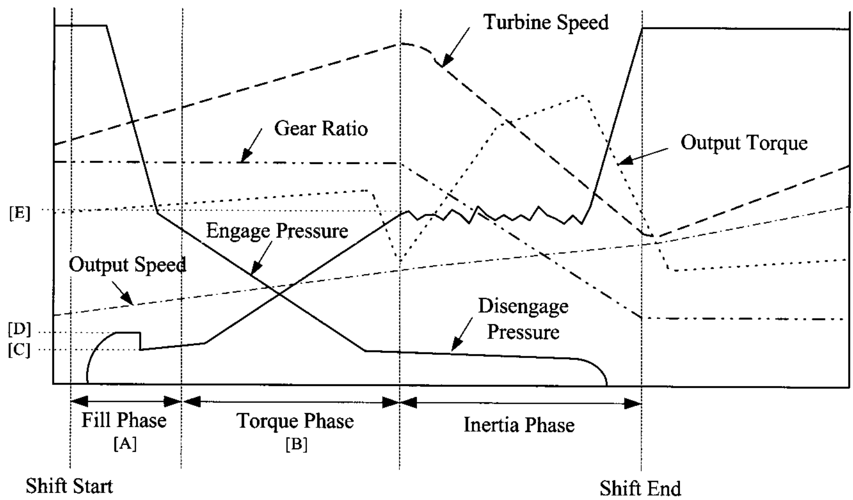
$$I_w\dot{\omega}_w = T_s - T_L, \tag{2.11}$$

where I_w is the vehicle inertia, T_s is the shaft torque, and T_L is the driving load.

3 Control strategies

A typical shift sequence and control parameters during a power-on upshift are illustrated in Figure 5. The power-on upshift begins with a decrease in output shaft torque in the torque phase, followed by a transition to an increase in output shaft torque in the inertia phase. The desired clutch pressure is adjusted by shift control parameters such as the fill time [A], the fill pressure [D], and the entry pressure to each phase and its duration [B], [C], and [E]. The desired clutch pressure is continuously updated by the updated control parameters. It is well recognized that the shift transient torque in output shaft is dominant in the inertia phase. Thus, supervision and control during the inertia phase shift are focused in this paper.

Figure 5 Typical transient characteristics of the variables involved during a shift

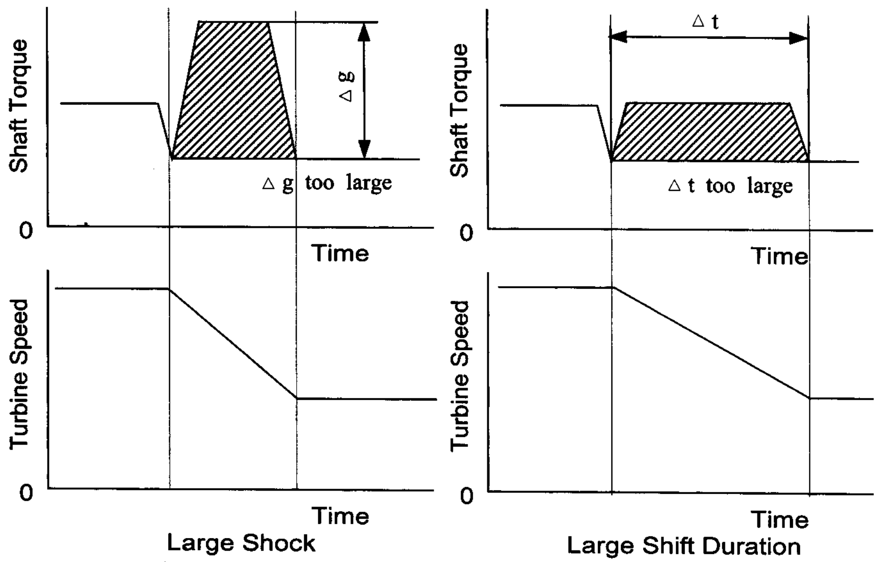


A typical pattern of an up-shift is shown in Figure 6. It is observed from the shift characteristics that there is a correlation between the shift shock and the inertia phase duration. During the inertia phase, if the torque increase is too large, passengers may experience an undesirable shock. On the other hand, if the torque increase is not sufficient, the total shifting time will increase. If the shifting time is too long, the clutch components may deteriorate. Thus, two control objectives are given as follows: One is to minimize the transmission output torque and jerk levels to enhance the ride comfort. Another is to minimize the clutch energy dissipation to enhance the durability of frictional elements.

The shift quality is very closely related to the output shaft torque and the engine torque. It has well been recognized that a nonlinear closed loop control algorithm can be developed, which yields a better shift quality provided that the accurate information on the shaft torque is available [26]. However, torque sensors cannot be used in production vehicles because they are expensive and their durabilities are poor in harsh environment. But, speed sensors for measuring the engine speed, the turbine speed, and the output speed are inexpensive and they are already being used. The operating torque of the torque converter depends on both the impeller speed and the turbine speed.

It is not simple to design an integrated engine–transmission controller for the purpose of enhancing shift quality due to the complexity, nonlinearity, and uncertainty of the system. Also, due to the actuator bandwidth limitation, achievable performance levels inevitably decrease as model uncertainties become larger. Thus, in this work a robust shift control strategy for the integrated engine and transmission system is investigated to overcome the unmodelled dynamics. Apart from the ease of handling nonlinear modelling results, a key attribute of the robust approach is the ability to incorporate modelling uncertainties directly into the control laws.

Figure 6 Examples of two extreme up-shift characteristics



4 Robust shift control

The purposes of shift control are primarily to reduce the shock occurring during the gear shifts and to prevent an excessive slip which may cause unnecessary wear of the frictional parts. Since the reaction carrier speed (or the transmission output speed), ω_{cr}, is fairly easy to measure and is directly related to the smooth acceleration of the vehicle, an error e_1 is first defined as follows:

$$e_1 = \omega_{cr} - \omega_{cr,des}, \tag{4.1}$$

where $\omega_{cr,des}$ represents the desired reaction carrier speed. To keep the reaction carrier jerk, ω_{cr}, at zero during the inertia phase, the carrier speed must have a constant slope during the inertia phase with the starting and end times, t_1 and t_2, respectively. Thus, $\omega_{cr,des}$ is given by

$$\omega_{cr,des}(t) = \omega_{cr,t_1} + \dot{\omega}_{cr,t_1}(t - t_1). \tag{4.2}$$

To improve durability of the frictional components, the clutch energy requirement and shift duration must also be considered. Since the shift duration and the slip speed directly influence the energy dissipation, control of the slip speed of the on-coming clutch must be considered. Therefore, another error e_2 is defined as

$$e_2 = \Delta\omega_{c2} - \Delta\omega_{c2,des}$$

$$= \left(\omega_t - \frac{\omega_{cr}}{R_2}\right) - \left(\omega_{t,des} - \frac{\omega_{cr,des}}{R_2}\right), \tag{4.3}$$

where $\Delta\omega_{c2,des}$ and $\omega_{t,des}$ denote the desired slip speed and the desired turbine speed, respectively. Note that the clutch slip speed is uniquely defined by the trajectory of the converter turbine and reaction carrier speeds. Since the reaction carrier speed is controlled by e_1 during the inertia phase, i.e., assuming $\omega_{cr} = \omega_{cr,des}$, e_2 can be modified to

$$e_2 = \omega_t - \omega_{t,des}\,. \tag{4.4}$$

When a shift is completed, the turbine speed and the reaction carrier speed must be synchronous to satisfy the kinematic constraint, $\omega_{t,t_2} \equiv \omega_{cr,t_2}/R_2$. Thus, $\omega_{t,des}$ is obtained by

$$\omega_{t,des} = \omega_{t,t_1} - \frac{\omega_{t,t_1} - \omega_{t,t_2}}{t_2 - t_1}(\,t - t_1)$$

$$= \omega_{t,t_1} - \frac{1}{\Delta\, t}\left(\frac{\omega_{cr,t_1}}{R_1} - \frac{\omega_{cr,des,t_2}}{R_2}\right)(\,t - t_1)$$

$$= \omega_{t,t_1} - \frac{1}{\Delta\, t}\left(\frac{\omega_{cr,t_1}}{R_1} - \frac{\omega_{cr,t_1} + \dot{\omega}_{cr,t_1}(\,t_2 - t_1)}{R_2}\right)(\,t - t_1)$$

$$= \omega_{t,t_1} + \left\{\frac{\omega_{cr,t_1}}{\Delta\, t}\left(\frac{1}{R_2} - \frac{1}{R_1}\right) + \frac{\dot{\omega}_{cr,t_1}}{R_2}\right\}(\,t - t_1)\,, \tag{4.5}$$

where $\Delta\, t = t_2 - t_1$ denotes the shift duration.

To minimize the shaft torque and jerk levels during the shifts, the reaction carrier speed error e_1 can be considered by assuming that the clutch torque is an input control variable. Also, to minimize the clutch energy dissipation, the turbine speed error e_2 can be considered by assuming that the engine torque is an input control variable.

Note that the engine torque is controlled normally by adjusting the throttle angle (α), and the clutch torque is controlled by adjusting the pressure command (or duty cycle) in practice. That is, in the power transmission system, a hydraulic control circuit applies pressure on the clutch plates, and the friction between the plates provides the control torque, so the actual control variable is the pressure. However, the hydraulic and clutch characteristics are specific to a given transmission and implementation method. If the clutch torque is assumed to be a control variable, all power transmissions share the same dynamic equations presented in Section 2, and the control problem is completely generic to all power transmissions. Therefore, the two control variables for the integrated engine-transmission control system are: the clutch torque (T_{c2}) to minimize the transmission output torque and the throttle angle (α) to minimize the clutch energy dissipation.

4.1 Transmission control: ride comfort

To minimize the jerk levels during the shifts, the reaction carrier speed error e_1 needs to be minimized.

From (2.7) $\dot{\omega}_{cr}$ is derived as

$$\dot{\omega}_{cr} = \frac{1}{I_{cr12}}\left(\frac{T_{c2}}{R_2} - R_d T_s\right). \tag{4.6}$$

Thus, in the ideal case (no uncertainties are involved), the desired command clutch torque, is analytically derived, by considering $V_{ideal} = e_1^2/2$, as follows:

$$T_{c2,\text{com}} = R_2(I_{cr12}\dot{\omega}_{cr,des} + R_d T_s - I_{cr12}e_1), \tag{4.7}$$

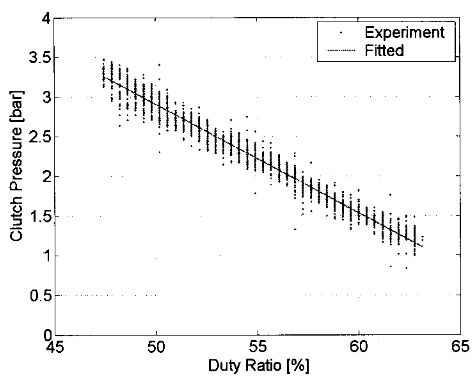
where $T_{c2,\text{com}}$ makes $\dot{V}_{ideal} = -e_1^2$.

However, the measurement of T_s in (4.7) is not possible in real vehicles. The uncertainties in clutch torque result from the variations in oil properties and friction characteristics of the clutch plates. As shown in Figure 7, the variations related to a solenoid valve can also cause a significant difference in the clutch pressure characteristics.

Although an ideal actuator assumption is made, an error in the actual control torque is inevitable (i.e., the difference between the command signal and the actual one). This inevitable error can be incorporated as follows:

$$\Delta T_{c2} \triangleq T_{c2,\text{actual}} - T_{c2,\text{com}}. \tag{4.8}$$

Figure 7 Variations in shift hydraulics



Furthermore, assume that this uncertainty is upper-bounded as follows:

$$\|\Delta T_{c2}\| \leq \Delta T_{c2,\max}. \tag{4.9}$$

With the above assumption, (4.6) is modified to

$$\dot{\omega}_{cr} = \frac{1}{I_{cr12}}\left(\frac{T_{c2,\mathrm{com}} + \Delta T_{c2}}{R_2} - R_d T_s\right)$$

$$= \frac{1}{I_{cr12} R_2} T_{c2,\mathrm{com}} + f\,, \tag{4.10}$$

where $f \triangleq 1/I_{cr12} R_2\, \Delta T_{c2} - R_d T_s / I_{cr12}$. Assume that $\left\| 1/I_{cr12} R_2\, \Delta T_{c2} - R_d T_s / I_{cr12} \right\|$ is uniformly bounded by ρ, i.e.,

$$\rho \geq \left\| \frac{1}{I_{cr12} R_2} \Delta T_{c2,\max} - \frac{R_d T_s}{I_{cr12}} \right\| \geq \|f\|, \tag{4.11}$$

where ρ is an unknown positive constant to be estimated.

Thus, the clutch torque control law for $T_{c2,\mathrm{com}}$ and the adaptation law to estimate the uncertainty bound ρ are developed so that the boundedness of all signals in the closed loop system and the convergence of the reaction carrier speed error e_1 near to zero can be assured.

The main idea is to consider the worst case of the uncertainties in the form of possible bounds. Based on the worst case, the following control algorithm is proposed:

$$T_{c2,\mathrm{com}} = \left(\frac{1}{I_{cr12} R_2}\right)^{-1} \left(\lambda e_1 + \dot{\omega}_{cr,des} + p_1 \right), \tag{4.12}$$

where $\lambda<0$, and p_1 is regarded as a new input signal to be determined based on the robust control strategy. Let the additional input p_1 be given by

$$p_1 = -\frac{e_1 \hat{\rho}^2}{\|e_1 \hat{\rho}\| + \varepsilon}, \tag{4.13}$$

where $\varepsilon > 0$, and $\hat{\rho}$ is the estimate of ρ.

The adaptation law of $\hat{\rho}$ is given by

$$\dot{\hat{\rho}} = \gamma \|e_1\| - \delta \hat{\rho}\,, \tag{4.14}$$

where $\gamma, \delta > 0$. Note that the control law (4.12) and the adaptation law (4.14) are implementable by sensing the reaction carrier speed. The term $-\delta\hat{\rho}$ in (4.14) is purposely inserted to enhance the convergence of $\hat{\rho}$.

Now, consider a Lyapunov function candidate V for the transmission part during the inertia phase as follows:

$$V = \frac{1}{2} e_1^2 + \frac{1}{2\gamma} \tilde{\rho}^2, \tag{4.15}$$

where $\tilde{\rho} = \hat{\rho} - \rho$. The differentiation of (4.15) with respect to t along (4.10) using (4.12)–(4.14) yields:

$$\begin{aligned}
\dot{V} &= e_1 \dot{e}_1 + \frac{1}{\gamma} \tilde{\rho} \dot{\hat{\rho}} \\
&= e_1 (\dot{\omega}_{cr} - \dot{\omega}_{cr,des}) + \frac{1}{\gamma} \tilde{\rho} \dot{\hat{\rho}} \\
&= e_1 \left\{ \frac{1}{I_{cr12} R_2} T_{c2} + f - \dot{\omega}_{cr,des} \right\} + \frac{1}{\gamma} \tilde{\rho} \dot{\hat{\rho}} \\
&= e_1 \left\{ \left(\lambda e_1 + \dot{\omega}_{cr,des} + p_1 \right) + f - \dot{\omega}_{cr,des} \right\} + \frac{1}{\gamma} \tilde{\rho} \dot{\hat{\rho}} \\
&\le \lambda e_1^2 - \frac{e_1^2 \hat{\rho}^2}{\|e_1 \hat{\rho}\| + \varepsilon} + \|f\| \, \|e_1\| + \frac{1}{\gamma} \tilde{\rho} \dot{\hat{\rho}} \\
&\le \lambda e_1^2 - \frac{e_1^2 \hat{\rho}^2}{\|e_1 \hat{\rho}\| + \varepsilon} + \rho \|e_1\| + \hat{\rho} \|e_1\| - \hat{\rho} \|e_1\| + \frac{1}{\gamma} \tilde{\rho} \dot{\hat{\rho}} \\
&\le \lambda e_1^2 + \varepsilon - \tilde{\rho} \|e_1\| + \tilde{\rho} \left(\|e_1\| - \frac{\delta}{\gamma} \hat{\rho} \right) \\
&\le \lambda e_1^2 - \frac{\delta}{2\gamma} \tilde{\rho}^2 + \left(\varepsilon + \frac{\delta}{2\gamma} \rho^2 \right).
\end{aligned} \tag{4.16}$$

From (4.16), all signals in the closed loop are uniformly ultimately bounded since $(\varepsilon + \delta\rho^2/2\gamma)$ is bounded. Furthermore, the uniform ultimate boundedness region and the uniform stability region can be made arbitrarily small by a suitable choice of ε, δ, and γ. Namely, if ε and δ are sufficiently small and γ is sufficiently large, then it is guaranteed that e_1 is uniformly ultimately bounded within an arbitrarily small neighbourhood of zero.

Thus, by using the clutch torque control law (4.12) and the adaptation law (4.14) developed, the trajectory following of the reaction carrier speed ω_{cr} to the desired speed $\omega_{cr,des}$ can be achieved. This implies that the first control objective, i.e., the minimization of the transmission output torque and jerk levels to enhance the ride comfort, is achieved.

4.2 Engine control: component durability and shift completion

To minimize the clutch energy dissipation, the turbine speed error e_2 needs to be zero by controlling the engine speed besides controlling T_{c2}.

From (2.6), ω_t is given by

$$\dot{\omega}_t = \frac{1}{I_t}\left(T_t - T_{c2} \right). \tag{4.17}$$

In the ideal case, the desired torque converter turbine torque, which can guarantee the convergence of e_2 to zero, can be computed as follows:

$$T_t = I_t\dot{\omega}_{t,des} + T_{c2} - I_t e_2 . \tag{4.18}$$

From the torque converter model (2.3) with (2.1), T_t is rewritten as follows:

$$T_t = T_r(\omega_e,\omega_t)T_p = T_r(\omega_e,\omega_t)\left(T_e(\omega_e,\alpha) - I_e\dot{\omega}_e \right). \tag{4.19}$$

Thus, from (4.18) and (4.19), if all variables are available with no modelling error, the desired throttle angle can then be derived as follows:

$$T_e(\omega_e,\alpha_{\mathrm{com}}) = \frac{1}{T_r(\omega_e,\omega_t)}\left(I_t\dot{\omega}_{t,des} + T_{c2} - I_t e_2\right) + I_e\dot{\omega}_e , \tag{4.20}$$

where α_{com} is the command throttle angle for the desired engine indicated torque.

However, there exist various uncertainties involved in equation (4.20), i.e., calculating the engine input torque. The input torque to the automatic transmission, which is generated by the engine, is delivered and amplified by the torque converter. Thus, the magnitude of this torque is mainly determined by a throttle operation and the torque converter characteristics. In particular, the turbine torque is transferred to the automatic transmission as a result of the oil-induced flow in the torque converter, and the oil properties depend on oil temperature.

Figure 8 shows the effect of oil temperature on the turbine torque. The torque-controlled AC motor in the experimental test setup, as shown in Figure 12, is used as an engine. Experimental tests were done for various AC motor torque commands. Test results for 50%, 60%, 70%, 80%, 90%, and 100% AC motor torque commands are presented in Figure 8. From Figure 8 it can be seen that the torque converter characteristics are considerably affected by the temperature of internally induced oil. Results from experiments indicate that the torque loss caused by the drag torque in the torque converter due to variations in oil temperature, 30°C and 80°C, is quite significant. Five to 25% variations of the turbine torque are observed for the oil temperature variations of 30°C and 80°C. Hence, the effect of solenoid valve characteristics and the drag torque can be handled by considering the oil temperature. Thus, instead of control law (4.20) described above, an adaptive learning control method is now pursued in this paper.

The nonlinear relationship between the input variables and the desired outputs (or control parameters) is modelled using an adaptive neuro-fuzzy supervisor. The control input pattern which generates throttle angle commands is updated through a learning process and adaptive compensation law to adjust for each subsequent shift based on continuous monitoring of shifting performance and environmental changes. The overall control scheme with the supervisor is shown in Figure 9.

Figure 8 Variations in turbine torque affected by torque converter oil temperature according to various driving torque from a torque-controlled AC motor: (a) at oil temperature 30°C, (b) at oil temperature 80°C

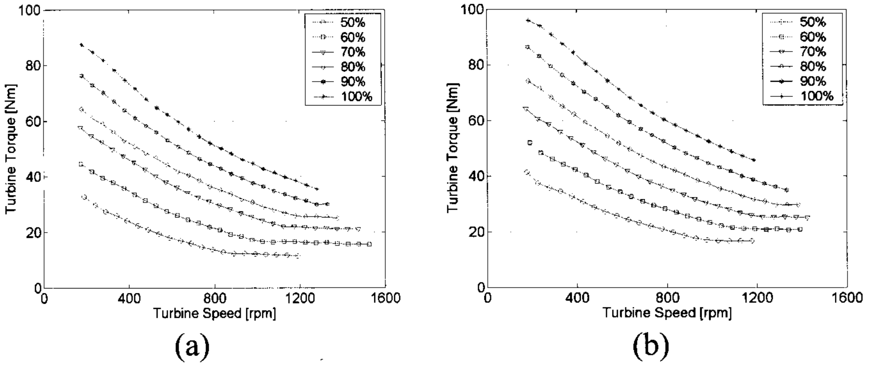
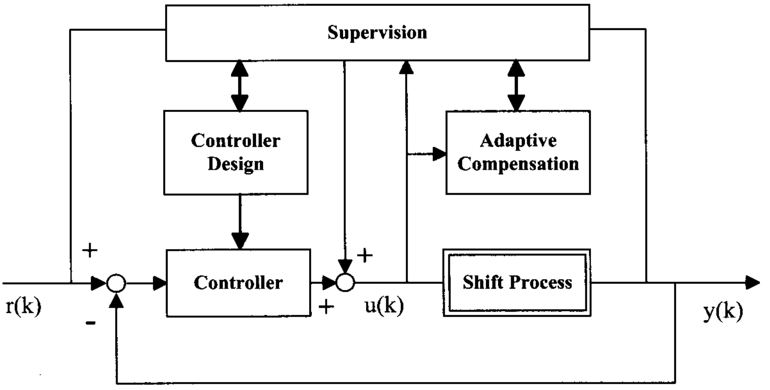


Figure 9 Supervision and control of the shift process



A neuro-fuzzy model that supervises the shift process based on the information of duty cycle and oil temperature is developed. The neuro-fuzzy system for modelling the nonlinear relationship between the input variables and the corresponding engine control is a fuzzy inference system built on the framework of a neural network, and the models for the nonlinear systems can be effectively established with this fuzzy system. It consists of five layers as shown in Figure 10. The first layer performs the fuzzification operation for the input variables, and the firing strengths are calculated in the second layer. The firing strengths are normalized in the third layer, and then the fourth layer performs the fuzzy inference operation. Finally, the defuzzification operation is carried out and the overall output of the fuzzy inference is provided in the fifth layer. The adaptive neuro-fuzzy inference system for supervising a shift control was constructed and is then trained using experimental data. The architecture of the intelligent supervisor is shown in Figure 10. The ANFIS used in this paper contains nine rules, with three membership functions being assigned to each input variables. It has two inputs, i.e., duty cycle and oil temperature, and Sugeno-type fuzzy inference scheme is used to generate the output, i.e., control parametric surface.

Figure 10 An adaptive neuro-fuzzy inference system architecture for supervising the engine control during the shifts

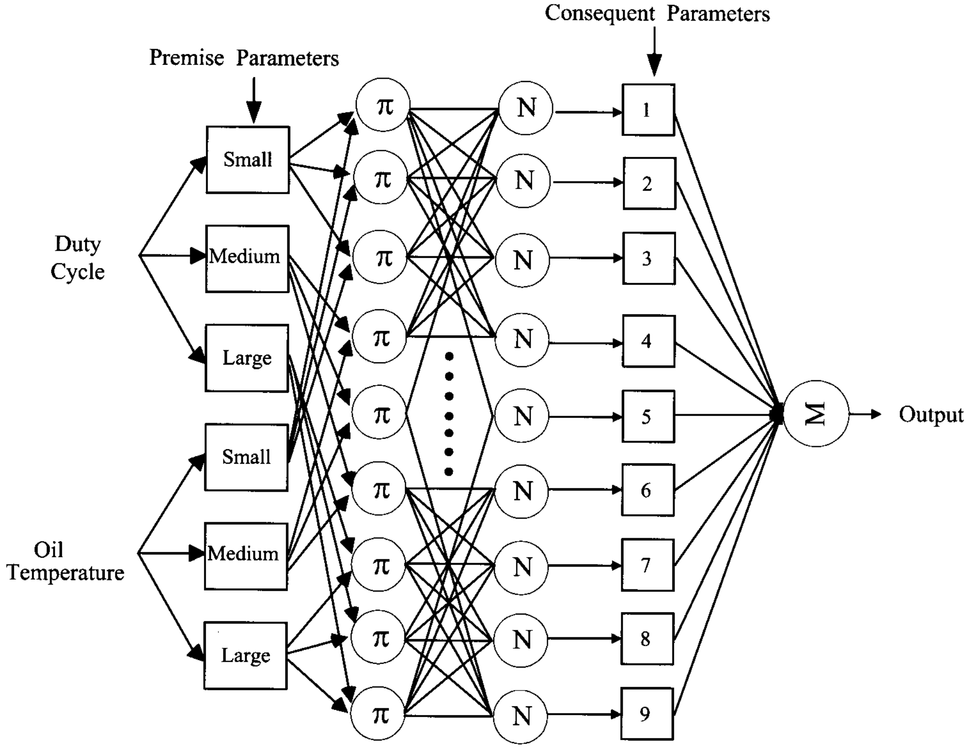
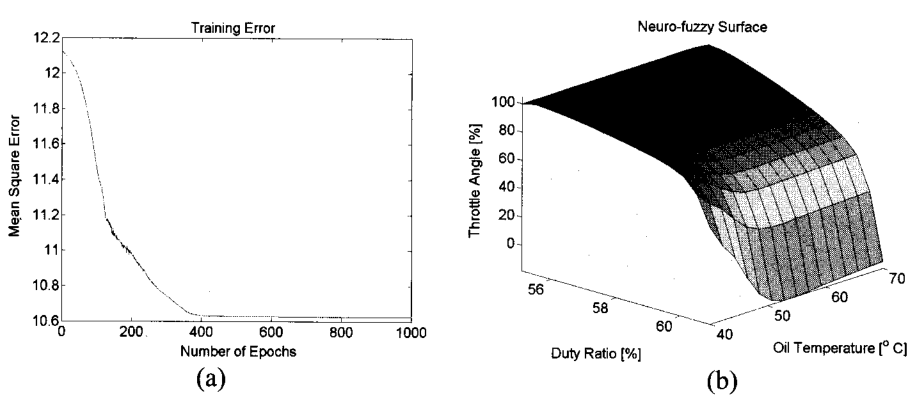


Figure 11 shows training results of the proposed intelligent supervisor. Figure 11(a) shows the learning curve in mean squares error, which indicates that most of the learning was done within the first 150 epochs. Figure 11(b) shows a control parametric surface depending on the duty cycle and oil temperature and demonstrates how the proposed ANFIS architecture can effectively model a highly nonlinear surface of the control input during the shifts.

Figure 11 Training results of the intelligent supervisor: (a) learning curve, (b) control surface



Variations in shift quality due to uncertain changes in the engine–transmission system can be recognized by monitoring the turbine speed error e_2 and the deviation of the shift duration from the desired range of shift duration. In order to keep the turbine speed and the shift duration within an allowable range in the presence of system variations, the desired engine torque is adjusted by updating the shift control parameters. In particular, the feed-forward throttle angle, which generates the engine torque during the inertia phase, is updated by applying the adaptive compensation law. The shift performance is continuously monitored in relation to a given desirable shift condition.

A performance index, as a quadratic function of the turbine speed error e_2 and the deviation of shift duration Δt from the desired range of shift duration Δt_{des}, is defined as follows:

$$J = \frac{1}{2}\mathrm{e}^T \mathrm{e}, \tag{4.21}$$

where $\mathrm{e} = (e_2, \Delta \tilde{t})^{\mathrm{T}}$ and $\Delta \tilde{t} = \Delta t - \Delta t_{des}$. Then, the performance index is minimized using the gradient descent method. The adaptive updated law is given by

$$\frac{du}{dt} = -\eta\left(\frac{dJ}{dt}\right). \tag{4.22}$$

For digital implementation, the adaptive updating law is modified as follows:

$$u(k+1) = u(k) - \eta\left(\left.\frac{dJ}{du}\right|_k\right), \tag{4.23}$$

where $u(k)$ is the command throttle angle for kth shift, and $\eta > 0$ is the adaptation gain.

From Jang [24], it is guaranteed that the error **e**, i.e., the turbine speed error e_2 and the deviation of shift duration Δt from the desired range of shift duration Δt_{des}, tends to a small value asymptotically. Thus, using the adaptive compensation control method based on the intelligent supervisor, the trajectory following of the turbine speed ω_t to the desired turbine speed $\omega_{t,des}$ as well as the shift duration Δt to the desired shift duration Δt_{des} can be achieved. This implies that the second control objective, i.e., the minimization of the clutch energy dissipation to enhance the durability of frictional elements, is achieved.

5 Experimental studies

5.1 Experimental setup

Experimental studies have been conducted to examine the proposed control method. Figure 12 shows a schematic diagram of the experimental setup. A photograph of the experimental test setup is shown in Figure 13. A torque-controlled AC motor is used as an engine. An inertia load is used for the external driving load. In addition, the direct clutch pressure control system using the proportional solenoid valve [27] has been installed to improve the controllability of the vehicle automatic transmission. By pulse-width-modulation of the voltage command to the solenoid valve, the real-time pressure control for each individual clutch can be achieved.

5.2 Experimental results

Bench pressure tests on the solenoid valve were performed. The pressure characteristics were measured for various pressure commands (i.e., duty cycle). The results are depicted in Figure 14. The pressure varies fairly linearly with the command duty cycle from 20% to 80%. The test condition is also shown in Table 1. The sampling time for the control loop is 10 ms and the oil temperature is around the nominal condition (about 70°C).

Figure 12 Schematic diagram of the experimental setup

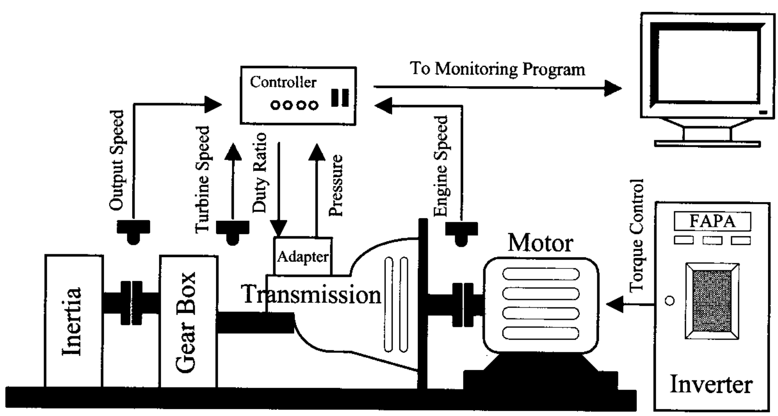


Figure 13 Picture of the experimental test setup

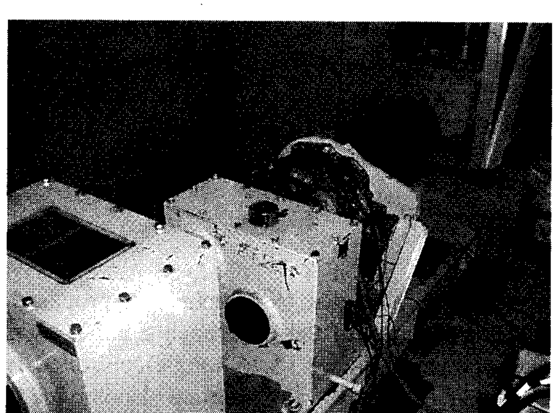


Figure 14 Hydraulic actuator characteristics

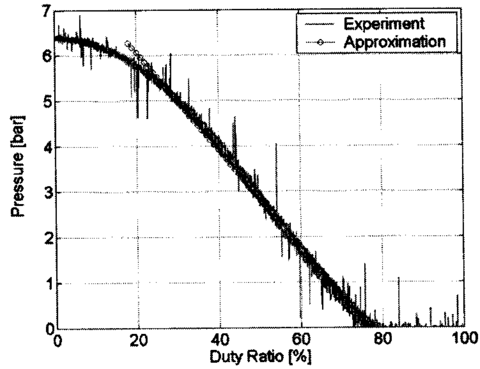
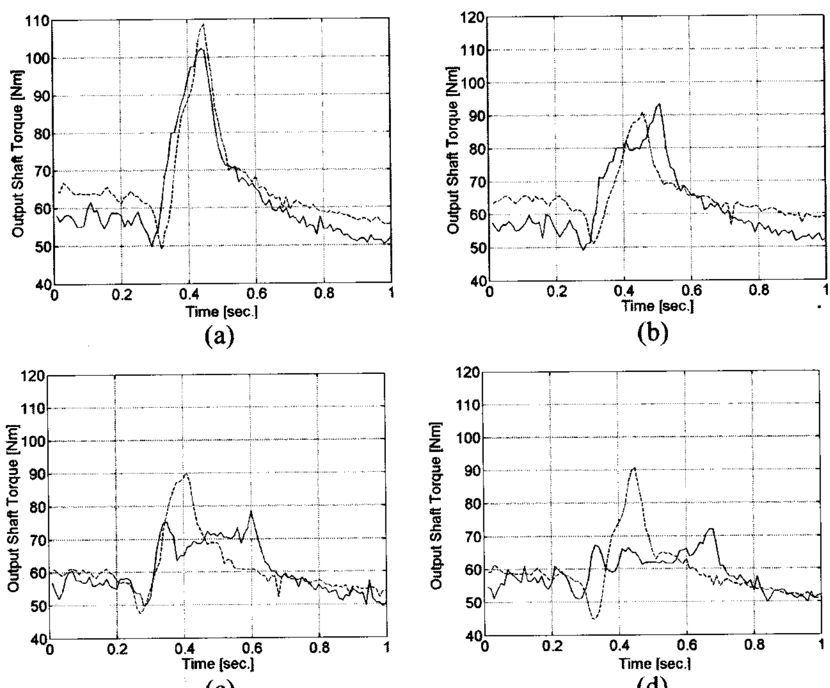


Figure 15 Comparison of shift transients: with the robust adaptive compensation (solid line) and without the robust adaptive compensation (dashed line): (a) 1st shift, (b) 3rd shift, (c) 5th shift and (d) 7th shift



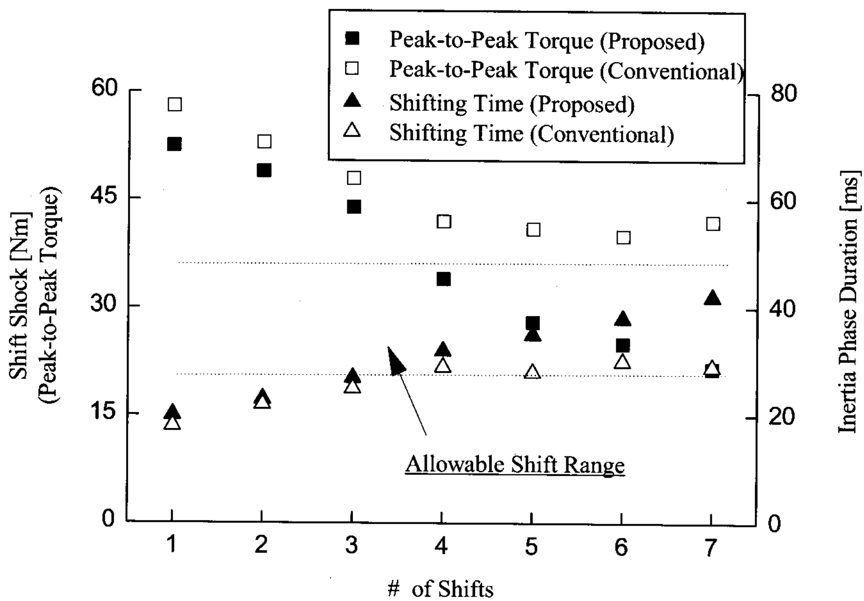
Experiments were performed for uncertain changes in the power transmission system, which result in degraded shift performances. Figure 15 compares the subsequent shift results with the robust adaptive compensation and without the robust adaptive compensation. As can be seen, actual output torque during the inertia phase is subsequently reduced when the proposed shift controller is applied, while the conventional controller cannot improve the undesirable shift performance.

Table 1 Experimental test conditions

Item	*Condition*
ATF temperature	60 ± 5°C
Line pressure	6.5 ± 0.5 bar
Sampling frequency	100 Hz
Filtering	3-pole LPF, 10 Hz cutoff

In terms of the shift shock and inertia phase duration, the shift performances are evaluated in Figure 16. It shows that a stable shift can be obtained by keeping the duration of the inertia phase within a certain desired range. It can be seen from the experimental results that the peak-to-peak values of the shift shocks in the inertia phase are subsequently reduced when the proposed controller is applied in the shift control.

Figure 16 Shift results: shift shock (N m) vs inertia phase duration (ms)



6 Conclusions

In this paper, to improve the shift transients and durability of the frictional parts of an automatic transmission system, a robust adaptive control of the clutch torque with an adaptive neuro-fuzzy inference system for supervising the engine torque control was investigated. The experimental results of the integrated control algorithm proposed reveal that the strategy of controlling the engine during the shifts provides better performance in shift transients than the conventional transmission control alone. Even though a control during the inertia phase has been focused in this paper, an integrated control of the output shaft torque as well as the engine indicated torque during both the torque and inertia phases would be more fruitful.

Acknowledgement

This work was supported by the National Research Laboratory (NRL) Program, grant no. M1-0203-00-0017-02J0000-00910, of the Ministry of Science and Technology, Korea.

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