

Gearshift Actuator Dynamics Predictions in a Dual Clutch Transmission

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ABSTRACT

Although hybrid variants of Automated Mechanical Transmissions such as the Dual Clutch Transmissions are less affected by driveline torque interrupts, actuator dynamics is very critical in the speed of gear pre-selection and during multiple gear shifts. To avoid torque interrupts, such systems require precise gearshift duration hence the actuators are expected to have fast, stable and predictable responses. However, actuator dynamics and controls remain barriers to attaining the full benefits of such complex systems, demanding precise timing and coordination of the gearshifts alongside the clutches engagement and disengagement. To overcome such challenges, a dynamic model of an electro-hydraulic gearshift actuator, the synchronizer and the shift fork has been developed. The model predicts the gearshift actuator dynamics for a given set of input parameters, which can be correlated against experimental data. The significance of the work is that an efficient and robust means of improving the gearshift quality of a DCT has been devised.

KEYWORDS: Actuator Dynamics, Dual Clutch Transmission, Gearshift Control, Synchronization

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I. INTRODUCTION

The emergence of tougher environmental and fuel efficient legislations have triggered exploration and advances towards new and better vehicle transmission technologies. Some of the emerging technologies have seen a growing trend towards increased drive comfort, fuel efficiency, and low CO₂ emission. The most recent additions have been variants or hybrids of the Automated Mechanical Transmission (AMT) including fully Automated Mechanical Transmissions (fAMT), Continuously Variable Transmissions (CVT), Hybrid Electric Vehicles (HEV), and Dual Clutch Transmissions (DCT) [1].

The AMT variants such as the DCT combines the convenience of an automatic transmission (AT) with high efficiency of a manual transmission (MT) by using twin clutches, one for the odd-numbered gears and the other for even-numbered gears. This allows uninterrupted and smooth upshifts or downshifts through the speed ratios without loss of traction hence no perceptible jolts during gearshifts. For instance, while the vehicle is meshed with the output shaft by the synchromesh mechanism to engage the odd-numbered gear clutch, as soon as the conditions for an upshift or

downshift are met, the even-numbered clutch is engaged simultaneously with the disengagement of the odd-numbered gear clutch in a process similar to the shift overlap function of an automatic transmission. However, unlike in automatic transmissions, the twin clutches of a DCT are subjected to higher loads due to the overlapping gearshifts, slip control and more frequent gearshift operations [2]. Precise actuators control during an upshift or downshift allows a seamless torque transfer between the two sub-transmissions. A DCT can work in two different ways: semi-automatic, when the gearshift process is initiated by the driver by means of a shift paddle/shift lever or fully automatic using sophisticated electronics attending to the strategy of the gear ratio selection, according to engine regime and the prevailing driving conditions.

Recent studies have focused primarily in the control and modeling of AMTs. These research works were intended to improve numerical modeling for accurate predictions of dynamic responses of such systems during the transmission prototype development and the control system optimization [3]. However, while conventional transmission technologies are well developed for both passenger and commercial vehicles, the hybrid technologies such as DCTs are still at the

emerging development stages, especially in high fidelity refinement of control algorithms for all driving conditions. There have been some researches on the analysis and modeling of a DCT dynamic characteristics. However, the methods suggested in several researches are open loop based control with experimental calibration [4]. Since a DCT does not have a torque converter, achieving sophisticated torque control under varying driving conditions poses a big challenge. Among the control approaches for the DCT gearshift seen in most literature, some strategies cannot be optimized without the precise control of the engine and twin clutches [5], whereas others allow tracking of the output torque trajectories. Although the former approach cannot guarantee the high shift quality, the latter present challenges in measurement of the output torque directly at output axle [6].

The present research is intended to develop a dynamic gearshift actuator and control algorithm for the gearshift quality optimization through accurate prediction of the actuator dynamic responses during the gear change transition. The presented gearshift strategy regulates the dual-clutch engaging forces and is implemented with feedback loops based on measurements of the clutches speeds and pressure, engine speed, gear position feedback and on dual clutches engaging torques estimation. As compared to the prior art, the study of gearshift transients and the shift quality optimization enables enhanced synchronization through the reduction of the peak post-engagement torsional vibrations.

2. GEARSHIFT ACTUATION

In order to achieve a smooth launch and a fast gearshift demand in a twin clutch transmission, shift actuation can be controlled electronically by hydraulics using valve assembly, with a complex maze of passages that distribute pressurized transmission fluid to the actuators inside the transmission [7]. Electronic transmission management is controlled by the transmission control module (TCM), which calculates optimum gear ratios and control the actuators to engage gears at the optimal time, increasing driving comfort and decreasing fuel consumption. The TCM initiates a gear change in response to the inputs from the driver, engine and the gearbox.

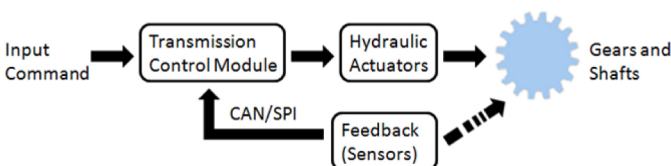


Fig. 1. Block diagram of an electro-hydraulic gearshift actuator system.

Interconnectivity is an important capability of modern transmissions. Vehicle and engine status data are transmitted through the control area network (CAN) or serial peripheral interface (SPI), both of which are international standard intra-

vehicle communication networks to the TCM. A block diagram of an electro-hydraulic actuator system is shown in Fig. 1, comprising of the TCM, hydraulic actuators, gears/shafts and sensors.

2.1. Hydraulic Actuator System

A schematic view of a gearshift actuator system is shown in Fig. 2. It consists of the following: 1) Hydraulic system, 2) Actuator piston, 3) Gear position sensor [y1], 4) Actuator fork, 5) Synchronizer sleeve and hub, 6) Lower gear, 7) Higher gear, 8) Input shaft, 9) Output shaft, 10) Input shaft angular velocity sensor [y2], 11) Output shaft angular velocity sensor [y3], 12) Clutch, and 13) Connection to wheels. The gear position, input shaft angular velocity and the output shaft angular velocity are the available measurements from the sensors y1, y2 and y3 respectively. The actuator comprises of two hydraulic lines, each connected to the piston, separated by the actuator piston. The end of the piston houses the gear position sensor while the end of the fork is attached to the synchronizer sleeve. The differential force on both sides of the hydraulic lines forces the piston and the fork to accelerate when the lines are pressurized [8].

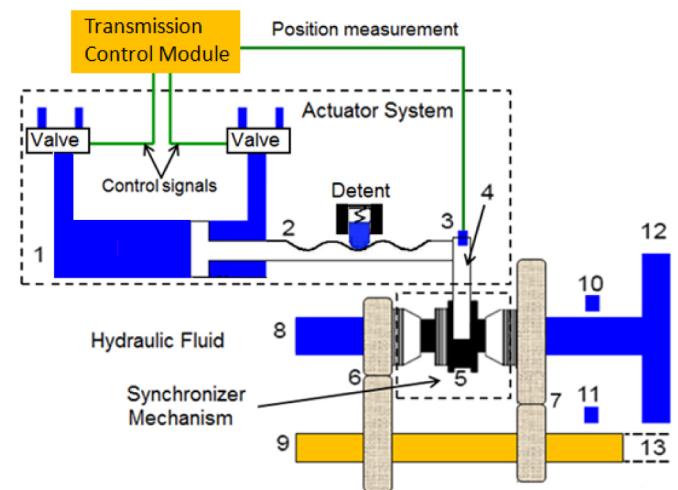


Fig. 2. An overview of the synchronization system showing the actuator, the input and output shafts, and the synchronizer mechanism.

The fluid flow distribution in a proportional valve is governed by non-linear equations which can be expressed in the simplest forms as follows [9]:

$$Q_i = \begin{cases} k_d w x_{sp} \sqrt{P_s - p_i} & x_{sp} \geq 0 \text{ (extension)} \\ k_d w x_{sp} \sqrt{P_i - p_r} & x_{sp} < 0 \text{ (retraction)} \end{cases} \quad (1)$$

$$Q_o = \begin{cases} k_d w x_{sp} \sqrt{P_o - p_r} & x_{sp} \geq 0 \text{ (extension)} \\ k_d w x_{sp} \sqrt{P_s - p_o} & x_{sp} < 0 \text{ (retraction)} \end{cases}$$

where Q_i and Q_o represent fluid flows into and out of the valve respectively. k_d is the metering coefficient, w is the orifice area gradient that relates the spool displacement, x_{sp} , to the orifice area. P_s is the pump pressure while P_i and P_o are the line input and output pressure respectively. P_r is the return pressure. A hydraulic proportional circuit highlighting P_s , P_i , P_o and P_r is shown in Fig. 3.

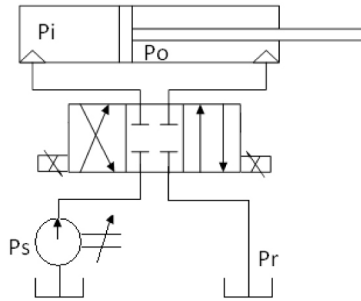


Fig. 3. Scheme of a Hydraulic proportional circuit.

Continuity equations for the fluid flow through the cylinder, neglecting any leakage flow across the actuator's piston can be expressed as follows:

$$Q_i = \frac{V_i(X)}{\beta} \dot{P}_i + A_i v \quad (3)$$

$$Q_o = \frac{V_o(X)}{\beta} \dot{P}_o + A_o v \quad (4)$$

where A_i and A_o are the effective piston areas, β is the effective bulk modulus of the hydraulic fluid, v is the actuator's linear velocity while $V_i(X)$ and $V_o(X)$ are the volumes of the fluid trapped on the sides of the actuator, which are functions of the actuator displacement, X . The relationship between the spool displacement, x_{sp} , and the input voltage, u , for a proportional valve can be expressed as a first order system as:

$$u = \frac{\tau}{k_{sp}} \dot{x}_{sp} + \frac{1}{k_{sp}} x_{sp} \quad (5)$$

where τ and k_{sp} are the gains describing the valve dynamics that defines the turbulent hydraulic fluid flow through an orifice. The electromagnetic valve dynamics is covered in great detail in [10]. The valve subsystem includes a mass, spring and damper that results from the combined effect of pressure and magnetic forces. This can be summed up using Newton's second law as follows:

$$m_s \frac{d^2}{dt^2} x(t) + b \frac{d}{dt} x(t) + k(x(t) + \delta) = F_m(t) + F_{prs} \quad (6)$$

where m_s is the spool mass, b and k are the damping and spring coefficients respectively, δ is the spring pre-tension and F_{prs} is the pressure force. Equation 6 is achieved using the magnetic force and feedback force created by the pressure inside the left and right chambers by the plunger movement. The equation describing the piston dynamics can be described as follows [11]:

$$F_{prs} = (A_1 - A_2)P_{sup} + (A_3 - A_4)P_{out} \quad (7)$$

where P_{sup} and P_{out} are the pressures at the supply and output valve ports, whereas A_1 , A_2 , A_3 and A_4 denotes the various spool lands areas which are affected by P_{sup} or P_{out} in different directions.

2.2. Electronic Gearshift Control

Electronic transmission management systems typically operate on event-driven basis by selecting the gear that provides the optimal torque to meet the prevailing driving conditions with the engine operating in the most favorable map range, such as the range that produces the lowest fuel consumption hence low CO₂ emissions. One of the major benefits of a twin clutch transmission is the shift quality improvement by balancing friction work done by the dual-clutches. An overall gearshift quality improvement strategy needs to meet the following objectives: minimization of the mean vehicle deceleration due to traction loss, reduction of the shifting time, and minimization of driveline oscillations resulting from the transmitted torque variations as well as clutch and gearbox operations. The control parameter adjustment for each shift schedule can be determined in an optimization calculation by means of an empirical model of the shift behavior [12].

2.2.1. Gearshift control logic

The gearshift control logic determines the optimal time to perform a gearshift during the drive cycle in attaining the best fuel economy. Studies have been conducted to fine tune the gearshift strategies. Most of these studies have focused predominantly on the vehicle speed, with the engine typically modeled as a mass inertia which accepts throttle angle as the input to produce a mean torque at the crankshaft [13]. In this work, gearshift logic has been constructed in Simulink incorporating a state-flow that drives the actuator based upon the vehicle speed and throttle position. For the actuator control scheme, a decoupled simulation model has been established in which the actuator parameters that have been calculated from both physical measurements and finite element analysis (FEA) such as current, pressure, force and displacement characteristics are incorporated in the form of look-up tables.

Gearshift quality dynamic models include the synchronizer, selector, and the driveline. The model utilizes the actuator displacement transfer function and solves for the actuator velocity and acceleration both axially and rotationally. The drive-train consists of component models of