

Electromagnetic Actuator Dynamic Response Prediction for an Automated Mechanical Transmission

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ABSTRACT

Among the many advantages of the hybrid variants of Automated Mechanical Transmissions (AMTs) such as the Dual Clutch Transmission are faster gearshifts and excellent acceleration that comes from reduced drive-train losses without torque interrupts which translates into improved drive quality through smoother shifts. However, actuator system dynamics and controls remain critical challenges to attaining the full benefits of such AMT variants, which demands precise timing and coordination of the actuators. This paper presents a method for modeling a solenoid, including its non-linear electromagnetic characteristics. The model has been validated against experimental measurements. The significance of the work is that an efficient and robust method that allows precise predictions of a hydraulic valve pressure, flow through the system and the position of the hydraulic elements has been devised.

INDEX TERMS: Actuator system, AMT, Gearshift process, Proportional Solenoid Valve, Synchronization

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I. INTRODUCTION

The gearshift process of a manual transmission is accomplished by moving the gearshift lever that effectively locks the input shaft to the output shaft via the required gear ratio. However, in an Automated Mechanical Transmission (AMT) hybrid variant such as a Dual Clutch Transmission (DCT), an electronic actuation controlled by a shift-by-wire system is employed. Presently, there are two variants of DCTs: a dry DCT in which the dual clutches can be actuated by an electromechanical system, which offers a simpler construction, fast dynamic response, and easier integration. However, such system may suffer from low power density due to limited electrical power availability within the vehicle system. A wet DCT on the other hand uses electro-hydraulic control system for actuation, with high torque density and smoother operation. However, the efficiency of a hydraulic system is impacted by the need for a pump and the energy expended (heat) in pressurizing the fluid while shifting is not taking place. Furthermore, the electro-hydraulic system can be quite complex due to the number of solenoids and spool valves that are required to deliver the high-pressure hydraulic fluid to the actuation points. Whichever method is employed, an effective control strategy is required to accurately regulate a seamless torque transfer with very precise gearshift durations, which are often dependent on the shaft/gear speed

differential, the power and capability of the synchronizer and the level of force that is applied to it [1].

The ever increasing demand for increased ride comfort, fuel efficiency, and low CO₂ emission has seen a soaring increase in sensing and actuation technologies for improved controllability and flexibility of intelligent transmission technologies. Such a mechatronic gearbox features embedded electronics packaged within the gearbox, with an integrated sensing and actuation for performance efficiency and cost reduction. The gearshift process usually entails four main subsystems namely; the actuator, the input and output shafts, the synchronizer mechanism, and the control software. The synchronizer subsystem includes a sleeve, a hub and friction rings placed between the sleeve and the gear wheels. During a gearshift event, the goal is to move the sleeve so that it locks with the gear wheel of the requested gear. For a hydraulic actuation system, this is performed by forces acting on different sides of the piston end due to the pressure differential within the hydraulic system. As soon as the synchronizer sleeve touches the friction rings, a torque is generated from the applied actuator force which accelerates or decelerates the input shaft angular velocity towards the angular velocity of the gear wheel; followed by the sleeve movement towards the gear wheel to complete the gearshift.

Precise control of the clutch fill time for an AMT variant such as a wet twin clutch gearbox is a major hurdle, which can hamper the coordination of the dual clutches in achieving a seamless torque transfer during a shift event. The oncoming clutch fill time can vary due to many factors such as solenoid valve characteristics, fluid temperature, line pressure variations, and time elapse between shifts. Both the command time and command fill pressure are absolute critical, with miniscule calculations errors of these two parameters likely to lead to clutch overfill or under fill. Clutch overfill can result in a shift tie-up, with potential pull down and a drop in the output torque. Conversely, an under fill can lead to a sluggish torque transfer between the engine and the driveline. Typical twin clutches coordination often implement open-loop, event driven control schemes with some feedback control, primarily from the transmission input and output speed sensors, gear position sensors, temperature and pressure sensors. As compared to a conventional AMT, a twin clutch gearbox requires extensive system calibration work to coordinate between the engine, Transmission Control Modules (TCM), and other driveline components. For instance, with increased gear ratios, such as in a 7 speed twin clutch gearbox, the calibration variable counts to achieve smooth shifts under all environmental conditions increases rapidly.

An automated and systematic, model-based calibration process is often employed to reduce the development time and for improved efficiency. However, the harsh and dynamic operating environments in a transmission, changes in fluid viscosity and hydraulic valve sensitivity offsets, hysteresis, drifts and thermal sensitivity pose some challenges to system robustness, demanding high fidelity hydraulic fluid flow control with high bandwidth demand. Adaptive learning control have been developed to accommodate the uncertain behavior of the hydraulic valves in different driving conditions [2], while detailed procedures for the development of position-control systems for proportional solenoid valve during a shift were presented in [3]. Recently, there have been research works done on position estimation based on observer principles [4]. Even though these past studies have had varying degrees of success, most of them employed quite expensive proportional directional control valves and the control schemes were highly non-linear and complex to model. This paper describes a method for modeling a solenoid, including its nonlinear magnetic characteristics. The significance of the work is that an efficient and robust method that allows precise predictions of the valve pressure, flow through the system and the position of the hydraulic elements has been devised.

2. SHIFT RAIL ACTUATION

2.1. Actuator System

The schematic view of the actuator system is shown in Fig. 1. The actuator consists of two hydraulic lines, each connected to the piston, separated by the actuator piston. The force differential in both sides of the hydraulic lines propels

the piston and the fork to accelerate when the lines are pressurized [5]. The end of the piston houses the gear position sensor while the end of the fork is attached to the synchronizer sleeve.

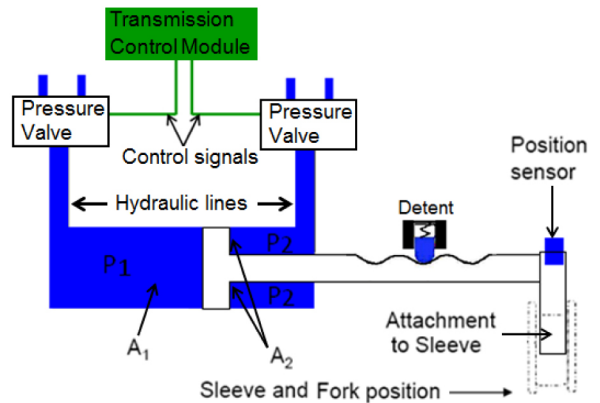


Fig. 1. A schematic view of the actuator system. A_1 and A_2 define the total area in contact with the hydraulic fluid on the opposite side, with A_2 smaller than A_1 .

An overview of the synchronization 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 (connection to the engine), and 13) Connection to the 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.

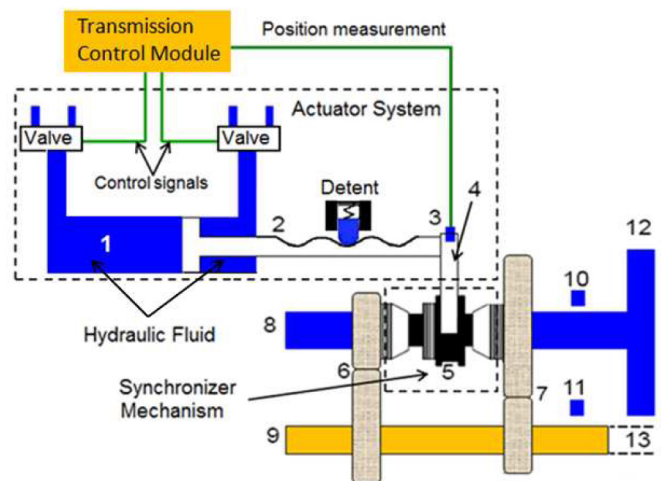


Fig. 2. An overview of the synchronization system showing the actuator, the input and output shafts, and the synchronizer mechanism.

2.1.1. Actuator fork

The actuator piston and fork are both attached to the hydraulic system on one side and to the synchronizer sleeve on the opposite side. The two may be treated as one object in the analysis of frictional forces. There are also static, coulomb and viscous frictions which can be denoted as $f_{a,static}$, $f_{a,coulomb}$ and $\mu_a v_f$ respectively. The friction coefficients are uncertain but their internal relationship is $f_{a,static} < f_{a,coulomb} \ll \mu_a v_f$. The static and coulomb friction can be described as follows [6]:

$$F_{a,sc}(t) = \begin{cases} -\text{sign}(F_{a,r}(t)) \min(f_{a,static}, |F_{a,r}(t)|) & \text{if } l_f(t) = 0 \\ -\text{sign}(l_f(t)) f_{a,coulomb}, & \text{else} \end{cases} \quad (1)$$

where l_f is the fork position, $F_{a,sc}$ is the coulomb friction force and $F_{a,r}$ is the sum of all other forces acting on the actuator piston and fork. The index sc denotes static coulomb. Similarly, the viscous friction effects can be modeled as follows:

$$F_{a,vf}(t) = -\mu_a v_f * l_f(t) \quad (2)$$

where $\mu_a v_f$ is the viscous friction and $F_{a,vf}$ is the viscous friction force on the piston and fork. A force f_{app} applied at the piston when the mechanism is in equilibrium causes a dislocation of x_{fork} between the position sensor and the sleeve attachment, as shown in Fig.3. This displacement can be modeled as a spring and a damper between the fork and the sleeve. The spring rate k_{fs} is calculated as $k_{fs} * x_{fork} = f_{app}$, where f_{app} is the applied force, hence $k_{fs} = f_{app}/x_{fork}$ and the damping rate can be approximated as $c_{fs} = k_{fs} \cdot 10^{-3}$. The force between the fork and the sleeve can therefore be approximated as follows:

$$F_{fs}(t) = k_{fs}(l_f(t) - l_s(t)) + c_{fs}(\lambda_f(t) - \sigma_s(t)) \quad (3)$$

where l_s is the synchronizer sleeve position, λ_f is the fork speed, σ_s is the synchronizer speed and $F_{fs}(t)$ is the force between the fork and the sleeve. The whole mass of the actuator piston and fork m_f can be assumed to be located on the hydraulic system side of the spring and damper effects, hence the equation of motion can be represented as follows:

$$m_f \Omega_f(t) = F_{a,static}(t) + F_{a,vf}(t) + F_{a,sc}(t) - F_{fs}(t) \quad (4)$$

where m_f is the mass of the actuator piston and fork, Ω_f is the fork acceleration, $F_{a,static}$, $F_{a,vf}$ and $F_{a,sc}$ represents the static, viscous and coulomb friction forces respectively. There may also be some backlash in the attachment between the fork and the sleeve which can be approximated as a variable offset error in the gear position sensor output signal as follows:

$$y_l(t) = l_f(t) + \text{sign}(\lambda_f(t)) - \sigma_s(t) \frac{L_b}{2} \quad (5)$$

where $y_l(t)$ is the gear position sensor output, l_f is fork position, λ_f is the fork speed, σ_s is the synchronizer speed and L_b is the backlash in the attachment between the fork and the sleeve.

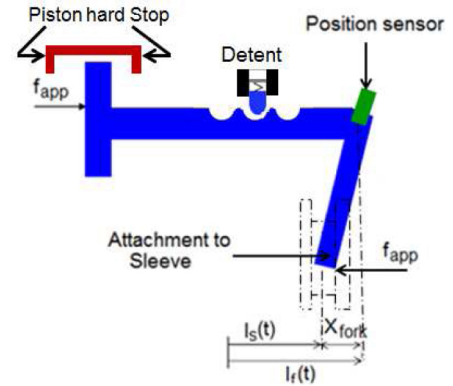


Fig. 3. Displacement between fork and sleeve at zero velocity of the fork and sleeve. The fork arm inclination is exaggerated to illustrate the backlash effect.

2.2. Synchronizer

The most common type of synchronizers used in modern gearboxes is the blocking strut-type. A pictorial view of a double cone strut-type blocking synchronizer is shown in Fig. 4. The hub and sleeve in the center of the synchronizer are fixed to the input shaft and the sleeve can slide in axial direction. During the synchronization process, the sleeve is moved through the blocker ring and gear wheel teeth so that it gets locked to the gear wheel. A gear wheel is placed after the inner ring on both sides. The struts are placed between the hub and the sleeve to pre-synchronize the blocker ring to its correct position relative to the sleeve. This also causes a pressure between the friction rings (the blocker, intermediate and inner rings) which wipes the oil between the two and enables the friction effects. The main synchronization takes place after the pre-synchronization [7].

A force applied at the blocker ring results in pressure between the three rings which in turn generates the torque required to accelerate or decelerate the input shaft. This torque results from the fixated blocker and inner rings to the input shaft and the intermediate ring that is attached to the output shaft, as shown in Fig. 4. Both the inner and outer layer of the intermediate ring is made of a special material having a large magnitude of friction. The choice of the number of friction surfaces for the synchronizer is a matter of wear and efficiency. During synchronization, the sleeve teeth are in contact with the blocker ring teeth. When the sleeve teeth moves beyond the blocker ring teeth, it meets the gear wheel teeth. The input shaft is then pushed aside so that the