

Clutch model and controller development in MATLAB Simulink

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Abstract. With rapid developing of automated gearboxes used in conventional, hybrid and also plug-in hybrid vehicles, clutches are inevitably controlled by transmission control unit. The effectiveness of the clutch control is based on how smoothly are engaged and disengaged in different situations like drive away from standstill, changing gears at high torque demand or changing between different modes in hybrid vehicles. In this paper a clutch model is developed in MATLAB Simulink environment taking in account physical properties of the clutch. For establishing a controller that is robust enough to take in consideration clutch wearing and is smooth enough for ride comfort, multiple control strategies are implemented. Finally the control strategies are compared in different situations.

1. Introduction

The purpose of this paper is to underline the importance of a model in MATLAB for further development use for a clutch and controller model. Especially for hybrid vehicles, this was also studied by Chen et. al. [1], by implementing a torque control during the transition between gears in order to ensure a smooth transition and a continuous torque. This was done by using a reference model, and a controller for the plant. The Lyapunov stability theory was used to make the system have an asymptotic stability. The method was also validated on a SPHEV (series-parallel hybrid electric vehicle) bus. Galvagno et. al. [2] implemented a dynamic and kinematic model of a dual clutch transmission (DCT) on a vehicle with front wheel drive and transversal internal combustion engine and gearbox. For the numerical simulation, several simplifying methods were used. The internal combustion engine model was a steady-state torque map as function of engine speed and throttle position. The implemented maneuvers were a sequence of upshifts and downshifts. By using the transmission model, the performance of specified speed profiles and shift transients were analyzed further and validated. The most problematic issue is the transient state, therefor Kim et. al. [3] implemented several control methods for EV (electric vehicle) / HEV (hybrid electric vehicle) change mode. An open and closed loop analysis was made and as results, methods were evaluated and the method with a slip-less control and ease of calibration was concluded as being the best. The best method was to control the clutch pressure in order to achieve the desired speed difference from each side of the clutch (with motor speed faster than engine idle). Kulkarni et. al. [4] simulated the shift dynamics and control of a DCT. The engine assembly was considered a 2 degree of freedom system (rotation inertia of moving parts and the inertia of the engine and transmission). Engine torque was considered as an interpolation from an engine map with engine speed and throttle position in this case as well. A shift control logic was implemented, with focus on detecting the time of

the shift initialization, creating a specific rate of engagement and disengagement of the clutches and determination of the shift completion. The importance of clutch pressure control signals and clutch pressure control were underlined. Letrouve et. al. [5] analyzed the influence of the clutch model in a simulation of a parallel HEV, where simulations were done with and without a clutch and fuel consumption and dynamic performance was underlined. For the NEDC (New European Driving Cycle) the two models gave similar results in terms of fuel consumption, but the models can be further used to implement different strategies, driver behavior and cycles. Liu et. al. [6] also implemented a DCT control model to analyze the dynamic behavior of vehicles for upshifting and downshifting in different situations like launching in first gear, creep and inching or uphill launch with heavy towing load. The implemented controller was a PID with feedback on the difference between a predesigned gear ratio change function and the actual speed ratio. The model was finally validated by measurements on a test vehicle. Smith et. al. [7] implemented a three PID loop control, with a clutch torque control loop, a motor torque control loop and a wheel torque control loop. The control system was implemented on a dSPACE MicroAutoBox on a HIL (hardware in the loop) bench. The experimental study showed that a flying engine start could be performed at 20 kph with little disturbance to vehicle acceleration. Van Der Heijden et. al. [8] simulated and optimized the control of a dry clutch for hybrid vehicles. After implementing the model, a piecewise linear quadratic (PWQL) control was implemented and compared to a PI controller. The PWQL controller was proven to be better since the clutch engages faster and smoother than the PI controller.

The dry clutch has been widely used in manual transmission vehicles and for automated manual transmissions, controlling automated manual transmissions in the most comfortable way for the passengers are one the most important control problem in this field. Various successful products such as the diaphragm spring clutch, self-adjusting clutch, travel adjusted clutch and pre-damped clutch damper have been developed by Valeo, LUK, SACHS and others. Although the development of a highly responsive system with the engine in low speed ranges makes the control task more difficult, the dry clutch is still widely used because of its efficiency, robustness and low manufacturing cost [9].

In order to simulate the clutch behavior all the powertrain must be modelled for more accurate results. The powertrain consists of the engine, clutch, gearbox, differential and vehicle. For a simple model the powertrain is developed based on components inertia and rotational speed.

2. Objectives

In this research a clutch system was developed in MATLAB Simulink environment and controlled by using different control systems. Controlling the engaging process of the powertrain with the engine by using the clutch is the focus of this paper. To achieve an acceptable control strategy two condition must be fulfilled: the engine speed must never get lower than the idle speed and have a smooth torque and rotation transfer from the engine to the powertrain. Different control types were used and compared in the following sections.

3. Mathematical model of the clutch

Therefore, in the clutch engagement and disengagement phase, there is a slip between the drive and the driven part of the clutch, making the clutch a two degrees of freedom mechanical system. When the clutch pedal is released, the pressure between the driving and driven disk increased and transferred torque and rotation is also increased, to the point where the driving disk and the clutch plate achieve synchronous speed, the clutch is locked as a whole and become a single degree of freedom mechanical system [11].

The maximum torque supported by the clutch can be expressed as:

$$T_{max} = \frac{1}{\pi(R_2^2 - R_1^2)} \int_0^{2\pi} \int_{R_1}^{R_2} r^2 \mu dr d\theta = \frac{2}{3} \frac{R_2^3 - R_1^3}{R_2^2 - R_1^2} \mu F \quad (1)$$

where, R_1 and R_2 are the inner and outer radius of the contact surface between the driving and driven disk; μ is the coefficient of friction between the driving and driven disk; F is the force to compressing the driven disk. When the rotation speed difference between the driven disk and the drive disk becomes close to zero, the slipping disappears.

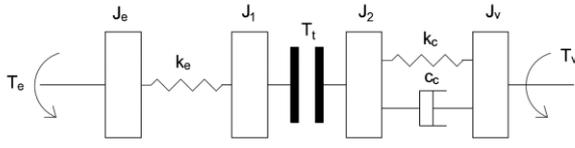


Figure 1. Clutch slipping state

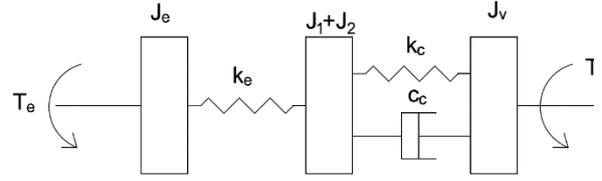


Figure 2. Locked state of clutch

The equations for slipping state of the clutch:

$$J_e \ddot{\theta}_e + k_e(\theta_e - \theta_1) - T_e = 0 \quad (2)$$

$$J_1 \ddot{\theta}_1 + k_e(\theta_1 - \theta_e) + T_t = 0 \quad (3)$$

$$J_2 \ddot{\theta}_2 + k_c(\theta_2 - \theta_v) + c_c(\dot{\theta}_2 - \dot{\theta}_v) - T_t = 0 \quad (4)$$

$$J_v \ddot{\theta}_v + k_c(\theta_v - \theta_2) + c_c(\dot{\theta}_v - \dot{\theta}_2) + T_v = 0 \quad (5)$$

The equations for locked state of clutch:

$$J_e \ddot{\theta}_e + k_e(\theta_e - \theta_{12}) - T_e = 0 \quad (6)$$

$$(J_1 + J_2) \ddot{\theta}_{12} + k_e(\theta_{12} - \theta_e) - k_c(\theta_{12} - \theta_v) - c_c(\dot{\theta}_{12} - \dot{\theta}_v) = 0 \quad (7)$$

$$J_v \ddot{\theta}_v + k_c(\theta_v - \theta_{12}) + c_c(\dot{\theta}_v - \dot{\theta}_{12}) + T_v = 0 \quad (8)$$

The parameters of the clutch model that were used are [9], [12]: $J_e=0.1$ [kgm²], $J_1=0.03$ [kgm²], $J_2=0.02$ [kgm²], $J_v=115$ [kgm²], $k_e=500$ [Nm/rad], $k_c=800$ [Nm/rad], $c_c=0.5$ [Nms/rad], $R_2=220$ [mm], $R_1=150$ [mm], $M=0.42$ [-].

4. Simulink model of the clutch

The Simulink model of the clutch system is a state dependent model, switching between the slipping state and the locked state is based on speed difference between the drive part and the driven part of the clutch. Equations (1) – (8) were implemented in Simulink. In figure 3 the lock logic block is responsible for switching between states of the clutch. The lock logic has inputs like gearbox speed and engine speed. After the lock, the output is clutch locked speed.

5. Control strategy

Requirements for controlling the clutch engagement are: no stalling of the engine or maintaining a minimal engine speed during the slipping phase, second condition is to maintain the locked clutch status after the driven and drive part of the clutch has the same speed. A closed loop controller is developed to satisfy to above conditions, main focus is on maintaining the engine speed above the idle speed. The control in first step is realized with a PI controller.

The PI controller integrator part is limited to last 50 time step error calculation, which is very helpful with long simulation times and highly nonlinear models.

The second controller used is a transfer function which functions the same as the PI and also the behavior is the same, but no limited integrator was used.

A fuzzy logic controller was implemented. The characteristics of this controller were established with trapezoidal and Gaussian functions. The trapezoidal function is the input function with the rotation speed difference, and the output controls the clutch pedal position.

To test the controllers, two setups were made, both drive-away from standstill situations, the difference between them being the gradient of accelerator pedal with 1 second from 0 to 1 for the high gradient and 5 seconds for the lower gradient. This situation is the most critical because of high demand of engine torque and the high torque at low engine speeds must be delivered fast and smooth without stalling the engine.

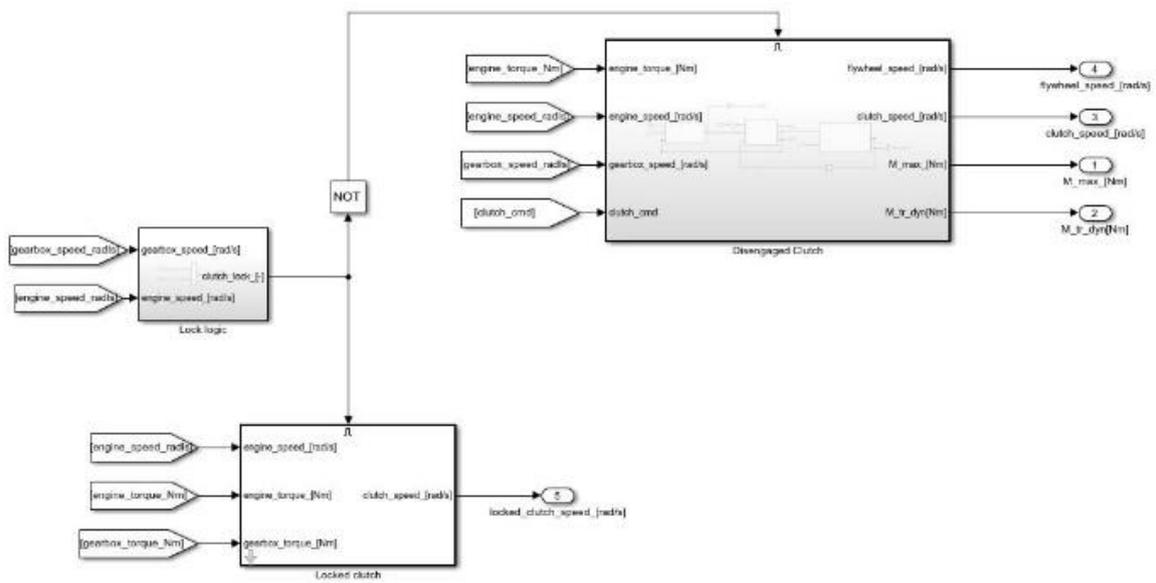


Figure 3. Simulink model of the clutch

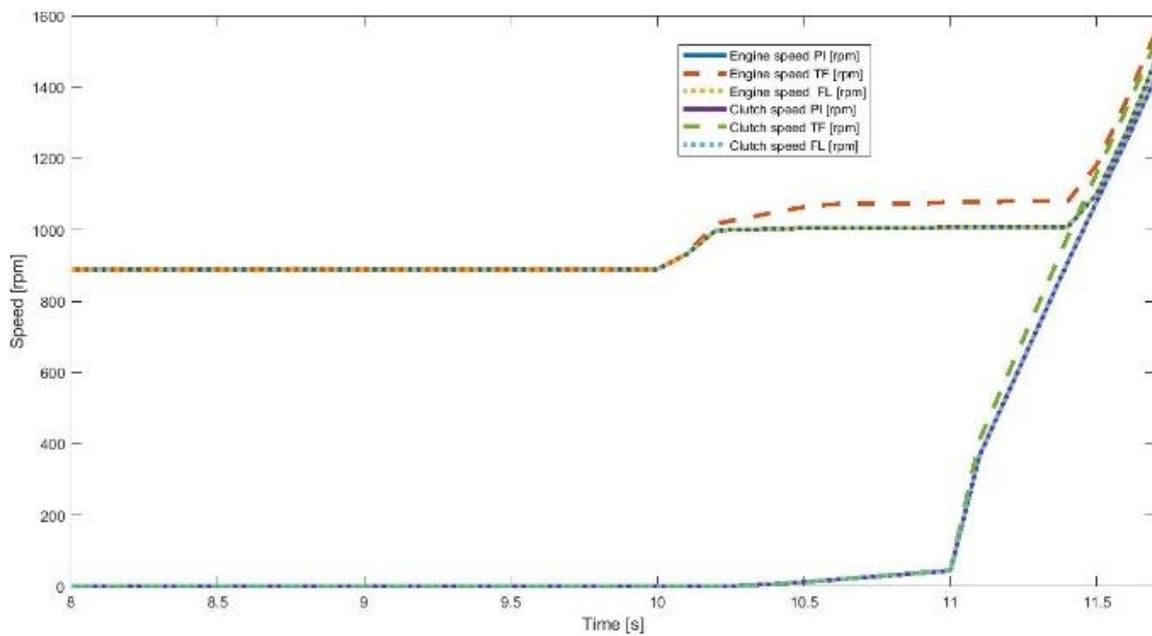


Figure 4. Clutch controller values for high gradient accelerator pedal variation

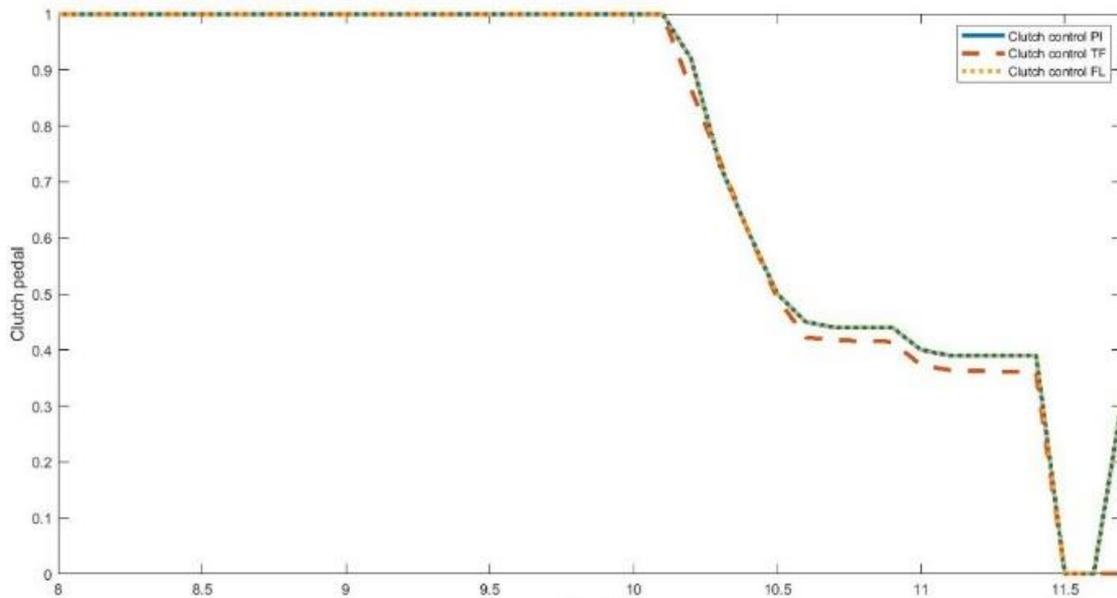


Figure 5. Engine and clutch speed in function of time for high gradient accelerator pedal variation

6. Conclusions

Previous work from Minh and Pumwa [13] has shown that fuzzy logic can be implemented to ensure the successful control of the clutch in HEVs. In this paper, the authors demonstrated that the clutch system model developed for simulation purpose has a similar behavior (figure 4, figure 5). The control strategy that was chosen for this comparison is based on not stalling the engine condition. The three different controllers that were implemented in the model have approximately the same results, engaging times have small differences in all cases. The reason why the three implementations were compared, is because of the computing time necessary to obtain similarly same results, and how that translates into different codes when trying to flash the controller onto a physical control unit.

7. References

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