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Fast and smooth clutch engagement control for dual-clutch transmissions

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ABSTRACT

Automotive dual-clutch transmissions use two gear shafts and two clutches to perform automated gear shifts at a high comfort level. The two objectives of the clutch engagement controller are to realize a fast clutch engagement to reduce the gear shifting time, and a smooth clutch engagement to accurately track the demanded torque without a noticeable torque dip. This research work presents a new controller design that explicitly separates the control laws for each objective by introducing clutch engagement phases. Simulations and experiments in a test vehicle show that the control objectives are realized with a robust and relatively simple controller.

1. Introduction

Automated transmissions have become increasingly popular in passenger vehicles for automatic gear selection and gear shifting (Wheals et al., 2007). There are several types of automated transmissions, such as the classic automatic transmission, the low-cost automated manual transmission, the continuously variable transmission, and the Dual Clutch Transmission (DCT). The DCT, sometimes referred to as a direct-shift gearbox, uses two gear shafts and clutches, as schematically depicted in Fig. 1. The inactive shaft can be prepared for the next gear shift in advance, so the two clutches can quickly switch between gear shafts when the actual gear shifting is demanded (Zhang, Chen, Zhang, Jiang, & Tobler, 2005). This gives the DCT some advantages over other automated transmission technologies: it overcomes the discomfort issue of torque interruption during gear shifting, without sacrificing much of the transmission efficiency. In addition, the fuel efficiency can be improved by using a *rapid upshifting* strategy without sacrificing much of the performance (Ngo, Hofman, Steinbuch, & Serrarens, 2012). The DCT is able to perform fast and comfortable shifts to a lower gear when the driver requests more engine power.

1.1. Control problem formulation

The performance of DCTs heavily relies on the quality of the clutch engagements: the driver expects a quick response of the powertrain without feeling a torque interruption and oscillations caused by the engine inertia. The task of the clutch engagement controller is to realize a fast and smooth clutch engagement, subject to the uncertainties in the actuator dynamics and sensor measurements. Furthermore, for its relevance in the competitive automotive industry, the controller is restricted to the use of only standard speed sensors to keep the cost low, and to the use of limited computation and memory resources for its implementation in real-time hardware. In order to enhance in-vehicle calibration, the design must be transparent to understand the impact of each calibration parameter, whereas the number calibration parameters needs to be as small as possible (Christen & Busch, 2012).

The design of clutch engagement controllers for automated transmissions is widely studied in the literature, resulting in several solutions. Decoupling (linear) controllers show promising simulation and experimental results with easily implementable control laws in Slicker and Loh (1996), Pettersson and Nielsen (2000), Fredriksson and Egardt (2000), Garofalo, Glielmo, Ianelli, and Vasca (2001), Serrarens, Dassen, and Steinbuch (2004) and Glielmo, Iannelli, Vacca, and Vasca (2006). Many of these designs, however, neglect relevant actuator dynamics or consider only their frequency domain characteristics, thereby neglecting relevant transient effects caused by initial conditions of the states. Heuristic and fuzzy logic controllers as presented in Tanaka and Wada (1995) and Kulkarni, Shim, and Zhang (2007) are based on multi-dimensional control maps and require ad hoc calibration of many parameters. Finally, optimal model predictive controllers with mixed cost functions, as described in Bemporad, Borrelli, Glielmo, and Vasca (2001), van der Heijden, Serrarens, Camlibel, and Nijmeijer (2007) and Dolcini and Béchartde Wit (2010), can be

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implemented in a vehicle after offline computation of the explicit control laws. In-vehicle calibration remains cumbersome as each adjustment in one of the calibration parameters requires a new, offline computation.

1.2. Main contribution and outline

Most of the controllers that aim at both fast and smooth clutch engagement can be captured in the same framework: first, the clutch slip speed is reduced at a high rate to achieve a fast clutch engagement, followed by a controlled reduction of the slip acceleration to achieve a smooth clutch engagement. The controlled reduction has to be fast, but cannot be faster than the dynamics of the actuator. This research work presents a new controller design that explicitly separates the control laws for each of these phases. The switches between the phases are chosen such that the desired slip acceleration is achieved at the time of clutch engagement. The latter can be used as the single calibration parameter to determine the trade-off between *fast* and *smooth* clutch engagement. Robustness analysis shows that the desired slip acceleration can be achieved for a range of time constants of the actuator dynamics, yet at the cost of a conservative clutch engagement time. The control framework uses only the slip speed and slip acceleration, so it can be used for the different driving modes of the DCT-based powertrain. It can also be used for a wider class of systems that require fast and smooth clutch engagement, such as described for a mechanical-hybrid powertrain equipped with a flywheel and a continuously variable transmission in van Berkel, Veldpaus, Hofman, Vroemen, and Steinbuch (in press).

This paper elaborates the clutch engagement controller for three relevant driving modes, which are gear upshifting and downshifting during propulsion, and vehicle launching from standstill. Other driving modes, such as gear shifting during braking, are not elaborated, yet the controller can be designed using the same framework. This paper will not consider the design issues related to the setpoint tracking of the clutch torque on component level, as in Horn, Bamberger, Michau, and Pindl (2003) and Montanari, Ronchi, Rossi, Tilli, and Tonielli (2004), yet will focus on the high-level setpoint generation for the powertrain components that influence clutch engagement, *i.e.*, the engine and the clutch. In summary, the main contributions of this paper are:

- the design of a stable and robust controller for fast and smooth clutch engagement that is tunable with a single calibration parameter; and
- a generic control framework elaborated for three relevant driving modes.

The outline is as follows: Section 2 presents the simulation model of the powertrain. Section 3 formalizes the control problem and describes the controller design. The performance and robustness of the controller are evaluated by simulations in Section 4 and by experiments in Section 5. Section 6 summarizes the results and conclusions.

2. Dynamic powertrain model

The simulation model describes the most important longitudinal dynamics of the DCT-based powertrain. The purpose is to simulate the engagement of either clutch in Fig. 1. Since the clutch engagement is not performed simultaneously with the "cross shift" switching between the two gear shafts, as will be discussed in Section 3.1.4, it is sufficient to model only one clutch and one gear shaft for the clutch engagement simulations. The remaining relevant components are the engine, clutch, gearbox, drive shaft, and the vehicle, as shown in Fig. 2. The powertrain parameters used throughout this paper are listed in Table 1, where the speed ratios are defined as the rotational speed on the vehicle side

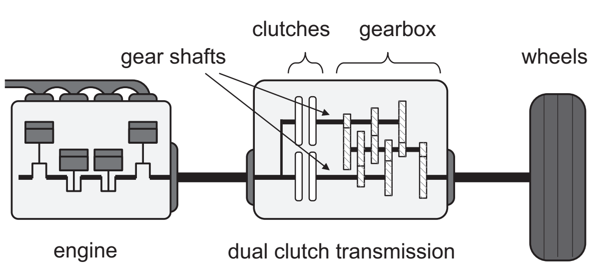

Fig. 1. Schematic representation of a DCT-based powertrain. The DCT enable fast and smooth gear shifting using two gear shafts and two clutches.

Table 1
Powertrain model parameters.

Symbol	Value	Unit	Description
J_e	0.15	kg m^2	Engine inertia
J_p	0.01	kg m^2	Primary inertia
J_s	0.01	kg m^2	Secondary inertia
J_v	90	kg m^2	Vehicle inertia
d_d	400	Nms/rad	Drive shaft damping
k_d	6000	Nm/rad	Drive shaft stiffness
r_d	0.19	–	Final drive ratio
r_g	$\{0.25, 0.39, 0.62, 0.97, 1.42\}$	–	Gearbox ratios
$\hat{\eta}_t$	0.92	–	Average gear efficiency
τ_e	$[-20, 140]$	Nm	Engine torque
τ_c	$[0, 140]$	Nm	Clutch torque
θ_e	0.08	s	Engine time constant
θ_c	0.04	s	Clutch time constant

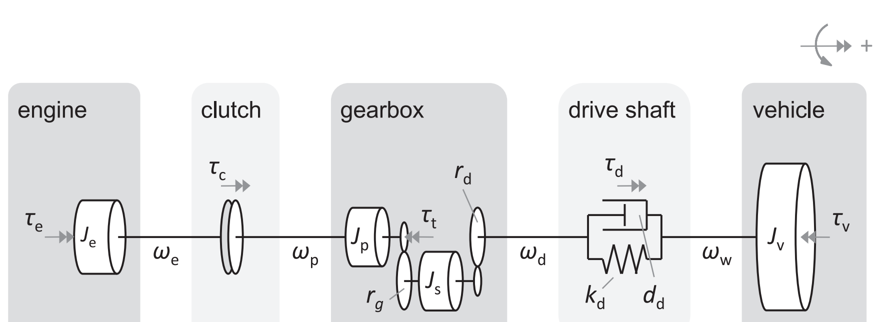

Fig. 2. Detailed dynamic powertrain model for simulation purposes.