

Model-Based Wheel Torque and Backlash Estimation for Drivability Control

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ABSTRACT

To improve torque management algorithms for drivability, the powertrain controller must be able to compensate for the nonlinear dynamics of the driveline. In particular, the presence of backlash in the transmission and drive shafts excites sharp torque fluctuations during tip-in or tip-out transients, leading to a deterioration of the vehicle drivability and NVH.

This paper proposes a model-based estimator that predicts the wheel torque in an automotive drivetrain, accounting for the effects of backlash and drive shaft flexibility. The starting point of this work is a control-oriented model of the transmission and vehicle drivetrain dynamics that predicts the wheel torque during tip-in and tip-out transients at fixed gear. The estimator is based upon a switching structure that combines a Kalman Filter and an open-loop prediction based on the developed model. The estimator relies only on measurements and signals that are commonly available in production vehicles, and in this paper is implemented in real-time using a rapid prototyping ECU. The estimator is verified experimentally on a test vehicle on a chassis dynamometer.

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INTRODUCTION

Engine downsizing, boosting and powertrain electrification are key technical solutions for improving fuel economy while complying with emissions standards. The implementation of sophisticated propulsion systems however requires careful control of the torque at the wheels, which has profound effects on drivability and, ultimately, on the customer acceptance. For this reason, the automotive industry is investing in new hardware and software systems to improve the drivability, for improved comfort and "fun to drive".

One of the critical cases where drive quality must be carefully managed is relative to the "backlash traversal" event, which occurs when the engine torque suddenly changes sign, for instance during a transition from coasting to acceleration (tip-in), or vice-versa (tip-out). Backlash is a phenomenon caused by gaps in the transmission linkages leading to zero torque transmitted through the drive shafts to the wheels when the sign of the traction torque changes. Traversing the backlash gap causes large shaft torque fluctuations and high-frequency longitudinal accelerations of the vehicle that are perceivable by the driver.

The design of engine and powertrain control algorithms that can improve drivability by compensating for backlash-induced torque fluctuations has received considerable attention in the literature, as mentioned for instance in [1], [2], [3], [4]. Since the backlash introduces a strong nonlinearity in the control loop for torque management, information about internal states of the system is required to improve the performance of the controller. In particular, the position within the backlash gap and the wheel torque are typically not available in production vehicles, due to the cost and reliability issues of current sensing technologies. For this reason, estimation algorithms are typically used to provide such information to the controller.

A seminal work in the field of backlash estimation is presented in [5] and [6], where an observer is designed to provide the instantaneous driveshaft position within the backlash gap based upon signals from wheels and engine sensors. Unfortunately, the proposed approach requires measuring both the velocity and angular position of the engine and wheel shafts, information that are typically unavailable in a production context. Moreover, [2] explains that when the system

crosses the lash region, the engine and the wheels are momentarily decoupled, which poses further challenges in the design of closed-loop estimator.

This paper proposes a model-based backlash and wheel torque estimator for real-time implementation in vehicle. The estimator is developed starting from a physics-based model of the vehicle drivetrain, which predicts the high-frequency, nonlinear behavior induced by the backlash and flexibility of the drive shafts. The model is validated on experimental data collected on a front wheel-drive passenger car on a chassis dynamometer. The proposed wheel torque estimator is designed to switch between a closed-loop observer (based on a Kalman Filter), and an open-loop model that is active during the backlash traversing events where loss of contact (and observability) occurs.

While the paper is inspired by prior research in this field ([3]), the work here described presents novel contributions in the system modeling, the formulation of the estimation problem, and in the implementation of the estimator. In addition, the paper describes the details of the implementation of the estimator in a rapid prototyping ECU, and the experimental verification in vehicle.

The paper is organized as follows. First, a description of the experimental setup is presented. Next, the transmission and driveline dynamics model is introduced. Then, the design process for the wheel torque estimator is illustrated, followed by a description of the experimental setup. Finally, implementation and verification of the estimator in vehicle are discussed.

DESCRIPTION OF THE EXPERIMENTAL SETUP

The experimental tests were conducted on a Chrysler Town and Country minivan with a FWD 6-speed automatic transmission, installed on a 150kW medium-duty chassis dynamometer. An ETAS ES1000.3 system was used to measure the relevant outputs (wheel speed, engine speed, gear, torque converter turbine speed and torque, wheel torque) and for real-time implementation of the estimator. An ETK interface allows for a direct reading of the ECU variables.

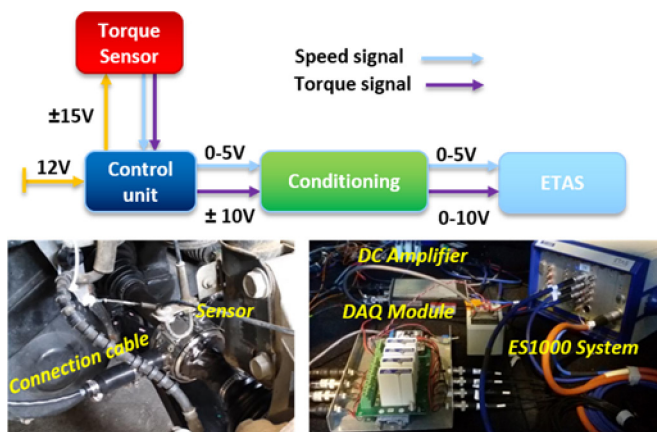


Figure 1. Overview of the Experimental Setup for Data Acquisition and Estimator Verification.

A schematic of the experimental setup designed for verification of the estimator is shown in Figure 1. The vehicle was modified by installing a strain gauge bridge torque sensor with encoder on the front left half shaft. The sensor allows for measuring the traction torque (or drive shaft torque) at the wheel and the wheel speed with 0.1ms sampling time. This information is used for model calibration and validation only. A Dataforth module and DC amplifier were used for conditioning the sensor output signal.

CONTROL-ORIENTED TRANSMISSION AND DRIVELINE MODEL

As a first step in the estimator design, a control-oriented drivetrain model was developed for a front wheel drive passenger car with automatic transmission. Figure 3 shows the structure of the model.

The model is designed to capture the transmission and driveline dynamics during transient operations at constant gear. In this context, four inertias are introduced to model the engine crankshaft dynamics, the transmission input shaft dynamics, the wheel slip dynamics and the vehicle longitudinal dynamics. The different components of the system characterized by backlash (for instance, the transmission gears and the drive shafts joints) are lumped together into one single term, which is considered a reasonable approximation for drive quality control [5], [7], [8].

Engine Dynamics and Torque Converter

The engine crankshaft rotational dynamics model is given by:

$$J_{eng} \frac{d\omega_{eng}}{dt} = T_{eng} - T_p \quad (1)$$

where J_{eng} represents the equivalent engine crankshaft inertia (inclusive of the torque converter pump), ω_{eng} is the engine angular velocity, and T_{eng} and T_p are the engine and torque converter pump torque, respectively.

A mean-value engine model is used to predict the engine brake torque [9], [10], [11]. In this model, the torque is considered a function of the air charge, residual gas concentration and spark timing, which are controlled by simple feed-forward maps scheduled based on the desired torque demand from the driver.

As suggested in [12], the torque converter has been modeled by separately consider its behavior in torque multiplication mode, fluid coupling mode and overrun mode. The point of distinction among the different modes of operation is defined by the value of the speed ratio $SR = \omega / \omega_{eng}$. The basic equations governing the torque converter in each of the three modes have the following structure:

$$\begin{aligned} T_t &= a_{1t} \omega_{eng}^2 + a_{2t} \omega_{eng} \omega_t + a_{3t} \omega_t^2 + T_{icc} \\ T_p &= a_{4t} \omega_{eng}^2 + a_{5t} \omega_{eng} \omega_t + a_{6t} \omega_t^2 + T_{icc} \end{aligned} \quad (2)$$

where the parameters a_{ji} are the coefficients resulting from a second order curve fitting of the torque converter data, and i refers to the torque converter operating mode (torque multiplication, fluid coupling, overrun).

The presence of the torque converter lockup clutch is accounted for in Eq. (2) by introducing the clutch torque T_{lcc} :

$$T_{lcc} = a_{TC} \cdot lock \cdot (\omega_{eng} - \omega_t) \quad (3)$$

where a_{TC} is a calibration parameter and *lock* is a discrete variable that indicates the state of the torque converter lockup clutch.

Transmission Dynamics

The drivetrain includes an automatic transmission. Since the focus of this work is on characterizing the driveline dynamics during in-gear operations, a single-inertia model is used to approximate the inertias of the torque converter turbine, gears and shafts on to the primary (input) shaft. This allows one to considerably simplify the dynamics of the transmission to a single equation:

$$J_{tr} \frac{d\omega_t}{dt} = \eta T_t - \frac{2}{\tau_{gear} \tau_{FD}} T_{shaft} \quad (4)$$

where τ_{gear} is the gear ratio of the transmission, τ_{FD} is the final drive ratio, and the term J_{tr} represents the equivalent inertia at the input shaft, which depends on the current gear and torque converter clutch state. Note that Eq. (4) assumes that an equal torque T_{shaft} is transferred to the left and right half-shafts.

The transmission efficiency $\eta = \eta(\omega_t, T_t, gear)$ is implemented in the model as a four-dimensional look-up table as function of the inputs and states. The final drive speed ω_{FD} is then calculated from Eq. (4) as:

$$\omega_{FD} = \frac{\omega_t}{\tau_{gear} \tau_{FD}} \quad (5)$$

Drive Shaft Dynamics with Backlash

As shown in [6], the nonlinear backlash dynamics is modeled by assuming an inertia-free flexible shaft that connects the transmission to the wheels. The approximation made consists of combining all the

backlash contributions together into a single term. This is a reasonable approximation since the inertias of the rotating masses between the backlash terms are generally small.

The drive shaft dynamics model is inspired by the work presented in [7]. Figure 2 shows a basic schematic of the backlash model and the essential notation.

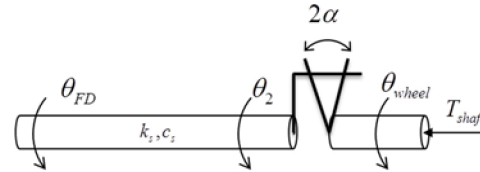


Figure 2. Simplified Model of Backlash (Adapted from [7]).

The torque exchanged by the flexible shaft is given by:

$$T_{shaft} = c_s (\dot{\theta}_d - \dot{\theta}_b) + k_s (\theta_d - \theta_b) \quad (6)$$

where $\theta_d = \theta_{FD} - \theta_{wheel}$ is the total shaft twist, c_s is the damping coefficient and k_s is the stiffness of the flexible shaft.

The above equation requires knowledge of the shaft position within the backlash gap, defined by $\theta_b = \theta_2 - \theta_{wheel}$. This variable is predicted by the equation:

$$\dot{\theta}_b = \begin{cases} \max(0, \dot{\theta}_d + (k_s / c_s) \cdot (\theta_d - \theta_b)) & \theta_b = -\alpha \\ \dot{\theta}_d + (k_s / c_s) \cdot (\theta_d - \theta_b), & |\theta_b| < \alpha \\ \min(0, \dot{\theta}_d + (k_s / c_s) \cdot (\theta_d - \theta_b)) & \theta_b = \alpha \end{cases} \quad (7)$$

which indicates that the shaft can only move within the backlash gap. When the shaft reaches one of the two boundaries (contact mode), the total displacement must be large enough in order for the shaft to move back into the gap.

Equations (6-7) essentially impose that the position θ_b is constrained to change within the backlash gap 2α , and that the shaft can transfer positive or negative torque only if the backlash angle is exactly at the corresponding boundary. Anytime the backlash angle falls within the gap, zero net torque will be transmitted [6], [13].

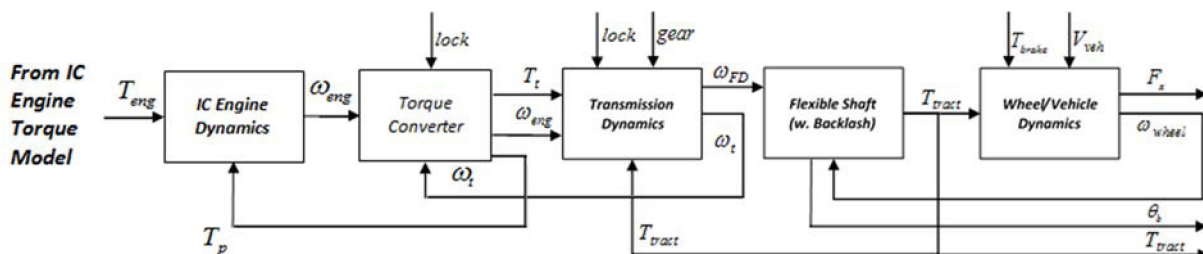


Figure 3. Block Diagram of the Vehicle Driveline Model.