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Nonlinear Estimation and Control of Automotive Drivetrains

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(汽车传动系的非线性估计与控制)

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Preface

Motivation

Electronic control has become the core technology in automotive industry to meet the increasingly stringent emission legislation and dynamic performance requirements. Accordingly, automotive electronics account for a larger and larger proportion of the manufacturing cost of the whole vehicle, including not only hardware cost but also the development cost of control software.

Although at present the widely used control algorithms are still based on event-driven (rule-based) feedforward and PID control, the question of how to design a high-performance control program efficiently using advanced control theories has become a hot topic in the fields of both control and automobile engineering. Since 2006, many academic journals, including IEEE T. Control Systems Technology, Control Engineering Practice, Int. J. Control, Vehicle System Dynamics, Int. J. Powertrain, etc. have published their special issues on automotive control. Besides, sessions on automotive control are organized every year at the annual conferences of IFAC, IEEE CDC and ACC, etc.

The application of advanced control theories is attractive because of its potential to reduce the calibration workload and improve the dynamic control performance under numerous driving conditions and large environmental variations.

This text presents an in-depth discussion on the control problems in automotive drivetrains, particularly the types of hydraulic Automatic Transmission (AT), Dual Clutch Transmission (DCT) and Automated Manual Transmission (AMT). The challenging estimation and control problems, such as driveline torque estimation and gear shift control, are addressed by applying the most up-to-date nonlinear control theories, including constructive nonlinear control (Backstepping, Input-to-State Stable) and Model Predictive Control (MPC). The estimation and control performance is improved while the calibration effort is reduced significantly. This book gives a detailed design process of many examples, and thus enables the readers to understand how to successfully combine the "purely theoretical methodologies" with "actual vehicle applications".

Intended Readers

This book should enable graduate and higher-level undergraduate students to understand the control and estimation problems in automotive drivetrains, and how to use control theories to solve these practical problems.

This book is also suited for the professional control engineers in the R&D centers of automobile manufacturers.

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Dr. Bingzhao Gao received the B.S. and M.S. degrees in vehicle engineering from Jilin University of Technology, China, in 1998 and Jilin University, China, in 2002, respectively. He received the Ph.D. degree in control engineering under the instructions of Prof. Chen in 2009, and his thesis was honored as an Excellent Doctoral Dissertation of Jilin Province, China. He is also a holder of the Doctor's degree in mechanical engineering of Yokohama National University, Japan. Dr. Gao is currently an associate professor at Jilin University. His research interests include vehicle powertrain control and vehicle stability control.

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Contents

Chapter 1
Introduction

1.1 Introduction of Automotive Drivetrain

Generally speaking, the terms of "powertrain" and "drivetrain" (see Fig. 1.1) refer both to the vehicle components which produce and deliver the power and torque. The term "powertrain" sometimes emphasizes the engine and the transmission, while "drivetrain" (or driveline) stresses the clutch (torque converter), transmission, driveshaft, differential gear box, axle shaft and wheels. The drivetrain delivers engine torque to the tires, and makes it possible for the vehicle to accelerate or climb a gradient. Figure 1.2 shows the function of a step-ratio transmission, where the engine torque characteristics are re-distributed, through different gear ratios, to approach a desired pattern of wheel torque.

1.1.1 Engine

Internal combustion engine generates power by converting chemical energy contained in the fuel into heat, and the heat produces then mechanical work. The engine torque T_e is determined by the flow rate of intake air and fuel, and influenced by combustion efficiency and friction losses. In modern vehicular powertrains, high-speed CAN (Controller Area Network) bus connects the control units of the engine and the transmission, and the shared information includes throttle angle, engine torque and engine speed, etc. On the other hand, the transmission sends torque request to the engine through CAN bus. Here, the detailed engine model will not be considered in the context of this book.

The dynamic equation of engine speed is described by

$$I_e\dot{\omega}_e + C_e\omega_e = T_e - T_c, \tag{1.1}$$

where I_e is the inertia moment of the engine crank shaft, C_e is the damping coefficient, ω_e is the engine rotational speed, T_e is the engine torque and T_c denotes the

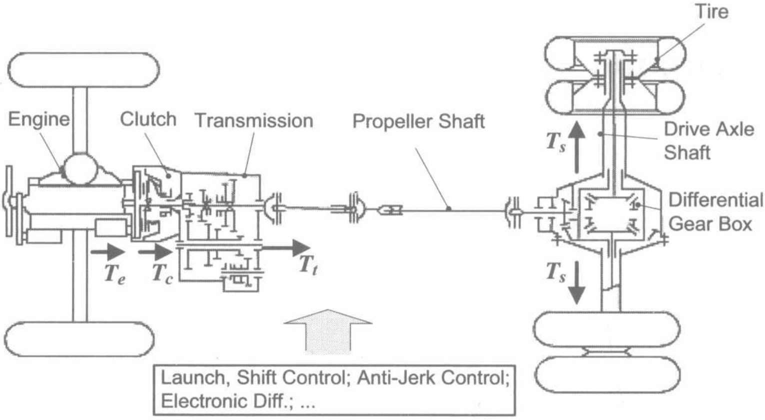

Fig. 1.1 Drivetrain of FR (Front Engine, Rear Wheel Drive) vehicle [30]: T_e, engine torque; T_c, clutch torque; T_t, transmission output torque; T_s, torque of drive axle shaft

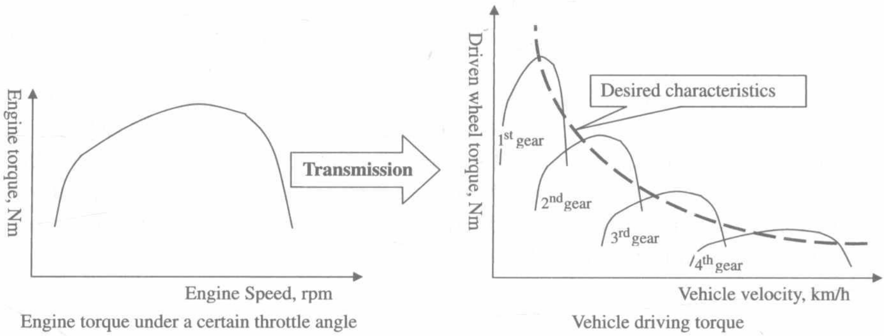

Fig. 1.2 Drive torque characteristics of a vehicle with a step-ratio transmission

clutch torque. The most simple engine model is the static torque map, which is a lookup table with the inputs of the throttle angle and the engine speed, and denoted as

$$T_e = T_e(\omega_e, \theta_{th}), \tag{1.2}$$

where θ_{th} is the engine throttle angle. Various maps in vehicle engineering are obtained from large numbers of experiments in the steady state. As an example, the torque map of a 2000 cc gasoline engine is shown in Fig. 1.3.

1.1.2 Clutch/Torque Converter

A dry clutch, shown in Fig. 1.4 [61], consists of a housing, pressure plates, friction plates, a clutch disc with torsion damper and a release mechanism. In manual

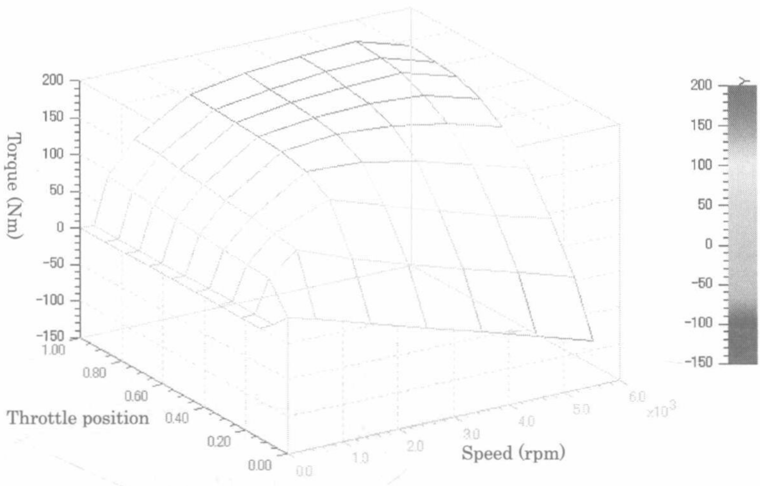

Fig. 1.3 Example of an engine torque map (lookup table)

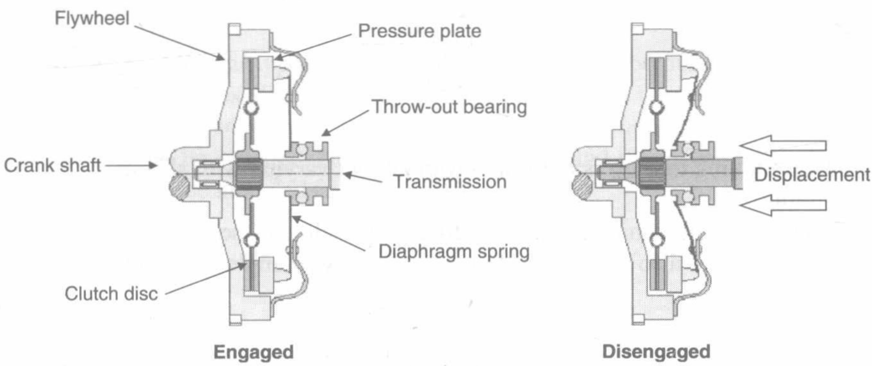

Fig. 1.4 Schematic overview of a dry clutch. Reprinted from [61], copyright 2007, with permission from Taylor & Francis

transmissions, when the vehicle is starting off from standstill, the clutch slips to compensate for the speed difference between the engine and the drivetrain. Moreover, when a gear shift operation takes place, the clutch disengages the engine from the transmission, and then engages them after gear shifting is over.

When the clutch is slipping, the torque delivered through the clutch T_c is determined by the clamping force F_c implemented on the friction disc:

$$T_c = F_c \mu_d R_c \operatorname{sign}(\Delta\omega), \tag{1.3}$$

where F_c is the clamping force, μ_d is the dynamic friction coefficient, R_c is the effective radius, and $\Delta\omega$ is the speed difference between the engine and the drivetrain.

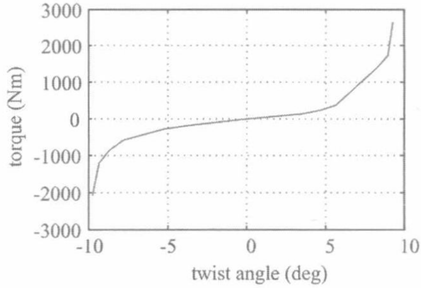

Fig. 1.5 Clutch spring characteristics

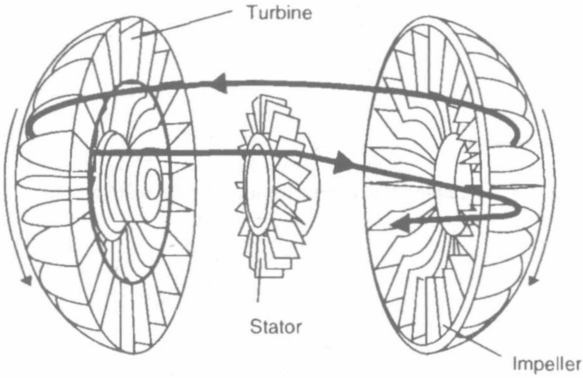

Fig. 1.6 Schematic overview of a torque converter [55]

It is worth noting that μ_d is not time-invariant, but varying with slip speed and the temperature.

Considering the damp spring embedded in the clutch, the torque of the clutch friction plate T_c is also a nonlinear function of the twist angle θ_c as follows:

$$T_c = T_c(\theta_c, \dot{\theta}_c), \tag{1.4}$$

and the equation is applicable for both slipping and locked-up state of the clutch.

The static characteristics of the torsion spring of a 4-ton truck clutch is shown in Fig. 1.5.

On hydraulic automatic transmissions, the torque converter, as shown in Fig. 1.6, assumes the functions of the clutch. When the turbine is driven forward, the dynamics of the torque converter are often characterized as [145]

$$T_p = C(\lambda)\omega_e^2 \tag{1.5}$$

and

$$T_{tb} = t(\lambda)T_p, \tag{1.6}$$

where T_p is the pump torque and T_{tb} is the turbine torque, λ is the speed ratio defined as

$$\lambda = \frac{\omega_{tb}}{\omega_e}, \tag{1.7}$$

with ω_{tb} being the turbine speed.

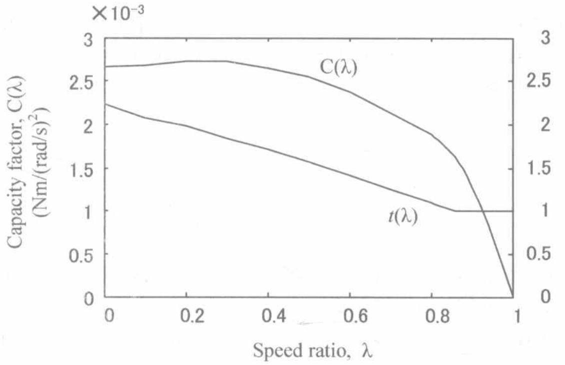

Fig. 1.7 Capacity factor and torque ratio of a torque converter

An example of the capacity factor $C(\lambda)$ and the torque ratio $t(\lambda)$ in a mid-size passenger car are given in Fig. 1.7.

1.1.3 Transmissions

There are many different types of transmissions which will be described in detail in the following sections. The function of the transmission is to modify the engine torque and engine speed with the ith gear ratio R_i, so that the momentary traction requirement could be satisfied. Neglecting the friction and the inertia torques, the transmission could be modeled as

$$T_t = T_c R_i, \tag{1.8a}$$

$$\omega_t = \frac{\omega_c}{R_i}, \tag{1.8b}$$

where T_t is the transmission output torque, T_c is the clutch output torque (transmission input torque), ω_t is the transmission output speed, ω_c is the clutch output speed (transmission input speed).

1.1.4 Propeller Shaft and Differential Gear Box

In FF (Front Engine, Front Wheel Drive) vehicles, the differential box is always combined with the transmission directly, while in FR (Front Engine, Rear Wheel Drive) vehicles, a propeller shaft connects the transmission and the differential box. The stiffness of the propeller shaft is comparatively larger, compared with that of the axle shaft and the clutch torsion spring. However, the clearance in the drivetrain shafts is an important element when modeling the propeller shaft precisely.

The differential unit compensates for the speed difference between the inside and the outside wheels when the vehicle is cornering. Generally speaking, the two output torques of the differential box are equivalent, while the two rotational speeds do not necessarily equal each other. If the twist deflection of the propeller shaft is ignored, we have

$$T_l = T_r, \tag{1.9a}$$

$$\omega_l + \omega_r = \frac{2\omega_t}{R_{df}}, \tag{1.9b}$$

where R_{df} is the gear ratio of the differential box, the subscripts l and r denote the left side and the right side.

At the same time, the rotational dynamic equation from the transmission to the differential is

$$I_p \dot{\omega}_t + C_p \omega_t = T_t - \frac{2T_{l,r}}{R_{df}}, \tag{1.10}$$

where I_p and C_p are the inertia and the damping of the propeller shaft, respectively.

1.1.5 Drive Axle Shaft

The two drive shafts between the differential gear and the driven wheels are represented as a torsion spring with stiffness coefficient K_s and a damping with coefficient C_s as follows:

$$T_s = 2T_{l,r}, \tag{1.11a}$$

$$T_s = K_s \theta_s + C_s \dot{\theta}_s, \tag{1.11b}$$

where T_s is the axle shaft torque and θ_s is the twist angle of the axle shaft satisfying

$$\dot{\theta}_s = \omega_{l,r} - \omega_w, \tag{1.12}$$

with ω_w being the wheel speed.

1.1.6 Tires and Vehicle

The longitudinal tire force F_x, which is usually simplified as a function of the longitudinal slip ratio S_x, rises fast when S_x increases under a threshold and declines slowly after that [55], see Fig. 1.8. The force F_z is vertical load of the tire, and the longitudinal slip is calculated as

$$S_x = \frac{R_w \omega_w - V}{R_w \omega_w} \quad \text{when driving,} \quad \text{and} \tag{1.13a}$$

$$S_x = \frac{V - R_w \omega_w}{V} \quad \text{when braking,} \tag{1.13b}$$

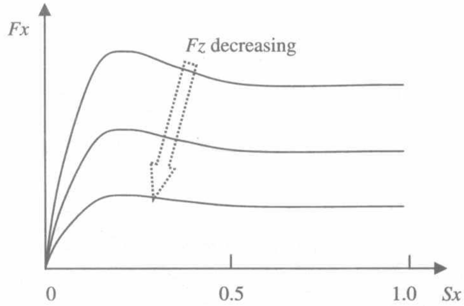

Fig. 1.8 Longitudinal force characteristics of tires

where R_w is the tire radius, ω_w is the wheel rotary velocity and V is the car body velocity.

The road load consists of three parts: the grade force F_G, the rolling resistant moment T_w of tires and the aerodynamics drag F_A. The resistant moment T_w of tires is regarded as constant here. The grade force is calculated as

$$F_G = mg \sin\theta_g, \tag{1.14}$$

where m is the vehicle mass, θ_g is the grade angle of the road. The aerodynamic drag is described as

$$F_A = \frac{1}{2}\rho C_D A_A V^2, \tag{1.15}$$

where C_D is the aerodynamic drag coefficient, A_A is the front area of the vehicle and ρ is the air density.

1.2 Overview of Automotive Transmissions

Automatic transmission, which relieves the driver from shift operation, changes the speed ratio of a drivetrain automatically according to the driver intent, current engine state and road surface condition, so that optimal drivability or fuel economy could be obtained. As mentioned before, many types of transmissions have been developed, and Fig. 1.9 shows the history of automotive transmissions. Different transmission has its own unique features and thereby its own control tasks.

1.2.1 Hydraulic Automatic Transmission (AT)

The predominant form of a hydraulic Automatic Transmission (AT) [1] uses a torque converter, and a set of planetary gearsets to provide a range of gear ratios. The torque converter consists of three rotating elements with curved blades: pump, turbine and stator. The pump and turbine hydraulically connect the engine to the transmission and the stator is used to enhance torque multiplication. The torque converter is followed by a set of planetary gearsets, usually including 2–4 planetary gearsets. Each