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Receding horizon online optimization for torque control of gasoline engines

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A B S T R A C T

This paper proposes a model-based nonlinear receding horizon optimal control scheme for the engine torque tracking problem. The controller design directly employs the nonlinear model exploited based on mean-value modeling principle of engine systems without any linearizing reformation, and the online optimization is achieved by applying the Continuation/GMRES (generalized minimum residual) approach. Several receding horizon control schemes are designed to investigate the effects of the integral action and integral gain selection. Simulation analyses and experimental validations are implemented to demonstrate the real-time optimization performance and control effects of the proposed torque tracking controllers.

1. Introduction

The modern gasoline engine is a sophisticated control system that involves many control loops to meet the increasing requirements for efficiency improvement and emissions reduction. In a basic control loop, torque control is crucial to guarantee drivability and ensure the coordinative operation with other control units such as the traction control system (TCS) [1,2], gearshift control system [3], and energy management system (EMS) of hybrid electric vehicles [4]. Many advanced control technologies have been applied to the torque control issue in past decades. Typically, the fuzzy robust controller was designed in [5] to achieve torque tracking with the throttle as the control input. Chamaillard et al. [6] proposed a simple PI and LQ torque controller with Smith predictor embedded to tackle the time-delayed control problem. In [7], H_∞ controller was proposed to control engine torque with zero steady-state error. Feedback linearizing and pole-placement methods were adopted in [8] based on a nonlinear state-space model to realize the torque tracking. Other torque control methods include neural sliding-mode control [9], and linear quadratic regulator (LQR) control [10]. Ashok et al. [11] reviewed the torque management strategies adopted in spark-ignition engines, which also indicates the variety and complexity of torque control methods. Although these methods provide feasible application examples on torque control, the optimal control performance for engine torque control is still challenging due to the presence of the physical boundary, nonlinearities and time-varying features in real engine systems.

In this context, receding horizon control (RHC), also known as model predictive control (MPC), has attracted much attention from the automotive industry during the past decade, due to its ability to handle the multi-variable, parameter-varying, and constrained optimal control problem [12–14]. RHC solves the open-loop optimization problem in each control period by minimizing the given performance index over a certain future horizon, in which the initial state is the current state of the plant and consequently the optimal control sequence is derived, while only the first control action of the derived optimal control sequence is applied to the plant. Indeed, such a control scheme has been proved quite effective for large classes of unit operations with slow response [15] and has already gained great success in the process industry [16]. However, a challenge of RHC is the intensive computation load for the online receding horizon optimization. Recent study efforts on RHC theory focus on the improvement of the real-time computing efficiency. Some optimization algorithms and packages have been developed, e.g., [17–20], most of which are still based on the extensive numerical iteration calculation. Hence, the real-

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time implementation for fast, nonlinear control plants such as engine systems is still challenging.

Recently, several studies have applied the receding horizon optimization concept to engine control issues, including the air-path control of the turbocharged gasoline engine [21] or diesel engine [22], air–fuel ratio control [23], and combustion phase control [24]. Typically, [25] presented simulation analysis for MPC-based engine torque control in which the online optimization algorithm relies on the computation of sensitivity function. In [26], a nonlinear speed controller was developed with the simulation validations, in which the particle swarm optimization (PSO) algorithm was adopted to complete the optimization. Furthermore, Cairano et al. [27] proposed experimental evaluations for a MPC-based idle speed controller, and the linearized model was adopted to reduce the computational load. In addition, Kang et al. [28] and Gagliardi et al. [29] applied the numerical solver referred to as the Continuation/GMRES (Generalized Minimum Residual Estimation) method [30] to the engine speed tracking issue and the air path control issue of diesel engines, respectively. The experimental results demonstrated the effectiveness of their control schemes. Nevertheless, studies have rarely applied the real-time RHC scheme to the torque control issue, which is especially straightforward using a nonlinear model rather than a linearized one because the latter results in the loss of model accuracy. Moreover, for combustion engines, the model parameters always vary according to operating conditions, which exacerbates the complexity of the RHC controller design. In practical application, constructing a model that can cope with a wider operating range will be quite important.

In this paper, the torque tracking control based on RHC scheme is investigated on a full-scale gasoline engine. The control scheme aims to achieve rapid torque tracking control under any given transient driving condition. To this end, the nonlinear model of the gasoline engine used for controller design (namely, control-oriented model) is first derived based on the mean-value modeling principle, and the model parameters are obtained by means of continuous-time recursive least square estimation algorithm and curve fitting method. Second, several receding horizon optimal control schemes, depending on whether an integral action is embedded or the integral gain is optimized, are proposed to investigate the effects of the integral action for the practical engine control issue. The control performance of the proposed control schemes is compared via both simulation and experimental analyses. Moreover, the real-time optimization achieved by the Continuation/GMRES algorithm is also assessed on the full-scale gasoline engine test bench. The transient control performance and the robustness of the proposed control schemes under driving conditions are further evaluated on an engine-in-the-loop simulation system.

The rest of the paper is organized as follows. The engine dynamic models including intake manifold air charging dynamics and engine torque production model are described in Section 2. Section 3 describes the RHC optimization problems and presents the torque tracking controller design within the framework of the Continuation/GMRES method. Then, simulation analyses and experimental validations are demonstrated in Section 4 to evaluate the proposed controllers. Finally, conclusions are presented in Section 5.

2. Modeling

Since the physics of the internal combustion engine are quite complicated involving thermodynamics, fluid mechanics and mechanical kinematics [32,33], exactly characterizing the engine dynamics will lead to high technical complexity. To simplify the model and avoid excessive complexity, the mean-value model is generally adopted for control-oriented development.

For a gasoline engine, air charging dynamics are mainly considered to represent the pressure change in the intake manifold. Basically, the dynamic behavior of the manifold pressure is characterized in the following form based on the ideal gas law and isothermal assumption:

$$\dot{p}_m = \frac{RT_m}{V_m}(\dot{m}_{in} - \dot{m}_{out}) \tag{1}$$

where p_m is the intake manifold pressure; R, T_m, and V_m denote the ideal gas constant, the air temperature in the intake manifold and the volume of intake manifold, respectively; $\dot{m}_{in}$ represents the air mass flow rate passing through the throttle valve and $\dot{m}_{out}$ is the air mass flow rate going into the cylinder.

Supposing no friction and no inertial effects work on the air fluid, the air mass flow rate passing through the orifice and entering the manifold can be described as follows according to Bernoulli's law [33,34]:

$$\dot{m}_{in} = \frac{\pi c_d D^2}{4}\left(1 - \frac{\cos(\phi)}{\cos(\phi_0)}\right)\sqrt{2\rho(p_a - p_m)} \tag{2}$$

where c_d is the discharge coefficient; ρ denotes the air density; p_a is the pressure in the upstream of the throttle valve, which can be regarded as the atmospheric pressure; D is the diameter of the throttle plate; and ϕ and ϕ_0 denote the throttle opening angle and the rest angle of the throttle plate, respectively.

In addition, the air flow rate going out from manifold $\dot{m}_{out}$ essentially involves the cycle-to-cycle transient. It can be simplified by a mean-value representation [33]:

$$\dot{m}_{out} = \frac{\eta_v V_d}{4\pi R T_m} p_m \omega \tag{3}$$

where V_d is the displacement volume; ω is the engine speed in radians per second; and η_v denotes the volumetric efficiency.

Thus, substituting (2) and (3) into (1), the air charging dynamics can be finally reduced in a compact form:

$$\dot{p}_m = a\left(1 - \frac{\cos(\phi)}{\cos(\phi_0)}\right)\sqrt{p_a - p_m} - b p_m \omega \tag{4}$$

where

$$a = \frac{\pi c_d D^2 R T_m}{4V_m}\sqrt{2\rho}, \quad b = \frac{\eta_v V_d}{4\pi V_m}.$$

In fact, a and b can be approximated as constants in consideration of the slight variations of the intake air temperature and the volumetric efficiency.

To deduce the torque production model, assuming that the aspirated air mass in the cylinder is mixed with fuel in a certain air–fuel ratio, the air–fuel mixture flow rate can be written as [32]:

$$\dot{m}_f = \frac{\dot{m}_{out}}{\gamma\lambda} \tag{5}$$

where γ is the air fuel equivalence ratio (it usually equals 1); and λ is the air fuel ratio. The air–fuel mixture in the cylinder is burned, and the indicated power P_i is also determined by the lower heating value of the fuel H_l and the indicated efficiency η_i [32]:

$$P_i = H_l \eta_i \dot{m}_f \tag{6}$$

Note that the indicated power in the above equation is effective in the mean value sense because it ignores the intake-to-power time delay. Combining (3), (5) and (6), the mean indicated torque τ_i can be obtained:

$$\tau_i = \frac{P_i}{\omega} = \frac{H_l V_d \eta_v \eta_i}{4\pi R T_m \gamma\lambda} p_m \tag{7}$$

Thus, the output torque τ_e can be deduced by removing the