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Direct multivariable controller tuning for internal combustion engine test benches

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ABSTRACT

Dynamical test benches are typically used in the development phase of engine systems and require tracking controllers with a high performance. Unfortunately, during such a work the components or operation parameters of the engine system are changed very frequently, making the use of classical model based control approaches very time-consuming. Against this background, this paper proposes a direct data-driven design approach for multivariable control of rotational speed and shaft torque of an internal combustion engine at a test bench based on an extended version of a recently introduced method for non-iterative direct data-driven tuning of multivariable controllers. This extension allows employing data collected in a closed-loop experiment in the direct identification of the controller parameters. The effectiveness of the proposed approach is shown on a test bench equipped with a production light duty Diesel engine. A comparison with the industrial state-of-the-art controller is provided on both a dynamically challenging test and a typical driving cycle as measured on an instrumented vehicle with the same internal combustion engine. The results confirm that the new method recovers the performance of the well-tuned industrial control, but can be developed in a fraction of the time as no explicit model of the system is needed.

1. Introduction

The speed of a vehicle is the result of the engine torque – set by the driver via the accelerator pedal – and the load as a direct consequence of the road and vehicle conditions. Test benches are today often used to simulate the operation of an internal combustion engine in a vehicle. Both the torque and the speed at the crankshaft of the internal combustion engine need to track the values that the internal combustion engine would experience in the vehicle. The load torque has to be computed and enforced by the dynamometer at a test bench. If the torque profiles delivered by the dynamometer are satisfactory, measurements in a vehicle may be replaced by measurements at a test bench with significant advantages in terms of reproducibility, reduced time and costs when calibrating maps of an engine control unit or developing new engine control concepts.

The accuracy of the control significantly affects the validity of the measurements at a test bench. Therefore, the subject has received attention in different ways. A digital controller for a turbocharged Diesel engine as well as a direct current dynamometer is developed in Tuken, Fullmer, and VanGerpen (1990) using a closed-loop pole assignment technique to test an internal combustion engine with transient test cycles. The model reference adaptive control approach using Lyapunov stability theory to derive the parameters update law is applied to engine speed and torque control in Yanakiev (1998). Multivariable controls of the engine-dynamometer system have become increasingly popular in recent times. The closed-loop reference tracking is maximized and excessive loop interactions are avoided in Bunker, Franchek, and Thomason (1997) by balancing the bandwidths of the loop transfer functions. In Gruenbacher and del Re (2008) a robust inverse tracking method is applied to control an internal combustion engine test bench achieving a high tracking performance. The inverse optimal control problem is solved in Gruenbacher, Colaneri, and del Re (2008), and in Passenbrunner, Sassano, and del Re (2011) an approximation of the solution of the optimization problem is calculated for an internal combustion engine test bench.

However, all the approaches mentioned above are based on a mathematical model of the system which consists of the test bench mechanics and the dynamometer, but also of the engine system under test. As dynamical test benches are mostly used for the development of engine systems, their behavior will change frequently due to different calibrations or to the exchange of components. Against this background, the derivation of new

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models can be quite time-consuming and is frequently omitted, under the assumption that the changes will not affect too much the performance of the closed-loop system.

Looking for a more efficient solution, an alternative approach that skips the explicit model phase and directly tunes the parameters of the controller from data, following the recent ideas in data-driven controller tuning, see e.g. Bazanella (2012) and Formentin, Hirsch, Savaresi, and del Re (2012), is suggested. The strategy employed herein is based on the method introduced in Formentin, Savaresi, and del Re (2012) and Formentin and Savaresi (2011a) for dealing with multivariable coupled controller design using a collection of open-loop data, which has already been applied on another challenging engine application in Formentin, Hirsch, et al. (2012).

In order to cope with the practical requirements of this setup, in particular with the integral behavior between the torque and the speed, these methods must be extended to work also with some of the data collected during the closed-loop operation. This extension makes the procedure suitable also for unstable systems. Notice that such an approach is different from classical direct methods like Hjalmarsson, Gevers, Gunnarsson, and Lequin (1998), in that it is non-iterative, and from standard direct adaptive control, see e.g. Wellstead and Zarrop (1991), in that it is an off-line method which allows for a fixed parameterization of the controller.

Measurements for the identification of the controller parameters are recorded at an internal combustion engine test bench equipped with a production light duty 2.0ℓ Diesel engine. The results are compared with the well tuned industrial standard, i.e. two gain scheduled single-input single-output PI controllers. The scheduling of the controller parameters of the industrial standard is done on the basis of the operating point that is defined from the engine speed and the accelerator pedal position. An adaptation of the controller parameters, learning approaches or similar is not provided by the industrial standard, see PUMA Open - avl.com (2014) for more details.

Both controllers show a comparable performance in tracking the references for the shaft torque and the engine speed. Disturbances caused by the couplings in the system are suppressed better by the proposed approach. Furthermore, the proposed approach accounts for a reduced effort and time required for tuning of the controller with respect to the industrial standard. The tuning of the test bench controller gets significantly simplified. Note that a preliminary version of this work has been presented in Passenbrunner, Formentin, Savaresi, and del Re (2012).

The paper is organized as follows: Section 2 presents the system and provides, for better understanding, a brief mathematical description of the main dynamics. Section 3 recalls the basics of direct multivariable control design and extends the method for the case where some of the data are collected during closed-loop operation. The experimental design of a multivariable controller for an internal combustion engine test bench and a comparison with the industrial state-of-the-art system are finally illustrated in Section 4 and discussed in Section 5. The paper is concluded by some remarks in Section 6.

2. Internal combustion engine test benches

The typical setup of an internal combustion engine test bench is depicted in Fig. 1, see Gruenbacher and del Re (2008) for further details. The internal combustion engine under test is connected via a flexible shaft with the electric dynamometer. The accelerator pedal position α and the set value $T_{D,set}$ of the dynamometer torque provide the inputs of the test bench. The engine speed ω_E, the dynamometer speed ω_D, the dynamometer torque T_D and

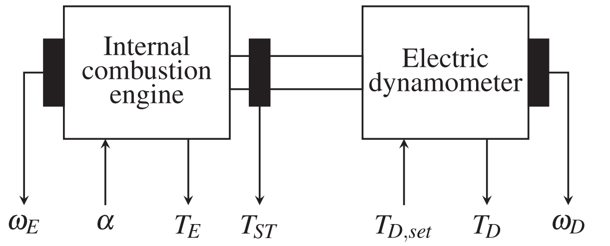

Fig. 1. Typical setup of an internal combustion engine test bench consisting of a dynamometer coupled to an internal combustion engine under test by a flexible shaft.

(sometimes) the shaft torque T_{ST} are measured. In case the torque signals are not measured, the shaft torque T_{ST} and the engine torque T_E can be estimated from the available measurements, see Ortner, Gruenbacher, and del Re (2008) and Passenbrunner, Trogmann, and del Re (2012).

The entire mechanical part of the test bench can be modeled as a two-mass-oscillator as follows:

$$\begin{aligned} \Delta\dot{\phi} &= \omega_E - \omega_D, \\ \dot{\omega}_E &= \frac{1}{J_E}(T_E - T_{ST}), \\ \dot{\omega}_D &= \frac{1}{J_D}(T_{ST} - T_D) \end{aligned} \tag{1}$$

with the shaft torque

$$T_{ST} = c\Delta\phi + d(\omega_E - \omega_D)$$

and $\Delta\phi$ being the torsion angle of the connecting shaft. J_E and J_D are the inertia of the internal combustion engine and of the dynamometer, respectively. The inertia of the adapter flanges, the damping element, the shaft torque measurement device and the flywheel are already included in these values. The constant c is the stiffness of the connecting shaft, d the damping of the connecting shaft.

In general, modeling the behavior of the internal combustion engine is rather complicated. However, a simplified torque model of a light duty Diesel engine can be described by

$$\dot{T}_E = -\tau(\omega_E, \alpha)T_E + \tau(\omega_E, \alpha)m(\omega_E, \alpha). \tag{2}$$

The varying parameter $\tau(\omega_E, \alpha)$ depends on the operating point defined by the accelerator pedal position α and the engine speed ω_E, whereas $m(\omega_E, \alpha)$ represents a nonlinear static map with respect to α and ω_E which is bounded from below and above. For accelerator pedal position α close to $\alpha = 0$, the drag torque of about -30 N m appears. The maximum engine torque T_E is limited to approximately 250 N m. Furthermore, for actual internal combustion engines, the static engine torque also depends on the parameterization of the engine control unit.

The dynamics of the employed electric dynamometer can be neglected as it is much faster than those of the other components. Within the range of maximum torque and maximum rate of change, the torque of the dynamometer can then be described as

$$T_D = T_{D,set}. \tag{3}$$

By exploiting the relationship $\Delta\omega = \omega_E - \omega_D$, the model (1)–(3) can be rewritten in its state-space representation as

$$\begin{aligned} \dot{x}_1 &= x_2, \\ \dot{x}_2 &= -c\frac{J_E + J_D}{J_E J_D}x_1 - d\frac{J_E + J_D}{J_E J_D}x_2 + \frac{1}{J_E}x_4 + \frac{1}{J_D}u_2, \\ \dot{x}_3 &= -\frac{c}{J_E}x_1 - \frac{d}{J_E}x_2 + \frac{1}{J_E}x_4, \\ \dot{x}_4 &= -\tau(x_3, u_1)T_E + \tau(x_3, u_1)m(x_3, u_1), \\ y_1 &= x_3, \end{aligned}$$