

Design and Validation of a Novel Model Reference Adaptive Algorithm to Control ETB for Drive-by-wire Applications

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ABSTRACT

In automotive industry the Electronic Throttle Body (ETB) plays a crucial role in drive-by-wire operations since it controls the incoming air into the engine and so the produced torque. This implies the performances of the vehicle in terms of traction, emissions, idle speed regime, cold starting management, thermal transient and smoother movement during tip/in tip/out, strongly depends on the precise control of this device [17]. Despite its apparent simplicity, the behavior of the ETB is affected by many nonlinearities and uncertain parameters which can dramatically alter its dynamics. In order to cope the unwanted nonlinear phenomena (stick-slip motion, hysteresis, hunting, impact, chaos), sophisticated model based control strategies and compensators are proposed in the literature. A time consuming identification parameters of the throttle is fundamental for these approaches and it is the main drawback for their application.

The aim of the paper is to show the efficiency of a model reference adaptive algorithm, named LQ-MCS (Linear Quadratic-Minimal Control Synthesis), to control the throttle plate position. The main feature of this controller is that minimal synthesis is needed to implement the strategy. Specifically only a rough nominal linear model of the



Figure 1: Picture of Bosch motorized throttle body (DV-E5)

plant is required to impose the dynamical behavior of the reference model. By means of a proper experimental set-up, the adaptive controller is synthesized and validated experimentally.

INTRODUCTION

Modern automotive systems increasingly need accurate control of injection timing and precise regulation of the air/fuel mixture fed to cylinders. This is fundamental for the design of vehicles with high performance in terms of traction, emissions, idle speed regime, cold starting management, thermal transient and smoother movement dur-

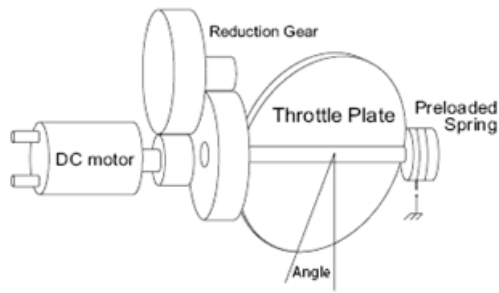


Figure 2: Schematic of the ETB

ing tip/in and tip/out manoeuvres [24]. The key idea is to affect directly the mixture formulation through appropriate electronic control of the fuel injection and the exact regulation of the air flow in the manifold (see for example [4]).

The ETB, shown in Fig. 1, is a mechatronic device dedicated to the regulation of the air mass flow rate (see Fig. 2 for a schematic of the system). In this system a shaped body duct regulates the relationship between the angular position of the throttle valve and the incoming air flow into the manifold. The desired plate position is imposed by a microcomputer in a drive-by-wire configuration. The reference signal is the solution of a trade off between the driver request (acceleration pedal position) and the effective traction possibilities depending upon driveability, safety and emission constraints. The control signal generated by the Electronic Control Unit (ECU) becomes, by means of a H-bridge power converter, the armature voltage of a DC-motor. The rotation motion is then transferred from the motor shaft to the plate shaft through a gear system. Despite its apparent simplicity, the behavior of the ETB is affected by many nonlinearities (see for example [23] and [5]) which can dramatically alter its dynamics. They can be briefly summarized as follows:

- *Friction.* Friction affects the transmission system and is mainly due to the low quality of the bearings. In mechanical systems the friction force is responsible for many unwanted nonlinear phenomena, such as stick-slip motion, hysteresis, hunting phenomena, chaos, etc.
- *Piece-wise linear restoring torque.* When a failure of the DC motor occurs, for safety reasons it is necessary to ensure that the valve comes back to a default position (called limp-home position). The limp-home angle corresponds to the supply of a small amount of air into the engine. This guarantees that the valve is not completely closed thus allowing the driver to "limp" until reaching the nearest car service [35]. To guarantee the limp-home, two additional springs, named return and default spring, are used. Since the stiffness of the default spring is always greater than the stiffness of the return spring, the resulting elastic torque is modeled as a piecewise linear function of the admissible angle.

- *Impacts.* Impacts occur when the plate hits the mechanical constraints, e.g. when the plate angle reaches the highest or lowest angle allowed.
- *Backlash.* This effect is due to a spacing between the gear teeth. As a consequence, extra dynamics can arise during the plate motion eventually including impacts between teeth.

Another important issue is the uncertainty on the system parameters values due to manufacturing tolerances, variable operating conditions or mechanical wears [24]. For example, the electrical lag due to the DC motor operation, could be five times greater than its nominal value. Also, the limp-home position is known at times with more of ten degrees of uncertainty and the friction torque is strongly dependent on temperature.

Notice that the necessity of minimizing the design and operation costs for mass production purposes makes the presence of nonlinearities and uncertainties unavoidable. At first sight, the control of the throttle body may appear as a simple position tracking control under a variable load torque. Unfortunately, the coupling with the engine, the presence of the above mentioned nonlinearities and uncertainties require to devise a much more sophisticated control strategy to ensure the best tradeoff between the controller complexity, technological feasibility and cost limitation.

To cope with the many sources of nonlinearity and uncertainties the latest literature reports several attempts to control the throttle body by means of variable-structure controllers (see e.g. [34], [19], [38], [6], [20]). Because of their robust nature, these controllers guarantee good performance in terms of tracking error and simplicity of the control structure [18]. However, the typical chattering phenomenon on the control variable restricts their implementation for mass production. This strong limitation implies that more classical approaches are still wide used in industry (see [21], [15], [8], [3]). These classical control schemes are often equipped with a nonlinear compensator [1]. Obviously, better performances can be achieved when both the controller and the compensator are tuned taking into account the mathematical model of the throttle valve and its parameters as well. In this case, a proper identification procedure becomes the key point to achieve good performances [37]. The intensive identification phase is the main drawback of the model based control strategies, not only because it is very time consuming but also because it should be performed periodically in order to take in account the variation of the parameters due to the wear of the mechanical device.

In order to deal with all the uncertainties and the nonlinear dynamics, it seems reasonable to implement a robust adaptive control strategy where the control gains are appropriately varied, or better adapted, according to the actual system behavior. The aim of this paper is to show the efficiency and minimal design effort of a novel approach based on a new adaptive strategy recently proposed.



Figure 3: Embedded resistive throttle plate position sensors

sented in [10]. The idea is to design and validate experimentally such a new strategy on a real ETB, confirming numerical results already presented in [9].

The novel strategy is based on a standard Minimal Control Synthesis (MCS) algorithm [25] where the model reference is the nominal linear model of the plant controlled by a classical LQ optimal strategy. The main idea behind the LQ-MCS approach is to seek a simple and alternative route to implement the LQ regulator via the MCS algorithm. In so doing, the derived controller has the benefits of the adaptive strategy (in terms of minimal knowledge of the plant, minimal design effort and robustness to uncertainties) while also matches the performance of the well known LQ regulator. Robustness is guaranteed by the MCS action which has been shown to be robust to nonlinear terms and slowly varying parameter variations especially for electro-mechanical systems like the throttle body [12].

Even though the performances numerically provided by the LQ-MCS strategy are excellent (see [10] and [9]) an experimental validation on industrial plant is now fundamental to enlarge the range of applications of MCS algorithm (see for example [2], [14], [28], [29], [30], [31], [32], [33], [36]).

As it will be shown in the rest of the paper, we found this novel approach particularly feasible to the ETB. Experimental evidence presented below clearly shows that the novel scheme can guarantee excellent tracking performance and transient behavior. All experiments are performed by using the reliable experimental setup based on a Rapid Control Prototyping (RCP) system.

The rest of the paper is outlined as follows. In section “mathematical model” a nonlinear model of the plant is given and validated. The incapability of classical control strategies with fixed gains to control the plate position is shown in section “Experimental results using a PI augmented with a model based compensator”. In section “LQ-MCS design” details on the design of the control strategy are provided, whereas the effectiveness of this novel adaptive controller is widely shown in section “Ex-

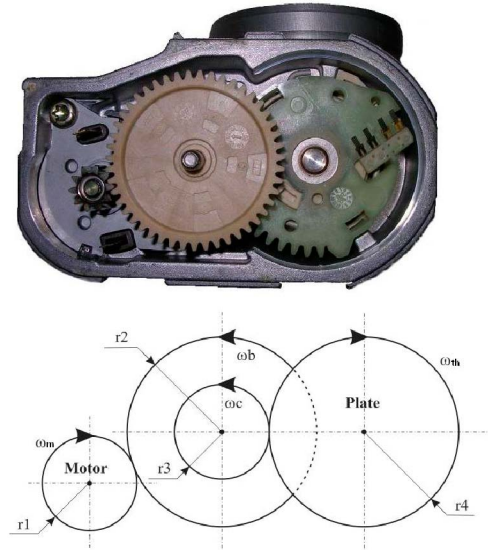


Figure 4: Gear box and its schematization

perimental results”. In appendixes the experimental setup and the theoretical background of the LQ-MCS algorithm are reported.

MATHEMATICAL MODEL

The ETB plant (shown in Fig. 1) mainly consists of a DC motor, a reduction gear and a plate where the springs necessary to lead the plate in the limp-home position are also located. A couple of potentiometers are used to measure the angular plate position (see Fig. 3).

The mathematical model of the plant can be constructed starting from the simple models that describe each part of the system (see Fig. 2) and then considering the interaction between them. Starting with the DC motor dynamics, we have:

$$\begin{cases} \frac{di}{dt} = \frac{1}{L}v_a - \frac{R}{L}i - \frac{K_v}{L}\omega_m, \\ \frac{1}{K}\frac{d\omega_m}{dt} = \frac{1}{J_m}T_m - \frac{\beta_m}{J_m}\omega_m - \frac{1}{J_m}T_{mp}, \\ T_m = K_t i. \end{cases} \quad (1)$$

where L [H] is the equivalent inductance of the armature coil; R [Ω] is the equivalent resistance of the armature coil; K_v [V·s/rad] is the velocity constant determined by the flux of the permanent magnets; v_a [V] is the voltage source across the coil of the armature; T_{mp} [N·m] is the torque of the mechanical load (the plate in our case); T_m [N·m] is the electromagnetic torque; J_m [Kg·m²] is the moment of inertia of the rotor; β_m [N·m·s/rad] is the damping coefficient; K_t [N·m/A] is the torque constant; ω_m [deg/s] is the angular velocity of the motor; i [A] is the current of the armature and $K = 180/\pi$ [deg/rad] is a dimensional constant to convert radians to degrees.

Moving to the gears, usually authors model them as a memoryless system placed between the actuator and the input of the DC motor, or, in other words, as a dead zone.

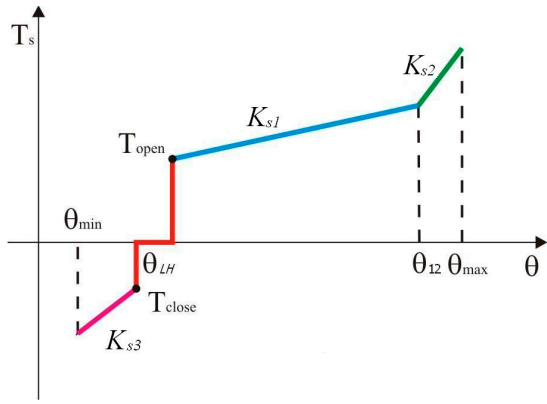


Figure 5: Piecewise linear restoring torque characteristic

with “output backlash structure” [20], [19], [16]. In the ETB context, often gears are modeled as a gain which relates the angular velocities of the motor and the plate shafts. Following this approach, the detachments between the teeth (and the following chattering sequence) cannot occur during the transmission of the motion from the driving gear to the driven gear. In our case of study the reduction gearbox (schematized as in Fig. 4) is composed by a set of four interlocking wheels. The first one is applied to the motor shaft whereas the last one is applied to the plate shaft. The intermediate gears allow to achieve a specified reduction rate. Assuming that the linear peripheral velocity (calculated by the radius of the pitch circles) is the same for each wheel, the relationship between the angular velocity of the first wheel (ω_m) and the last one (ω_{th}) is given by

$$G_r = \frac{\omega_m}{\omega_{th}} = \frac{r_2 r_4}{r_1 r_3} = \frac{z_2 z_4}{z_1 z_3}, \quad (2)$$

where G_r is the gears ratio; r_i and z_i $i \in \{1, 2, 3, 4\}$ are respectively the radius of the circular pitch and the number of teeth of the i -th wheel.

The plate motion can be derived by applying Newton's law. From the balance of the rotational torques applied to the system, it follows that

$$\frac{1}{K} \frac{d\omega_{th}}{dt} = -\frac{T_s(\theta)}{J_{th}} - \frac{T_{f_{th}}(\omega_{th})}{J_{th}} + \frac{T_{pm}}{J_{th}}, \quad (3)$$

where θ [deg] is the plate angular position ($\theta \in [\theta_{min}, \theta_{max}]$ for mechanical constraints); ω_{th} [deg/s] is the plate angular velocity; J_{th} [Kg·m²] is the moment of inertia of the plate; $T_{f_{th}}$ [N·m] is the friction torque; T_{pm} [N·m] is the torque provided by the DC motor on the plate shaft; T_s [N·m] is the torque due to the presence of the springs which gives the restoring torque.

Taking into account Eqs. (1), (2), (3) (see Fig. 2), the overall model can be then derived as:

$$\begin{cases} \frac{di}{dt} = -\frac{R}{L}i - \frac{K_v}{L} \frac{G_r}{K} \omega_{th} + \frac{1}{L} v_a \\ \frac{1}{K} \frac{d\omega_{th}}{dt} = \frac{K_t}{J} G_r i - \frac{T_s(\theta)}{J} - \frac{T_f(\omega_{th})}{J} \\ \frac{d\theta}{dt} = \omega_{th}, \end{cases} \quad (4)$$

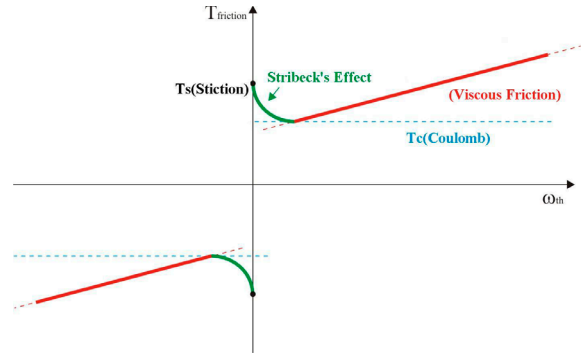


Figure 6: Friction torque characteristic

where $J = J_{th} + J_m G_r^2$ is the equivalent moment of inertia and $T_f(\omega_{th}) = T_{f_{th}}(\omega_{th}) + G_r^2 \beta_m \omega_{th} / K$ represents all friction torques.

The analytical expression of the restoring torque characteristic is derived from the direct observation of the experiments (see [11]) and it is a PWL (PieceWise Linear) function (see Fig. 5) given by:

$$T_s(\theta) = \begin{cases} T_{S3}(\theta) & \text{if } \theta \in [\theta_{min}; \theta_{LH} - \frac{\Delta\theta}{2}] \\ 0 & \text{if } \theta \in [\theta_{LH} - \frac{\Delta\theta}{2}; \theta_{LH} + \frac{\Delta\theta}{2}] \\ T_{S1}(\theta) & \text{if } \theta \in [\theta_{LH} + \frac{\Delta\theta}{2}; \theta_{12}] \\ T_{S2}(\theta) & \text{if } \theta \in [\theta_{12}; \theta_{max}] \end{cases} \quad (5)$$

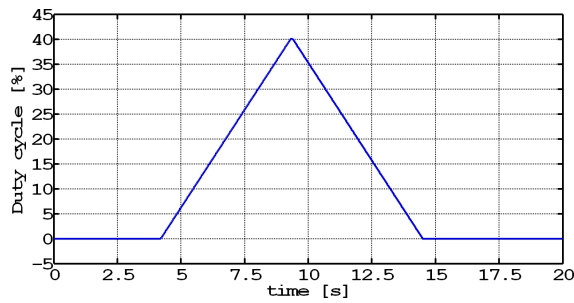
$$T_{S1}(\theta) = K_{s1} \left[\theta - \left(\theta_{LH} + \frac{\Delta\theta}{2} \right) \right] + T_{open}$$

$$T_{S2}(\theta) = K_{s2}(\theta - \theta_{12}) + T_{S1}(\theta_{12})$$

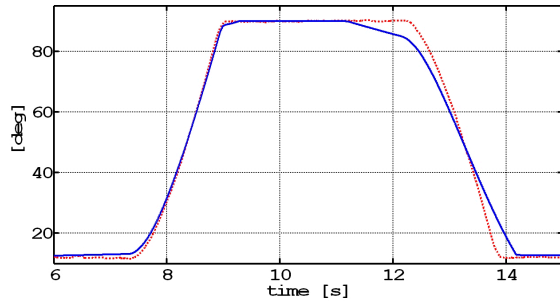
$$T_{S3}(\theta) = -K_{s3} \left[\left(\theta_{LH} - \frac{\Delta\theta}{2} \right) - \theta \right] - T_{close}$$

where K_{s1} , K_{s2} , K_{s3} [N·m/deg] are the stiffness coefficients in each region of interest; T_{close} [N·m] is the minimum torque necessary to close the valve; T_{open} [N·m] is the minimum torque necessary to start opening the valve; θ_{LH} [deg] is the limp-home angular position; $\Delta\theta$ [deg] angle is the mechanical play of the plate at limp-home position due to clearance between the teeth of the gear; θ_{12} [deg] angle is the discontinuity point of the slope of the elastic torque when the plate is nearly WOT (Wide Open Throttle) position.

Friction torque, due to the low quality of the gearbox and bearings, plays an important role since it is the source of self-sustained oscillations that often have undesired effects. In the literature on ETB control, authors refer to different modeling approaches. For example, in [15] or [38], friction is merely modeled as a continuous viscous torque. A classical Coulomb model is used in [19], [20], [6], whereas a dynamical LuGre model is used in [19], [7]. Hybrid models are instead used in [21], [8], [22] where commutations between a dynamic Dahl model and a generalized Stribeck static model are explicitly taken into account. This more complex approach is aimed at overcoming the obvious limitation of the LuGre model, which is not



(a)



(b)

Figure 7: Model validation: (a) time history of the duty cycle (b) comparison between the experimental data (red dotted line) and model prediction (blue solid line)

able to predict the experimental ETB behavior characterized by the drop of the breakaway torque when the rate of the applied DC motor torque increases. Other authors (see for example [35], [34]) adopt different phenomenological hybrid models to get a simple PWA (Piecewise Affine) friction characteristic.

To conjugate the simplicity of the model with its accuracy, in what follows the model of the friction torque is based on a static Coulomb model modified in order to include the Stribeck effect as (see Fig. 6)

$$T_f(\omega_{th}) = \left[T_c + (T_{st} - T_c) e^{-\left| \frac{\omega_{th}}{\omega_s} \right|} \right] \text{sign}(\omega_{th}) + \beta \frac{\omega_{th}}{K}, \quad (7)$$

where T_c [N·m] is the Coulomb friction torque; T_{st} [Nm] is the stiction friction torque; ω_s [deg/s] is the Stribeck velocity $\beta = G_r^2 \beta_m$ [N·m·s/rad] is the damping coefficient.

EXPERIMENTAL MODEL VALIDATION In order to validate the nonlinear complex ETB model 4 a proper parameters identification procedure has been performed. It is important to highlight that from the control viewpoint this expensive stage of the analysis can be skipped (or minimized) when robust adaptive control strategies, like LQ-MCS algorithm, are used. We have performed it only to confirm that the nonlinear discontinuous model 4 is able to capture the complex dynamics of the plant.

Details on the identification procedures and the related experimental activity are reported in [11]. Here, we just highlight that the parametric identification of model (4) has

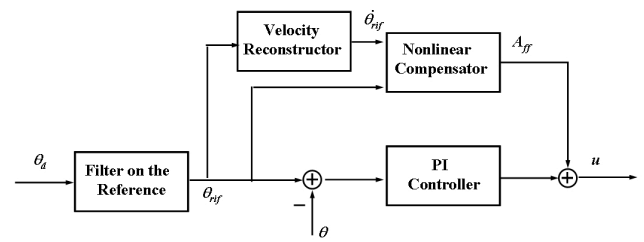


Figure 8: PI control scheme

been performed by following different approaches. In particular the parameters R , L , K_v , K_t have been identified according to the procedures in the work of Pavkovic et al. [22]. Values of the parameters J , β , T_c , ω_s have been estimated applying an optimization technique based on a least-square algorithm. All the remaining parameters have been derived by the knowledge of the experimental set-up and on the basis of straight-forward geometrical considerations. The value of the parameters identified are reported in appendix in Tab. 1.

The effectiveness of the identification procedure is shown from the validation results in Fig. 7.b, where it is evident that the synthesized model is able to capture the system behavior not only in steady-state conditions, but also during the valve opening and closing.

EXPERIMENTAL RESULTS USING A PI AUGMENTED WITH A MODEL BASED COMPENSATOR

In this section we experimentally investigate the performance achievable via a properly tuned PI controller equipped with a non linear compensator and a filter on the reference signal to limit the tracking error during rapid

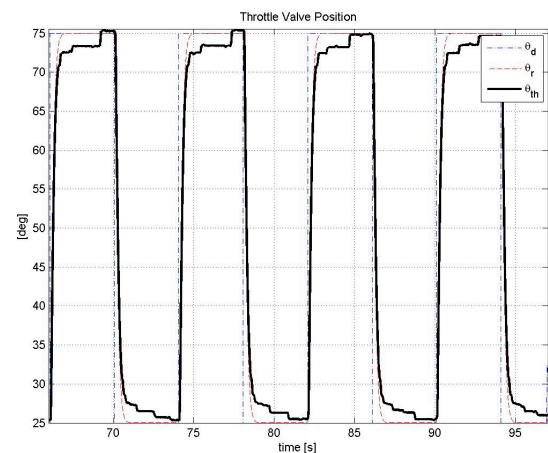


Figure 9: Experimental results, PI with a model based feed-forward action. θ_d reference plate position (dashdot blue); θ_r filtered reference signal to be tracked (dashed red); θ_{th} angular plate position (solid black), when the reference is a square wave with period 5 [s] and amplitude 50 [deg]

tip-in/tip-out maneuvers. The compensator is based on the identified model previously presented. Further details on the control design can be found in [9].

Even though the compensator attempts to contrast the nonlinear torques 5 and 7, experiments confirm the lack of robustness and the poor tracking performances of the PI strategy. In fact, figures 9, 10 clearly show the classical unwanted stick and slip motion [3] due to the friction torque, while figure 11 shows the incapability of the classical controller to keep the throttle position at a constant position. In fact in this case the unwanted hunting phenomenon [13] affect the closed loop dynamics.

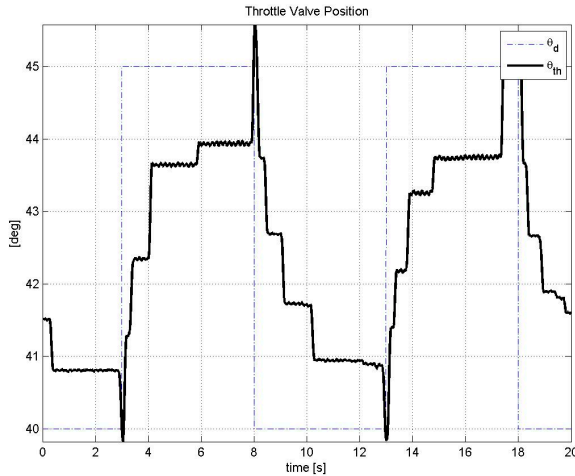


Figure 10: Experimental results, PI with a model based feed-forward action. θ_d reference plate position (dashdot blue); θ_{th} angular plate position (solid black), when the reference is a square wave with period 10 [s] and amplitude 5 [deg]

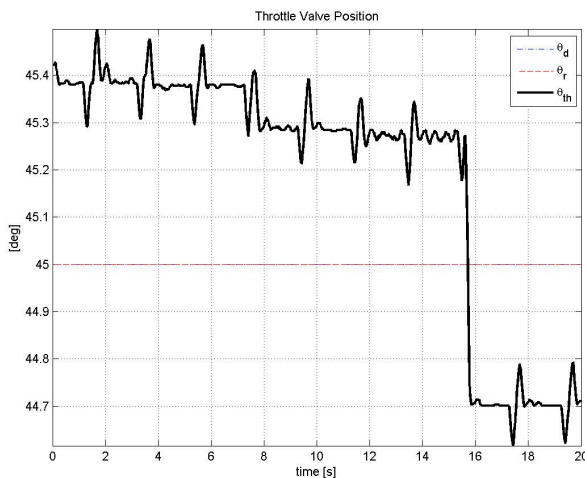


Figure 11: Experimental results, PI with a model based feed-forward action. θ_d reference plate position (dashdot blue); θ_r filtered reference signal to be tracked (dashed red); θ_{th} angular plate position (solid black), when the reference is constant at 45 [deg]

LQ-MCS DESIGN

In order to design the control strategy 14 (described in appendix), it is necessary to have a linear model of the plant. Neglecting the electrical dynamics, we adopt as nominal linear model a second order LTI system with state variable $x = [x_1 \ x_2]^T = [\theta \ \omega_{th}]^T$.

Generally speaking a nominal second order LTI system can be characterized by means of the following matrices:

$$A_0 = \begin{bmatrix} 0 & 1 \\ a_1 & a_2 \end{bmatrix}, \quad B_0 = \begin{bmatrix} 0 \\ b \end{bmatrix}. \quad (8)$$

We note explicitly that the parameters of the matrices 8 can be obtained using: 1) a black box approach by means of a rough and fast identification activity; 2) the ETB model Eq. 4, if available, neglecting current dynamics and all nonlinear torque terms.

Since the nonlinear model is available it is convenient to use the second approach. In particular the linear part of the complete model is found assuming:

1. the plate position usually works in the range $\left[\theta_{LH} + \frac{\Delta\theta}{2}, \theta_{12}\right]$, thus the expression of the elastic force in Eq. (5) simply reduces to:

$$T_{sp}(\theta) = K_{s1}\theta$$

2. the friction action, whose mathematical expression is in eq. (7), is approximated just by the term $\beta\omega_{th}$.

In so doing the matrices of the nominal LTI model (19) are given by:

$$A_0 = \begin{bmatrix} 0 & 1 \\ -K \frac{K_{s1}}{J} & -\frac{\beta}{J} \end{bmatrix}, \quad B_0 = \begin{bmatrix} 0 \\ G_r K \frac{K_t}{J} \end{bmatrix} \quad (9)$$

The weight matrices of the LQ control (20) have been chosen heuristically as a trade off between the settling time of the closed loop system and the amount of the energy of the control action. Choosing:

$$Q = \begin{bmatrix} 500 & 0 \\ 0 & 1 \end{bmatrix}, \quad R = 30, \quad (10)$$

the actual settling time is approximately 200 [ms] in according with the performances required to industrial ETB systems [24].

As it usually happens when implementing the MCS strategy, the scalar quantities α and β (Eqs. (15a) and (15c) in appendix), which scale the adaptive gains in [26], have been chosen heuristically as a trade off between convergence time and reactivity of the control action as $\alpha = 0.005$ and $\beta = 0.1\alpha$.