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Idle Mode Control on a Combustion Engine Test Bench via Internal Model Control

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Abstract— **In this paper we present an adaptive output regulation approach for idle mode simulation on combustion engine test benches. The combustion oscillations are modeled applying a parameter dependent exosystem whose parameter is related to the mean value engine speed which is a partly unknown quantity. Due to this reason we adapt existing literature on redundant adaptive internal model design and reduced order internal model design in order to achieve a fast converging output regulator. Furthermore we present the idea of an non-adaptive internal model based regulator in the case when the combustion oscillations are modeled by a reduced order nonlinear exosystem. At the end the methods are compared to a standard H_∞ feedback controller and the advantages of the presented approaches are discussed.**

I. INTRODUCTION

Idle mode control on a combustion engine test bench means that the engine is - like the engine idling in a car - in idle mode which means that the combustion engine is declutched from the power train and hence its only load is the drag torque, which include the friction torque of the combustion engine, any additional requested torque (*e.g.* from the air condition or power steering or any other accessories), but not by the power train (see *e.g.* [1]). A typical combustion engine test bench usually consists of the combustion engine and a dynamometer, which is linked to the combustion engine by a stiff shaft (see [2], [3], [4]). The problem of idle mode control on a combustion engine test bench is to control the dynamometer in such a way that the shaft torque, the load torque of the combustion engine, is regulated to zero. This means that the additional inertia and friction caused by the test cell (*e.g.* dynamometer, connection shaft and adapter flanges) has to be compensated by applying a sufficient dynamometer torque. If this is well done, internal control loops of the combustion engine, which are active in this operating mode, can be tuned or parameterized already in the pre-development using the test bench without changing the hardware set up. One of these control loops is the idle speed control loop (see *e.g.* [5] and the references therein). The validation of the idle speed controller is often done by set point tracking of the desired idle speed and by disturbance rejection of an additional torque (*e.g.* due to power steering) (see also [6]). To this end it is necessary to simulate the declutched power train even under changing conditions.

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The challenging task of idle mode simulation is the suppression of the torque ripples caused by the combustion oscillations which can only be compensated whenever the controller ensures sufficient high differential property which means a high gain at high frequencies. In practice this states an real disadvantage and it will mean that the requested power for compensating the torque ripples will be quite huge. Due to the robustness property of the internal model as it will be shown in the paper this can be improved significantly.

In the past the problem of output regulation has been extensively studied for linear systems starting with the work of [7] and [8] and for nonlinear systems beginning with [9]. For the present problem we adapt existing literature (see [10], [11] and [12]) for compensating the torque ripples. The frequency of this periodic disturbance, as the torque ripples are interpreted, is related to the mean value engine speed, which cannot be measured in general. Filtering using *e.g.* crank angle based techniques (see [13]) would be a possible solution but it would also increase the complexity of the closed loop and it would be difficult to prove stability and convergence. For this reason an adaptive output regulation approach is the better choice.

The paper is organized as follows. In the following section the system is explained and the regulation problem is formulated. Afterwards we discuss three different versions of the output regulation applying the redundant adaptive internal model design, the reduced order adaptive internal model design and a non-adaptive internal model base regulator in which a reduced order but nonlinear exosystem is applied. After this section the adaptive control approaches are applied in a very precise simulation. The validation of this simulation model was done in three steps. In the first step the mechanical system was evaluated (see [14]), then the mean value engine model was developed and validated too (see [4]). Finally a combustion oscillation simulation was added [15]. At the end the conclusion and an outlook finalize the work.

II. COMBUSTION ENGINE TEST BENCH SYSTEM

A typical engine test bench, as shown in Figure 1 consists of two main power units – the dynamometer and the combustion engine – which are connected via a shaft consisting of the adapter flanges, the real connection shaft and the shaft torque measurement device.

The physical system is modeled as a two mass oscillator. The first mass is described by the combustion engine and the second by the dynamometer. The inertias of the shaft and the flanges are added either to the inertia of the dynamometer θ_D or the inertia of the combustion engine θ_E. Hence the

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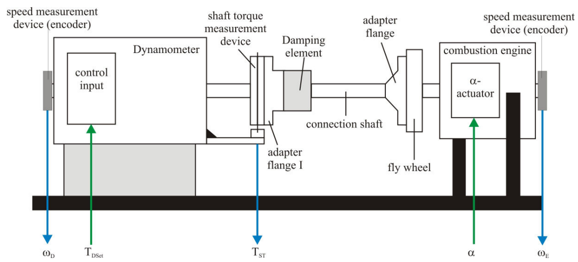

Fig. 1. Structure of the engine torque estimation

full physical model can be described by

$$\begin{aligned}\dot{\varphi} &= \omega_E - \omega_D \\ \dot{\omega}_E &= \frac{1}{\theta_E}\left(T_E - c\varphi - d\left(\omega_E - \omega_D\right)\right) \\ \dot{\omega}_D &= \frac{1}{\theta_D}\left(c\varphi + d\left(\omega_E - \omega_D\right) - T_D\right)\end{aligned} \quad (1)$$

where c is the stiffness of the connection shaft and d the damping constant (both parameters can be identified as shown in [14]). The state of the system (1) is given by the torsion φ, the engine speed ω_E and the dynamometer speed ω_D. The outputs of the system are the engine speed ω_E and the shaft torque

$$T_{ST} = c\varphi + d\left(\omega_E - \omega_D\right) \quad (2)$$

The input to the system thus are the dynamometer torque T_D and the combustion engine torque T_E. It should be mentioned that the fast dynamics of the dynamometer can be neglected so that it is possible to consider the dynamometer torque T_D as an input to the system. The second input, which from a control viewpoint is a disturbance input, is the combustion engine torque which can be modeled by a parameter varying exosystem.

A. Modeling the combustion engine torque via parameter varying exosystem

Analyzing the engine torque signal by using the power spectral density function - see Figure 2 for an operating point at $\bar{\omega}_E = 1000rpm$ and α (position of the accelerator pedal) equal to 0% (close to idle mode) - one can detect three main frequencies which can be interpreted as the harmonics of the torque ripples in the engine torque signal. All the other visible frequencies are neglected at this time since it is difficult to see and to explain the relationship to the combustion oscillations.

Due to this in the following the torque ripples will be described by linear but frequency depending harmonic oscillators

$$\begin{aligned}\dot{w}_i &= S_i\left(\beta\left(t\right)\right)\omega_i \\ T_{Ei} &= \Gamma_{Si} w_i\end{aligned} \quad (3)$$

where $w_i \in I\!R^2$ is the state, T_{Ei} the output of the i^{th} oscillator and

$$S_i\left(\beta\right) = \begin{pmatrix} 0 & -i\beta\left(t\right) \\ i\beta\left(t\right) & 0 \end{pmatrix} \quad \forall i = 1..3 \quad (4)$$

$$\Gamma_{Si} = \begin{pmatrix} 1 & 0 \end{pmatrix} \quad (5)$$

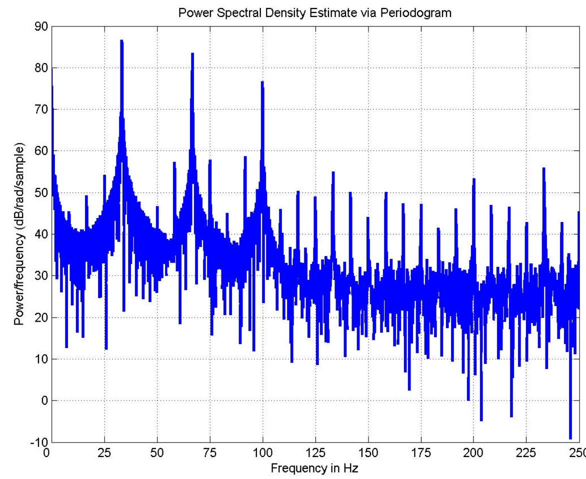

Fig. 2. Power spectral density plot for the engine torque at $\bar{\omega}_E = 1000rpm$, α (position of the accelerator pedal)$= 0\%$

are the dynamic matrix and the output map respectively. The amplitude and the phase shift of system (4) are defined by the initial state of the of the i^{th} harmonic. In (3) the parameter $\beta\left(t\right)$ defines the frequency of the first harmonic of the combustion oscillations and it is directly related to the mean value engine speed $\bar{\omega}_E(t)$ (mean value over one full combustion cycle). It further depends on the number of cylinders $\#Cyl$ and the number of strokes v, so that

$$\beta(t) = \frac{\#Cyl}{v}\bar{\omega}_E(t). \quad (6)$$

For the mean value engine torque a simple integrator

$$\dot{w}_0 = 0, \quad T_{E0} = \omega_0 \quad (7)$$

can be assumed. Hence the overall system for describing the engine torque (mean value plus all the harmonics) of the combustion oscillations with $w = \begin{pmatrix} w_0 & w_{11} & w_{12} & w_{21} & w_{22} & w_{32} & w_{32} \end{pmatrix}'$ becomes

$$\begin{aligned}\dot{w} &= S\left(\beta\left(t\right)\right)w \\ T_E &= \Gamma w\end{aligned}$$

with

$$S\left(\beta\left(t\right)\right) = \begin{pmatrix} 0 & 0 & 0 & 0 \\ 0 & S_1\left(\beta\left(t\right)\right) & 0 & 0 \\ 0 & 0 & S_2\left(\beta\left(t\right)\right) & 0 \\ 0 & 0 & 0 & S_3\left(\beta\left(t\right)\right) \end{pmatrix}$$

and

$$\Gamma = \begin{pmatrix} 1 & 1 & 0 & 1 & 0 & 1 & 0 \end{pmatrix}$$

For steady state condition an identification of the amplitudes of the harmonics is possible. Figure 3 shows the validation of the exosystem in one single operating point by applying real measurements.

Assumption 1: In the following we assume a constant speed, hence we assume that $\dot{\beta}(t) = 0$.

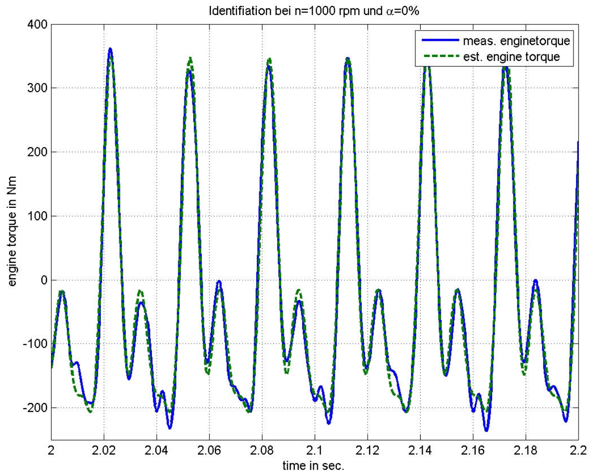

Fig. 3. Validation of engine model at $\bar{\omega}_E = 1000rpm$, α (position of the accelerator pedal)$= 0\%$

III. Formulation of the regulation problem

For the formulation of the regulation problem the physical model of the combustion engine test bench is considered (see (1)). In the present application the control output is the shaft torque (2), the control input is the dynamometer torque and the disturbance input is the combustion engine torque. Since speed is no longer a considered system output, the minimal realization of the system is

$$\begin{aligned} \dot{\varphi} &= \omega_r \\ \dot{\omega}_r &= -c\left(\frac{1}{\theta_E}+\frac{1}{\theta_D}\right)\varphi - \\ &\quad -d\left(\frac{1}{\theta_E}+\frac{1}{\theta_D}\right)\omega_r + \frac{1}{\theta_D}T_D + \frac{1}{\theta_E}T_E \\ T_{ST} &= c\varphi + d\omega_r \end{aligned}$$

where $\omega_r = w_E - \omega_D$. The problem at hand can be now formulated as an adaptive output regulation problem and, specifically, as the problem of stabilizing the origin of the previous system while compensating the effect of the disturbance $\frac{\theta_D}{\theta_E}T_E$, given by the sum of three periodic signals of uncertain amplitudes, phases and frequencies. Note that the "regulation error", given by T_{ST}, is vanishing at the origin. In the following we shall present how the existing literature on adaptive output regulation can be successfully used to solve the problem. For notational convenience we set $x = \begin{pmatrix}\varphi & \omega_r\end{pmatrix}'$ we shall rewrite the overall system as

$$\begin{aligned} \dot{w}' &= S(\beta)w' \\ \dot{x} &= Ax + B(T_D - \Gamma w') \end{aligned}$$

where $w' = \frac{\theta_D}{\theta_E}w$, $\frac{\theta_D}{\theta_E}T_E = -\Gamma w'$, and where (A, B) and Γ can be easily obtained from the above expressions.

A. Design of a Redundant Adaptive Internal Model

In this part we specialize the theory of adaptive output regulation, originally proposed in [10], to the problem at hand. By following the main lines of [10] we focus on the regulator of the form

$$\begin{aligned} \dot{\xi} &= F\xi + GT_D + Nx \qquad \xi \in I\!R^7 \\ T_D &= \hat{\psi}\xi + v_{st} \end{aligned} \tag{8}$$

in which (F, G) is a controllable pair with F Hurwitz, N is a 7×2 matrix to be chosen, and v_{st} is a residual control input which will be designed later. The vector $\hat{\Psi}$, of dimension 1×7, contains further regulator state variables which have to be adapted so that the proposed regulator structure has desired asymptotic properties. The regulator structure is thus completed by an adaptation law for $\hat{\Psi}$ of the form

$$\dot{\hat{\Psi}} = v_{ad}$$

where v_{ad} will be chosen later.

Ideally, the vector $\hat{\Psi}$ should be chosen equal to the vector $\Psi := \Gamma T^{-1}$ where $T \in I\!R^7 \times I\!R^7$ is the nonsingular solution of the Sylvester equation

$$TS(\beta) - FT = G\Gamma \tag{9}$$

which always exists as the spectrum of F and $S(\beta)$ are disjoint (see [10]). As a matter of fact, it is easy to realize that the matrices $S(\beta)$ and $F + G\Psi$ are similar under T and the regulator (8), if $\hat{\Psi}$ were chosen equal to Ψ and it were initialized at $\xi(0) = Tw'(0)$, is able to reproduce the ideal steady state control input $\Gamma w'(t)$ needed to compensate for the engine torque disturbance T_E. This is what, in the terminology recently proposed in [16], has been referred to as "internal model property". Unfortunately the ideal choice $\hat{\Psi} = \Psi$ can not be implemented as Ψ depends, via T, on the uncertain frequency β. Hence an adaptive law for $\hat{\Psi}$ will be sought.

In the forthcoming analysis we choose the pair (F, G) in the canonical form

$$F = \left(\begin{array}{c|c} \mathbf{0} & I_6 \\ \hline \multicolumn{2}{c}{\mathrm{f}} \end{array}\right) \qquad G = \begin{pmatrix} \mathbf{0} \\ 1 \end{pmatrix} \tag{10}$$

in which I_6 denotes the identity matrix of dimension 6 and $f = \begin{pmatrix} f_1 & \dots & f_7 \end{pmatrix}$ contains the coefficients of an Hurwitz polynomial. Since $(F + G\Psi)$ and $S(\beta)$ are similar and the latter has an eigenvalue at the origin, it is easy to realize that the scalar ΨG is independent of β. This fact will turn out useful later on.

Following [10], consider the preliminary change of variables $\tilde{\xi} = \xi - Tw'$ and note that the regulator dynamics (8) in the new coordinates read as

$$\dot{\tilde{\xi}} = (F + G\Psi)\tilde{\xi} + Gv_{st} + G\tilde{\Psi}\xi + Nx$$

where $\tilde{\Psi} = \hat{\Psi} - \Psi$. Furthermore, by bearing in mind that $\Psi T = \Gamma$, the system dynamics transform as

$$\dot{x} = Ax + B(\Psi\tilde{\xi} + v_{st} + \tilde{\Psi}\xi)$$

We consider now the additional change of variable, meant to eliminate the control variable v_{st} from the $\tilde{\xi}$ dynamics, given by

$$\tilde{\xi} \mapsto \chi := \tilde{\xi} - \theta_D^2 GB^T x$$

which, by choosing the degree-of-freedom N in the regulator (8) as

$$N = \theta_D^2 (GB^T A - FGB^T) \quad (11)$$

transforms the overall $(x, \tilde{\xi}, \tilde{\Psi})$ dynamics as

$$\begin{array}{rcl} \dot{\chi} & = & F\chi \\ \dot{x} & = & Jx + B\Psi\chi + Bv_{st} + B\tilde{\Psi}\xi \\ \dot{\tilde{\Psi}} & = & \dot{\hat{\Psi}} = v_{ad} \end{array}$$

where $J := A + \theta_D^2 B\Psi G B^T$. This system has a cascade structure with the driving system $\dot{\chi} = F\chi$ which is Hurwitz. Furthermore observe that, since ΨG is β-independent, the state matrix J is not dependent on the uncertain frequency β. By bearing in mind the previous facts, the regulator design can be completed as follows. Let K be such that $J + BK$ is Hurwitz and denote by P the positive definite matrix solution of the Lyapunov equation $P(J+BK)+(J+BK)^T P = -I$. As J is not dependent on β, P is a known matrix. Choose

$$v_{st} = Kx \qquad v_{ad} = -\mathrm{dzn}_\ell(\hat{\Psi}) - \gamma x^T PB\xi^T \quad (12)$$

where γ is an arbitrary positive and $\mathrm{dzn}_\ell(\cdot)$ is a "dead-zone" vector function[1] defined as $[\mathrm{dzn(s)}]_i = s_i - \ell \mathrm{sgn} s_i$ if $|s_i| \geq \ell$ and $[\mathrm{dzn(s)}]_i = 0$ otherwise, were ℓ is any positive number such that $\ell \geq \max \Psi_i$. Note that, since Ψ depend on β, the tuning of ℓ requires the knowledge of an upper bound on the uncertain frequency. By considering the candidate Lyapunov function $V(\chi, x, \tilde{\Psi}) = \kappa \chi^T P_F \chi + x^T P x + \frac{1}{\gamma} \tilde{\Psi} \tilde{\Psi}^T$ where $P_F = P_F^T > 0$ is such that $P_F F + F^T P_F = -I$ and $\kappa > 0$ is a sufficiently large positive number, standard completion of squares arguments lead to the following bound on $\dot{V}$

$$\dot{V} \leq -\frac{1}{2} x^T x - q\chi^T \chi$$

where q is a positive number. Then, by invoking standard LaSalle arguments, it is concluded that the overall closed-loop trajectories are bounded and attracted by the set $\{(\chi, x, \tilde{\psi}) : I\!R^7 \times I\!R^2 \times I\!R^7 \quad : \quad x = 0, \chi = 0\}$. The previous analysis proves that the regulator (8), (11), (12) solves that problem at hand, namely it is able to reject asymptotically the effect of the engine torque for any possible *constant* frequency β. According to the general theory presented in [10], the analysis above is not conclusive as far as the possible convergence of $\tilde{\Psi}$ to 0 is concerned. Additional persistence of excitation conditions should be assumed to formally prove asymptotic properties of $\tilde{\Psi}$.

B. Design of a Reduced-Order Adaptive Internal Model

The regulator designed in the previous section turns out to be redundant as it comes from an off-the-shelf application of the general theory in [10] without taking advantage from the fact that the three harmonics which characterize the uncertain exosystem have indeed multiple frequencies. It turns out that the order of the regulator, previously equal to 14, can be reduced to 10 by a mild adaptation of the analysis presented in the previous section. As a matter of fact note that the characteristic polynomial of the matrix $S(\beta)$ is given by

$$\det(sI - S(\beta)) = s^7 + 14\beta^2 s^5 + 49\beta^4 s^2 + 36\beta^6$$

Hence, given an arbitrary pair (F, G) of the form (10), it turns out that the β-dependent vector Ψ having the property that $(F + G\Psi)$ and $S(\beta)$ are similar is necessarily of the form

$$\Psi = \Psi_3(\beta) L - f$$

in which

$$\Psi_3(\beta) = \begin{pmatrix} -36\beta^6 & -49\beta^4 & -14\beta^2 \end{pmatrix},$$

L is a $I\!R^3 \times I\!R^7$ matrix with all zeros except the elements $(1, 2)$, $(2, 4)$ and $(3, 6)$ which are set equal to 1. According to the previous facts, all the arguments used in the previous section can be repeated with mild modifications by estimating not the entire 7-order vector Ψ but rather the 3-order vector Ψ_3. In particular the proposed regulator is of the form

$$\begin{array}{rcll} \dot{\xi} & = & F\xi + GT_D + Nx & \xi \in I\!R^7 \\ \dot{\hat{\Psi}}_3 & = & -\gamma x^T PB\xi^T L^T & \hat{\Psi}_3 \in I\!R^3 \\ T_D & = & \hat{\Psi}_3 L\xi - f\xi + Kx & \end{array} \quad (13)$$

in which K is such that $J + BK$ is Hurwitz, γ is an arbitrary positive design parameter and N is chosen as in (11). The arguments used in the previous analysis allow one to conclude that the previous controller guarantees that the overall closed-loop trajectories are bounded and that $\lim_{t\to\infty} T_{ST}(t) = 0$ as required by the regulation problem.

C. A non-adaptive internal model-based regulator

In this part we present the main steps which lead to design an internal model-based regulator which is not characterized by adaption laws like the ones presented in the previous two-subsections. The idea is to model the behavior of the engine torque as a linear oscillator with uncertain fundamental frequency with a nonlinear output function rather than as a linear exosystem composed by three oscillators (with uncertain multiple frequencies β, 2β, 3β) and linear output. Specifically, the idea is consider the matched disturbance $-\frac{\theta_D}{\theta_E} T_E$ in system (8) as generated by the output of the 4th order nonlinear exosystem $\dot{w} = s(w)$ given by

$$\begin{array}{rcl} \dot{w}_1 & = & 0 \\ \dot{w}_2 & = & 0 \\ \dot{w}_3 & = & w_2 w_4 \\ \dot{w}_4 & = & -w_2 w_3 \end{array} \qquad y_w = \kappa(w)$$

where $\kappa : I\!R^3 \to I\!R$ is a properly tuned nonlinear function. As before, the state of the exosystem lives in a compact invariant set $W \subset I\!R^4$. The specific form of the function $\kappa(\cdot)$ is under investigation and will be not presented in this paper.

Following the recent theory in [11] and [12] the idea is to choose a regulator of the form

$$\begin{array}{rcll} \dot{\xi} & = & F\xi + Gu + Nx & \xi \in I\!R^m \\ u & = & \gamma(\xi - \frac{1}{\theta_D} G\omega_r) + v_{st} & \end{array}$$

[1]The presence of the deadzone function does not play any role in the forthcoming stability analysis and only considered for practical numerical reasons in the simulations.

where m, the dimension of the regulator, is to be fixed, (F, G) is an arbitrary controllable pair with F Hurwitz, N is a matrix to be designed, and v_{st}, a residual control input, and $\gamma(\cdot)$, a nonlinear continuous function, are terms to be designed. By changing coordinates as

$$\xi \ \mapsto \ \chi := \xi - \frac{1}{\theta_D^2} G B^T x$$

and by choosing $N = \frac{1}{\theta_D^2}\left(GB^T A - FGB^T\right)$ it turns out that the closed-loop system has the form

$$\begin{array}{rcl} \dot{w} & = & s(w) \\ \dot{\chi} & = & F\chi + G\kappa(w) \\ \dot{x} & = & Ax + B(v_{st} + \gamma(\chi) - \kappa(w)) \end{array} \qquad (14)$$

namely it presents a cascade structure with the controlled plant driven by the signal $\gamma(\chi) - \kappa(w)$ which is generated by the autonomous system described by the first two dynamics in (14) (which, in turn, have a cascade structure as well). If v_{st} is chosen as $v_{st} = Kx$ with K designed so that $A + BK$ is Hurwitz, it turns out that the trajectories of the overall system are ultimately bounded (for any possible choice of m and F, with the latter Hurwitz). Furthermore, by following [11], the state (w, χ) reaches a "steady-state" set described by

$$\mathcal{A} = \text{graph}\ \tau|_W = \{(w, \chi) \in W \times I\!R^m \quad : \quad x = \tau(w)\}$$

in which

$$\tau(w) = \int_{-\infty}^{0} e^{-Ft} G\kappa(\Phi_s(t, w))dt$$

where $\Phi_s(t, w)$ denotes the solution of (14) at time t with initial condition w at time $t = 0$. Hence, the problem at hand is solved if $\gamma(\cdot)$ can be designed so that

$$\gamma \circ \tau(w) = \kappa(w) \qquad \forall\, w \in W\,.$$

As a matter of fact, under the previous circumstances, the controlled plant is a linear Hurwitz system driven by an asymptotically vanishing signal and the asymptotic regulation objective is met. The (positive) answer to the question if a γ with the above property exists has been given in [11] in which it has been proved that a continuous $\gamma(\cdot)$ always exists provided that $m \geq 2\text{dim}w + 2 = 10$. A possible explicit expression of the function in question is given by (see [12])

$$\gamma(\xi) = \inf_{w \in W} \{\kappa(w) + \rho(|\xi - \tau(w)|)\}$$

where ρ is an appropriate continuous function as described in [12]. From a practical viewpoint, the previous expression of γ can be approximated as proposed in [12] in a context of uniform practical output regulation. We briefly summarize in the following the main steps of the algorithm by referring the reader to [12] for details.

Step 1: Select a set of points $\{w_i,\, i \in I\}$ of W that cover the latter in a sufficiently "dense" way;

Step 2: Compute a set of points $\{\tau_i,\, i \in I\}$ by numerically integrating the second equation of the system

$$\dot{w} \ = \ s(w)\ , \quad \dot{\tau} \ = \ e^{-Ft}\, G\, \kappa(w)$$

initialized at $(w_i, 0)$, $i \in I$, by taking the integration time t_* sufficiently large.

Step 3: Compute the approximate expression of $\gamma(\cdot)$ as

$$\gamma(\xi) \ = \ \min_{i \in I} \{\, \kappa_0(w_i) \ + \ \hat{\rho}(|\tau_i - \xi|)\,\}$$

where $\hat{\rho}$ is a sufficiently large number.

The density of the sets $\{w_i,\, i \in I\}$ (and as consequence of $\{\tau_i,\, i \in I\}$) and the value of t_* are precisely the two degree-of-freedom which should be used by the designer to improve the approximation of the real γ and, as a consequence, to improve the asymptotic performance of the regulator. Details can be found in [12].

IV. Simulation results

For the following simulation result the engine speed is controlled by an external controller, the idle speed controller of the ECU (Engine Control Unit). For testing the control approach, a step of the desired idle speed is performed. even in this operation, the shaft torque should be close to zero. Figure 4 shows the shaft torque controlled by the redundant adaptive internal model based controller, when the controller is switched on at $t = 1sec$ and the speed step, which results in a change of the frequency is at $t = 3sec$. For the reduced order internal model approach, the result looks qualitatively similar. In comparison to a well tuned standard feedback control such as *e.g.* H_∞ - control the performance of the presented output regulator is much better. Table II gives an overview of the performance indices

$$J_{T_{ST}} \ = \ \int_{t_1}^{t_2} T_{ST}(\tau)^2\, d\tau$$

$$J_{T_D} \ = \ \int_{t_1}^{t_2} T_D(\tau)^2\, d\tau$$

for the control methods (Redundant Adaptive Output Regulation (AOR), Reduced - Order Adaptive Output Regulation (ROAOR) and $H\infty$ - control ($H\infty$ C)). The parameters of the AOR and ROAOR are shown in Table I where f is chosen such that from $F = \Phi - G\Psi$, where Φ has the same canonical form than F and is similar to S, it results that Ψ has proper entrees. Furthermore we set $\ell = 10 \max_i \Psi_i$. In this comparison it is necessary to stress that it is possible to tune a H_∞ controller so that it leads to a better disturbance suppression (less $J_{T_{ST}}$) but this would also mean an increased value of J_{T_D} and for practical reasons (limited torque and torque rate of dynamometer) this might be a problem. Furthermore in case of the output regulation approaches the shaft torque is more or less regulated to zero. The performance index mainly results from the transients after switching on the controller. The ROAOR approach in comparison to the AOR approach shows a greater $J_{T_{ST}}$ - index and a smaller J_{T_D} - index. It should be mentioned that this is a result which can be explained by slightly different tuning. Figure 5 furthermore shows the shaft torque applying a H_∞ controller.

TABLE I

TUNING PARAMETERS OF THE AOR AND THE ROAOR APPROACHES

f_0	f_1	f_2	f_3	f_4
$-1.39e19$	$-1.01e17$	$-5.82e14$	$-1.66e12$	$-3.64e9$
f_5	f_6	ℓ	γ	
$-4.87e6$	-3400	$1.39e20$	1000	

TABLE II

COMPARISON OF OUTPUT REGULATION AND H_∞ CONTROL USING PERFORMANCE INDICES

	AOR	ROAOR	H_∞ C
$J_{T_{ST}}$	2.3	6.7	32.12
J_{T_D}	1.78×10^5	1.4×10^5	2.65×10^5

V. CONCLUSION AND OUTLOOK

In this paper adaptive output regulation is applied to the idle mode control problem for engine test benches. The main challenge of this control mode is the compensation of the torque ripples caused by the combustion oscillations. To this end, the control problem is first formulated in the adaptive output regulation setup where the torque ripples are assumed to be generated by an linear parameter dependent exosystem and then solved using a redundant adaptive internal model design and a reduced order internal model design. Both approaches has been tested an compared to standard feedback control such as a H_∞ controller. Furthermore the application of recent theory in the case of reduced order but nonlinear exosystem has been discussed as well. Further work will consider the implementation of the presented control algorithms at a real engine test bench.

VI. ACKNOWLEDGEMENTS

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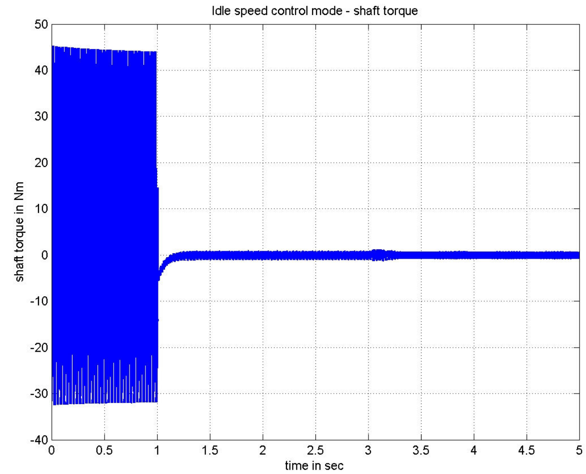

Fig. 4. Shaft torque during idle mode operation applying adaptive output regulation (AOR)

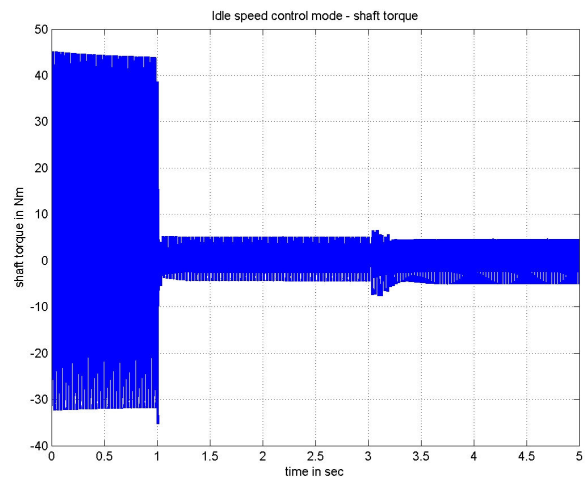

Fig. 5. Shaft torque during idle speed operation applying H_∞ - control

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