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Adaptive Control of Engine Torque with Input Delays

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Abstract: Control of the inner engine torque of a combustion engine is very crucial for the overall performance of a dynamical combustion engine test bench. The main problem thereby is the usually unknown system behavior of the combustion engine, the time delay of the accelerator actuator which is used to control the combustion engine. In general the combustion engine is mounted on a combustion engine test bench in order to adjust the parameters of the engine control unit (ECU). Hence the system behavior can change quite fast. In this paper we will present an adaptive approach to control the combustion engine torque. Measurements on a dynamical combustion engine test bench will verify the proposed approach.

1. INTRODUCTION

The use of combustion engine test benches is very helpful for adjusting the parameters of the ECU. Especially dynamical engine test benches are increasingly used in order to improve the performance of a combustion engine which means, to reduce exhaust emissions and fuel consumption while increasing the power of the combustion engine. Combustion engine test benches are used for *e.g.*, vehicle simulations or simulation of different load patterns. Therefore MIMO control structures had been published ([Gruenbacher et al, 2006a, 2006b], [Bunker et al, 1997], or an adaptive structure has been presented in [Yanakiev, 1998]). For engine testing it is further useful to control the inner engine torque of the combustion engine. This is quite complicated, since the inner engine torque in practice consists of the superposition of the combustion expansions of every stroke. Furthermore it cannot be measured directly and for closed loop control only the mean value of it is necessary. So it is necessary to do a fast estimation of this internal virtual quantity. For torque estimation but also for engine torque control it has to be mentioned that a mathematical model of the combustion engine will not be fully available. One possibility to describe the behavior of a combustion engine is to use an approximate model, which can be identified rather fast as shown in [Gruenbacher et al, 2006a, 2006b]. The main problem thereby is that the combustion engine behavior will change due to deterioration or parameter updates and the approximate model has to be adapted. Using an approximate model also means that it is necessary to run an identification process in order to identify and validate the model. To prevent such measurements, in this paper we focus on an adaptive strategy that is able to start with zero information and which achieves sufficient performance for the full operating range. Furthermore if the combustion engine changes its behavior the controller since it is continuously adapted, will not loose performance.

In this paper we present a Model Reference Adaptive System (MRAS) control scheme which is similar to the well known MRAS controller (see *e.g.* [Astrom and Wittenmark, 1995]) but which is extended such that it is useful also for input delayed LPV systems.

The paper is organized as follows. In the next section we explain the system and the approximate model of the combustion engine and then in the third section we discuss the full control structure. Furthermore in that section we present the main theoretical results and we further explain the extension from a standard MRAS controller to the MRAS controller for input delayed LPV systems. In section four we discuss some implementation issues and show the effectiveness of the proposed methods using measurements on a real combustion engine test bench. Finally we conclude the paper with final statements and comments to the result and to the quite useful control structure

2. APPROXIMATE MODEL OF THE COMBUSTION ENGINE

In the following we consider a part system of a combustion engine test bench where the input is the accelerator pedal, and the output is the engine torque. However as mentioned above the combustion engine torque cannot be measured and therefore an observer is necessary. Especially for this kind of problem it is possible to design an observer since on engine test bench it is possible to measure or at least to estimate the shaft torque which is used to estimate the inner engine torque as well.

For an approximate model of the combustion engine we apply an interpolation of local linear models. Several trials have shown that first order models are sufficient in order to describe the local behavior of the combustion engine. The structure of the engine model is fixed as shown in Fig.1.

The system consists of a nonlinear static map and a dynamical part which depends on the actual operating point that is defined by the static torque T_{EStat} and the engine speed ω_E (consider Fig.1). The static behavior is described by

$$T_{EStat} = T_{EStat}(\omega_E, \alpha), \tag{1}$$

where α is the accelerator position which is the control input of the combustion engine. This nonlinear static map can be easily identified (see [Gruenbacher et al., 2007]). The dynamical subsystem in Fig.1 is a first order LPV – model. Hence it is locally linear but the time constant depends on the operating point. Thus with the two scheduling variables, the engine speed and the stationary engine torque, the resulting mathematical model of the dynamical subsystem is

$$\begin{aligned} \dot{T}_E &= -\rho(T_{Estat}, \omega_E) T_E + \rho(T_{Estat}, \omega_E) T_{EStat} \\ T_{Edyn} &= T_E \end{aligned} \tag{2}$$

where $\rho(T_{Estat}, \omega_E)$ is the operating point depending time constant. The dynamical engine torque is defined by T_{EDyn} which is equal to the state T_E. At this point for a validation of the model we want to refer the reader to [Gruenbacher et al., 2006c].

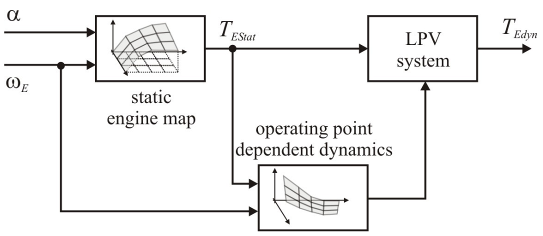

Fig.1: Structure of the simplified engine model

The accelerator pedal is operated by a mechanical accelerator pedal actuator. Usually the test benches are fitted by a mechanical accelerator pedal actuator which pushes the accelerator pedal. This actuator can be approximated by a time delay, a rate limiter and a first order dynamical system. This system is assumed to be well known.

3. MRAS – STRUCTURE FOR UNKOWN ENGINE BEHAVIOR

Because of the simple structure of the engine system (see (1) and (2)) it may be possible to apply a MRAS control scheme with some extensions. Extensions are necessary since the model parameters depend on its own operating point and there is a time delay of the accelerator pedal actuator which has to be considered.

The time delay is considered in a smith – predictor like structure whereas the simulated state instead of the measured state of the combustion engine is applied to the state feedback control law. As it can be seen from Fig. 2 and as it will be discussed in the sequel this yields an adaptive feedforward control scheme at which the state of the reference system instead of the actual state of the system is used in the controller. The system state is used for adapting the controller gains. The parameter update then is done using the delayed signal. Therefore it is necessary to know the time delay of the accelerator pedal signal. Since this actuator is a standard product this time delay is well known and for this considered system it is 20ms.

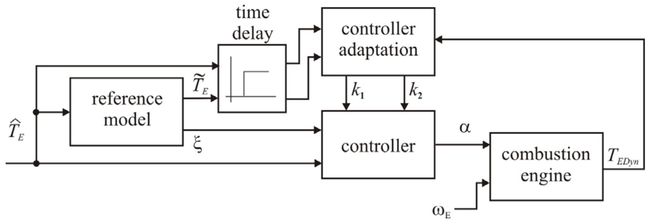

Fig. 2: Structure of the modified MRAS Control structure

In the following the MRAS structure is described and discussed. For a better explanation the standard solution will be recalled first and afterwards the necessary extensions will be explained.

3.1. MRAS Structure for engine torque control:

A model reference adaptive controller for a first order system as it is discussed here is an adaptive pole placement controller. The aim of the control problem is to design a feedback controller such that the closed loop control system behavior converges to a reference system which is:

$$\begin{aligned} \dot{\xi} &= -\hat{b}\xi + \hat{b}\hat{T}_E \\ \tilde{T}_{EDyn} &= \xi \end{aligned} \tag{3}$$

where $\hat{b}$ is the reference time constant, ξ the state of the reference system, $\hat{T}_E$ the input to the reference system and $\tilde{T}_E$ the reference output. For the proposed method we assume a simplified engine model similar to (2)

$$\begin{aligned} \dot{T}_E &= -\rho(\omega_E, T_{EStat}) T_E + \rho(\omega_E, T_{EStat}) \kappa(\omega_E, \alpha) \alpha \\ T_{EDyn} &= T_E \end{aligned} \tag{4}$$

where

$$\kappa(\omega_E, \alpha) = \frac{\partial T_{EStat}(\omega_E, \alpha)}{\partial \alpha} \tag{5}$$

It is further assumed that there exist two positive constants κ_L and κ_U such that $\kappa_L < \kappa(\omega_E, \alpha) < \kappa_U$ for all $(\omega_E, \alpha) \in W \times A$ where W and A define the range of possible ω_E and α respectively. It should be mentioned that this is not a general assumption but in order to achieve drivability of the combustion engine the engine torque should increase if the accelerator is pushed or if α increases. Hence the gradient with respect to the accelerator position should be positive. With this assumption there always exists the local inverse of (5) which is $\alpha = \kappa^{-1}(\omega_E, T_{EStat}) T_{EStat}$. Hence the local pole placement controller for system (4) to achieve a closed loop behavior equal to (3) is

$$\alpha = \kappa^{-1}(\omega_E, T_{EStat})\left(-\frac{\hat{b}-\rho(\omega_E,T_{EStat})}{\rho(\omega_E,T_{EStat})}T_E + \frac{\hat{b}}{\rho(\omega_E,T_{EStat})}\hat{T}_E\right). \quad (6)$$

However the control law (6) is not applicable since the real time constant and the nonlinear static map are not exactly known or are even totally unknown if there is no identification of the engine's input-output behavior at first.

If we assume a constant values for both $\rho(\omega_E,T_{EStat})=\rho$ and $\kappa(\omega_E,T_{EStat})=\kappa$, it is possible to define an adaptive control law using the MRAS theory (see *e.g.* [Astrom and Wittenmark, 1995]) which is

$$\begin{aligned}\dot{k}_1 &= -\gamma e T_E \\ \dot{k}_2 &= \gamma e \hat{T}_E\end{aligned} \quad (7)$$

$$\alpha = -k_1 T_E + k_2 \hat{T}_E \quad (8)$$

where $e = T_E - \xi$ is the difference of the state of the reference system (3) and the system (4). With (8) the error differential equation is

$$\dot{e} = -\hat{b}e + \left(-\rho + \hat{b} - \rho\kappa k_1\right)T_E + \left(\hat{b} + \rho\kappa k_2\right)\hat{T}_E \quad (9)$$

Note that (7) and (9) define one system and it is possible to prove asymptotic stability by applying the Lyapunov function

$$V(e,k_1,k_2) = \frac{1}{2}e^2 + \frac{1}{2}\frac{\left(\hat{b}-\rho\kappa k_2\right)^2}{\gamma\rho\kappa} + \frac{1}{2}\frac{\left(\left(-\rho+\hat{b}\right)-\rho\kappa k_1\right)^2}{\gamma\rho\kappa} \quad (10)$$

where γ is a positive constant (tuning variable) and $\rho\kappa > 0$ since $\rho > 0$ and $\kappa > 0$ for the full operating range. For a detailed proof of this standard solution we want to refer the reader to [Astrom and Wittenmark, 1995].

However system (7) and (9) is a nonlinear parameter varying system since the assumptions $\rho(\omega_E,T_{EStat})=\rho$ and $\kappa(\omega_E,T_{EStat})=\kappa$ are constant values are certainly not true, furthermore the stability proof will fail in this case. Hence in unsteady state conditions the controller may diverge and the error dynamics differential equation might be unstable. To ensure stability even in that case we extend the adaptive controller (7) and (8) such that the tracking error will be bounded even in dynamical operating. For proving this we will exploit the notation of practical stability referred in *e.g.* [Sontag and Ingalls, 2002] and [Yang, 2000]. Furthermore without loss of generality since the change rate of the operating point is bounded and the time constant is constant for this application we do the following assumptions (for convenience we omit the arguments (ω_E,T_E) in ρ and κ)

$$\rho_L < \rho < \rho_U \qquad \dot{\rho}_L < \dot{\rho} < \dot{\rho}_U \quad (11)$$

$$\kappa_L < \kappa < \kappa_U \qquad \dot{\kappa}_L < \dot{\kappa} < \dot{\kappa}_U \quad (12)$$

Now we are ready to introduce the first extensions to the proposed controller given by (7) and (8).

Preposition 1: *If (11) and (12) is true, the model reference adaptive controller for system (4) which guarantees practical stability is given by*

$$\begin{aligned}\dot{k}_1 &= \gamma_1 e T_E - \gamma_2 dz(k_1) \\ \dot{k}_2 &= -\gamma_1 e \hat{T}_E - \gamma_3 dz(k_2)\end{aligned} \quad (13)$$

$$\alpha = -k_1 T_E + k_2 \hat{T}_E \quad (14)$$

with

$$dz(k_i) = \begin{cases} 0 & k_{i\_L} \le k_i \le k_{i\_U} \\ k_i - k_{i\_M} & otherwise \end{cases} \quad (15)$$

where γ_1,γ_2 and γ_3 are positive constants and $k_{i\_M} = \frac{k_{i\_L}+k_{i\_U}}{2}$ (k_{i_L} and k_{i_U} define the bounds of the allowed range of controller gains).

Proof: With $V(e,k_1,k_2) \to V(e,k_1,k_2,\rho,\kappa)$ the time derivative of (10) is

$$\dot{V}(e,k_1,k_2,\rho,\kappa) = V_e\dot{e} + V_{k_1}\dot{k}_1 + V_{k_2}\dot{k}_2 + V_\rho\dot{\rho} + V_\kappa\dot{\kappa} \quad (16)$$

Applying (13) and (14) to (16) after some computations we get (for convenience we write $\dot{V}$ instead of $\dot{V}(e,k_1,k_2,\rho,\kappa)$)

$$\dot{V} = -\hat{b}e^2 + \Delta\dot{V} \quad (17)$$

for which if $k_{i\_L} \le k_i \le k_{i\_U}$ it can be shown that

$$\begin{aligned}\Delta\dot{V} = &\frac{\left(\hat{b}-\rho-\rho\kappa k_1\right)\left(-\dot{\rho}(1+\kappa k_1)-\dot{\kappa}\rho k_1\right)}{\gamma_1\kappa\rho} \\ &-\frac{\left(\hat{b}-\rho-\rho\kappa k_1\right)^2(\dot{\rho}\kappa+\dot{\kappa}\rho)}{2\gamma_1\kappa^2\rho^2} + \frac{\left(\hat{b}-\rho\kappa k_2\right)(\dot{\rho}\kappa+\dot{\kappa}\rho)}{\gamma_1\kappa\rho} \\ &-\frac{\left(\hat{b}-\rho\kappa k_2\right)^2(\dot{\rho}\kappa+\dot{\kappa}\rho)}{2\gamma_1\kappa^2\rho^2}\end{aligned} \quad (18)$$

and if $k_i < k_{i\_L} \vee k_i > k_{i\_U}$ the result is

$$\begin{aligned}\Delta\dot{V} = &\frac{\left(\hat{b}-\rho-\rho\kappa k_1\right)\left(-\dot{\rho}(1+\kappa k_1)-\dot{\kappa}\rho k_1+\gamma_2 dz(k_1)\right)}{\gamma_1\kappa\rho} \\ &-\frac{\left(\hat{b}-\rho-\rho\kappa k_1\right)^2(\dot{\rho}\kappa+\dot{\kappa}\rho)}{2\gamma_1\kappa^2\rho^2} - \frac{\left(\hat{b}-\rho\kappa k_2\right)^2(\dot{\rho}\kappa+\dot{\kappa}\rho)}{2\gamma_1\kappa^2\rho^2} \\ &+\frac{\left(\hat{b}-\rho\kappa k_2\right)\left(\dot{\rho}\kappa+\dot{\kappa}\rho+\gamma_3 dz(k_2)\right)}{\gamma_1\kappa\rho}\end{aligned} \quad (19)$$

In a first consideration equation (17) with (18) and (19) looks very complicated, but if this is considered in more detail one can see, that the square parts in (18) and (19) are always negative and hence are not critical. So we will concentrate on the rest. If $k_i < k_{i\_L} \vee k_i > k_{i\_U}$ it is possible to show that there exist sufficient large positive constants γ_2 and γ_3 such that $\dot{V}$ with (19) is negative definite. Hence

$$\frac{\left(\hat{b}-\rho-\rho\kappa k_1\right)\left(-\dot{\rho}(1+\kappa k_1)-\dot{\kappa}\rho k_1+\gamma_2 dz(k_1)\right)}{\gamma_1\kappa\rho} < 0 \quad (20)$$

$$\frac{\left(\hat{b}-\rho\kappa k_2\right)\left(\dot{\rho}\kappa+\dot{\kappa}\rho+\gamma_3 dz(k_2)\right)}{\gamma_1\kappa\rho} < 0 \quad (21)$$

For proving this, we consider (21) and introduce the desired controller gain which is $\tilde{k}_2 = \frac{\hat{b}}{\rho\kappa}$. Hence it is necessary to prove that

$$\left(\tilde{k}_2 - k_2\right)\left(\dot{\rho}\kappa + \dot{\kappa}\rho + \gamma_3\left(k_2 - k_{2\_M}\right)\right) < 0 \quad (22)$$

Now it is easy to see that if $k_2 > \tilde{k}_2$ then $k_2 > k_{2\_M}$ too, since we assumed that $k_i < k_{i\_L} \vee k_i > k_{i\_U}$, and if $k_2 < \tilde{k}_2$ then $k_2 < k_{2\_M}$ too and hence it is clear that with (11) and (12) there exist a constant γ_3 such that (22) is true. For (20) we do the same considerations.

This proves that $\Delta\dot{V} < 0$ if the controller gains are outside the considered or allowed range. However, the desired values of the controller gains are inside this interval. In this case it is possible to check that $\dot{V}$ is not negative definite for all k_i's inside $k_{i\_L} \le k_i \le k_{i\_U}$, but it is bounded and it is possible to show that

$$\begin{gathered} D^+V < 0, \quad \forall k_i < k_{i\_L} \vee k_i > k_{i\_U} \\ V\left(e,\hat{k}_1,\hat{k}_2,\rho,\kappa\right) < V\left(e,k_1^*,k_2^*,\rho,\kappa\right) \end{gathered} \tag{23}$$

where D^+ is the Dini derivative, $k_i^* < k_{i\_L} \vee k_i^* > k_{i\_U}$ and $k_{i\_L} \le \hat{k}_i \le k_{i\_U}$. And this matches the condition of practical stability as it is recalled in [Yang, 2000]. Hence we can guarantee that the tracking error and the adapted controller gains are bounded. ♦

From the considered application, practical stability is sufficient as long as the performance is still acceptable. However since in the standard operating conditions the rates of the static gain and the time constant of the engine system are small the positive value of $\Delta\dot{V}$ in (17) will be small too. If $\Delta\dot{V}$ is positive the set (e,k_1,k_2) will diverge as long as $\hat{b}e^2 < \Delta\dot{V}$. Note then since $\Delta\dot{V}$ is bounded from above the tracking error will be bounded too. Furthermore in steady state $\Delta\dot{V}$ become zero and therefore $\dot{V}$ is negative.

As we mentioned above the parameters of the combustion engine model are unknown. Hence it is not possible *a priori* to calculate the bounds of the controller gains k_{i_L} and k_{i_U}. But these bounds are not critical. The constraints for the controller constants can directly be calculated from the assumption (11) and (12).

$$k_{1\_L} = \begin{cases} \dfrac{\hat{b}-\rho_U}{\kappa_L\rho_L}, & \hat{b} < \rho_U \\ \dfrac{\hat{b}-\rho_U}{\kappa_U\rho_U}, & else \end{cases}, \quad k_{1\_U} = \begin{cases} \dfrac{\hat{b}-\rho_L}{\kappa_U\rho_U}, & \hat{b} < \rho_L \\ \dfrac{\hat{b}-\rho_L}{\kappa_L\rho_L}, & else \end{cases} \tag{24}$$
$$k_{2\_L} = \frac{\hat{b}}{\kappa_U\rho_U}, \quad k_{2\_L} = \frac{\hat{b}}{\kappa_L\rho_L}$$

However the constraints for the local static gains (ρ_L,ρ_U) and the local time constants (κ_L,κ_U) are unknown too but in general they can be roughly estimated. In the worst case if there is no knowledge about these constants one has to choose them carefully such that the range $k_{i\_L} \le k_i \le k_{i\_U}$ is sufficiently large. In that case the constants γ_2 and γ_3 should be chosen very large such that the stabilizing part in the adaptation law (13) acts as a soft restriction of the controller gains.

3.2. Extension of the MRAS to the time delay problem:

Because of the input time delay τ of the accelerator actuator the adaptive controller (13) and (14) cannot directly be applied. Without further considerations the controller gains tend to wind up.

Assuming $\dot{\rho} = 0$ and $\dot{\kappa} = 0$ the controller gains converge to

$$\tilde{k}_1 = \frac{\hat{b}-\rho}{\rho\kappa}, \qquad \tilde{k}_2 = \frac{\hat{b}}{\rho\kappa} \tag{25}$$

Hence, it is possible to reconstruct the system parameters from the estimated controller gains.

$$\tilde{\rho} = \hat{b} - \frac{\hat{b}\tilde{k}_1}{\tilde{k}_2} \qquad \tilde{\kappa} = \frac{\hat{b}}{\tilde{\rho}\tilde{k}_2} \tag{26}$$

Using these parameters, the system's output and the system's state without delay time can be simulated using the system

$$\dot{\xi} = -\tilde{\rho}\xi + \tilde{\rho}\tilde{\kappa}\alpha \tag{27}$$

where ξ is the state and u the input of the simulated system. Applying the control law $\alpha = -k_1\xi + k_2\hat{T}_E$ the closed loop system results in

$$\dot{\xi} = -(\tilde{\rho} + \tilde{\rho}k_1)\xi + \tilde{\rho}\tilde{\kappa}k_2\hat{T}_E\,. \tag{28}$$

With (26) the system (28) exactly matches the reference system (3).

$$\dot{\xi} = -\hat{b}\xi + \hat{b}\hat{T}_E\,. \tag{29}$$

Hence for considering the time delay we will now exchange the feedback law (8) with the adaptive feedforward control described in the following lemma. Instead of the measured state and output of system (4) we apply the state of the reference system. The measured values (T_E and T_{EStat} (4)) are used in the adaptation law of the controller gains. This is control structure is now similar to a smith predictor control structure.

Lemma 1: *Given the system described by (4) and a control input delay τ, then, if (11) and (12) is true, the model reference adaptive controller which guarantees practical stability is given by*

$$\begin{aligned} \dot{\xi} &= -\hat{b}\xi + \hat{b}\hat{T}_E \\ \dot{k}_1 &= \gamma_1\tilde{e}(T_E - \tilde{e}) - \gamma_2 dz(k_1) \\ \dot{k}_2 &= -\gamma_1\tilde{e}\hat{T}_E(t-\tau) - \gamma_3 dz(k_2) \\ \alpha &= -k_1\xi + k_2\hat{T}_E \end{aligned} \tag{30}$$

where $\tilde{e} = T_E(t) - \xi(t-\tau)$ and $dz(k_i)$ is defined by (15).

Proof: For the first step of the proof we again assume that $\dot{\rho} = 0$ and $\dot{\kappa} = 0$. If in this case it is possible to proof asymptotic stability then in a second step it is possible to ensure practical stability in the same way as done in the proof of Preposition 1 in the case if the assumption ($\dot{\rho} = 0$, $\dot{\kappa} = 0$) is not valid. For $\dot{\rho} = 0$ and $\dot{\kappa} = 0$ the error differential equation is:

$$\dot{\tilde e} = \dot T_E(t) - \dot\xi(t-\tau) = \left(\hat b - \rho - \rho\kappa k_1\right)T_E(t) - \rho e(t-\tau) \\ -\left(\hat b - \rho - \rho\kappa k_1\right)\tilde e(t-\tau) - \left(\hat b - \rho\kappa k_2\right)\hat T_E(t-\tau) \quad (31)$$

With

$$\tilde V(\tilde e, k_1, k_2) = \frac{1}{2}\tilde e^2 + \frac{1}{2}\frac{\left(\hat b - \rho\kappa k_2\right)^2}{\gamma\rho\kappa} + \frac{1}{2}\frac{\left(\left(-\rho + \hat b\right) - \rho\kappa k_1\right)^2}{\gamma\rho\kappa} \quad (32)$$

it is possible to show that with (30)

$$\dot{\tilde V} = -\rho\tilde e(t-\tau)^2 \\ \underbrace{-\frac{\gamma_2}{\gamma_1}\left(\rho(1+\kappa k_1) - \hat b\right)dz(k_1) - \frac{\gamma_3}{\gamma_1}\left(\rho\kappa k_2 - \hat b\right)dz(k_2)}_{\Delta} \cdot \quad (33)$$

For sufficiently large γ_2 and γ_3 as mentioned above the term Δ in (33) is either zero if $k_{i\_L} \le k_i \le k_{i\_U}$ or negative if $k_i < k_{i\_L} \vee k_i > k_{i\_U}$. The first term in (33) is negative since without loss of generality the engine system is a stable system and $\rho > 0$.

The second step of the proof consists of repeating the second part of the proof of Preposition 1. Hence with

$$\dot{\tilde V}(\tilde e, k_1, k_2, \rho, \kappa) = \tilde V_e \dot{\tilde e} + \tilde V_{k_1}\dot k_1 + \tilde V_{k_2}\dot k_2 + \tilde V_\rho \dot\rho + \tilde V_\kappa \dot\kappa < 0 \quad (34)$$

we get

$$\dot{\tilde V} = -\rho\tilde e(t-\tau)^2 + \Delta\dot{\tilde V} \quad (35)$$

and since $\Delta\dot{\tilde V}$ has the same structure than shown in (18) and (19) the proof is identical to the proof of Preposition 1 and practical stability can be shown in the same manner. ♦

4. IMPLEMENTATION AND MEASURMENT RESULTS

For implementing the proposed algorithm some further extensions are necessary in order to consider the dynamics of the control input actuator as well.

4.1. Dynamics of the control input actuator

Because of the control input actuator the real system order is higher than one and hence the output cannot exactly follow the output of the reference system. Thus the controller gains cannot converge and furthermore the controller will – as usual in such a case – generate an overshoot in the step response. Our approach for the present system is very simple. The control idea is kept equal while the output of the reference system is simply filtered using a first order low pass system with the same cut off frequency than the control input actuator. This frequency is again a well known constant, since the actuator is a standard device with a defined dynamical behavior. The cut off frequency of the accelerator pedal actuator is 50Hz and the structure of a reference model can be seen in Fig. 3.

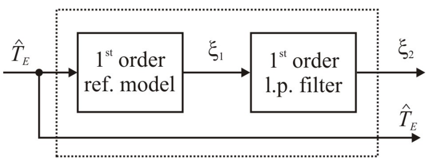

Fig. 3: Structure of the reference model

According to the structure in Fig. 3 the reference model is

$$\dot\xi_1 = -\hat b\xi_1 + \hat b\hat T_E \\ \dot\xi_2 = -a_{act}\xi_2 + a_{act}\xi_1 \\ \tilde T_E = \xi_2 \quad (36)$$

where ξ_1 is the state of the first order reference model, ξ_2 is the state of the first order low pass filter and a_{act} is the time constant of the actuator. Hence the final controller is

$$\dot\xi_1 = -\hat b\xi_1 + \hat b\hat T_E \\ \dot\xi_2 = -a_{act}\xi_2 + a_{act}\xi_1 \\ \dot k_1 = \gamma_1\tilde e(T_E - \tilde e) - \gamma_2 dz(k_1) \\ \dot k_2 = -\gamma_1\tilde e\xi_1(t-\tau) - \gamma_3 dz(k_2) \\ \alpha = -k_1\xi_1 + k_2\hat T_E \quad (37)$$

where $\tilde e = T_E(t) - \xi_2(t-\tau)$.

4.2. Measurement results

The following measurements have been done using a conventional combustion engine, a turbo charged diesel engine, which is fitted with an EGR (exhaust gas recirculation) and which is mounted on a dynamical combustion engine test bench. A test bench consists of the dynamometer and the combustion engine which are connected via a shaft (see Fig. 4). The dynamometer can be used either to control the speed of the combustion engine or to simulate different load patterns to the combustion engine.

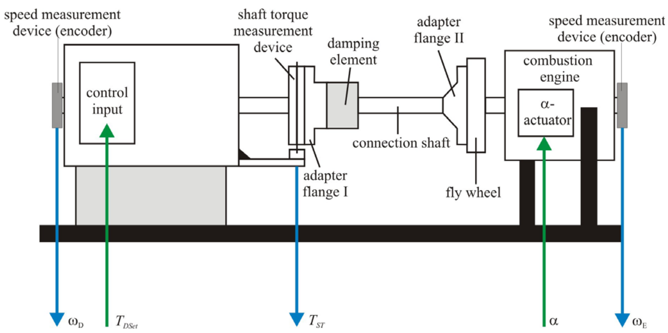

Fig. 4: Dynamical engine test bench system

The general system's inputs are the desired torque of the dynamometer T_{DSet} and the accelerator pedal α. The engine test bench is fully controlled by a processor system. The measured quantities are the speed of the combustion engine and the dynamometer (ω_E and ω_D respectively) and the shaft torque T_{ST}.

As mentioned in the introduction for the controller implementation the mean value inner engine torque has to be estimated. Therefore a high gain Kalman filter in combination with an internal model observer is used (see [Gruenbacher and del Re, 2007]).

In Fig. 5 we show the tracking result using the proposed MRAS controller for an engine torque control. The engine speed during this measurement is kept constant at 3000 rpm. In this operating area the EGR valve is usually fully closed. The control structure works quite good but surely not perfect. The main reason for this is that the model assumption (4) is not fully valid. The real system of course shows nonlinear behavior which is mainly caused by the ECU and which cannot be exactly controlled using linear control theory. In Fig. 6 the same tracking result for an engine speed of 2000 rpm is shown. Here one can see that the result is still good although in this speed region the EGR valve is controlled and due to this, the input – output behavior is strongly nonlinear. This effect can mainly be seen in increasing torque steps. From diesel engines it is well known that if the accelerator is pushed rapidly then the EGR valve is closed in order to generate more sudden power and to avoid a soot cloud as it is well known from old diesel engines. This is why the behavior totally changes and why the MRAS controller for increasing torque steps does not work as well as for decreasing torque steps and in Fig. 5.

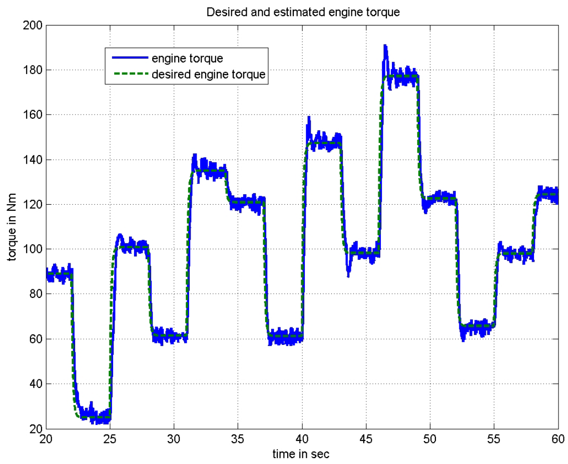

Fig. 5: Engine torque tracking at a speed of 3000 rpm using the proposed MRAS controller

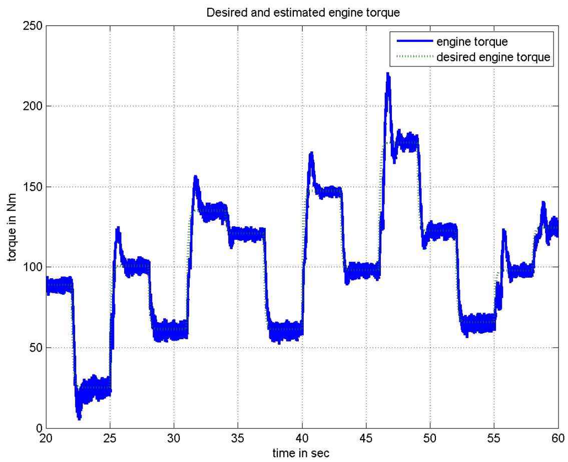

Fig. 6: Engine torque tracking at a speed of 2000 rpm using the proposed MRAS controller

5. CONCLUSIONS

In this paper we have presented a model reference adaptive control structure for controlling the inner engine torque of combustion engine. Therefore it is shown how to adapt a standard MRAS control structure to a LPV system which further includes input delay times. It has been proven that if the desired trajectory is in steady state the controller will converge and the tracking error will tend to zero as time goes to infinity. In the dynamical operation mode it has been shown that the tracking error is bounded. Hence the closed loop system is practically stable.

Future work will consider a mapping update in order to store already learnt information. This will achieve a better convergence if the operating point of the combustion engine changes.

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REFERENCES

Bunker B. J., Franchek M. A., Thomason B. E., Robust Multivariable Control of an Engine-Dynamometer System, IEEE Transactions on Control Systems Technology, Vol. 5, No. 2, 1997

Yanakiev D., Adaptive Control of Diesel Engine-Dynamometer Systems, Proceedings of the 37th IEEE Conference on Decision & Control • Tampa, Florida USA • December 1998

Astrom K. J, Wittenmark B., Adaptive Control, Prentice Hall, ISBN 978-0201558661, 1995

Sontag E. D., Ingalls B., A small-gain theorem with applications to input/output systems, incremental stability, detectability, and interconnections, Journal of the Franklin Institute 339 (2002) 211–229, 2002

Gruenbacher E., del Re L., Kokal H., Schmidt M., Paulweber M., Online Trajectory Shaping Strategy for Dynamical Engine Test Benches, CCA 2006

Gruenbacher E., Langthaler P., del Re L., Kokal H., Schmidt M., Paulweber M.; Goal oriented experiment planning for the optimal use of dynamical engine test benches, MTZ worldwide, 2006

Gruenbacher E., Langthaler P., del Re L., Kokal H., Schmidt M., Paulweber M.; Trade-Offs in Highly Dynamical Transient Simulation on Engine Test Benches, SAE World Congress, 2006

Langthaler P., del Re L.; Identification of Driveline Parameters using an Augmented Nonlinear Model, 16th IFAC World Congress, 2005

Gruenbacher E., del Re L.; Adaptive Mean Value Engine Torque Estimation on Engine Test Benches, CACS 2007

Yang X.; Practical stability in dynamical systems, Chaos, Solitons and Fractals 11, 1087-1092, 2000