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Brief Paper

Coordinated passivation designs☆

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Abstract

In two-input (or multi-input) nonlinear systems it may be possible to achieve feedback passivation of a chosen input using the second input (other inputs) to improve the stability properties of the first input's zero dynamics. This 'coordinated passivation' approach is illustrated on a simplified model of a turbocharged diesel engine.

Keywords: Nonlinear passivation; Zero dynamics; Turbocharged diesel engine

1. Introduction

During the last decade feedback passivation has become a popular approach to nonlinear control design: Kokotović and Sussman (1989), Byrnes, Isidori, and Willems (1991) van der Schaft (1996), Sepulchre, Janković, and Kokotović (1997) Ortega, Loría, Nicklasson, and Sira-Ramírez, (1998). These references and related works surveyed in Kokotović and Arcak (2001), show that passivation designs often exploit inherent system properties and tend to require less control effort. However, the conditions under which a system can be made passive, Byrnes et al. (1991), are restrictive, especially for MIMO systems: the vector relative degree must exist and be zero or one and the zero dynamics must be stable (weak minimum phase). Under these conditions along with a detectability property, the system can be stabilized with output feedback. When vector relative degree fails to exist, stabilization may still be possible under a weaker input–output invertibility condition (Schwartz, Isidori, & Tarn, 1999).

In this paper, a less demanding 'coordinated passivation' approach is pursued for systems with two (or more) inputs. First, an input–output pair is chosen for which the relative degree is one (or zero). Then the stabilization of its zero dynamics is pursued using the remaining input (or inputs). This task is important not only in the case of unstable zero dynamics, but also when the behavior of stable zero dynamics is not acceptable and must be modified by feedback. Finally, when satisfactory zero dynamics behavior is achieved, the design proceeds with feedback passivation of the chosen input–output pair. Preliminary 'master-slave' and 'indirect passivation' versions of 'coordinated passivation' were introduced by the authors in conference papers Larsen and Kokotović (1998) and Larsen, Janković, and Kokotović (2000). A similar two-input coordination approach was used by Kolmanovsky, Sun, and Wang (1999) for combined torque and ratio control of a powertrain with a continuously variable transmission.

In Section 2, we outline the coordinated passivation approach, presenting two alternative designs for zero dynamics stabilization. In Section 3 we describe a third-order simplification of a higher order diesel engine model. In Section 4, the coordinated passivation design is applied to the third-order model and then tested by simulation on the more realistic higher order model.

2. Coordinated passivation

A system with state $x \in \mathbb{R}^n$ is feedback passive for an input–output pair $u, y \in \mathbb{R}$ if there exist a positive definite

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storage function $V(x)$ and feedback $u=\kappa(x)+\delta(x)v$ such that

$$\dot{V} \leqslant vy. \tag{1}$$

Feedback passivation is a useful preliminary step in a stabilization design because additional output feedback

$$v=-\phi(y), \tag{2}$$

where ϕ is a sector-nonlinearity satisfying

$$y\phi(y)>0 \quad \text{for } y \neq 0 \text{ and } \phi(0)=0, \tag{3}$$

achieves $\dot{V} \leqslant -y\phi(y) \leqslant 0$, which ensures stability. Moreover, if the system is zero state detectable, this also guarantees asymptotic stability Byrnes et al. (1991).

We recall that the zero dynamics are the system dynamics in the manifold $y=h(x)=0$, when this manifold is rendered invariant by the control u. The system is said to be minimum phase (weak minimum phase) if the zero dynamics are asymptotically stable (stable). In the coordinated passivation approach for a two-input system

$$\dot{x}=f(x)+g_1(x)u_1+g_2(x)u_2, \tag{4}$$

an input–output pair (u_1, y) is selected for which the relative degree is one.[1] The approach is feasible if the zero dynamics subsystem associated with this pair is stabilizable with the input u_2. If the zero dynamics are already stable, u_2 can be used to improve performance.

For clarity, we express system (4) in the normal form of Isidori (1995)

$$\dot{z}=q(z,y)+p(z,y)u_2, \tag{5a}$$

$$\dot{y}=\alpha(z,y)+\beta_1(z,y)u_1+\beta_2(z,y)u_2 \tag{5b}$$

and assume that its zero dynamics subsystem, that is (5a) with $y=0$, is stabilizable by u_2, where $z\in\mathbb{R}^{n-1}$.

The coordinated passivation design is done in two steps: zero dynamics stabilization and feedback passivation. We present two versions of the basic design which differ in how the input u_2 is used to stabilize or otherwise modify the behavior of the zero dynamics. In the first version the input u_2 is assigned the task of stabilizing the zero dynamics, whereas in the second version, the input u_2 is given the more demanding task of rendering the zero dynamics input-to-state stable with respect to y.

In this presentation, we assume that the stability and input–output properties are to be achieved globally. In applications, this is required only in the region of practical interest, denoted by $\mathscr{D}$. We also assume the system is zero-state detectable, so that asymptotic stability can be achieved.

[1] Henceforth, we will only deal with the relative degree one case because the results are readily extended to relative degree zero systems.

In the *first version* of the design we start by using u_2 to stabilize, or modify the behavior of, the zero dynamics:

$$\dot{z}=q(z,0)+p(z,0)u_2. \tag{6}$$

This entails finding a control Lyapunov function (CLF) for (6), that is, a positive definite function $W(z)$ for which there exists a control law $u_2=\gamma(z)$ such that

$$\dot{W}=\frac{\partial W(z)}{\partial z}(q(z,0)+p(z,0)\gamma(z))<-\alpha(\|z\|)$$

for some class $\mathscr{K}$ function[2] α. Having found $\gamma(z)$, we proceed with feedback passivation of the whole system (5) for the pair (u_1, y). To this end we rewrite (5a) with $u_2=\gamma(z)$ as

$$\dot{z}=q(z,y)+p(z,y)\gamma(z)=\tilde{q}(z)+\tilde{p}(z,y)y,$$

where $\tilde{q}(z)=q(z,0)+p(z,0)\gamma(z)$. Taking, for example,

$$u_1=\beta_1^{-1}(z,y)(-\beta_2(z,y)u_2(z)-\alpha(z,y) \\ -\frac{\partial W}{\partial z}\tilde{p}(z,y)+v), \tag{7}$$

it can easily be verified that the storage function $V=W(z)+\frac{1}{2}y^2$ satisfies $\dot{V}\leqslant vy$ for system (5), which proves passivity of the pair (v,y). Additional output feedback $v=-\phi(y)$ satisfying (3) then achieves asymptotic stability of the overall system (5).

Instead of the control law (7) which relies on cancellation, more robust control laws can be designed by domination as in Sepulchre et al. (1997). Clearly, the terms contributing to the negativity of $\dot{V}$ need not be cancelled.

In the *second version* of the design we use the input u_2 to render (5a) ISS with respect to y, which is a stronger stability property. We seek a positive definite function $W(z)$, called an ISS-CLF, for which there exists $u_2=\gamma(y,z)$ such that $\|z\|\geqslant\alpha_1(\|y\|)$ implies that

$$\frac{\partial W}{\partial z}(q(z,y)+p(z,y)\gamma(z,y))\leqslant-\alpha_2(\|z\|), \tag{8}$$

for some class $\mathscr{K}$ functions α_1 and α_2. This also ensures asymptotic stability of the zero dynamics because (8) with $y=0$ implies

$$\frac{\partial W}{\partial z}(q(z,0)+p(z,0)\gamma(z,0))\leqslant-\alpha_2(\|z\|).$$

An equivalent characterization of the ISS property is the existence of a class $\mathscr{KL}$ function[3] β and a class $\mathscr{K}$ function

[2] A function $\alpha:\mathbb{R}_{\geqslant 0}\to\mathbb{R}_{\geqslant 0}$ belongs to class $\mathscr{K}$ if it is continuous, positive definite, and strictly increasing.

[3] A continuous function $\beta:\mathbb{R}_{\geqslant 0}\times\mathbb{R}_{\geqslant 0}\to\mathbb{R}_{\geqslant 0}$ belongs to class $\mathscr{KL}$ if for each fixed r, $\beta(r,s)$ belongs to class $\mathscr{K}$ with respect to r, and $\beta(r,s)$ is decreasing to zero as $s\to 0$.

λ such that $z(t)$ exists for all $t \geqslant 0$ and is bounded by

$$\|z(t)| \leqslant \beta(\|z(0)\|, t) + \lambda\left(\sup_{0\leqslant t\leqslant \tau} \|y(\tau)\|\right) \tag{9}$$

for any initial condition $z(0)$ and bounded $y(t)$. (See Teel, 1996; Sontag & Wang, 1996.)

A distinguishing feature of this design is that the ISS property of subsystem (5a) allows us to disregard (5a) in the second design step. System (5) with $u_2 = \gamma(y, z)$ is the cascade of ISS subsystem (5a) driven by

$$\dot{y} = \alpha(z, y) + \beta_1(z, y)u_1 + \beta_2(z, y)\gamma(z, y). \tag{10}$$

Thus (5) can be stabilized by stabilizing (10) alone. With the control law

$$u_1 = \beta_1^{-1}(z, y)(-\alpha(z, y) - \beta_2(z, y)\gamma(z, y) + v)$$

(10) becomes $\dot{y} = v$, which is passive. Stabilization of (10), and, hence, (5), is accomplished by choosing $v = -\phi(y)$ as in (2).

3. Diesel engine model

Several recent studies Guzzella and Amstutz (1998), van Nieuwstadt, Kolmanovsky (2000), Stefanapoulou, Kolmanovsky, and Freudenberg (2000), Janković, Janković, and Kolmanovsky (2000) indicate that advanced control designs promise to both improve performance and reduce emissions of modern diesel engines. In this way, the turbocharged (TC) diesel engine has become one of the benchmark MIMO control problems on which to test new control designs.

A schematic of a TC diesel engine with exhaust gas recirculation (EGR) is shown in Fig. 1. The TC is used to boost the pressure in the intake manifold, increasing the mass air charge delivered to the engine cylinders and the amount of fuel that can be burnt. As a result, TC diesel engines can develop greater power and torque than non TC engines. The turbocharger consists of a variable geometry turbine (VGT) connected to a compressor via a common shaft. The energy extracted by the turbine from the exhaust gas flow drives the compressor. Variable inlet guide vanes alter the turbine's geometry to improve transient performance at low engine speeds and to avoid over-speeding at high engine speeds. Recirculating inert exhaust (EGR) gas lowers combustion temperature which reduces NO_x formation and tail-pipe emissions.

One task of diesel engine control is air fuel ratio (AFR) and EGR ratio (EGRR) regulation to set points which have been selected to achieve a tradeoff between emissions and fuel economy. The set points are functions of engine speed, N, and fuelling rate $W_{\mathrm{f}}^{\mathrm{d}}$:

$$\overline{\mathrm{AFR}} = \mathrm{AFR}_{\mathrm{ref}}(N, W_{\mathrm{f}}^{\mathrm{d}})\ \overline{\mathrm{EGRR}} = \mathrm{EGRR}_{\mathrm{ref}}(N, W_{\mathrm{f}}^{\mathrm{d}}).$$

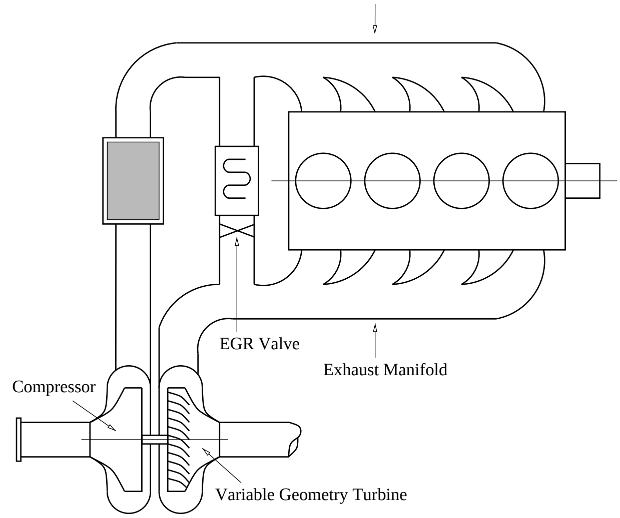

Fig. 1. TC diesel engine with EGR and VGT.

We will use this problem to demonstrate the coordinated passivation design. The design is based on a reduced-order model derived in Janković et al. (2000) that captures the main flow relationships of the engine. A design based on this low-order model provides a starting point for a more realistic control design. The design model is

$$\dot{p}_1 = k_1(W_{\mathrm{c}} + \tilde{u}_1 - k_{\mathrm{e}} p_1), \tag{11a}$$

$$\dot{p}_2 = k_2(k_{\mathrm{e}} p_1 + W_{\mathrm{f}} - \tilde{u}_1 - \tilde{u}_2), \tag{11b}$$

$$\dot{P}_{\mathrm{c}} = \frac{1}{\tau}(-P_{\mathrm{c}} + \eta_{\mathrm{m}} k_{\mathrm{t}}(1 - p_2^{-\mu})\tilde{u}_2) \tag{11c}$$

where k_1, k_2, k_{t}, k_{e}, η_{m}, τ, and μ are positive constants and the control inputs are $\tilde{u}_1$, the EGR flow, and $\tilde{u}_2$, the VGT flow. The flow through the compressor is given by

$$W_{\mathrm{c}} = k_{\mathrm{c}} \frac{P_{\mathrm{c}}}{p_1^{\mu} - 1}. \tag{12}$$

For the design model (11), the AFR, EGRR regulation problem translates into a feedback stabilization problem for the EGR flow $\tilde{u}_1$ and the compressor flow W_{c}. The required steady-state compressor flow $\bar{W}_{\mathrm{c}}$ and EGR flow $\bar{u}_1$ needed to achieve a desired $\overline{\mathrm{AFR}}$ and $\overline{\mathrm{EGRR}}$ are given by the relationships in Janković et al. (2000)

$$\bar{W}_{\mathrm{c}} = \frac{W_{\mathrm{f}}}{2}(\theta + \sqrt{\theta^2 + 4(1 - \overline{\mathrm{EGRR}})\overline{\mathrm{AFR}}})$$

where

$$\theta = (\overline{\mathrm{AFR}}(1 - \overline{\mathrm{EGRR}}) + 15.6\overline{\mathrm{EGRR}} - 1),$$

$$\bar{u}_1 = \frac{\overline{\mathrm{EGRR}}}{1 - \overline{\mathrm{EGRR}}}\bar{W}_{\mathrm{c}} \quad \bar{u}_2 = \bar{W}_{\mathrm{c}} + W_{\mathrm{f}}.$$

Hence, the steady-state intake and exhaust manifold pressures and compressor power are, respectively,

$$\bar{p}_1 = \frac{\bar{W}_c + \bar{u}_1}{k_e},$$

$$\bar{p}_2 = \left(1 - \frac{\bar{W}_c}{\bar{u}_2 \eta_m k_t k_c}\left(\left(\frac{\bar{W}_c + \bar{u}_1}{k_e}\right)^{\mu} - 1\right)\right)^{-1/\mu},$$

$$\bar{P}_c = \frac{\bar{W}_c}{k_c}\left(\left(\frac{\bar{W}_c + \bar{u}_1}{k_e}\right)^{\mu} - 1\right).$$

In the set-point centered coordinates, with the over-bar denoting steady-state values, $x_1 = p_1 - \bar{p}_1$, $x_2 = p_2 - \bar{p}_2$, $x_3 = P_c - \bar{P}_c$, $u_1 = \tilde{u}_1 - \bar{u}_1$, and $u_2 = \tilde{u}_2 - \bar{u}_2$, the model (11) takes the form

$$\dot{x}_1 = -k_1 k_e x_1 - \varphi_1(x_1) + \psi_1(x_1)x_3 + k_1 u_1, \tag{13a}$$

$$\dot{x}_2 = k_2 k_e x_1 - k_2(u_1 + u_2), \tag{13b}$$

$$\dot{x}_3 = -\frac{1}{\tau} x_3 + \varphi_2(x_2) + \psi_2(x_2)u_2. \tag{13c}$$

where φ_1, φ_2, ψ_1, and ψ_2 are given by

$$\varphi_1(x_1) = \frac{k_1 k_c \bar{P}_c}{1 - \bar{p}_1^{\mu}}\left(\frac{\bar{p}_1^{\mu} - (x_1 + \bar{p}_1)^{\mu}}{(x_1 + \bar{p}_1)^{\mu} - 1}\right),$$

$$\psi_1(x_1) = \frac{k_1 k_c}{(x_1 + \bar{p}_1)^{\mu} - 1},$$

$$\varphi_2(x_2) = \frac{\bar{P}_c}{\tau}\left(\frac{\bar{p}_2^{-\mu} - (x_2 + \bar{p}_2)^{-\mu}}{1 - \bar{p}_2^{-\mu}}\right),$$

$$\psi_2(x_2) = \frac{\eta_m k_t}{\tau}(1 - (x_2 + \bar{p}_2)^{-\mu}).$$

In the set $\mathscr{D} = \{p_1 > 1, p_2 > 1, P_c > 0\}$, which is the region of interest for the design, the nonlinearities $\varphi_1(x_1)$ and $\varphi_2(x_2)$ satisfy the sector-condition (3). Furthermore, $\psi_1(x_1) > 0$, and $\varphi_2(x_2)$ and $\psi_2(x_2)$ are bounded

$$-\frac{\bar{P}_c}{\tau} < \varphi_2(x_2) < \frac{\bar{P}_c \bar{p}_2^{\mu}}{\tau(1 - \bar{p}_2^{\mu})}, \tag{14}$$

$$0 < \psi_2(x_2) < \frac{\eta_m k_t}{\tau}. \tag{15}$$

4. Controller design and simulations

In the coordinated passivation design, a single output $y = -x_2$, the deviation of the exhaust gas pressure from the desired set point, is selected. The choice is motivated by the fact that the exhaust manifold pressure strongly influences the flow relationships in the engine. Other choices are possible and merit further investigation. This approach is in contrast to Janković et al. (2000), where x_2 and $W_c - \bar{W}_c$ are selected as the outputs for a two-input two-output MIMO design. Furthermore, whereas feedback linearization is used in Janković et al. (2000) to generate a CLF for the design model, our method differs in that we systematically arrive at a CLF using the flexibility afforded by multiple inputs. A domination redesign with this CLF could be performed as in Janković et al. (2000), but is not pursued here.

4.1. Controller design

As VGT was introduced to the engine with the goal of improving transient performance, we assign u_1 as the input for which passivation will be pursued, assisted by the input u_2. By this assignment, the role of the input u_2 is to enhance stability and improve the performance of the zero dynamics of the pair (u_1, y).

Introducing new state variables

$$z_1 = x_1 - \frac{k_1}{k_2}y, \quad z_2 = x_3, \quad y = -x_2$$

we rewrite the design model (11) as

$$\dot{z}_1 = -\varphi_1\left(z_1 + \frac{k_1}{k_2}y\right) + \psi_1\left(z_1 + \frac{k_1}{k_2}y\right)z_2 - k_1 u_2, \tag{16a}$$

$$\dot{z}_2 = -\frac{1}{\tau}z_2 + \varphi_2(-y) + \psi_2(-y)u_2, \tag{16b}$$

$$\dot{y} = -k_1 k_e y - k_2 k_e z_1 + k_2(u_1 + u_2). \tag{16c}$$

With $u_2 = 0$, the z-subsystem (16a) and (16b) is

$$\dot{z}_1 = -\varphi_1\left(z_1 + \frac{k_1}{k_2}y\right) + \psi_1\left(z_1 + \frac{k_1}{k_2}y\right)z_2, \tag{17a}$$

$$\dot{z}_2 = -\frac{1}{\tau}z_2 + \varphi_2(-y). \tag{17b}$$

It is significant that this subsystem has the cascade structure

$$\dot{\xi}_1 = f(\xi_1, \xi_2, u), \tag{18a}$$

$$\dot{\xi}_2 = f_2(\xi_2, u), \tag{18b}$$

for which useful tools exist for verifying the ISS property and constructing ISS-Lyapunov functions. This motivates us to pursue the stronger ISS design for (16).

It was shown in Krstić, Kanellakopoulos, and Kokotović (1995, Lemma C.4) that if ξ_1 is ISS with respect to ξ_2 and u and if ξ_2 is ISS with respect to u, then (18) is ISS with respect to u. Furthermore, if $V_1(\xi_1)$ and $V_2(\xi_2)$ are ISS-Lyapunov functions for (18a) and (18b), respectively, then $V(\xi) = V_1(\xi_1) + V_2(\xi_2)$ can be taken as an ISS-Lyapunov function for (18). This result is now applied to (17).

Ignoring the constraints on the state variables, it is immediately evident that (17b) is ISS with respect to y as the input. It easy to show that this remains true with the constraints $z_2, y \in \mathscr{D}$. As the ISS-Lyapunov function we use $W_2 = \alpha_2 z_2^2$. Because for $y \in \mathscr{D}$, $\varphi_2(-y) > -\bar{P}_c/\tau$, it follows that for

$z_2 > -\bar{P}_c$, i.e. $z_2 \in \mathscr{D}$, $\|z_2\| > \rho(\|y\|)$ implies that $\dot{W}_2 < 0$, where

$$\rho(\|y\|) = \max(\tau\varphi_2(-\|y\|), \tau\varphi_2(\|y\|)).$$

With the ISS-Lyapunov function $W_1(z_1) = \alpha_1 z_1^2$, a similar argument shows that (17a) is ISS with respect to y, z_2 for z_1, z_2, and y restricted to $\mathscr{D}$.

This proves that (17) is ISS with respect to y and an ISS-Lyapunov function is

$$W(z_1, z_2) = W_1(z_1) + W_2(z_2) = \alpha_1 z_1^2 + \alpha_2 z_2^2.$$

When $y = 0$ subsystem (17) is asymptotically stable, and, thus, the whole system (16) is minimum phase *with* $u_2 = 0$.

In this example the minimum phase property is already present, however the transient behavior is not acceptable. Therefore, the task of the assisting input u_2 is to enhance the stability and improve performance of the zero dynamics. One way to do this is to use W as an ISS-CLF and design a control law for u_2 which increases the negativity of $\dot{W}$. Rewriting (16a) and (16b) as

$$\dot{z} = q(z, y) + p(z, y)u_2,$$

we select a sector nonlinearity ψ and

$$\begin{aligned} u_2 &= -\psi\left(\frac{\partial W}{\partial z}\, p(z, y)\right) \\ &= -\psi(-\alpha_1 k_1 z_1 + \alpha_2 \psi_2(-y) z_2) \end{aligned} \tag{19}$$

which contributes a term that increases the negativity of

$$\dot{W} = \frac{\partial W}{\partial z}\, q(z, y) - \left(\frac{\partial W}{\partial z}\, p(z, y)\right) \psi\left(\frac{\partial W}{\partial z}\, p(z, y)\right).$$

We now proceed to the feedback passivation step of the design, in which the task of the input u_1 is to stabilize (16c). This is accomplished, for example, with

$$u_1 = -u_2 + k_e z_1 + v \tag{20}$$

and a subsequent choice of $v = -\phi(y)$ satisfying (3). The design is completed by making specific choices of ψ in (19) and v in (20) to guarantee that the constraint set $\mathscr{D}$ is positively invariant for (11), that is the solutions of (11) do not leave $\mathscr{D}$.

It is easy to verify that VGT flow needs to be positive, $\tilde{u}_2 > 0$, that is $u_2 > -\bar{u}_2$ for $P_c(t) > 0$. Hence ψ must be selected so that

$$\psi > -\bar{u}_2. \tag{21}$$

To ensure that $p_2(t) > 1$ for the third-order design model, we need that $\tilde{u}_1 + \tilde{u}_2 < k_e p_1 + W_f$ when $p_2 = 1$. For the control law (20), a straightforward calculation shows that this is achieved if

$$\phi(\bar{p}_2 - 1) > k_e \frac{k_1}{k_2}(\bar{p}_2 - 1). \tag{22}$$

Using the fact that $W_c \to \infty$ as $p_1 \to 1$, it can be shown that $p_1(t) > 1$ when u_2 is chosen according to (19).

For implementation, we return to the original coordinates (11). The control laws for the VGT and EGR flows are

$$\begin{aligned} \tilde{u}_1 &= \psi(-c_1\lambda + c_2\psi_2(-p_2 + \bar{p}_2)(P_c - \bar{P}_c)) \\ &\quad + k_e\lambda - \phi(-p_2 + \bar{p}_2) + \bar{u}_1, \end{aligned} \tag{23}$$

$$\tilde{u}_2 = -\psi(-c_1\lambda + c_2\psi_2(-p_2 + \bar{p}_2)(P_c - \bar{P}_c)) + \bar{u}_2,$$

$$\lambda = \left(p_1 - \bar{p}_1 + \frac{k_1}{k_2}(p_2 - \bar{p}_2)\right),$$

where the choice of constants $c_1, c_2 > 0$ and the functions ψ, ϕ satisfying (21) and (22) provide the design freedom to tune performance. Initially, this tuning was performed by simulations with the design model (11). For simplicity, ϕ and ψ were chosen as

$$\phi(y) = c_3 y, c_3 > k_e \frac{k_1}{k_2} \qquad \psi(\theta) = \begin{cases} \theta & \text{if } \theta \geqslant -\bar{u}_2, \\ -\bar{u}_2 & \text{if } \theta < -\bar{u}_2. \end{cases}$$

4.2. Simulations

To validate this low-order design, simulations were performed using the seven-state model described in Kolmanovsky, Moraal, van Nieuwstadt, and Stefanopoulou (1997) and shown in the block diagram of Fig. 2. For this more realistic model, the control objective is to regulate the AFR, and the exhaust gas recirculation ratio, EGRR, to set-points $\overline{\text{AFR}}$ and $\overline{\text{EGRR}}$, respectively. The control

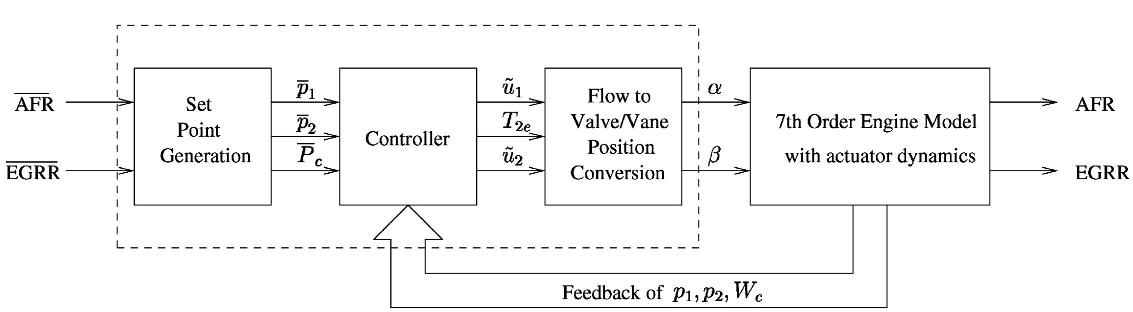

Fig. 2. Simulation model.

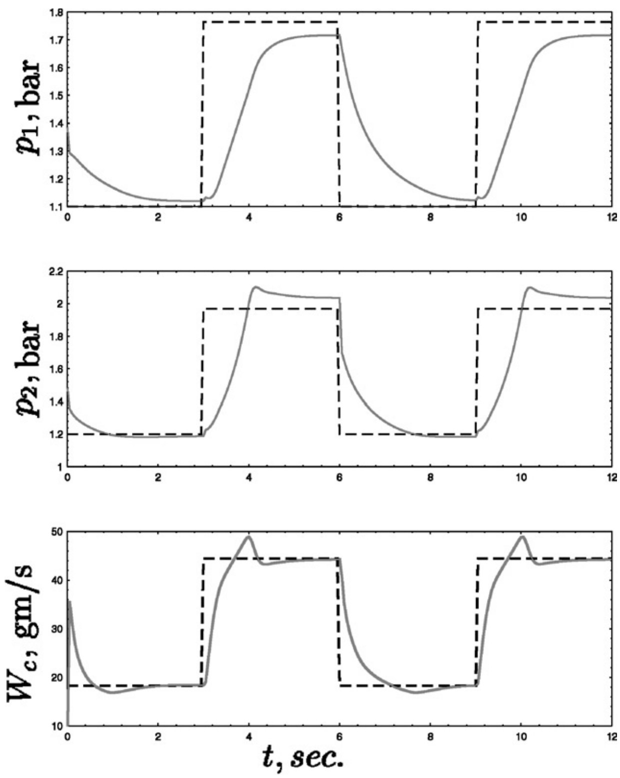

Fig. 3. State responses to a sequence of fueling step changes.

signals for this model are the normalized EGR valve lift α and normalized VGT vane setting β. They were generated from the controls laws (23) for the EGR and VGT flows designed for the third-order design model (11) using approximate static maps. EGR and VGT actuator dynamics and magnitude saturation, as well as a fuel limiter, were also added to the model. The compressor power was derived from the compressor flow using (12).

Fig. 3 shows the response of the system to a step sequence of fuelling changes from 2 to 6 kg/h at an engine speed of 2000 RPM. The dashed curves of Fig. 3 indicate the desired set-points. The response of the compressor flow is rapid, and exhibits overshoot. The exhaust manifold pressure responds more slowly and also exhibits overshoot.

Fig. 4 shows the set-point regulation of EGRR and AFR. The traces show fast response of the AFR to sudden changes in fuel-flow dictated by driver's torque demand. As expected, the EGR response shows a nonminimum phase character (AFR and EGRR are nonminimum phase outputs for this system), but settles close to the desired steady-state value. In Fig. 5, a comparison the normalized commanded and actual actuator settings is shown.

The small steady-state regulation errors present in Figs. 3 and 4 are mostly due to imperfect set point calculation. Furthermore, the clipping in the AFR plot at AFR = 18 is caused by the fuel limiter.

The simulations performed show the controller achieves satisfactory performance and demonstrates the utility of the

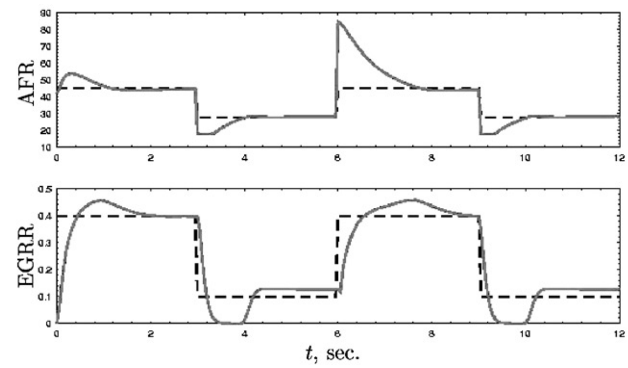

Fig. 4. Nonminimum phase behavior of output signals of the seven-state simulation.

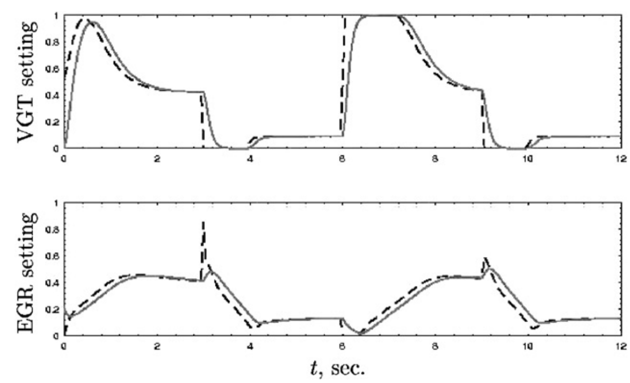

Fig. 5. Control signals of the seven-state simulation.

coordinated passivation design based on a low-order model. Further improvement is possible with additional parameter tuning.

5. Conclusion

We have introduced coordinated passivation as a design tool for multi-variable nonlinear systems. Two versions of the design were presented to achieve a minimum phase system for which feedback passivation can be applied. Coordinated passivation retains the user-friendly SISO reasoning to overcome the single-input structural obstacles to passivity and improve performance by redesigning the zero dynamics. The methodology was used to design a controller for a low-order model of TC diesel engine.

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