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MULTI-MODEL ADAPTIVE CONTROL FOR TURBOCHARGED DIESEL ENGINES

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Scopul acestui articol este acela de a aduce in discutie cateva aspecte privind functionalitatea motoarelor cu combustie interna in general, si a motoarelor diesel in particular, etape in elaborarea modelelor pentru diferite conditii de functionare, precum si analiza comenzii multimodel asupra motorului. Lucrarea propune un multicontroller pentru motorul diesel. Verifica consistenta caracteristicilor controllerului in cazul diverselor variatii ale parametrilor modelului ales; sunt aplicate perturbatii, si apoi este studiat atat modelul nominal cat si cel perturbat. Scopul devin acela de a dovedi stabilitatea sistemului folosind acest tip de controller – si anume stabilitatea interschimbarii comenzilor in directa relatie cu referintele si indiferent de perturbatiile aplicate modelului.

The purpose of this article is to discuss several aspects regarding the functionality of the internal combustion engines in general, and that of the diesel engines in particular, stages in model construction for different operation conditions, and those of analyzing the multi model command of the engine. The work proposes a multi stage controller for the diesel engine. It checks the constancy of the characteristics of this controller for diverse variations of the chosen model's parameters; perturbations are being applied, and then we study both the nominal model and the perturbed one. The purpose is to prove the stability of the system by using this type of controller which allows the so called switch of commands depending of certain functionality references and regardless of the disturbances applied to the model's parameters.

Keywords: diesel engine, robust control, turbocharger, multivariable system, model identification

1. Introduction

In the diesel engine, fuel is electronically injected into the cylinders at a desired angle. The fuel combusts and the released energy is transformed into mechanical force which forces the crankshaft to rotate. The crank-shaft has a flywheel attached on the side closest to the drive line. A sensor measures the rotating speed of the flywheel. The flywheel speed oscillates and its behavior depends on several factors where the cyclic torque from the cylinders is

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considerate. By changing the injection on individual cylinders, the behavior of the flywheel speed oscillation alters. This makes it possible to balance the engine by controlling the electronic injection with feedback from the speed signal. Thus small injection errors in individual cylinders can be corrected as well as unwanted orders of the oscillations can be removed. At present engines, the engine management system (EMS) alters out a few interesting orders of the oscillating speed signal. The unwanted orders, typically the half and the first engine order, are used as feedback to the injectors in order to balance the engine.
Modern diesel engines are typically equipped with variable geometry turbochargers (VGT) and exhaust gas recirculation (EGR), which both introduce feedback loops from the exhaust to the intake manifold. This leads to a multivariable nonlinear control problem. Among the control methodologies previously applied to this problem are control Lyapunov function based nonlinear control [1], minimum-time optimal control [2], adaptive control [3], robust control [4], passive design [5], or gain-scheduled PI control with dc gain-based directional compensation [6]. A comprehensive performance comparison for various control schemes applied to a similar hardware setup to the one in this paper can be found in [7]. These problems are addressed by the approach presented here. This paper describes the design and implementation of a robust, gain scheduled multi-model controller based on recently developed results on the control of linear parameter-varying plants applied to a multi variable model.

The multi-model approach in the study for nonlinear control strategies is a relatively new current. In recent studies you can often find the need to monitor the functionality fot nonlinear systems or multi regime processes, like diesel engines, in different operating stages that cannot be approximated but roughly through a single global model. First works of this kind that proposed stability and robustness methods using classic algorithms are those of Bal Krishnan and Narenda in the 90s [5]. Later the Kalman filter and the hysteresis algorithms based models appeared. Studies made by Petridis, Kehagias and Toscano use nonlinear systems with time variables, while Landau and Karimi use the so called CLOE (Closed Loop Output Error) procedure of parameter adaptation [6]. After that studies evolve concerning the use of neural networks and fuzzy procedures. In general in multi model control one must consider the following aspects: modulation based on the particularities of the system of study, the choosing of the right control algorithm for a certain stage of the dynamic process, the switching from one algorithm to another while the dynamic process stages alternate, and the ensuring stability of the whole ensemble while switching [11]. In general, the multi model structure looks like in fig.1 [5]:

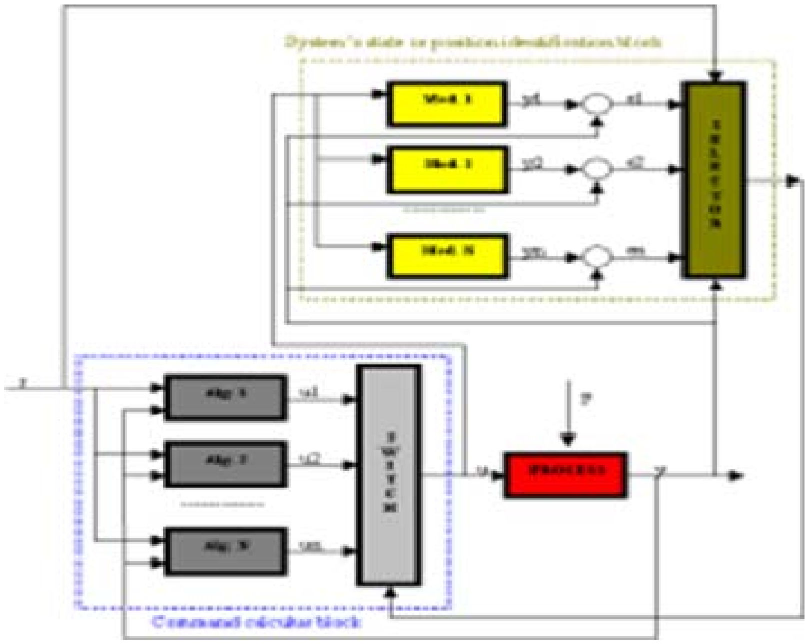
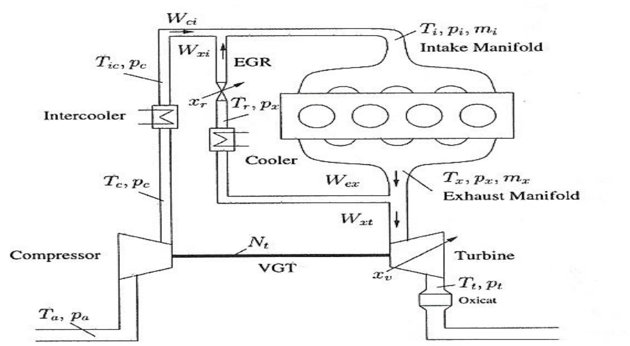


Fig. 1. The Classic Multi Model Structure

2. Analysis of the Dynamic Model

The plant to be controlled is a turbocharged passenger car diesel engine equipped with exhaust gas recirculation (EGR) as seen in the following figure. The turbocharger increases the power density of the engine by forcing air into the cylinders, which allows injection of additional without reaching the smoke limit. The turbine, which is driven by the energy in the exhaust gas, has a variable geometry (VGT) that allows the adaptation of the turbine efficiency based on the engine operating point. The second feedback path from the exhaust to the intake manifold is due to exhaust gas recirculation, which is controlled by an EGR valve. The recirculated exhaust gases replace oxygen in the inlet charge, thereby reducing the temperature profile of the combustion and hence the emissions of oxides of nitrogen. The interactions are relatively complex; a detailed description can be found in [5] and the references therein. While the VGT actuator is typically used to control the intake manifold absolute pressure (MAP), the EGR valve controls the mass air flow (MAF) into the engine. Both the EGR and VGT paths are driven by the exhaust gases and hence constitute an inherently multivariable control problem.



Schematic diagram of a turbocharged diesel engine with EGR.

Fig. 2. Schematic diagram

2.1 Mathematical structure of the model

In this section a mean-value model of the air path of a turbocharged diesel engine with EGR is described. For a detailed derivation of the mean-value model see [13, 15] and the reference quoted therein. A third order nonlinear model can be derived using the conservation of mass and energy, the ideal gas law for modeling the intake and exhaust manifold pressure dynamics, and a first order differential equation with time constant τ for modeling the power transfer dynamics of the VGT. Under the assumption that the intake and exhaust manifold temperatures, the compressor and turbine efficiencies, the volumetric efficiency and the time constant τ of the turbocharger are constant, this modeling approach results in the nonlinear model [12, 13].

$$\begin{aligned}
\dot{p}_i &= \frac{RT_i}{V_i}\left(W_{ci} + W_{xi} - W_{ie}\right) \\
\dot{p}_x &= \frac{RT_x}{V_x}\left(W_{ie} - W_{xi} - W_{xt} + W_f\right) \\
\dot{P}_c &= \frac{1}{\tau}\left(-P_c + \eta_m P_t\right).
\end{aligned} \tag{1}$$

pi – intake manifold pressure
px – exhaust manifold pressure
Pc – power transferred by the compressor
τ = 0.11 s – time constant
nium = 0.98 – mechanical efficiency

Vi = 0.006m3 – intake manifold volume
Vx = 0.001m3 – exhaust manifold volume
Wci – describes the relationship between the flow through the compressor and the power
niuc = 0.61 – compressor efficiency
Ta = 298K – ambient temperature
cp = 1014.4 J/kgK – heat at constant pressure
cv = 727.4 J/kgK – heat at constant volume
miu = (cp-cv) / cp = 0.286 – constant
pa = 101.3 kPa – ambient pressure
Pt = power transferred by the turbine
Wxi – flow through the EGR Valve
Wie – flow from the intake manifold into the cylinders
Wxt – turbine flow
Wf – fuel rate

$$W_{ci} = \frac{\eta_c}{c_p T_a} \frac{P_c}{\left(\frac{p_i}{p_a}\right)^{\mu} - 1} \tag{2}$$

$$W_{xi} = \frac{A_{egr}(x_{egr}) p_x}{\sqrt{RT_x}} \sqrt{\frac{2p_i}{p_x}\left(1 - \frac{p_i}{p_x}\right)} \tag{3}$$

$$W_{ie} = \eta_v \frac{p_i N V_d}{120 T_i R} \tag{4}$$

$$W_{xt} = (a x_{vgt} + b)\left(c\left(\frac{p_x}{p_a} - 1\right) + d\right)\frac{p_x}{p_{ref}} \times \sqrt{\frac{T_{ref}}{T_x}} \sqrt{\frac{2p_a}{p_x}\left(1 - \frac{p_a}{p_x}\right)} \tag{5}$$

$$P_t = W_{xt} c_p T_x \eta_t \left(1 - \left(\frac{p_a}{p_x}\right)^{\mu}\right) \tag{6}$$

Wxi – describes the flow through the EGR valve. Aegr(Xegr) is the effective area of the EGR valve, Tx=509K is the exhaust manifold temperature and R=287 J/kgK is the gas constant. Wie is the flow from the intake manifold into the cylinders, modeled by the speed-density equation. Niuv=0.87 is the volume efficiency, N-the engine speed, the intake manifold temperature Ti=313K, and the displacement volume Vd=0.002m3. Wxt – the turbine flow, with its parameters, a=-0.136, b=0.176, c=0.4, d=0.6, the reference pressure pref=101.3 kPa, and the reference temperature Tref=298K. Finally, the turbine pressure is modeled with the turbine efficiency niut=0.76. Furthermore, the engine speed N and the fueling rate Wf are considered as known external parameters.

2.2 Global model representation and nonlinear model predictive control

In the following the finite horizon optimal control problem for the one described earlier is set up. In the set point centered and normalized coordinates,

$$\begin{aligned} x_1 &= \frac{p_i - p_i^s}{p_i^s},\ x_2 = \frac{p_x - p_x^s}{p_x^s},\ x_3 = \frac{P_c - P_c^s}{P_c^s} \\ u_1 &= \frac{A_{egr} - A_{egr}^s}{A_{egr}^s},\ u_2 = \frac{A_{vgt} - A_{vgt}^s}{A_{vgt}^s}, \end{aligned} \tag{7}$$

where set point variables are in the following denoted by a superscript s. Thus the diesel engine model can be rewritten as

$$\begin{aligned} \dot{x}_1 &= k_1\left(\phi_1(x_1,x_2) + \phi_2(x_1,x_2,u_1) - \phi_3(x_1)\right) \\ \dot{x}_2 &= k_2\left(\phi_3(x_1) - \phi_2(x_1,x_2,u_1) - \phi_4(x_2,u_2)\right) \\ \dot{x}_3 &= k_3\left(\phi_5(x_2,u_2) - (x_3 P_c^s + P_c^s)\right) \end{aligned} \tag{8}$$

with the nonlinearities and constants

$$\phi_1 = \frac{\eta_c}{c_p T_a} \frac{x_3 P_c^s + P_c^s}{\left(\frac{x_1 p_i^s + p_i^s}{p_a}\right)^{\mu} - 1},$$
$$\phi_2 = \frac{(u_1 A_{egr}^s + A_{egr}^s)(x_2 p_x^s + p_x^s)}{\sqrt{RT_x}} \times \sqrt{\frac{x_1 p_i^s + p_i^s}{x_2 p_x^s + p_x^s}\left(1 - \frac{x_1 p_i^s + p_i^s}{x_2 p_x^s + p_x^s}\right)},$$
$$\phi_3 = \eta_v \frac{NV_d}{120 T_i R}(x_1 p_i^s + p_i^s),$$
$$\phi_4 = (u_2 A_{vgt}^s + A_{vgt}^s)\left(c\left(\frac{x_2 p_x^s + p_x^s}{p_a} - 1\right) + d\right) \times \sqrt{\frac{T_{ref}}{T_x}} \sqrt{\frac{2p_a(x_2 p_x^s + p_x^s)}{p_{ref}^2} - \frac{2p_a^2}{p_{ref}^2}},$$
$$\phi_5 = \phi_4 c_p T_x \eta_t \left(1 - \left(\frac{p_a}{x_2 p_x^s + p_x^s}\right)^{\mu}\right),$$
$$k_1 = \frac{RT_i}{V_i p_i^s},\ k_2 = \frac{RT_x}{V_x p_x^s},\ k_3 = \frac{1}{\tau P_c^s}. \tag{9}$$

The result is a nonlinear model in state representation. But the state control strategy is difficult. Thus we propose the extraction of a nonlinear I/O model and as a control procedure, a method based on multi model configuration.

2.3 Multi Model Approach

Future work involves finding a complex model of superior order with nonlinear characteristics and trying the control strategy imposed on a PI controller characteristic. Regarding the particular study for the diesel engine, in [7] a CLOE type identification method with dynamic loop is proposed.
The discret model of the system is presented:

$$A(q^{-1})y(k) = B(q^{-1})u(k) \tag{10}$$

where:

$$A(q^{-1}) = 1 + a_1 q^{-1} + \ldots + a_{n_A} q^{-n_A} \tag{11}$$
$$B(q^{-1}) = b_0 + b_1 q^{-1} + \ldots + b_{n_B} q^{-n_B}$$

with nA<=nB. We imply an RST algorithm:

$$S(q^{-1})u(k)+R(q^{-1})y(k)=T(q^{-1})y^{*}(k) \tag{12}$$

And the polynoms are:

$$\begin{aligned}
S(q^{-1}) &= s_0 + s_1 q^{-1} + \ldots + s_{n_S} q^{-n_S} \\
R(q^{-1}) &= r_0 + r_1 q^{-1} + \ldots + r_{n_R} q^{-n_R} \\
T(q^{-1}) &= t_0 + t_1 q^{-1} + \ldots + t_{n_T} q^{-n_T}
\end{aligned} \tag{13}$$

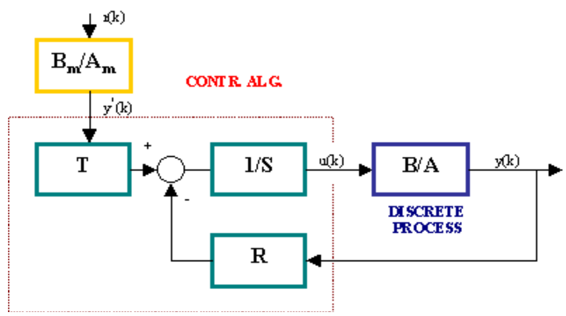


Fig. 3. RST algorithm, two freedom-degrees closed-loop canonical form

For simulation purposes we consider three functionality points on the nonlinear process diagram, mainly P1, P2 and P3, in which we identify three models, meaning M1, M2 and M3.

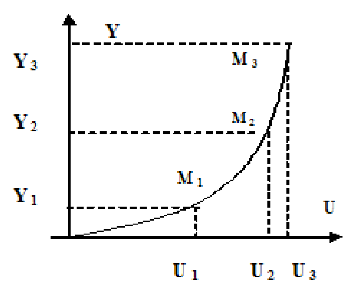


Fig. 4. RST algorithm, two freedom-degrees closed-loop canonical form (U-command, Y-reference)

According to the models-algorithms matching zones, we have identified the models M1, M2 and M3, as being connected to the following intervals (considered as set points): 1000 rpm, 1600 rpm, 2000 rpm, with Wf – fuel flow respectively 4, 5 and 6 kg/h. Through the least squares identification method and using a sample time period Te = 0.5 sec (usual sample period – chosen for simplifing the calculus), the models are obtained in the WinPim platform and then through pole placement method the controllers in the WinReg platform. For the stability of the switching process, a Cvi – National Instruments real time software platform will be developed.

3. Conclusions

In this paper, a multi model approach is considered for controlling the air path of a turbocharged diesel engine with EGR. An I/O model evaluation will be made through experimental methods. As shown, the multi model controller is a well suited control approach to regulate a diesel engine due to the nonlinear multivariable structure of the process and the present constraints on inputs and states. In simulation studies it is shown that the multi model approach can improve the transient behaviour of a diesel engine. Future work involves the implementation of an explicit multi model controller on a diesel engine.

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