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Modeling and Control Solutions for the Turbocharged Diesel Engines

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Abstract*— *The purpose of this paper is to present two types of models for the turbocharged diesel engine: dynamic I/O and steady state models. The simulation results conclude the righteousness behind the choosing of the models and thus validating them. For the control part we propose a LQR controller and a RST controller respectively, concluding on some simulation results on behalf of each part in order to be optimizing the combustion process in future work.

I. INTRODUCTION

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.

II. DIESEL ENGINE (DE) TECHNOLOGICAL ASPECTS

The objective to be controlled is a turbocharged passenger car diesel engine equipped with exhaust gas recirculation (EGR) as seen in figure 1. 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.

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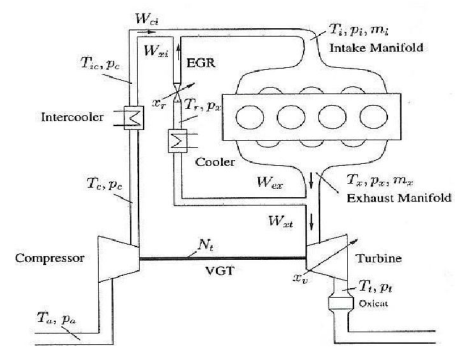

Fig. A. Schematic diagram of the turbocharged diesel engine.

A. *The DE Dynamic Equations*

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] 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; xv – turbine blade position.

The parameters Wci, Wxi, Wie, Wxt and Pt are calculated with the relations:

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

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

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

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

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

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.

III. Dynamic Models

A. The DE LPV model structure

Taking in consideration the notes and parameters' values from above, the standard steady state model representation becomes:

$$\begin{pmatrix} \dot{p}_i \\ \dot{p}_x \\ \dot{P}_c \end{pmatrix} = A(\rho(t)) \begin{pmatrix} p_i \\ p_x \\ P_c \end{pmatrix} + B(\rho(t)) \begin{pmatrix} A_r \\ x_v \\ N \\ W_f \end{pmatrix}$$
$$\begin{pmatrix} W_{ci} \\ p_i \end{pmatrix} = C(\rho(t)) \begin{pmatrix} p_i \\ p_x \\ P_c \end{pmatrix} + D(\rho(t)) \begin{pmatrix} A_r \\ x_v \\ N \\ W_f \end{pmatrix} \tag{7}$$

$$\dot{x} = A(\rho(t)) * x + B(\rho(t)) * u$$
$$y = C(\rho(t)) * x + D(\rho(t)) * u \tag{8}$$

where A,B,C,D are matrix functions of the exogenous parameter $\rho(t)$, which has a variation of $\rho: [0, \infty) \to \mathcal{R}^k$. Both $\rho(t)$ as well as its ratio $\dot{\rho}$ are contained in the pre specified sets Γ and Γ_d. The parameter vector ρ of ρ_i components, each varying between $\underline{\rho_i}$ and $\overline{\rho_i}$

$$\rho_i(t) \in \left[\underline{\rho_i}, \overline{\rho_i}\right], t \geq 0, i = 1, \dots, k$$

One can see out of the differential equations that only the intake and the exhaust manifold pressures (pi and px) have nonlinear forms. Thus they become parameters to those equations. Bearing that in mind, each term contains a highlighted state variable. The multiplying terms now contain only fix or time variable parameters, ρ_1 or ρ_2. In some terms, pxp_x is highlighted, which means that there are no stats or other variables that can multiply it, so it becomes a state.

The equation that defines the model can be easily deducted and represented as follows:

$$\begin{pmatrix} W_{ci} \\ p_i \end{pmatrix} = \begin{pmatrix} 0 & 0 & \frac{\eta_c}{c_p T_a} \frac{1}{\left(\frac{p_i}{p_a}\right)^{\mu} - 1} \\ 1 & 0 & 0 \end{pmatrix} \begin{pmatrix} p_i \\ p_x \\ P_c \end{pmatrix}$$

B. Dynamic models simulation

The simulation is made on an invariant state model, available around a functioning point defined by pi=133 kPa and px=141 kPa. This is a constant parameters model representation. To switch to a state model approach, we had to find a static functioning point. This was chosen after the previous simulation results. The point stands at an intake pressure of 133 kPa and an exhaust pressure of 141 kPa. The initial point was considered zero. The state model, invariant in time, was analytically deducted, by following the differential equations of the initial model and after grouping the terms in order to obtain the following form:

$$\begin{pmatrix} \dot{p}_i \\ \dot{p}_x \\ \dot{P}_c \end{pmatrix} = A \begin{pmatrix} p_i \\ p_x \\ P_c \end{pmatrix} + B \begin{pmatrix} A_r \\ x_v \\ N \\ W_f \end{pmatrix}$$
$$\begin{pmatrix} W_{ci} \\ p_i \end{pmatrix} = C \begin{pmatrix} p_i \\ p_x \\ P_c \end{pmatrix} + D \begin{pmatrix} A_r \\ x_v \\ N \\ W_f \end{pmatrix} \tag{9}$$

One can now easily observe pi, px and Pc, the vector of command measures made of EGR area (depending of the position of the EGR valve) Ar, the turbine's blades position xv, engine speed N, as well as the fuel flow Wf (still considered zero). The process outputs are the compressor air flow WciW_{ci} and the intake pressure pip_i.

In these conditions, the A, B, C and D matrixes become:

$$A = \begin{bmatrix} -3.625 & 0 & 373.0841 \\ 35.3698 & -93.4776 & 0 \\ 0 & 0.0260 & -9.0909 \end{bmatrix}$$

$$B = 1.0e+005 * \begin{bmatrix} 0.5714 & 0 & -0.000003 & 0 \\ -5.5755 & 0.1020 & 0.00003 & 1.4608 \\ 0 & -0.0002 & 0 & 0 \end{bmatrix}$$

$$C = \begin{bmatrix} 0 & 0 & 0.0249 \\ 1 & 0 & 0 \end{bmatrix}$$

$$D = \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix}$$

The model was simulated using State-Space Model which works by relations to the characteristics matrixes A, B, C, D. The initial state is considered null. The output vector consisting of the turbine air flow and the intake pressure corresponds to the simulated data due to the variable parameters model.

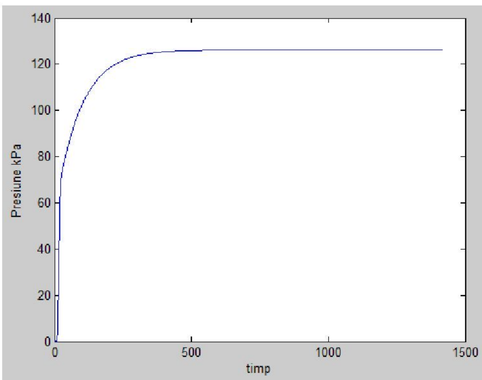

Fig 4. Engine model output – intake pressure

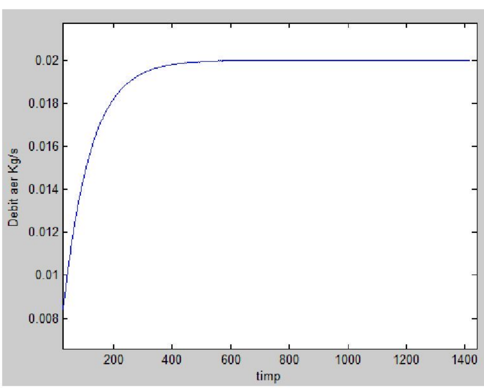

Fig 5. Engine model output – compressor airflow

The model respects the imposed levels of pressure and airflow. The model answer is approximately 1 second (the time axis is illustrated by executed sections with a 300 step per second), and the curve is asymptotic. The Ar and xv parameters are constant. We still need to make further improvement of the model answer, so we need to design and simulate a square linear controller.

IV. DE CONTROLLERS DESIGN AND SIMULATION

A. LQR Controller design and simulation

We take into consideration the dynamic state model represented in (9).

In order to obtain the square linear controller for the invariant model we used the platform provided by Mat Lab Simulink, which has already implemented the function *lqr()* that allows the calculus of a square linear controller in relation to the system's characteristics.

$$[K, S, E] = LQR(A, B, Qn, Rn)$$

where KK is the optimal matrix, QnQ_n and RnR_n are sharing matrixes and AA and BB are the model description matrixes. Here's how we've chosen the matrixes:

$$Q_n = \begin{bmatrix} 0.0001 & 0 & 0 \\ 0 & 0.01 & 0 \\ 0 & 0 & 10 \end{bmatrix} \qquad R_n = \begin{bmatrix} 100 & 0 & 0 & 0 \\ 0 & 100 & 0 & 0 \\ 0 & 0 & 100 & 0 \\ 0 & 0 & 0 & 100 \end{bmatrix}$$

The closed loop system controller matrix is:

$$K = \begin{bmatrix} 0.0002 & -0.0095 & -0.0854 \\ 0.00003 & 0.0002 & -0.0986 \\ 0 & 0.0000 & 0.0000 \\ 0.0009 & 0.0026 & 0.0427 \end{bmatrix}$$

The closed loop model behavior can be approximated with the help of matrix An:

$$A_n = A - B * K$$

Thus resulting the matrix:

$$A_n = 10^4 * \begin{bmatrix} -0.0016 & 0.0542 & 0.5251 \\ 0.0023 & -0.5765 & -5.2822 \\ 0.00007 & 0.00002 & -0.0011 \end{bmatrix}$$

The system's response is shown in picture 6 and 7.

One can notice the much faster response rate, of approximately 0.5 seconds for both outputs, regardless of an override of 10%, pretty high if to consider the performances obtained in open loop. The results are directly linked to the choosing of the Qn and Rn matrixes, so through the careful variation of the calibration matrixes parameters, one can improve the performances. We used a step of 20% for the xv parameter and a step of 50% for the Ar parameter.

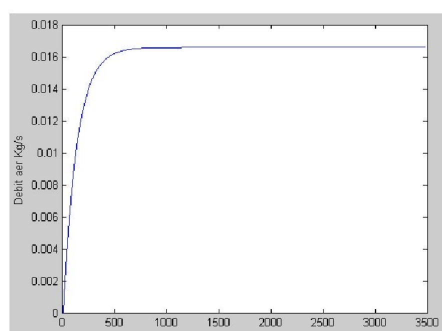

Fig 6. Closed loop system output – compressor air flow

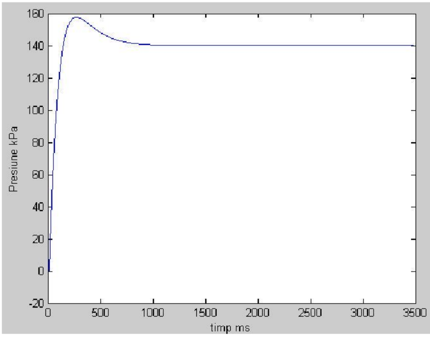

Fig 7. Closed loop system output - pressure

B. *RST Controller design and simulation*

For the implementation of a RST control algorithm, the invariable model must be transformed into a matrix of transfer functions. This description allows the control of each input over each desired output. This principle allows the possibility of controlling a single interest measure using a SISO command state.

Because an invariable model with 4 inputs and 2 outputs holds no less than 8 trasfer functions that can describe its functionality through SISO systems, and the effect that the EGR valve has over the turbine air flow is low, we've decided that the engine speed N and the fuel flow Wf implications are null. So the only functions that make the object of further study and further control are those that link the turbine blades position and the function that describes the effect of the EGR valve position has over the intake pressure.

Those discrete functions are:

$$Hd_{EGR-p_i}(z) = \frac{280.8 * z^3 - 102.9 * z^2 - 273.6 * z - 99.22}{z^3 - 2.271 * z^2 + 1.617 * z - 0.3458} = H_1$$

$$Hd_{VGT-w_{ci}}(z) = \frac{0.03711 * z^3 - 0.0785 * z^2 + 0.09375 * z + 0.02185}{z^3 - 2.271 * z^2 + 1.617 * z - 0.3458} = H_2$$

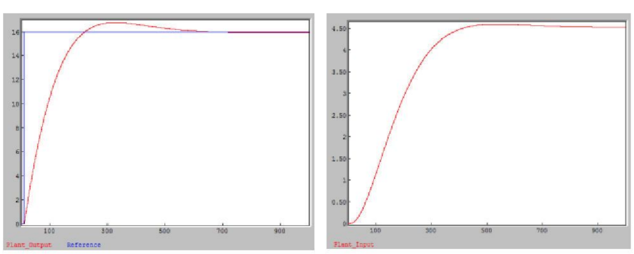

Fig 8. The H1 model response linked with a RST command

The sampling period is 0.01 seconds. For the controlling performances we used w0=2, xi=0.8 and for the follow up performances w0=1, xi=0.8. The answer shows an override of approximately 4.5% and a controller response of 2 seconds. The simulation was made in Win Reg.
The designated H1 controller, in polynomial representation is:

R = [-0.007, 0.016, -0.014, 0.0059, -6.184e-04]
S = [1, -0.9739, -1.2290, 1.3804, -0.1774]
T = [0.284, -0.559, 0.275]

and the polynomials of the trajectory generator are:
Am = [-1.984, 0.984]
Bm = [4.973e-05, 4.947e-05]

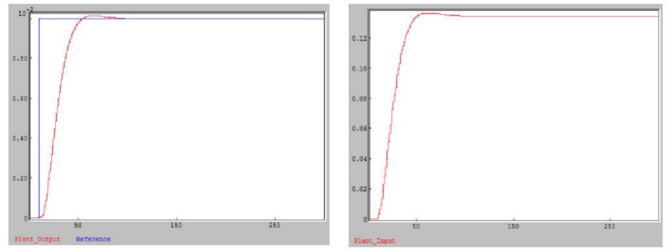

Fig 9. The H2 model response linked with a RST command

This case takes in consideration the xv input and the Wci output. Due to the physical link between the two, there is a direct correlation shown here. The response is very fast, with a transitorily time of only 0.5 seconds and an override less than 2% at a sampling period of 0.01 seconds. For the controlling performances we used w0=5, xi=0.8 and for the follow up performances w0=10, xi=0.8. The simulation was made in WinReg.

The designated H2 controller, in polynomial representation is:
R = [-22.9276, 49.6353, -40.4257, 16.2586, -2.5083]
S = [1, -2.0698, 1.622, -0.3938, -0.1584]
T = [13.4752, -25.8821, 12.4392]

and the polynomials of the trajectory generator are:
Am = [-1.8429, 0.8521]
Bm = [0.0047, 0.0044]

V. CONCLUSION

In this paper, an approach is considered for modeling and control of a turbocharged diesel engine with EGR valve. The dynamic state and I/O models have been developed. Then, state and RST polynomial I/O controllers have been computed. As shown, the controllers presented are a good way to control a diesel engine and offer suitable performances for the diesel engine functioning. In simulation studies it is shown that the proposed techniques can improve the dynamic behavior of a diesel engine. Future work involves the implementation of an explicit multi model controller on a diesel engine.

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