DECENTRALIZED AND MULTIVARIABLE DESIGNS FOR EGR-VGT CONTROL OF A DIESEL ENGINE

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Abstract: This paper is a study into the potential benefits of a coordinated EGR-VGT control strategy for a high speed diesel engine equipped with EGR and a variable nozzle geometry turbocharger (VGT). Traditionally, control strategies for this problem use SISO techniques or use one actuator at a time. Since the effect of the EGR and VGT actuators is coupled through the pressure in the exhaust manifold, it can be expected that a coordinated approach will yield a performance benefit. We will investigate in this paper to what extent this claim holds true.

Keywords: nonlinear control, engine control, diesel engines

1. INTRODUCTION

In recent years, more stringent requirements on performance, fuel economy and emissions have led to increasingly complicated engine configurations. Techniques like exhaust gas recirculation and turbocharging have been devised to face the stringent requirements. They give a great deal of freedom to control the behavior of the engine. Current practice often exploits these techniques only in a suboptimal way, since the devices used to control these features affect many different parts of the engine, through the cross-couplings in the system. The development of an optimal coordinated strategy often takes more time than available in a production cycle. Conventional strategies often resort to treating these devices and their desired effects as single input-single output systems, or by using only one actuator at a time, thereby ignoring the coupled nature of the system, (Truscott and Porter 1997, Dekker and Sturm 1996, Acton *et al.* 1994, Watson and Banosilo 1988, H. Jelde and Woelke 1997). In order to fully exploit the potential of these devices, one needs to consider the problem in the context of multivariable control. It is expected that a multivariable approach allows for better performance. Since a complete model based controller calibration is unrealistic with the present state of the art, and on-vehicle tuning cannot be bypassed, this multivariable strategy comes at the cost of a larger number of controller gains to tune. Typically, the gains of diesel engine controllers are scheduled on speed and fuel quantity (fuel may be replaced by load or boost). Hence, each controller gain results in as many calibration values to be tuned, as there are speed-load points in the scheduling grid. In this paper we investigate the performance benefits of a multivariable control strategy for EGR-VGT control of a high speed diesel engine, compared to a decentralized (SISO) approach.

2. ENGINE MODEL

We developed a mean value model for the diesel engine, which takes into account the gas flow dynamics of the engine, but ignores sub-cycle and in-cylinder events. The derivation of mean value models for diesel engines has been described in detail in (Kao and Moskwa 1995, Tsai and Goyal 1986, Jensen *et al.* 1991, Amstutz and Re 1995), hence we will restrict ourselves to the resulting equations. The general diesel engine configuration is depicted in Figure 1. The particular configuration studied here does not have the intercooler and the EGR throttle.

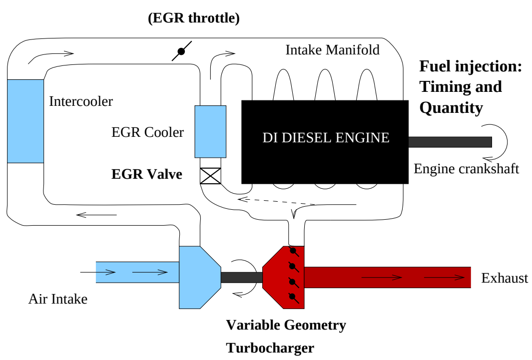

Fig. 1. Generic diesel engine configuration,

To model the engine, we use a combination of first principles, and static maps identified from experimental data. Some simplifications concerning the thermodynamics are made. We assume that gas thermodynamic properties do not change with composition and temperature, that pressure and temperature are uniform over a volume and, since the flow velocities are small, we ignore differences between static and total enthalpy. Based on these assumptions the gas dynamics of the intake and exhaust manifolds are described by the ideal gas law, and their respective mass and energy balances. For each volume we end up with three equations:

$$\begin{aligned}\dot{m} &= \sum W_{in} - \sum W_{out}\\ \dot{p} &= \frac{R\gamma}{V}\left(W_{in}T_{in} - W_{out}T\right)\\ \dot{F} &= \frac{1}{m}\sum W_{in}(F_{in} - F),\end{aligned} \tag{1}$$

where p is the pressure, V is the (constant) volume, m is the mass of gases in the volume, T is the temperature in Kelvin, $R = 0.2870$ kJ/kg/K is the gas constant, $\gamma = \frac{c_p}{c_v}$ is the ratio of specific heats, F is the combustion product fraction, and W represents a mass flow. Flows through valves are modelled with the orifice flow equation

$$\begin{aligned}W &= C_d(u)\frac{p_2}{\sqrt{RT_2}}\Psi\left(\frac{p_1}{p_2}\right)\\ \Psi(r) &= \sqrt{\frac{2\gamma}{\gamma-1}\left(r^{\frac{2}{\gamma}} - r^{\frac{\gamma+1}{\gamma}}\right)} \text{ for } r > \left(\tfrac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}\\ \Psi(r) &= \gamma^{\frac{1}{2}}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma-1)}} \text{ for } r \le \left(\tfrac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}},\end{aligned} \tag{2}$$

where p_1 is the downstream pressure, p_2 is the upstream pressure, T_2 is the upstream temperature, and $C_d(u)$ is the effective flow area which is dependent on an actuator (valve) setting u.

The compressor and turbine are each modelled with an expression for the flow, the outlet temperature and the power transfer from or to the rotor. For the turbine this takes the form:

$$\begin{aligned}W_t &= \phi_t\frac{p_2}{\sqrt{T_2}}\\ P_t &= W_t c_p T_2 \eta_t \left(1 - \left(\frac{p_x}{p_2}\right)^{\frac{\gamma-1}{\gamma}}\right)\\ \eta_t &= \eta_t\left(\theta_{vgt}, N_{tc}, T_2, \frac{p_x}{p_2}\right)\\ \phi_t &= \phi_t\left(\frac{p_x}{p_2}, \theta_{vgt}\right),\end{aligned} \tag{3}$$

where the subscript t refers to the turbine, *2* refers to the exhaust manifold, x refers to post turbine, N_{tc} is the turbocharger speed, P_t is the power transferred to the turbine and θ_{vgt} is the VGT vane setting, η is an efficiency, and ϕ is a mass flow parameter. For the compressor the equations take the form:

$$\begin{aligned}W_c &= \phi_c\frac{p_a}{\sqrt{T_a}}\\ T_c &= T_a\left(1 + \frac{1}{\eta_c}\left(\left(\frac{p_1}{p_a}\right)^{\frac{\gamma-1}{\gamma}} - 1\right)\right)\\ P_c &= W_t c_{pa}(T_c - T_a)\\ \eta_c &= \eta_c(N_t, T_a)\\ \phi_c &= \phi_c\left(\frac{N_t}{\sqrt{T_a}}, \frac{p_1}{p_a}\right),\end{aligned} \tag{4}$$

where the subscript c stands for the compressor, a stands for the airbox, and *1* stands for the intake manifold. Finally, the turbine speed is integrated from power transferred to the rotor

$$\dot{N}_{tc} = \frac{P_t - P_c}{I_{tc} N_{tc}}, \qquad (5)$$

where I_{tc} is the inertia of the rotor.

Among the static maps used to model the engine are a temperature rise map, a torque map, a map for the effective area of the EGR valve, volumetric efficiency, EGR cooler effectiveness, and maps for the efficiency and mass flow parameter of the compressor and turbine.

In this paper we look at the system with EGR and VGT actuators as inputs and mass air flow (MAF, kg/h) and intake manifold air pressure (MAP, kPa) as outputs. The inputs are duty cycles (in percent) to the vacuum controlled actuators (CVTs). The effect of an increase in EGR duty cycle is to open the valve, resulting in more EGR flow, whereas the effect of an increase in VGT duty cycle is to close the vanes, resulting in higher boost pressure. In (Moraal *et al.* 1997, Kolmanovsky *et al.* 1997) it was shown that the engine model thus obtained is highly nonlinear. The steady state gain varies two orders of magnitude over the operating range. The steady state gain from EGR to MAP and VGT to MAF can be positive or negative depending on the operating point. The system from EGR to MAF and VGT to MAP can be non minimum phase. Analysis of the DC gain over the operating range shows that EGR has a dominant effect on MAF, and VGT a dominant effect on MAP. Hence the industrial practice to control MAF with EGR and MAP with VGT.

3. THE CONTROL PROBLEM

Since PID controllers are widely used in the automotive industry, and have an intuitively attractive interpretation, we restrict our attention to PID controllers. We assume that performance can be expressed as setpoints for the mass air flow into the intake manifold, and for the boost pressure. The generation of these setpoints is by no means a trivial task, and depending on the particular performance objective, different setpoints will result. In general, one wants to make sure that enough air is available for combustion without generating smoke, and that intake manifold conditions are such as to not exceed emission limits. The pumping losses are proportional to the difference in pressure between intake and exhaust manifold. For a given speed-load point, this difference has a minimum for medium boost pressure. Therefore, excessive boost pressure may result in poor fuel economy due to pumping losses, whereas too low a boost pressure will restrain accelerations due to lack of air. The mass air flow indirectly governs the EGR flow, in the sense that setpoints for boost and mass airflow for a given operating point uniquely determine the nominal actuator settings, and hence the setpoint for EGR flow. The EGR rate is of great importance to reduce NOx emissions. In this paper, we opt for setpoints generated to achieve minimum specific fuel economy (fuel per kWh), subject to an air fuel ratio constraint to prevent visible smoke. Both the MAF and MAP setpoints and the controller gains are scheduled on engine speed and injected fuel. One common approach to the EGR-VGT control problem is to use the EGR valve to control the MAF setpoints, and the VGT to control the boost setpoints. Both controllers are typically PID controllers. Due to the coupled nature of the system, it is expected that a multivariable approach, PID or otherwise, will yield better performance. To compare the two, we use the same design technique to design SISO and MIMO PI (not PID) controllers. The control design technique used in this paper, for purpose of illustration, is the fixed structure approach, (Takashi *et al.* 1997). We set up the control problem as a norm minimization problem, and use numerical optimization to minimize that norm over a set of PI gains. In particular, we formulate the problem as a weighted sensitivity $\mathcal{H}_\infty$ minimization problem as depicted in Figure 2. This formulation penalizes tracking errors in MAF and MAP and actuator effort. The weighting functions W_{track} and W_{act} allow us to tune the bandwidths of tracking error and actuator effort, and the magnitude of the steady state tracking errors and actuator effort. Alternatively, the weighting function W_{act} can be interpreted as input multiplicative plant uncertainty. The weighting functions W_{track} have high gain at low frequency, and low gain at high frequency, indicating that we want good steady state tracking, and relax the tracking requirements for high frequency signals. The weighting functions W_{act} are have low gain at low frequency and high gain at high frequency, indi-

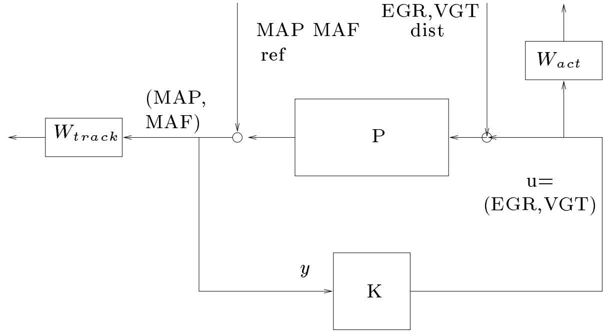

Fig. 2. $\mathcal{H}_\infty$ formulation of EGR/VGT control problem

cating that we do not have actuator authority at high frequency, or alternatively, that we have high plant uncertainty at high frequency. Figure 2 does not show the MAP and MAF tracking signals to be generated by fuel and RPM. We do this to make the controller design independent of the setpoints.

We use the Matlab *constr* function (Grace 1995) to locally minimize the $\mathcal{H}_\infty$ norm of the linearized system depicted in Figure 2. We add the constraint that the linearization of the closed loop system must be stable, and to facilitate the optimization, we add sign constraints to the gains from MAF to EGR and from MAP to VGT. As observed above, these gains are dominant, and their required sign can be determined from the steady state gain of the plant. If we have a positive MAF error, for example, measured MAF is higher than desired, hence we need to open the EGR valve to decrease MAF, which requires an increase in duty cycle. This indicates that the gains from MAF error to EGR valve duty cycle should be positive. By a similar argument, the gains from MAP error to VGT duty cycle should be negative. The optimal off-diagonal gains can change sign depending on the operating point, as a function of the steady state gain matrix of the plant. The initial condition of the optimization was chosen as a negative scalar times the sign of the dc-gain for the diagonal gains, and zeros for the off-diagonal gains. This guarantees that the initial condition satisfies the sign constraint imposed above, and leaves the off-diagonal gains free to take their sign depending on the operating point. The sign constraint and the stability constraint make this problem discontinuous in the first derivative, hence the Matlab *constr* function seemed an appropriate

	SISO	MIMO
cost	2.3628	1.3328
L_2 MAP error	1.3652	1.3263
L_2 MAF error	3.6052	2.9188

Table 1. Comparison of errors for SISO and MIMO controllers at a mid load, mid speed point.

choice, since it bypasses the need for analytic gradient computations. We apply this procedure to the linearization of the plant at $N = 2500$ and $W_f = 6$ kg/h. Figure 3 shows that the resulting responses of the SISO and MIMO controllers are quite similar. Table 1 shows some statistics regarding these plots: the value of the cost function after minimization, and the L_2 errors of the MAP and MAF responses resulting from a fuel step profile as depicted in Figure 3. Indeed, the values are quite close. Inspection of the gain matrices resulting from the optimization shows us that the off diagonal gains (i.e. from VGT to MAF and EGR to MAP) are very small, explaining the similarity of the responses. These results seem to indicate that a SISO controller matches the performance of a MIMO controller, at least in part of the operating range. Experiments on the real engine validated this conclusion. We also experimented with using an $\mathcal{H}_2$ norm instead of an $\mathcal{H}_\infty$ norm, and with adding derivative terms to the controller. The resulting controllers performed in a similar manner. For the $\mathcal{H}_2$ norm, the direct feedthrough D-term was initially set to zero for the $\mathcal{H}_2$ norm computation, and then added at the end as the sum of its singular values.

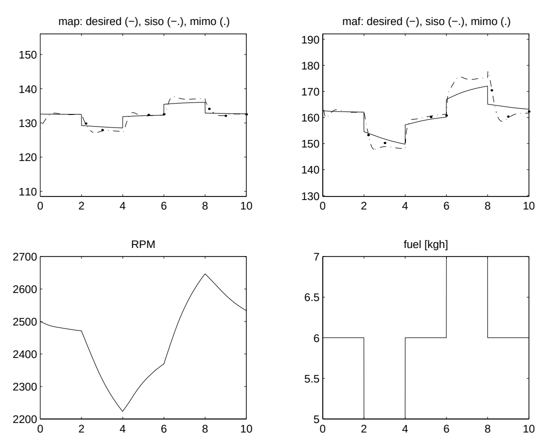

Fig. 3. Simulated MAF and MAP responses at $W_f = 6$ kg/h, RPM = 2500

4. COORDINATED CONTROL IN THE LOW SPEED, LOW LOAD RANGE

In this section we will investigate under what conditions coordinated EGR-VGT control might be beneficial. One prime concern for performance is to get fast MAF dynamics. The fuel quantity we can inject into the cylinders is limited by the available air. If the increase in MAF resulting from a tip-in is fast, than we get fast torque response, and hence good acceleration. In the mid-speed/mid-load region, the exhaust gas has enough energy to guarantee a fast increase in MAF, just by the actions of increased RPM and fuel alone. The acceleration of the engine and the increased power from the fuel help to increase the mass air flow. Only at low load and low engine speed, the rise in MAF is slower. As mentioned above, the MAF setpoints are generated by table lookup from fuel and RPM, so we are really dealing with a system with three inputs. The third input is fuel, and has a much larger gain at higher power levels. This can be checked in Figure 4, which shows experimental MAP and MAF responses in a medium and a low speed/load range. In the low power range, we can discern more delay in MAF response than in the medium power range.

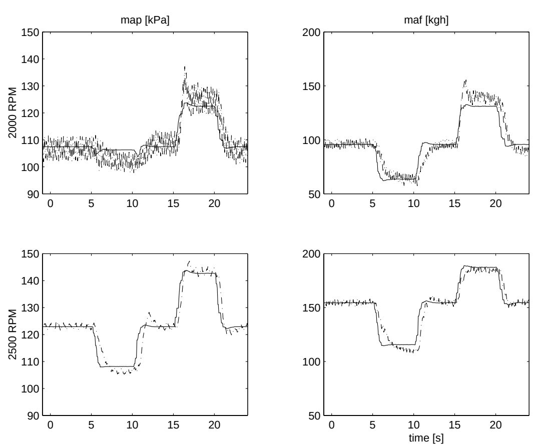

Fig. 4. Experimental MAP and MAF responses at low speed (upper) and medium speed (lower).

In the low speed/low load region we can use a coordinated approach to speed up the MAF response. By using a nonzero term from VGT to MAF in the controller, we let both the EGR valve and the VGT vanes work to increase MAF, albeit at the expense of the MAP response. In the low speed/low load regime, the steady state gain from VGT to MAF is negative, i.e., opening the VGT will increase MAF, but also decrease MAP. Since this sign changes outside of the low speed/low load range, and the exact location of the sign change is uncertain, one should limit this coordinated strategy to low speed and low load.

Figure 5 shows simulation results of the coordinated strategy at 2000 RPM and a fuelling staircase profile centered at 3 kg/h. The left column shows MAP, the right column shows MAF. The first row shows the response of MAP and MAF when the strategy is uncoordinated. The delayed MAF response for low fuel is clearly visible. The second row shows the responses when the VGT-MAF term is added indiscriminately. This leads to a fast MAF response at low loads, but also to high overshoot for higher loads. The third row shows the response of MAP and MAF when the VGT-MAF term is applied for low loads only. The latter strategy allows one to keep the naturally fast MAF and MAP response for higher loads, while speeding up the MAF response for low loads. It can also be seen that the boost response deteriorates with improving MAF response.

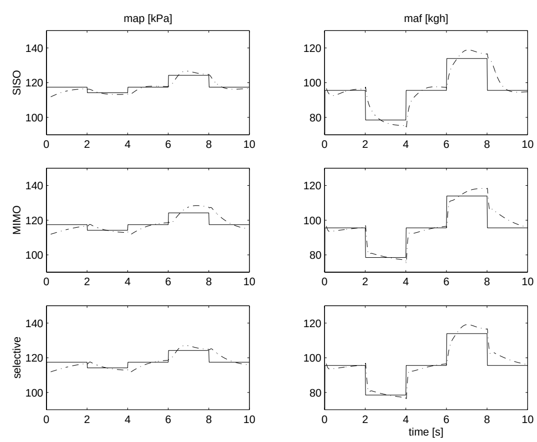

Fig. 5. Simulated MAP and MAF responses for coordinated control. First row: no coordination for all loads. Second row: coordination for all loads. Third row: coordination for low loads only.

Figure 6 shows experimental results of the same fuel profile at the same speed. Although the exact numerical behavior is different than in the simulation, the qualitative behavior and the progression between the strategies is the same as in the model. In the first row, it can be observed that the MAF response for higher fuelling levels is much faster than for lower fuelling levels. The experimental boost response exhibits a lot of overshoot with the coordinated controller.

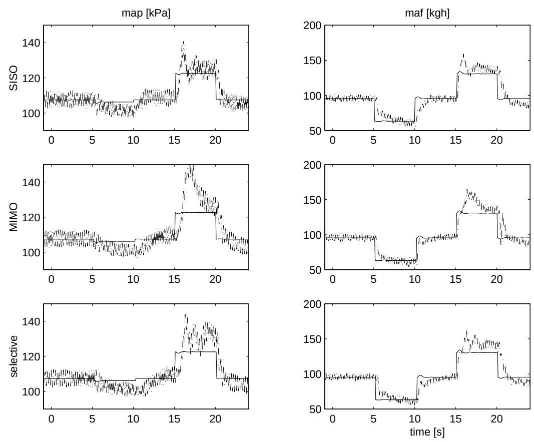

Fig. 6. Experimental MAP and MAF responses for coordinated control. First row: no coordination for all loads. Second row: coordination for all loads. Third row: coordination for low loads only.

5. CONCLUSIONS

This paper investigated the benefits of coordinated VGT-EGR control for a high speed diesel engine. While in most operating regimes, the coordinated strategy gave only very small improvement, it was shown that in low speed/low load regions, a coordinated approach could yield a faster transient MAF response, and hence a more rapid acceleration.

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