PID CONTROLLERS AND THEIR TUNING FOR EGR AND VGT CONTROL IN DIESEL ENGINES

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Abstract: A PID structure is proposed and investigated for coordinated control of EGR-valve and VGT-position in heavy duty diesel engines. Control goals are to fulfill the legislated emission levels and safe operations of the engine and the turbocharger. These goals are achieved through regulation of the following performance variables: normalized air/fuel ratio λ, intake manifold EGR-fraction as well as turbocharger speed. A systematic tuning strategy for the PID controllers is also developed and the tuning rules and their performance is successfully illustrated on a demanding part of the European Transient Cycle. Further, it is demonstrated that the VGT-position to turbocharger speed loop does benefit from a derivative part in order to predict high turbocharger speeds. This is due to the large time constant in the corresponding open-loop transfer function.

Keywords: emission control, air-fuel ratio control, turbocharger speed control

1. INTRODUCTION

Legislated emission limits for heavy duty trucks are constantly reduced. To fulfill the requirements, technologies like Exhaust Gas Recirculation (EGR) systems and Variable Geometry Turbochargers (VGT) have been introduced. To reach the legislated emissions limits, primarily NO_x and smoke, it is necessary to have coordinated control of the EGR and the VGT. Various control approaches regarding emission control have been published and can be found in Jankovic *et al.* (1998), Nieuwstadt *et al.* (2000), Stefanopoulou *et al.* (2000), and Rückert *et al.* (2004). In this paper the approach and goal is to develop a simple control structure together with systematic tuning rules that an engineer can use on an engine test bench to tune the controllers.

1.1 Foundation for the controller

Load control is necessary, since the drivers demand must be actuated, and this is achieved through basic fuel control using feed forward. The primary emission reduction mechanisms utilized in the controller is that NO_x can be reduced by increasing the intake manifold EGR-fraction and smoke can be reduced by increasing λ (Heywood, 1988). Thus the emission limits are formulated as set-points in EGR-fraction and λ. It is also important to monitor and control turbocharger speed since aggressive transients can cause damage through overspeeding.

1.2 Control objectives

The engine torque, normalized air/fuel ratio λ, intake manifold EGR-fraction and turbocharger

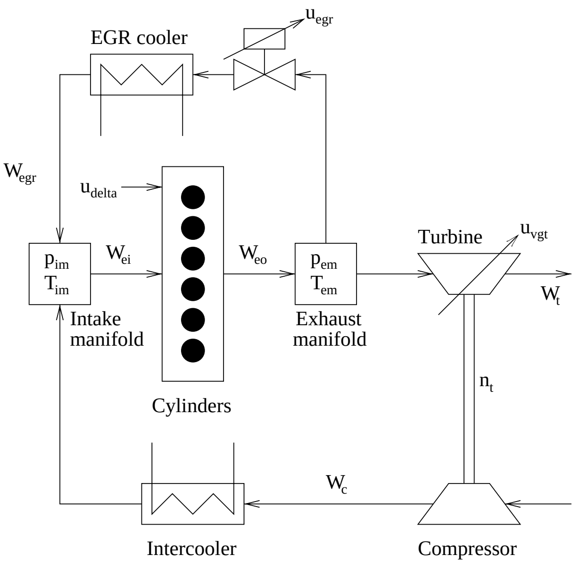

Fig. 1. The structure of the diesel engine model.

speed are chosen as performance variables to be controlled. To follow a driving cycle while maintaining low emissions and suitable turbocharger speeds during a test cycle the performance variables above are given following control objectives.

(1) Follow the set-point engine torque from the driving cycle.
(2) λ is not allowed to go below a minimum limit, otherwise there will be too much smoke.
(3) Minimize the error between the EGR-fraction and its set-point. There will be more NO_x if the EGR-fraction is too low and there will be more smoke if the EGR-fraction is too high.
(4) The turbocharger speed is not allowed to exceed a maximum limit, otherwise the turbocharger can be damaged.

The aim of this paper is to achieve these control objectives using a PID structure, and to develop a tuning method for the PID parameters. The tuning strategy is developed and illustrated using simulations of the closed-loop system.

2. DIESEL ENGINE MODEL

The structure of the model can be seen in Figure 1. The engine model is a mean value model with three main states: intake manifold pressure p_{im}, exhaust manifold pressure p_{em}, and turbocharger speed n_t.

The inputs to the model are fueling rate u_δ, EGR-valve position u_{egr}, VGT actuator position u_{vgt}, and engine speed n_e. The EGR-valve is closed when $u_{egr} = 0\,\%$ and open when $u_{egr} = 100\,\%$. The VGT is closed when $u_{vgt} = 100\,\%$ and open when $u_{vgt} = 0\,\%$.

The model has been verified against test cell measurements and gives good agreement. For a detailed description of the modeling and validation see Wahlström (2004).

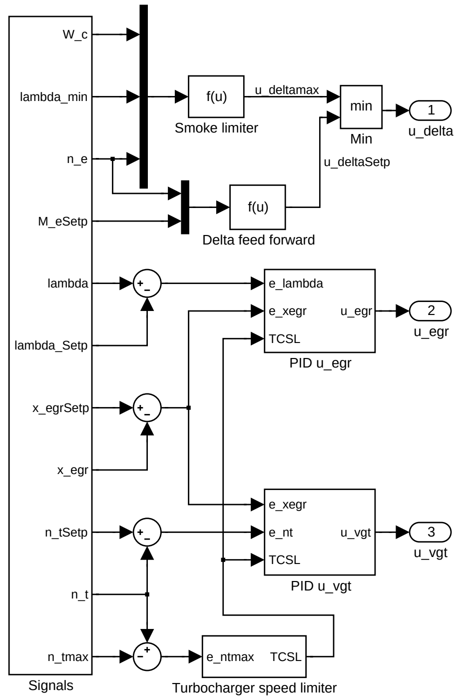

Fig. 2. The control structure in MATLAB/SIMULINK.

3. CONTROL STRUCTURE

The control design objective is to coordinate u_δ, u_{egr}, and u_{vgt} in order to achieve the control objectives in Section 1.2 using a PID structure. In Figure 2 a MATLAB/SIMULINK schematic of the control structure is shown.

3.1 Signals

The signals needed for the controller can be seen in the block "Signals" in Figure 2. Compressor air mass flow W_c, engine speed n_e, and turbocharger speed n_t are measured. It is assumed that λ and x_{egr} can be estimated using observers. The observer design is very important, but it is not the focus in this paper.

The set-points and limits for the controllers are obtained from stationary measurements with emissions just below the legislated requirements, and represented as look-up tables as a function of operation condition. In the tuning section 4 these set-points are held constant.

3.2 Feed forward

The engine torque is controlled to its set-point M_{eSetp} with the control signal u_δ using feed forward. The block "Delta feed forward" in Figure 2 calculates the set-point value for u_δ.

$$u_{\delta Setp} = c_1\, M_{eSetp} + c_2\, n_e^2 + c_3\, n_e + c_4$$

The parameters c_1 to c_4 are estimated from stationary measurements.

3.3 Feedback loops

λ is controlled to a set-point λ_{Setp} with the control signal u_{egr}. This causes a closing of the EGR-valve during a load transient in order to speed up the turbocharger and increase the air mass flow into the engine. It is not convenient to control λ with u_{vgt} with a linear control strategy since the corresponding transfer function has a sign reversal when the operating point is changed (Kolmanovsky *et al.*, 1997).

The intake manifold EGR-fraction x_{egr} is controlled to its set-point $x_{egrSetp}$ with the control signal u_{vgt}. This feedback loop causes a closing of the VGT during a load transient in order to speed up the turbocharger.

If the VGT is closed too much n_t may exceed the maximum limit n_{tmax}. This can be avoided by controlling n_t with u_{vgt} to a set-point n_{tSetp} which has a value slightly lower than the maximum limit. The appropriate value for u_{vgt} is then the smallest value of the outputs from the two different controllers.

In some operating points there is too much EGR, although the VGT is fully open. Therefore x_{egr} is also controlled by the control signal u_{egr}. The appropriate value for u_{egr} is then the smallest value of the outputs from the two different PID controllers.

The two PID controller blocks in Figure 2 have the following main equations

$$\begin{aligned} u_{egr} &= \min\left(\text{PID}\left(e_\lambda\right), \text{PID}\left(e_{xegr}\right)\right) + TCSL \\ u_{vgt} &= \min\left(\text{PID}\left(e_{xegr}\right), \text{PID}\left(e_{nt}\right)\right) - TCSL \end{aligned} \tag{1}$$

The PID controllers are implemented in differential form with anti-windup. Each PID controller has the following structure

$$F(s) = K_j \left(1 + \frac{1}{T_{ij}\, s} + T_{dj}\, s\right) \tag{2}$$

where the index j is the feedback loop number according to the tuning order in Section 4.1.

3.4 Safety functions

Aggressive transients can cause λ to go below its lower limit and the turbocharger speed n_t to exceed its maximum limit. This is avoided by using safety functions.

The block "Smoke limiter" in the top of Figure 2 calculates the maximum value of u_δ in order to avoid overshoots in λ. The calculation is based on engine speed n_e, air mass flow through the compressor W_c, and lower limit of lambda λ_{min}.

$$u_{\delta max} = \frac{W_c\, 120}{\lambda_{min}\, (A/F)_s\, 10^{-6}\, n_{cyl}\, n_e}$$

The block "Turbocharger speed limiter" in the bottom of Figure 2 is a PD controller that calculates the signal $TCSL$ in order to reduce overshoots in n_t.

$$TCSL = \max\left(\text{PD}\left(e_{ntmax}\right), 0\right)$$

Apart from no integral part, the PD controller has the same structure as Equation (2). When the output from the PD controller is positive, $TCSL$ opens the EGR-valve and the VGT according to Equation (1).

4. CONTROLLER TUNING

The tuning parameters K_j, T_{ij}, and T_{dj} in Equation (2) are obtained by first initializing them (without derivative part) using the Åström-Hägglund step-response method for pole-placement (Åström and Hägglund, 1995). Then the parameters are fine tuned using the methods described below in order to achieve the control objectives.

4.1 Tuning order

After the initialization the following tuning order is applied:

(1) u_{egr} to λ loop (Section 4.3)
(2) u_{egr} to x_{egr} loop (Section 4.3)
(3) u_{vgt} to x_{egr} loop (Section 4.4)
(4) u_{vgt} to n_t loop (Section 4.4)
(5) $TCSL$ to n_t loop (Section 4.5)

This order follows the causality of the system, i.e. the order that the signals in Figure 2 start changing in a load transient

$$\begin{aligned} &M_{eSetp} \rightarrow u_\delta \rightarrow \lambda \rightarrow u_{egr} \rightarrow \\ &x_{egr} \rightarrow u_{vgt} \rightarrow n_t \rightarrow TCSL \end{aligned}$$

which is obtained by taking the smallest time constants of the different loops into consideration.

4.2 Fine tuning method

The final tuning is obtained by adjusting the parameters iteratively until the control objectives are achieved. The gain K_j is adjusted in order to change the speed of the controller and T_{dj} is adjusted in order to improve the performance of

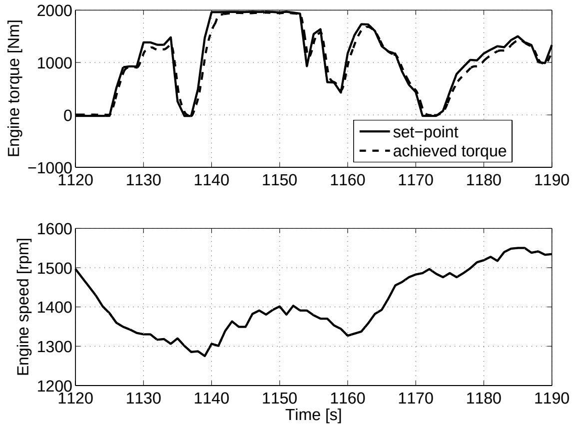

Fig. 3. A part of the European Transient Cycle (ETC) which is used for simulation of the closed-loop system and tuning of the parameters in the controllers. A critical part of the cycle is the aggressive load transient at 1139 s.

the closed-loop system (see Section 4.3-4.5). If it is possible, a derivative part is avoided. The tuning strategy for T_{ij} is to increase it until oscillations appear in the control signal, then decrease it until the oscillations disappear.

The rest of this section describes how the tuning results can be evaluated on an important part of the European Transient Cycle (ETC), see Figure 3. There are stationary errors between the set-point and the achieved torque since there is no feedback of the torque. But the errors are relatively small and have no effect on the results in this paper.

In Figures 4 to 8 K_j and T_{dj} represent the final result from the tuning. From this value the parameters are perturbed yielding information and showing important features in the signals that a control tuner should look for to determine if the parameters K_j and T_{dj} are too high or too low.

4.3 EGR-valve

The value of the control signal u_{egr} for the EGR-valve is based on both λ and x_{egr} according to Equation (1). Thus, there are two PID controllers that must be tuned for best performance of the EGR-valve control. No derivative part is needed in these controllers in order to predict the performance variables since the time constants in these loops are relatively small.

Normalized air/fuel ratio The closed-loop system is simulated for three different values of the gain K_1 in the u_{egr} to λ loop (see Figure 4). The tuning strategy is that the gain should be increased until the opening speed of the EGR-valve is saturated at the high load transients. If

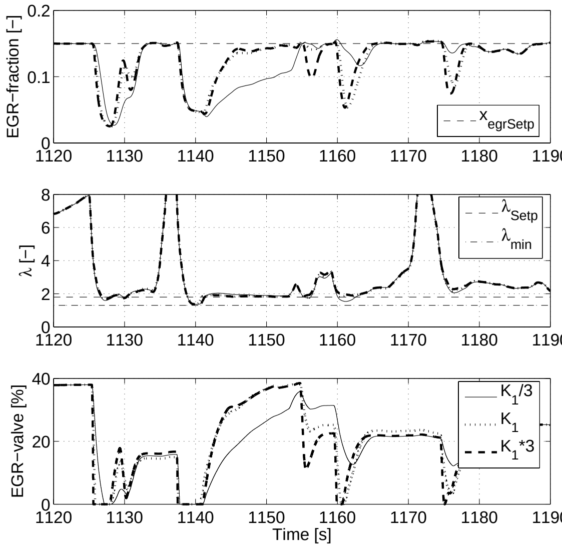

Fig. 4. Tuning of the gain K_1 in the EGR-valve to λ loop.

the gain is increased even more, there is too much closing at small load transients.

If the gain is decreased ($K_1/3$), the opening and the closing speed of the EGR-valve is decreased when λ is near the set-point. A slow opening is most obvious in a high load transient (between 1140 and 1153 s), since λ is low at high loads. This slow opening leads to a slow control of x_{egr}. A low value of the gain also leads to a slow closing of the EGR-valve at medium load transients (at 1160 s). This leads to lower acceleration of the turbocharger during the transient, a later opening of the EGR-valve and a slower control of x_{egr}.

A too high gain ($K_1 * 3$) in the u_{egr} to λ loop, leads to an aggressive closing of the EGR-valve at small load transients (at 1155 and 1175 s) that is not necessary, since λ is large enough. In addition, there are no improvements in the closing or opening speed at the other load transients.

EGR-fraction In Figure 5 the closed-loop system is simulated for three different values of the gain K_2 in the u_{egr} to x_{egr} loop. The tuning strategy is that the gain should be increased until the slope of the torque starts to decrease for high load transients that are close to each other.

If the gain is low ($K_2/3$), the opening of the EGR-valve is slow and therefore the control of x_{egr} is slow during a load transient. But if the gain is increased ($K_2 * 3$), the stationary value of u_{egr} is increased after a positive load transient (at 1134 s). This leads to a lower flow through the turbocharger, lower turbocharger speed and therefore less air for the next load transient. The result is a slower torque increase for example at 1139 s. This drawback only appears when the load transients are close to each other.

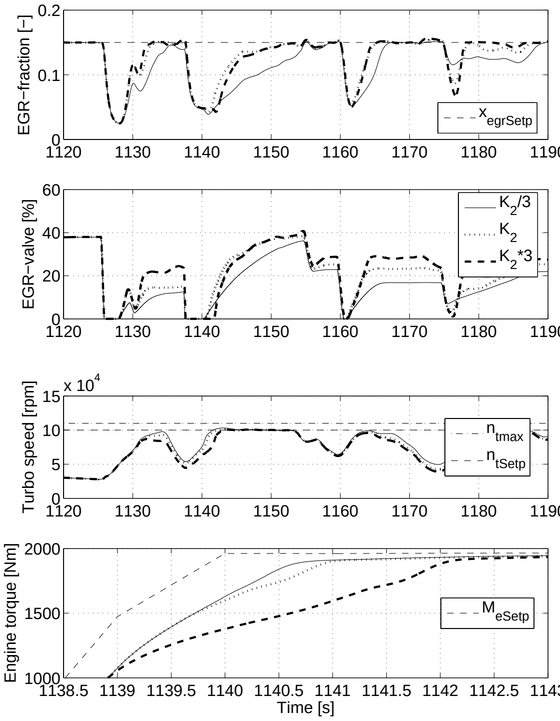

Fig. 5. Tuning of the gain K_2 in the EGR-valve to EGR-fraction loop.

4.4 VGT-position

There are two PID controllers that must be tuned for best performance of the VGT-position control, since the control signal u_{vgt} is based on both x_{egr} and n_t according to Equation (1).

EGR-fraction In Figure 6 the closed-loop system is simulated for three different values of the gain K_3 in the u_{vgt} to x_{egr} loop. The tuning strategy is that the gain should be increased until the closing speed of the VGT-position is almost saturated at the load transients. If the gain is increased even more, oscillations appear in the u_{vgt} signal. No derivative part is needed in order to predict x_{egr} since the time constant in this loop is relatively small.

The disadvantage with a low gain ($K_3/3$) is a slow closing of the VGT during a load transient. This leads to a slow control of x_{egr} (at 1130 s), a lower turbocharger speed, and a slower control of the engine torque (at 1139 s). The disadvantage with a high gain (K_3*4) is oscillations in the u_{vgt} signal, which can be seen at 1130 and 1157 s.

Turbocharger speed In Figure 7 the closed-loop system is simulated for three different values of the derivative time T_{d4} in the u_{vgt} to n_t loop. The tuning strategy is that the derivative time should be increased until the control speed of x_{egr} starts to decrease or just before oscillations start to appear in the u_{vgt} signal.

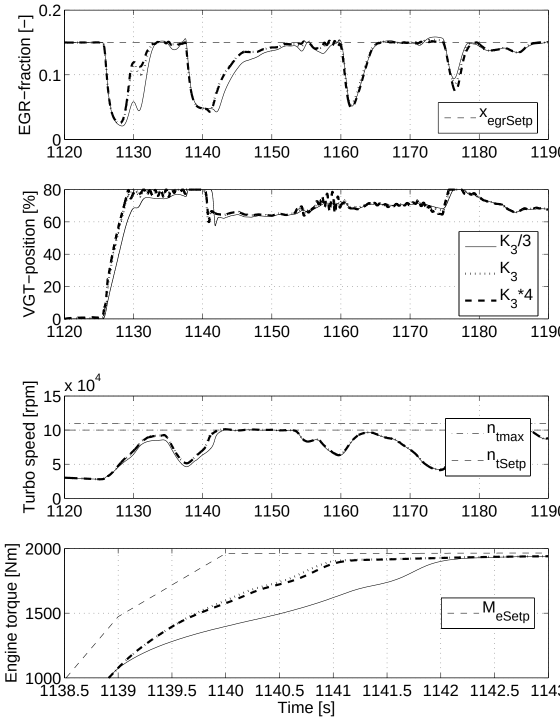

Fig. 6. Tuning of the gain K_3 in the VGT-position to EGR-fraction loop.

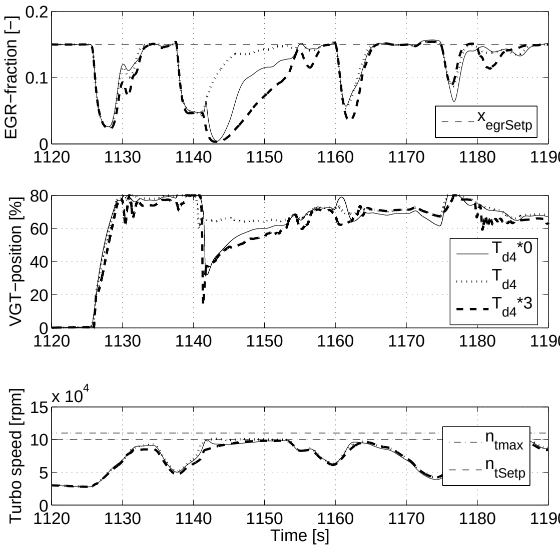

Fig. 7. Tuning of the derivative time T_{d4} in the VGT-position to turbocharger speed loop.

There must be a derivative part in the PID controller for the the u_{vgt} to n_t loop in order to predict the turbocharger speed, since the corresponding open-loop transfer function has a large time constant (about 2 s compared to 1 s for the other transfer functions that are used as feedback loops).

If no derivative part is used ($T_{d4} * 0$), u_{vgt} opens the VGT to late in order to decrease the turbocharger speed (at 1141 s), this leads to a high acceleration of the turbocharger and a positive $TCSL$ (the result is an opening of EGR-valve

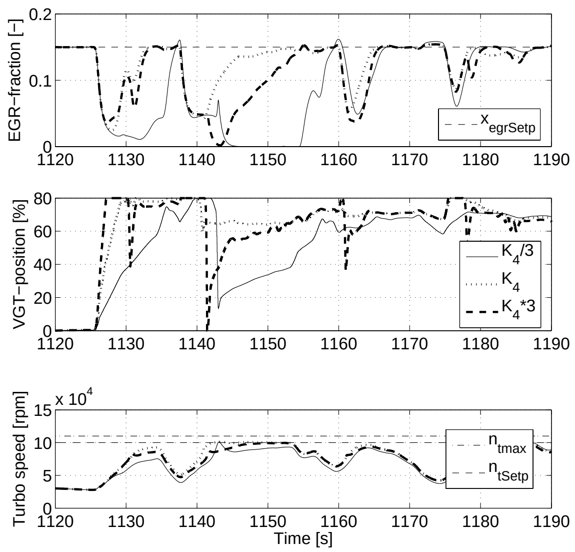

Fig. 8. Tuning of the gain K_4 in the VGT-position to turbocharger speed loop.

and VGT). The drawback with a high derivative part is slow control of x_{egr}, oscillations in the u_{vgt} signal and too much opening of the VGT (at 1141 s) that is not necessary

In Figure 8 the closed-loop system is simulated for three different values of the gain K_4 in the u_{vgt} to n_t loop. The tuning strategy for the gain is the same as for the derivative time, i.e. the gain should be increased until the control speed of x_{egr} starts to decrease or just before oscillations start to appear in the u_{vgt} signal.

4.5 Turbocharger speed limiter, TCSL

The turbocharger speed limiter consists of two parameters: the gain K_5 and the derivative time T_{d5}. The tuning strategy is that the gain and the derivative time should be decreased until just before the turbocharger speed starts to exceed its maximum limit during a high load transient when the u_{vgt} to n_t control is insufficient.

$TCSL$ is positive for the solid line in Figure 8 at 1143 s and the turbocharger speed is below the maximum limit.

5. CONCLUSION

A PID structure is proposed and investigated for control of air/fuel ratio λ and intake manifold EGR-fraction. These are chosen as performance variables since they are strongly coupled to the emissions. It is also important to control the turbocharger speed in order to avoid overspeeding. The key idea behind the structure is that λ is controlled by the EGR-valve and EGR-fraction by the VGT-position, which handles the sign reversal in the system from VGT to λ. A simple straightforward tuning strategy for the PID structure is also developed.

The proposed PID structure shows promising results in simulation on a demanding part of the European Transient Cycle. Further, it is demonstrated that the VGT-position to turbocharger speed loop does benefit from a derivative part in order to predict high turbocharger speeds. This is due to the large time constant in the corresponding open-loop transfer function.

6. ACKNOWLEDGMENTS

The Swedish Energy Agency are gratefully acknowledged for their financial support.

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