

Nonlinear Input Transformation for EGR and VGT Control in Diesel Engines

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ABSTRACT

In diesel engines with EGR and VGT, the gas flow dynamics has significant nonlinear effects. This is shown by analyzing DC-gains in different operating points showing that these gains have large variations. To handle these nonlinear effects, a nonlinear state dependent input transformation is investigated. This input transformation is achieved through inversion of the models for EGR-flow and turbine flow. It is shown that the input transformation handles the nonlinear effects and decreases the variations in DC-gains substantially. The input transformation is combined with a new control structure that has a pumping work minimization feature and consists of PID controllers and min/max-selectors for coordinated control of EGR-fraction and oxygen/fuel ratio. The EGR flow and the exhaust manifold pressure are chosen as feedback variables in this structure. Further, the set-points for EGR-fraction and oxygen/fuel ratio are transformed to set-points for the feedback variables. In order to handle model errors in this set-point transformation, an integral action on oxygen/fuel ratio is proposed in an outer loop. An experimental validation and comparison with a control structure without input transformation shows that the proposed structure reduces EGR-errors at the expense of increased pumping losses. In addition the comparison shows that the input transformation improves the performance and achieves the same step response for different flow conditions, thus handling the nonlinear effects.

1. INTRODUCTION

Legislated emission limits for heavy duty trucks are constantly reduced while at the same time there is a significant drive for good fuel economy. To fulfill the requirements, technologies like Exhaust Gas Recirculation (EGR) systems and Variable Geometry Turbochargers (VGT) have been introduced. The primary emission reduction

mechanisms utilized are that NO_x can be reduced by increasing the intake manifold EGR-fraction and smoke can be reduced by increasing the air/fuel ratio [1, 2]. However the EGR fraction and air/fuel ratio depend in complicated ways on the EGR and VGT actuation and it is therefore necessary to have coordinated control of the EGR and VGT to reach the legislated emission limits. Various approaches have been published, and an overview of different control aspects is given in [3]. A multi-variable controller is presented in [4], some approaches that differ in the selection of performance variables are compared in [5], and in [6] decoupling control is investigated. Other control approaches are described in [2, 7, 8, 9, 10, 11, 12, 13, 14].

Nonlinear effects are significant in the system and this paper analyzes and clarifies the benefits of using a nonlinear state dependent input transformation for the gas flow control. The input transformation is developed by inverting the models for EGR-flow and turbine flow having actuator position as input and flow as output. Examining the literature one can identify the input transformation as sub-component in other controllers that have been proposed, e.g. in the comprehensive paper [4], but it is here identified as an important component for handling the nonlinear effects. The main contributions of this paper are:

- It is illustrated and described why and how the control performance is improved using the nonlinear input transformation.
- A new control structure with PID controllers and min/max-selectors that minimizes pumping losses is proposed. It is similar to the structure in [13] and PID controllers are used due to their well known robustness properties. The robustness of the proposed structure is further analyzed in [14].

- A new integral action on oxygen/fuel ratio is proposed in an outer loop in order to handle model errors in a set-point transformation.

Fuel control and injection timing control is also important, but they are not the focus in this paper. Here the focus is on gas flow control.

The control approach for the complete control system and the control objectives related to EGR and VGT control are described in Sec. 2. Sec. 3 describes a mean value diesel engine model that is first used for system analysis in Sec. 3.1 and later used for development and analysis of the input transformation and the proposed control structure. The control structure without input transformation in [13] is described in Sec. 4. The input transformation is developed and analyzed in Sec. 5, while Sec. 6 combines the input transformation with a control structure similar to the structure in [13]. The control structure without input transformation in [13] and the proposed control structure are compared in an engine test cell in Sec. 7.

2. CONTROL APPROACH

To deliver low fuel consumption, fast response to driver commands, and fulfill emission requirements are the goals for engine control. The control of EGR and VGT for emission abatement is considered first, and then the other goals are considered as they are also important for a successful application. The selection of performance variables is an important first step [15], and for emission control it should be noted that exhaust gases, present in the intake from EGR, also contain oxygen. This makes it more suitable to define and use the oxygen/fuel ratio instead of the traditional air/fuel ratio. The main motive is that it is the oxygen content that is crucial for smoke generation, and the idea is to use the oxygen content of the cylinder instead of air mass flow, see e.g. [16]. The exact definition of the normalized oxygen/fuel ratio λ_O is given by (5) in Sec. 3. Thus, EGR-fraction x_{egr} and oxygen/fuel ratio λ_O are a natural selection for performance variables as they are directly related to the emissions. These performance variables are equivalent to cylinder air/fuel ratio and burned gas ratio which are a frequent choice for performance variables in many papers [2, 4, 5, 12].

2.1. ADVANTAGES OF THIS CHOICE

There are three main advantages with the choice of EGR-fraction x_{egr} and oxygen/fuel ratio λ_O as performance variables. These advantages are described in [13] and they are reviewed below.

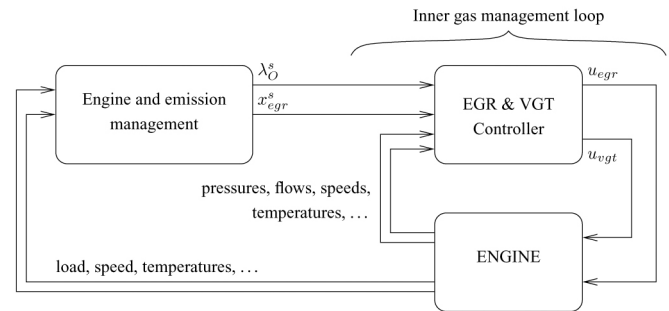


Figure 1. A cascade control structure, with an inner loop where EGR and VGT actuators are controlled using the main performance variables EGR fraction x_{egr} and oxygen/fuel ratio λ_O . This sketch is a simplified illustration of the main idea that will be completed in Sec. 6 to also include turbo protection and fuel control.

The first advantage is that these variables provide direct information about when it is possible to minimize the pumping work, compared to e.g. manifold pressure and air mass flow. To facilitate improved fuel economy the proposed control structure has a simple mechanism for optimizing the fuel consumption by minimizing the pumping work. In diesel engines a large λ_O is allowed and there is thus an extra degree of freedom, when λ_O is greater than its set-point, that can be used to minimize the pumping work. Pumping minimization is an important feature, however the performance variables x_{egr} and λ_O are always controlled as they are the major variables in the controller.

The second advantage is as mentioned above that these variables are strongly connected to the emissions and give a natural separation within the engine management system. The performance variables are handled in a fast inner loop, whereas trade-offs between e.g. emissions and response time for different operating conditions are made in an outer loop. The idea with two loops is depicted in Fig. 1.

The third follows from the second in that it fits well into industry's engineering process where the inner control loops are first tuned for performance. Then the total system is calibrated to get stable combustion and to meet the emission limits by adjusting set-points for different operating conditions, different hardware configurations, and different legislative requirements depending on the measured emissions during the emission calibration process.

Normally, neither x_{egr} nor λ_O are measured and have to be estimated using observers. The observer design is important, but it is not the focus in this paper. Here it is assumed that an observer exists similar to that in [12]. This means that the known issues about oxygen estimation are handled. In the experiments such an observer of industrial production type is available and used. Engines could in the future be equipped

with a sensor for λ_O as suggested in [3, 8], and if so, then nothing has to be changed in the proposed controller structure, which is an additional advantage.

2.2. CONTROL OBJECTIVES

In addition to control of x_{egr} and λ_O it is also necessary to have load control, since the driver's demand must be actuated. This is achieved through basic fuel control using feedforward. Furthermore it is also important to monitor and control turbocharger speed since aggressive transients can cause damage through over-speeding.

The primary variables to be controlled are engine torque M_e , normalized oxygen/fuel ratio λ_O , intake manifold EGR-fraction x_{egr} , and turbocharger speed ω_t . The goal is to follow a driving cycle while maintaining low emissions, low fuel consumption, and suitable turbocharger speeds, which together with the discussion above gives the following control objectives for the performance variables.

3. x_{egr} should follow its set-point x_{egr}^s . 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 engine torque, M_e , should follow the set-point M_e^s from the drivers demand.

5. The turbocharger speed, ω_t , is not allowed to exceed a maximum limit ω_t^{max} , preventing turbocharger damage.

6. The pumping losses, M_p , should be minimized in order to decrease the fuel consumption.

The aim is now to develop a control structure that achieves all these control objectives when the set-points for EGR-fraction and engine torque are reachable.

3. MODELING AND SYSTEM PROPERTIES

A model for a heavy duty diesel engine is used for system analysis and control design. This model includes the fuel path and torque generation but its focus is on the gas flows, see Fig. 2, and it is a mean value model with eight states: intake and exhaust manifold pressures (p_{im} and p_{em}), oxygen mass fraction in the intake and exhaust manifold (X_{Oim} and X_{Oem}), turbocharger speed (ω_t), and three states describing the actuator dynamics for the two control signals (u_{egr} and u_{vgt}) where there are two states for the EGR-actuator to describe an overshoot. These states are collected in a state vector x

$$x = [p_{im} \quad p_{em} \quad X_{Oim} \quad X_{Oem} \quad \omega_t \quad \tilde{u}_{egr1} \quad \tilde{u}_{egr2} \quad \tilde{u}_{vgt}]^T$$

There are no state equations for the manifold temperatures, since the pressures and the turbocharger speed govern the most important system properties, such as non-minimum phase behaviors, overshoots, and sign reversals, while the temperature states have only minor effects on these system properties [17].

The resulting model is expressed in state space form as

$$\dot{x} = f(x, u, n_e)$$

where the engine speed n_e is considered as a measured disturbance, and u is the control input vector

$$u = [u_\delta \quad u_{egr} \quad u_{vgt}]^T$$

which contains mass of injected fuel u_δ , EGR-valve position u_{egr} , and VGT actuator position u_{vgt} .

A detailed description and derivation of the model together with a model tuning and a validation against test cell

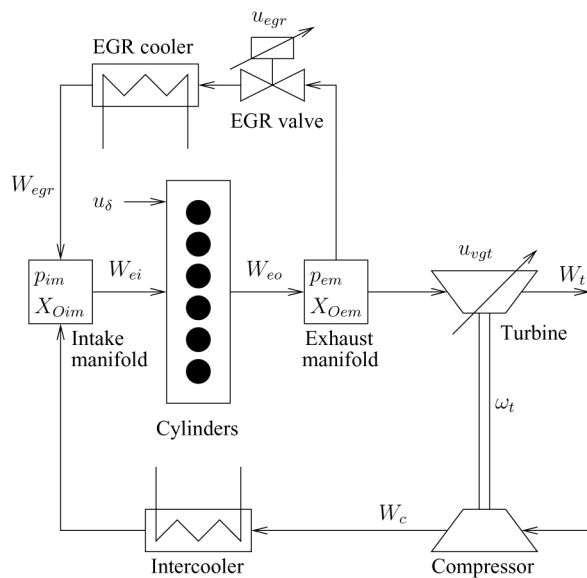


Figure 2. Sketch of the diesel engine model used for system analysis and control design. It has five states related to the engine (p_{im} , p_{em} , X_{Oim} , X_{Oem} , and ω_t) and three for actuator dynamics.

1. λ_O should be greater than a soft limit, a set-point λ_O^s , which enables a trade-off between emission, fuel consumption, and response time.

2. λ_O is not allowed to go below a hard minimum limit λ_O^{min} , otherwise there will be too much smoke. λ_O^{min} is always smaller than λ_O^s .

measurements is given in [17]. The derivatives of the engine state variables are given by

$$\frac{d}{dt} p_{em} = \frac{R_e T_{em}}{V_{em}} (W_f + W_{ei} - W_t - W_{egr}) \quad (1a)$$

$$\frac{d}{dt} \omega_t = \frac{P_t \eta_m - P_c}{J_t \omega_t} \quad (1b)$$

$$\frac{d}{dt} p_{im} = \frac{R_a T_{im}}{V_{im}} (W_c + W_{egr} - W_{ei}) \quad (1c)$$

$$\frac{d}{dt} X_{Oim} = \frac{R_a T_{im}}{p_{im} V_{im}} ((X_{Oem} - X_{Oim}) W_{egr} + (X_{Oc} - X_{Oim}) W_c) \quad (1d)$$

$$\frac{d}{dt} X_{Oem} = \frac{R_e T_{em}}{p_{em} V_{em}} (X_{Oe} - X_{Oem}) (W_f + W_{ei}) \quad (1e)$$

the dynamics of the actuators is given by

$$\tilde{u}_{egr} = K_{egr} \tilde{u}_{egr1} - (K_{egr} - 1) \tilde{u}_{egr2} \quad (2a)$$

$$\frac{d}{dt} \tilde{u}_{egr1} = \frac{1}{\tau_{egr1}} (u_{egr}(t - \tau_{degr}) - \tilde{u}_{egr1}) \quad (2b)$$

$$\frac{d}{dt} \tilde{u}_{egr2} = \frac{1}{\tau_{egr2}} (u_{egr}(t - \tau_{degr}) - \tilde{u}_{egr2}) \quad (2c)$$

$$\frac{d}{dt} \tilde{u}_{vgt} = \frac{1}{\tau_{vgt}} (u_{vgt}(t - \tau_{dvgt}) - \tilde{u}_{vgt}) \quad (3)$$

the oxygen concentration in the exhaust gas is calculated in

$$X_{Oe} = \frac{W_{ei} X_{Oim} - W_f (O/F)_s}{W_f + W_{ei}} \quad (4)$$

and the main performance variables are defined by

$$x_{egr} = \frac{W_{egr}}{W_c + W_{egr}}, \quad \lambda_O = \frac{W_{ei} X_{Oim}}{W_f (O/F)_s} \quad (5)$$

The EGR flow model is given by (6) where the density variations in the mass flow are considered using the quotient $p_{em}/\sqrt{T_{em} R_e}$ in the expression

$$W_{egr} = \frac{p_{em}}{\sqrt{T_{em} R_e}} \Psi_{egr} A_{egrmax} f_{egr}(\tilde{u}_{egr}) \quad (6a)$$

where Ψ_{egr} is modeled as a simplification of a compressible flow restriction according to

$$\Psi_{egr} = 1 - \left(\frac{1 - \Pi_{egr}}{1 - \Pi_{egropt}} - 1 \right)^2 \quad (6b)$$

where the pressure quotient Π_{egr} is limited when the flow is choked at sonic condition and when $1 < p_{im}/p_{em}$ according to

$$\Pi_{egr} = \begin{cases} \Pi_{egropt} & \text{if } \frac{p_{im}}{p_{em}} < \Pi_{egropt} \\ \frac{p_{im}}{p_{em}} & \text{if } \Pi_{egropt} \leq \frac{p_{im}}{p_{em}} \leq 1 \\ 1 & \text{if } 1 < \frac{p_{im}}{p_{em}} \end{cases} \quad (6c)$$

and the effective area ratio f_{egr} is modeled as

$$f_{egr}(\tilde{u}_{egr}) = \begin{cases} c_{egr1} \tilde{u}_{egr}^2 + c_{egr2} \tilde{u}_{egr} + c_{egr3} & \text{if } \tilde{u}_{egr} \leq \frac{-c_{egr2}}{2c_{egr1}} \\ c_{egr3} - \frac{c_{egr2}^2}{4c_{egr1}} & \text{if } \tilde{u}_{egr} > \frac{-c_{egr2}}{2c_{egr1}} \end{cases} \quad (6d)$$

The turbine flow model is given by (7) where the density variations are considered in the same way as in (6a) using the expression

$$W_t = \frac{p_{em}}{\sqrt{T_{em} R_e}} f_{\Pi t}(\Pi_t) A_{vgtmax} f_{vgt}(\tilde{u}_{vgt}) \quad (7a)$$

where the choking function $f_{\Pi t}$ is limited when the flow is choked at sonic condition according to

$$f_{\Pi t}(\Pi_t) = \sqrt{1 - \Pi_t^{K_t}}, \quad \Pi_t = \frac{p_{es}}{p_{em}} \quad (7b)$$

and the effective area ratio f_{vgt} is modeled as

$$f_{vgt}(\tilde{u}_{vgt}) = c_{f2} + c_{f1} \sqrt{\max\left(0, 1 - \left(\frac{\tilde{u}_{vgt} - c_{vgt2}}{c_{vgt1}}\right)^2\right)} \quad (7c)$$

The engine torque model is given by

$$M_e = M_{ig} - M_p - M_{fric}, \quad M_p = \frac{V_d}{4\pi} (p_{em} - p_{im}) \quad (8a)$$

$$M_{ig} = \frac{1}{4\pi} u_\delta 10^{-6} n_{cyl} q_{HV} \eta_{igch} \left(1 - \frac{1}{r_c^{\gamma_{cyl}-1}}\right) \quad (8b)$$

$$M_{fric} = \frac{V_d}{4\pi} 10^5 (c_{fric1} n_e^2 + c_{fric2} n_e + c_{fric3}) \quad (8c)$$

In (7b), $p_{es} > p_{amb}$ if there is a restriction (e.g. a particle filter) after the turbine. However, for this engine there is no restriction after the turbine and therefore it is assumed that $p_{es} = p_{amb}$.

3.1. SYSTEM PROPERTIES

An analysis of the characteristics and the behavior of a system aims at obtaining insight into the control problem. This is known to be important for a successful design of a EGR and VGT controller due to non-trivial intrinsic properties such as non-minimum phase behaviors and sign reversals, see for example [18]. Therefore, a system analysis of the model in Sec. 3 is performed in [17]. The analysis shows that the DC-gains for the channels $u_{vgt} \rightarrow \lambda_O$, $u_{egr} \rightarrow \lambda_O$, and $u_{vgt} \rightarrow p_{em}$ change sign with operating point.

3.1.1. Mapping of sign reversal

Knowledge about the sign reversal in the entire operating region is important when developing a control structure. Therefore, the sign reversal is mapped in [17] by simulating step responses in the entire operating region. In Fig. 3 the sign reversals in $u_{vgt} \rightarrow \lambda_O$, $u_{egr} \rightarrow \lambda_O$, and $u_{vgt} \rightarrow p_{em}$ are mapped by calculating the DC-gain in the step responses and then plotting the contour line where the DC-gain is equal to zero. The step responses are simulated at 20 different u_{vgt} points, 20 different u_{egr} points, 3 different n_e points, and 3 different u_δ points. The size of the steps in u_{vgt} is 5% of the difference between two adjoining operating points. A system analysis also shows that the engine frequently operates in operating points where the sign reversal occurs for the

channels $u_{vgt} \rightarrow \lambda_O$ and $u_{vgt} \rightarrow p_{em}$ [17]. Consequently, it is important to consider the sign reversal for $u_{vgt} \rightarrow \lambda_O$ and $u_{vgt} \rightarrow p_{em}$ in the control design.

These nonlinear effects are also analyzed by calculating the quotient between the maximum and minimum absolute DC-gain for the operating region in Fig. 3. The result is that

$$\frac{\max(|K_{u_{vgt} \rightarrow \lambda_O}|)}{\min(|K_{u_{vgt} \rightarrow \lambda_O}|)} = +\infty \quad (9)$$

$$\frac{\max(|K_{u_{vgt} \rightarrow p_{em}}|)}{\min(|K_{u_{vgt} \rightarrow p_{em}}|)} = +\infty \quad (10)$$

where $K_{u_{vgt} \rightarrow \lambda_O}$ and $K_{u_{vgt} \rightarrow p_{em}}$ are the DC-gains for $u_{vgt} \rightarrow \lambda_O$ and $u_{vgt} \rightarrow p_{em}$. These quotients are equal to $+\infty$ since $\min(|K_{u_{vgt} \rightarrow \lambda_O}|) = \min(|K_{u_{vgt} \rightarrow p_{em}}|) = 0$ which occur at the corresponding lines in Fig. 3. For linear systems, these quotients are equal to 1, and consequently the model in Sec. 3 has large nonlinear effects.

4. CONTROL STRUCTURE WITH PID CONTROLLERS

Based on the control objectives in Sec. 2.2, a control structure with PID controllers and min/max-selectors is proposed in [13] with the following algorithm

$$u_{egr}(t_i) = \begin{cases} \min(-PI_1(e_{\lambda_O}), PI_2(e_{x_{egr}})) & , \text{ if } u_{vgt}(t_{i-1}) = 100 \\ -PI_1(e_{\lambda_O}) & , \text{ else} \end{cases} \quad (11)$$

$$u_{vgt}(t_i) = \begin{cases} 100 & , \text{ if } (u_{vgt}(t_{i-1}) = 100) \\ & \& (e_{x_{egr}} < 0.01) \\ \max(-PI_3(e_{x_{egr}}), -PID_4(e_{\omega_t})) & , \text{ else} \end{cases} \quad (12)$$

where $e_{\lambda_O} = \lambda_O^s - \lambda_O$, $e_{x_{egr}} = x_{egr}^s - x_{egr}$ and $e_{\omega_t} = \omega_t^s - \omega_t$. This structure handles the sign reversal in $u_{vgt} \rightarrow \lambda_O$ because u_{egr} is used to control λ_O , and it also minimizes the pumping work by opening the EGR valve and

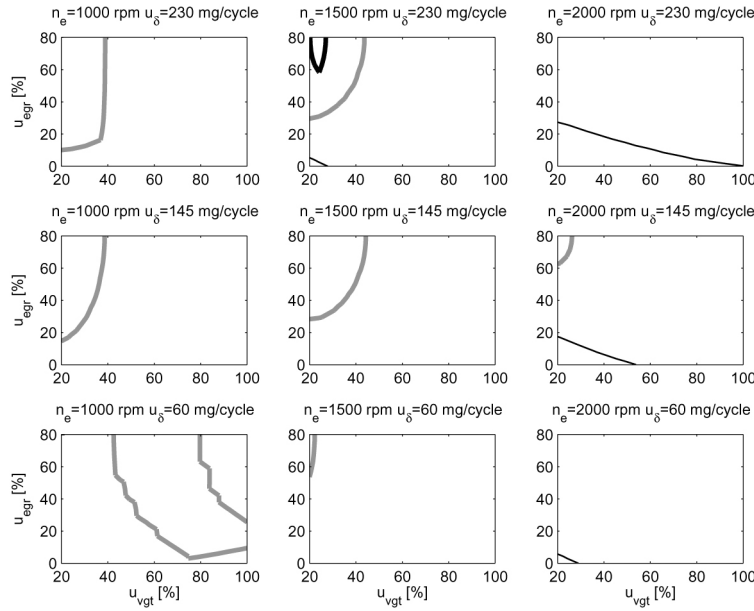


Figure 3. Contour lines where the DC-gain is equal to zero for the system in Sec. 3 for the channels $u_{vgt} \rightarrow \lambda_O$ (thick gray line), $u_{egr} \rightarrow \lambda_O$ (thin black line), and $u_{vgt} \rightarrow p_{em}$ (thick black line) showing that sign reversals occur in these operating points.

the VGT as much as possible while achieving the control objectives for λ_O and x_{egr} [13].

4.1. ENGINE TEST CELL EXPERIMENTS

The controller (11)-(12) is in [13] compared to the current production controller in an engine test cell. This comparison shows that these two controllers have approximately the same control performance in λ_O and x_{egr} during the ETC cycle and that the production controller has at least 26% higher pumping losses.

The control structure (11)-(12) is also validated here in an engine test cell during steps in λ_O^s in the following way. An available production observer, similar to the one in [12], is used to estimate the oxygen mass fraction X_{Oim} . Once X_{Oim} is estimated, the mass flow into the engine W_{ei} , λ_O and x_{egr} are calculated. The engine speed (n_e), intake and exhaust manifold pressure (p_{im} , p_{em}) and turbocharger speed (ω_t) are measured with production sensors. Due to measurement noise, all measured and observed variables are filtered using low pass filters with a time constant of 0.1 s. The PID parameters are initially tuned using an optimization method that minimizes a cost function according to [19] with $\gamma_{Me} = 3/2$ and $\gamma_{egr} = 1/2$. Then, the parameters are manually fine tuned in the engine test cell experiments. The experiment in Fig. 4 shows that the control structure (11)-(12) gives slow control at the first step and oscillations at the third step. This is due to that the DC-gains in $u_{egr} \rightarrow \lambda_O$ and $u_{vgt} \rightarrow x_{egr}$ (the two loops that are used as feedbacks in (11)-(12)) increase

when λ_O increases. This could be handled using gain scheduling, but it is time consuming to tune the parameters for each operating point. Instead, these nonlinear effects are handled using a nonlinear input transformation that will be described in the following sections.

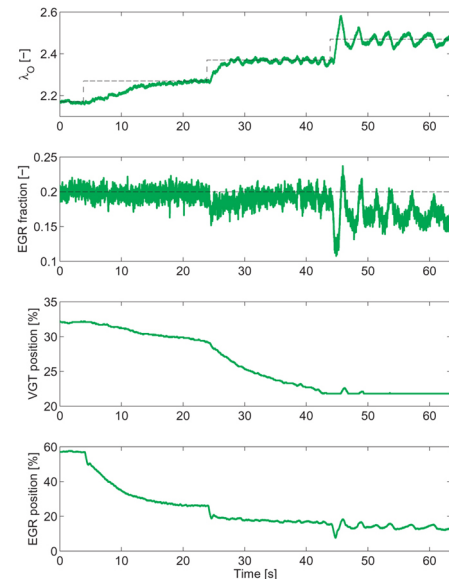


Figure 4. Step responses for the control structure (11)-(12) in an engine test cell showing slow control and oscillations at different steps, i.e. this control structure does not handle nonlinear effects in the diesel engine. Operating point: $n_e = 1200$ rpm and $u_\delta = 136$ mg/cycle.