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Modeling and Control of Diesel Engines with a High-Pressure Exhaust Gas Recirculation System ⋆

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Abstract: This work presents an alternative way of exhaust gas recirculation (EGR) systems implementation in the turbocharged compression ignition engines. This sort of EGR has been studied as well as the most commonly used EGR system, where the intake and the exhaust manifolds are directly connected with a hose. However, in authors opinion the setup described in this paper has not been investigated enough, as most of the research papers concentrate on a conventional configuration modeling and control design. A problem of overcoming a positive scavenging pressure drop and delivering high amount of EGR is addressed in this article. It comes from the fact that in this kind of engines the exhaust pressure is lower than the intake, making it difficult to deliver high and stable portion of EGR over the engine operating range. A generic mean value engine model is developed in this work and an initial simple control structure is proposed.

Keywords: engine modeling, EGR, VGT.

1. INTRODUCTION

The emission reduction problem has become important during past several decades due to a significant growth of transportation sector. It is, therefore, crucial for manufacturers to produce engines that can meet strict emission regulation requirements. Internal combustion engines used in cars are sources of pollution gases, among which the following are known to be harmful: hydrocarbons, carbon dioxide (CO_2) and monoxide (CO), nitrogen oxides (NO_x) and particulate matter (PM) (see Reifarth [2010]).

This work describes one way of reducing NO_x emission formation, which is called an exhaust gas recirculation (EGR) system and was introduced in 1970's. Since NO_x is formed at high temperatures, the basic idea was to re-route part of the engine exhaust gases back to the intake manifold to lower the combustion temperature inside the cylinders. The methodology turned out to be quite efficient and the variety of configurations was suggested by R&D companies to improve EGR performance.

Modeling and control of a charged intake combustion engines equipped with a conventional EGR system is presented by many authors, for example Wang et al. [2011], Kolmanovsky et al. [1997], Jankovic et al. [1998], Wahlstrom and Eriksson [2011], Jung et al. [2002]. The required amount of EGR in this system is delivered by simply connecting intake and exhaust manifolds with a hose. The amount of EGR is regulated with a valve incorporated in that hose. Unlike normally aspirated engines, in these engines pressure in the exhaust manifold is lower than in the intake, which does not let them to operate with a high EGR fraction without additional control valves, such as the intake throttle, which is used to temporarily lower the pressure in the intake manifold. This, however, can result in a performance drop or in fuel consumption increase.

Grondin et al. [2009] describe an EGR system composed of two loops: conventional (which is called a high pressure EGR (HPEGR) in the article) and a low pressure (LP). In a LP loop exhaust gas is taken from the downstream of variable geometry turbocharger (VGT) and fed back to the intake compressor upstream. Although double loop EGR looks promising, it requires a complicated control system, due to the presence of the five control inputs (intake and exhaust throttles, two EGR valves and VGT). Another approach is presented by Haber [2010], where a two-stage turbocharger is incorporated in a LP-loop to increase the compression level in the intake.

This paper presents a different configuration of the EGR system from those in the above described papers. However, similar kind of system was found among patents, where a problem of positive scavenging pressure drop is addressed.

The proposed configuration is very close to the one described in patent by Shao et al. [2001], with a difference that an electrically or crankshaft driven compressor in the

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EGR path is used, which is advantageous to the proposed scheme being independent of the exhaust mass flow. It can therefore deliver the required amount of EGR, avoiding such problems as a turbine "spool-up" (or turbo lag) in case of a turbocharger. However, it either consumes engine power or requires a separate electrical motor be installed. In our case the rotational speed of a turbocharger is in the range of $76 \cdot 10^3 \ldots 172 \cdot 10^3$ rpm and it might be problematic or inefficient to achieve these speeds using electrical motor.

Another similar configuration is shown in patent by Codan et al. [1996], where two different turbochargers are used for establishing an intake boost pressure and for delivering high EGR fraction. It is claimed that this system is capable of providing almost constant EGR rate.

However, as shown before, most of the authors concentrate on modeling and control of EGR systems different from described in patents by Shao et al. [2001] and Codan et al. [1996]. Therefore a need for generic modeling algorithm of the proposed embodiment exists. This paper presents mean value modeling and initial control design of a turbocharged HPEGR system on a charged intake combustion engine, which can be then used for advanced control system design.

The proposed HPEGR configuration is described in Section 2 and the mean-value modeling is done in Section 3. Simulation results and preliminary control system design are shown in Section 4. Section 5 proposes further work and conclusion.

2. SYSTEM DESCRIPTION

The proposed HPEGR system is able to deliver higher and more stable portion of the exhaust gas to the intake manifold. This system is constructed on the compression ignition (CI) diesel engine laboratory testbed, which is a flexible research engine and not a production one. The modeling configuration of the engine is shown in Fig. 1. It consists of the following basic components:

- four control volumes (marked by p_i,p_x,p_{egr} and p_4)
- intake compressor C_{int}
- cylinder head (CH)
- VGT in the EGR path
- exhaust valve, EGR valve 1 and EGR valve 2

The intake manifold is charged with an electrical motor driven rotary screw compressor, which has a separate control system and is not a subject of this work. The idea is to set and to sustain a certain pressure in the intake manifold.

It should be noted that a separate electrical compressor obviously consumes extra engine power. This cannot be circumvented due to the testbed multi-objective configuration. However, a mathematical model of a turbocharger is incorporated in the intake pressure control system, which means that compressor power depends on the amount of exhaust gas available. This prevents compressor from generating unrealistic air mass flow rate.

High EGR fraction is achieved by incorporating a high-speed Borg Warner VGT into the EGR loop.

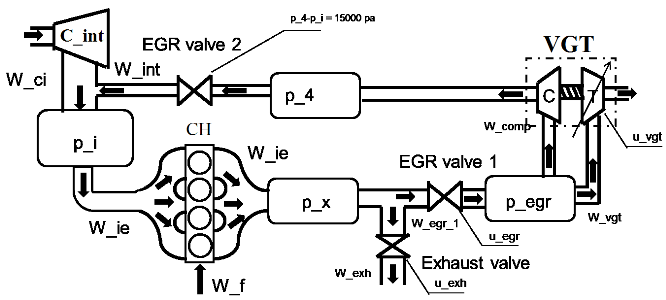

Fig. 1. Internal combustion diesel engine equipped with high-pressure EGR system modeling configuration. C and T stand for compressor and turbine, respectively.

There are three controllable components in this setup. First, the exhaust valve is used to keep a back-pressure on a predefined level. It is controlled by a simple proportional-integral (PI) controller and is not a subject of this work. Second, EGR valve 1 is used to regulate a portion of the exhaust gas flowing into the loop. Its operation is similar to the one in a standard EGR configuration. Third, VGT has to be controlled to operate with a reasonable efficiency and at the same time to prevent a compressor from surging.

EGR valve 1 and the exhaust valve are the ball valves and valve 2 is the poppet valve. Valve 2 opens automatically when a pressure difference across the valve gets greater than $\Delta p = p_4 - p_i = 0.1 \ldots 0.2$ bar, which is determined by the spring stiffness. In this setup mixing of the intake air and the exhaust gas occurs at the compressor downstream, where the pressure is high. Therefore, to prevent a reverse mass flow (when p_4 is lower than p_i), EGR valve 2 is installed.

A two-stage cooling system (not shown in Fig. 1) is installed in the EGR path. First stage occurs before the turbocompressor and the second right after it. Second cooler is needed because there is a temperature increase across the turbocompressor. The recirculated exhaust gas is cooled from $\approx 600°C$ to $\approx 100°C$ during the first stage and from $\approx 250°C$ to $\approx 95°C$ during the second stage.

A CO_2 analyzer is used to determine the EGR fraction in the intake manifold.

The control objectives in this setup are as follows: to track the EGR fraction to the desired set-point and to operate VGT in a stable region. The EGR fraction and the VGT operating point depend in a complicated way on valve 1 and VGT actuation. Therefore an advanced control system is required.

Since the EGR fraction is governed by the turbocharger, the nonminimum phase and sign-reversal behavior of the system is guaranteed to occur (see Kolmanovsky et al. [1997], Wahlstrom and Eriksson [2011]). It is therefore of importance to construct a model that catches these dynamics.

3. SYSTEM MODELING

This section presents mean-value modeling (see Heywood [1988]) of the CI engine equipped with an HPEGR system. The main dynamics of the system are defined by the pressure in the four control volumes, turbocharger shaft speed and the engine rotational speed. Modeling of similar systems has been presented by several authors, see Jung [2003], Wahlstrom and Eriksson [2011], Stefanopoulou et al. [1998].

Assumption: this engine testbed has rather long connecting pipes and volumetric balancing vessels, which allows considering temperatures in control volumes constant for simplification.

3.1 Intake manifold

The change of pressure $p_i(t)$ in the intake manifold is described by the ideal gas law

$$V_i\dot{p}_i(t) = R_iT_iW_i \Rightarrow \dot{p}_i(t) = \frac{R_iT_i}{V_i}(W_{ci}+W_{int}-W_{ie}), \quad (1)$$

where R_i is the specific gas constant of the air, T_i and V_i are the intake manifold temperature and volume, respectively and $W_i = (W_{ci} + W_{int} - W_{ie})$ is the change of mass in the intake manifold. The mass flow into the cylinders W_{ie} is described with the volumetric efficiency and with the engine's ability to pump the air

$$W_{ie} = \frac{V_d\omega_e(t)p_i(t)}{v2\pi R_iT_i}\eta_v(\omega_e(t), p_i(t), \ldots), \quad (2)$$

where V_d is the piston displacement, $\omega_e(t)$ is the angular engine speed, v is the number of revolutions per cycle (2 for 4-stroke engines), $\eta_v(\ldots)$ is the engine's volumetric efficiency, that depends on several variables, but mostly on the engine speed.

The compressor mass flow W_{ci} into the intake manifold can be calculated from the equation for the compressor power $P_c(t)$

$$P_c(t) = W_{ci}c_{p,i}(T_c - T_a), \quad (3)$$

where $c_{p,i}$ is the air specific heat in constant pressure. The state of the gas at compressor inlet and outlet is defined as (T_a, p_a) and (T_c, p_i), respectively. For the engineering purposes the transition from input to output of the compressor is considered isentropic, which means that

$$\frac{T_{c,is}}{T_a} = \left(\frac{p_i(t)}{p_a}\right)^{\frac{\gamma_c}{\gamma_c-1}}, \quad (4)$$

where γ_c is the ratio of the air specific heats $c_{p,i}$ and $c_{v,i}$ at constant pressure and at constant volume, respectively. The compressor efficiency η_c comes from the fact that there is a temperature increase across the compressor and can be calculated as

$$\eta_c = \left(\frac{T_{c,is} - T_a}{T_c - T_a}\right), \quad (5)$$

where $T_{c,is}$ is the isentropic compressor temperature. Combining (4) and (5) and taking into account that $\mu_c = \frac{\gamma_c}{\gamma_c-1}$, the temperature T_c of the gas at compressor output is expressed as

$$T_c = T_a\left(1 + \frac{1}{\eta_c}\left(\left(\frac{p_i(t)}{p_a}\right)^{\mu_c} - 1\right)\right). \quad (6)$$

Substitution of the expression (6) for the compressor temperature into (3) yields the expression for the mass flow into the intake manifold

$$W_{ci} = \frac{\eta_c P_c(t)}{T_a c_{p,i}\left(\left(\frac{p_i(t)}{p_a}\right)^{\mu_c} - 1\right)}. \quad (7)$$

Exhaust gas mass flow through the EGR valve 2 is defined by the standard orifice flow equation

$$W_{int} = A_{valve2}(u_{valve2})\frac{p_4(t)}{\sqrt{R_xT_4}}\psi_4(p_r), \quad (8)$$

where $A_{valve2}(u_{valve2})$ is the effective area of the EGR valve 2, u_{valve2} is the corresponding control signal, that defines how much the valve is opened, $p_4(t)$ and T_4 are the intermediate volume pressure and temperature, respectively and $\psi_4(p_r)$is the pressure ratio correction factor, which is defined as

$$\psi_4(p_r) = \begin{cases} \sqrt{\dfrac{2\gamma_e}{\gamma_e - 1}\left(p_r^{\frac{2}{\gamma_e}} - p_r^{\frac{\gamma_e+1}{\gamma_e}}\right)} & if\ p_r > r_c \\ \gamma_e^{\frac{1}{2}}\left(\dfrac{2}{\gamma_e + 1}\right)^{\frac{\gamma_e+1}{2(\gamma_e-1)}} & if\ p_r \le r_c \end{cases} \quad (9)$$

where the pressure ratio is $p_r = \frac{p_i(t)}{p_4(t)}$, critical pressure ratio is $r_c = \left(\frac{2}{\gamma_e+1}\right)^{\frac{\gamma_e}{\gamma_e-1}}$ and γ_e is the ratio of the gas specific heats c_p and c_v at constant pressure and at constant volume, respectively.

3.2 Exhaust manifold

The change of pressure $p_x(t)$ in the intake manifold is described by the ideal gas law

$$\begin{aligned} V_x\dot{p}_x(t) &= R_xT_xW_x \Rightarrow \\ \dot{p}_x(t) &= \frac{R_xT_x}{V_x}(W_{ie} + W_f - W_{egr1} - W_{exh}), \end{aligned} \quad (10)$$

where R_x is the gas constant of the exhaust gas, T_x and V_x are the exhaust manifold temperature and volume, respectively and $W_x = (W_{ie} + W_f - W_{egr1} - W_{exh})$ is the change of mass in the exhaust manifold. Fuel mass flow W_f is considered as a control variable. The equations describing the mass flows W_{egr1} through the EGR valve1 and W_{exh} through the exhaust valve are modeled using orifice flow equation

$$W_{egr1} = A_{egr}(u_{egr})\frac{p_x(t)}{\sqrt{R_xT_x}}\psi_{egr}(p_{r,egr}), \quad (11)$$

$$W_{exh} = A_{exh}(u_{exh})\frac{p_x(t)}{\sqrt{R_xT_x}}\psi_{exh}(p_{r,exh}), \quad (12)$$

where $A_{egr}(u_{egr})$ and $A_{exh}(u_{exh})$ are the effective areas of the EGR valve 1 and exhaust valve, respectively, u_{egr} and u_{exh} are the corresponding control signals, that defines how much the valves are opened, $p_{egr}(t)$ is the pressure in the EGR manifold and $\psi_{egr}(p_{r,egr})$ and $\psi_{exh}(p_{r,exh})$are the pressure ratio correction factors, which are defined in the same way as $\psi_4(p_{r,4})$, but $p_{r,egr} = \frac{p_{egr}(t)}{p_x(t)}$ and $p_{r,exh} = \frac{p_a}{p_x(t)}$.

3.3 EGR manifold

The change of pressure $p_{egr}(t)$ in the EGR manifold is described by the ideal gas law

$$\begin{aligned} V_x\dot{p}_{egr}(t) &= R_xT_xW_{egr} \Rightarrow \\ \dot{p}_{egr}(t) &= \frac{R_xT_3}{V_{egr}}(W_{egr1} - W_{comp} - W_{vgt}), \end{aligned} \quad (13)$$

where $W_{egr} = (W_{egr1} - W_{comp} - W_{vgt})$ is the change of mass in the EGR manifold. The mass flow through the variable geometry turbine is calculated with an orifice flow equation

$$W_{vgt} = A_{vgt}(u_{vgt}) \frac{p_{egr}}{\sqrt{R_x T_3}} \psi_{vgt}(p_{r,vgt}), \tag{14}$$

where $A_{vgt}(u_{vgt})$ is the effective area of the turbine nozzle, u_{vgt} is the corresponding control signal, T_3 is the temperature in the EGR manifold and $\psi_{vgt}(p_{r,vgt})$ is the pressure ratio correction factor, which is defined in the same way as $\psi_4(p_{r,4})$, but $p_{r,vgt} = \frac{p_a}{p_{egr}}$.

3.4 Intermediate manifold

The change of pressure $p_4(t)$ in the EGR manifold is described by the ideal gas law

$$\begin{aligned} V_x \dot{p}_4(t) &= R_x T_x W_4 \Rightarrow \\ \dot{p}_4(t) &= \frac{R_x T_4}{V_4}(W_{comp} - W_{int}), \end{aligned} \tag{15}$$

where $W_4 = (W_{comp} - W_{int})$ is the change of mass in the intermediate manifold and V_4 is its volume. The EGR compressor mass flow W_{comp} is calculated in a similar manner as W_{ci}

$$W_{comp} = \frac{\eta_{c,egr} P_{c,egr}(t)}{T_2 c_{p,egr} \left(\left(\frac{p_4(t)}{p_{egr}(t)} \right)^{\mu_e} - 1 \right)}, \tag{16}$$

where $\eta_{c,egr}$ is the compressor efficiency, $P_{c,egr}(t)$ its power, T_2 is the temperature at the compressor upstream, $c_{p,egr}$ is the specific heat in constant pressure.

3.5 Turbocharger dynamics

Turbocompressor power $P_{c,egr}(t)$ is modeled as a power transfer between the turbine and the compressor.

$$\dot{P}_{c,egr}(t) = \frac{1}{\tau}(\eta_{vgt} P_{vgt}(t) - P_{c,egr}(t)), \tag{17}$$

where τ is the turbocharger time constant and the turbine power $P_{vgt}(t)$ is calculated as

$$P_{vgt} = W_{vgt} c_{p,vgt} T_3 \left(1 - \left(\frac{p_a}{p_{egr}(t)} \right)^{\mu_e} \right), \tag{18}$$

where $c_{p,vgt}$ is a specific heat in a constant pressure.

3.6 Actuators dynamics

The dynamics of the actuators (valve 1, valve 2, exhaust valve, VGT guide vanes) are approximated as a first-order systems

$$\frac{d}{dt} x_{ind}(t) = \frac{1}{\tau_{ind}}(-x_{ind}(t) + u_{ind}(t)), \tag{19}$$

where τ_{ind} is the actuator time constant, $x_{ind}(t)$ and $u_{ind}(t)$ are the actual and the desired control signals, respectively. Index *ind* can be equal to *egr*, *vgt*, *exh* or *valve2*, depending on the actuator modeled.

3.7 Prime-mover dynamics

The rotational speed of the combustion engine can be regulated through the fuel injection rate. The increase of the fuel mass flow causes engine to produce more torque.

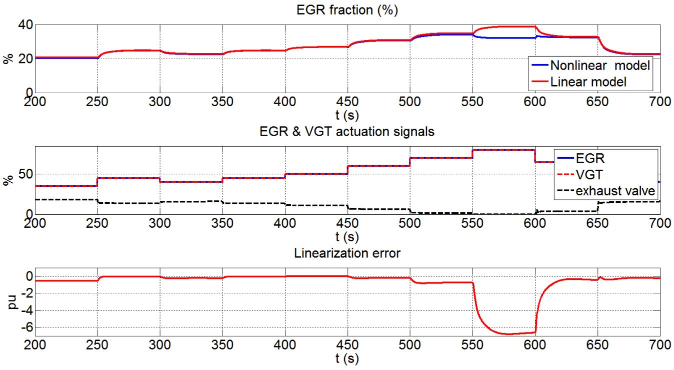

Fig. 2. Comparison of the original and linearised plant open-loop responses.

The dynamics between the engine's speed and torque are described by the Newton's second law

$$J \frac{d\omega_e(t)}{dt} = M_i - M_f - M_l, \tag{20}$$

where J is the moment of inertia of the engine, M_i is the indicated torque, M_f is the friction torque and M_l is the load torque. The latter three torque components are calculated as follows

$$\begin{aligned} M_i &= \frac{\eta_i W_f Q_{HV}}{\omega_e(t)} \qquad M_f = \frac{f_{mep} V_d}{2\pi v} \\ M_l &= \frac{load \cdot P_{e,max}}{\omega_e(t)}, \end{aligned} \tag{21}$$

where η_i is the indicated thermal efficiency, Q_{HV} is the lower heating value of the fuel, $f_{mep} = (C_1 + 48(N_e(t)/1000) + 0.4S_p^2) \cdot 10^3$ is the friction mean effective pressure experimental equation, where C_1 is a constant, $N_e(t)$ is the engine speed (rpm), S_p is the mean piston speed and $P_{e,max}$ is the maximum engine power (see Malkhede et al. [2005]).

4. LINEARIZATION AND SIMULATION

4.1 Linearization

The model is represented by a set of nonlinear differential equations, which have to be linearized to get transfer function or a state-space representation of the system. Linearization by hand is tedious due to the high order of the system. Another problem is when the model is slightly changed, the linearization has to be done again, which takes much time. Therefore it is done using Simulink linear analysis tool, which gets the linear model at a steady state operating point. The comparison of the open-loop step responses of the original and linear model is shown in Fig. 2. The inputs are the EGR valve 1 and VGT actuation signals and the output is the EGR fraction. The linear model follows the original rather accurately in the vicinity of the operating point and deviates when the model goes far from it, which can be seen from the linearization error subplot in Fig. 2. This model will be used for the initial control design.

4.2 Open-loop step response

The nonlinear model of a turbocharged EGR diesel engine is implemented in Simulink. The model has two control

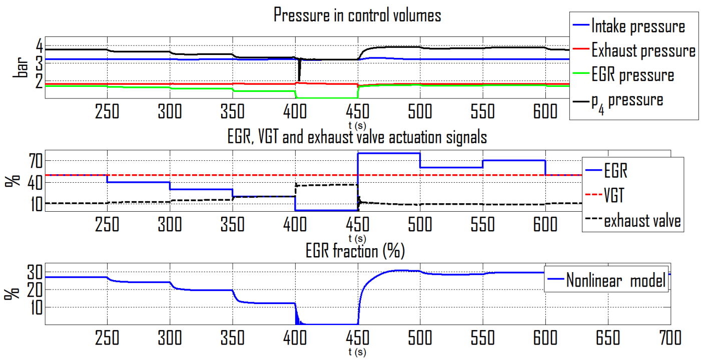

Fig. 3. Nonlinear model open-loop simulation. Change of pressures p_i, p_x, p_{egr} and p_4 in the control volumes and of the EGR fraction depending on the actuation signals. VGT control signal is kept constant while EGR is changing. Exhaust valve position is controlled with a PI controller.

inputs (EGR and VGT actuation signals u_{egr} and u_{vgt}, respectively) and one controllable output — EGR fraction, which is calculated as $\chi = \frac{W_{int}}{W_{ci}+W_{int}}100\%$. Exhaust and intake pressures are controlled with the exhaust valve and the intake compressor, respectively. The aim is to regulate them to a predefined level. Simple PI controllers are used for these purposes.

The open loop simulation is shown for different combinations of the control inputs. The change of pressures in control volumes, actuation signals and the EGR fraction are shown in Fig. 3 when the VGT is kept constant and EGR valve 1 varies. It can be seen that the EGR fraction depends a lot on EGR valve position.

The same parameters are shown in Fig. 4 but EGR valve 1 is constant and VGT varies. It is noticeable that exhaust and EGR pressure stay almost equal despite changes in VGT. This is due to the EGR valve being constant (exhaust and EGR manifold can be considered as one reservoir).

Note: an interesting phenomenon was noticed during nonlinear simulations. When the EGR fraction gets low ($\approx$ 5%) the pressure p_4 in the intermediate manifold also gets low. As it approaches Δp, chattering of the EGR fraction occurs. This happens due to modeled valve rapid opening/closing while the pressure is in the vicinity of $|\Delta p|$. It can be observed in Fig. 3 at time $t = 400s$. The EGR fraction does not drop to 0 smoothly, but more like 'chatters' to it. This, however, should not happen in reality, since the poppet valve is adjusted in a way that it starts leaking the mass flow as the pressure difference get close to Δp and opens fully when it gets higher.

The important issue is that the constructed model catches a non-minimum phase behavior of the system when a VGT step change is applied (Fig. 5), which is necessary for a proper control system design. Physically it can be explained as follows: when negative step change is applied to VGT, a sudden increase of p_{egr} occurs, which leads to an EGR fraction initial rise. However, the turbine power then drops causing EGR fraction to decrease.

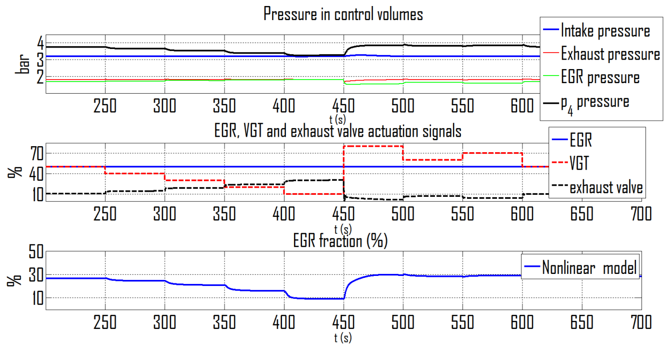

Fig. 4. Nonlinear model open-loop simulation. Change of pressures p_i, p_x, p_{egr} and p_4 in the control volumes and of the EGR fraction depending on the actuation signals. EGR control signal is kept constant while VGT is changing. Exhaust valve position is controlled with a PI controller.

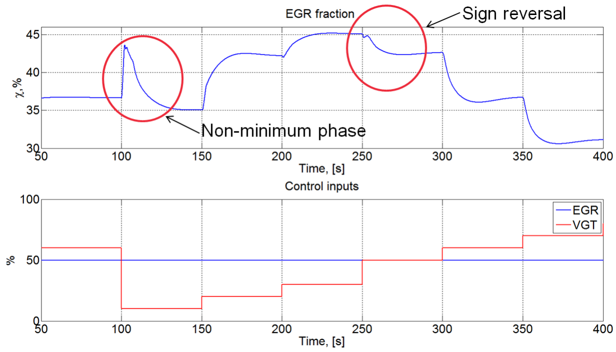

Fig. 5. Nonlinear model open-loop simulation. Non-minimum phase behavior of the EGR fraction when a VGT step change is applied. The operating point is $u_{egr} = 50\%$.

A sign reversal takes place when the VGT opening is high, because exhaust gases start flowing through the turbine, rather than through the EGR compressor. It means that further increase in VGT decreases EGR fraction.

4.3 Closed-loop step response

Two PI controllers of the form $C(s) = P(1 + I\frac{1}{s})$ are designed as a reference control system for further improvements. The closed-loop system and the simulated step response are shown in Fig. 6 and Fig. 7, respectively. In this design most of the control is done by the EGR valve 1, whereas VGT has rare control authority. This is explained by the corresponding controllers tuning, where the gains of PI controller for VGT input are smaller than for EGR. The purpose here is to demonstrate that a wide range of the EGR fraction can be covered using active EGR valve 1 control, keeping VGT almost constant. This should give more freedom for VGT to control compressor surge in the future design.

5. CONCLUSION AND FURTHER WORK

The problem of delivering high and stable portion of EGR over the engine operating range is addressed in this work. A generic mean-value modeling of a turbocharged HPEGR

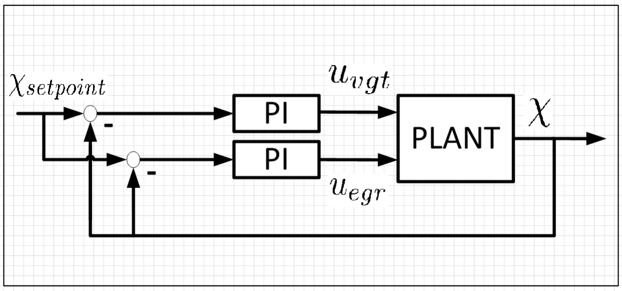

Fig. 6. Control configuration of the HPEGR.

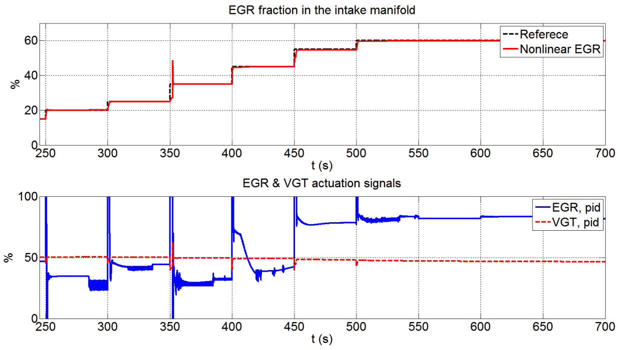

Fig. 7. Nonlinear model closed-loop simulation. PI-controllers are used for EGR and VGT actuators control.

system installed on a compression ignition engine is presented. In authors opinion this type of an EGR structure has not been researched enough and it is, therefore, important to investigate it.

Further work will include model experimental verification and improvements with data obtained in a laboratory engine testbed described in section 2. Specifically, these improvements include model enhancement to a 2×2 multi-input multi-output (MIMO) system and a turbocharger look up tables incorporation to track its operating point.

After model verification is done, more thorough analysis should be performed to design a control system. This should include the investigation of a system two-way interaction and proper input-output pairings selection (using relative gain analysis (RGA) technique, for example).

The measurement and control system for the engine is to be implemented using LabView$^{\text{TM}}$ software.

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