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RESEARCH ARTICLE

Control oriented air path model for compressed air boosted Diesel engines

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Abstract

In this paper an air path model is presented for control system design. The model was developed for direct injected, turbocharged and intercooled commercial vehicle diesel engines which are equipped with compressed air booster system (PBS® – Pneumatic Booster System) [12], *high pressure exhaust gas recirculation (EGR) with EGR-cooler and exhaust brake (EB). Current and next generation emission standards introduced significant limitations for NO_x and soot. It is challenging to handle these components, especially at transient engine operations. Nitric oxide formation can be limited with an appropriate amount of exhaust gas recirculation. Soot formation is influenced mainly by the air-fuel ratio of the mixture which can be affected by the intake manifold pressure. Therefore with the targeted design of a suitable air path controller the modeled engine setup is able to handle both the NO_x and soot formation in transient cases. The reported model is the first step of this work.*

Keywords

Diesel engine · Air path system · EGR · Compressed air booster · Turbo-lag · Model-based control

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Nomenclature

Notation of variables

A	area [m^2]
B_t	fuel flow [kg/s]
c	specific heat [J/kgK]
H	internal energy of gas [J]
H_l	diesel lower heating value [J/kg]
K_{L0}	stoichiometric air-fuel ratio [-]
m	mass [kg]
n	rotational speed [RPM]
P	power [W]
Q	heat transfer [J]
R	specific gas constant [J/kgK]
t	time [s]
T	absolute temperature [K]
U	internal energy of gas [J]
V	volume [m^3]
V_d	engine displacement [m^3]
ε	heat exchanger efficiency [-]
η	efficiency [-]
κ	adiabatic exponent [-]
λ	air-fuel ratio [-]
Π	pressure ratio [-]
ρ	density [kg/m^3]
σ	mass flow [kg/s]
τ	time constant [s]

Notation of indices

1	refers to compressor inlet
4	refers to exhaust brake outlet
a	refers to air
amb	refers to ambient
c	refers to compressor
e	refers to engine
eb	refers to exhaust brake
eff	refers to effective value
egf	refers to exhaust gas fraction
egr	refers to exhaust recirculation
$egrc$	refers to EGR cooler

ei	refers to engine inlet
em	refers to exhaust manifold
eng, cool	refers to engine coolant
eo	refers to engine outlet
ind	refers to indicated
ic	refers to intercooler
im	refers to intake manifold
in	refers to inlet
out	refers to outlet
p	refers to constant pressure
pbs	refers to the compressed air booster
red	refers to reduced
t	refers to turbine
tc	refers to turbocharger
th	refers to PBS® throttle valve
to	refers to turbine outlet
v	refers to constant pressure

1 Introduction

In case of diesel engines fresh air only is inducted into the cylinders and the fuel will be injected near to the firing top dead center (FTDC) position of the piston. With a desired amount of fuel which can appropriately delivered by a high pressure fueling system (mainly a common-rail) a non-premixed combustion takes place shortly after the begin of the mixture building. The torque production of the engine can be adjusted by the fuel amount which is limited by the available fresh air (oxygen) in the cylinders. It could be improved (in case of constant engine displacement) by increasing the pressure (more correct the density) of the charge air. It is achieved most frequently and efficiently by turbocharging which means the usage of the engine out enthalpy. With this method the size and weight of modern engines could be decreased by a given power output, the efficiency and the manufacturing costs could also be reduced. It is the so called down-sizing concept which is in focus of nowadays engine development. Besides of the above mentioned positive effects turbocharging results worse transient responses and reduced driveability due to the lack of charge pressure at the start of an acceleration which is caused by the finite dynamics of the turbocharger rotor (due to its inertia) and the intake and exhaust manifold (due to their volume). This is the well-known turbolag. Several arrangements have been done to improve driveability by transmission development [16] and by accelerating the engine transient behavior in reactive ways such as wastegated and variable nozzle turbines but in order to completely eliminate this phenomenon and ensure good response and emission one needs proactively control the charge air pressure. The most applicable solution approach is the compressed air injection into the intake manifold. Commercial vehicles are provided with compressed air system for brake air supply, air suspension etc. and the air stored in reservoirs can be used to replace the lacking air mass flow for the engine. With this amount of compressed air, which was produced beforehand by a reciprocating compressor, arbitrary boost pressure can be reached immediately after the torque demand of the driver. The detailed description of the compressed air booster system can be found in [12].

Beside of the torque production in transient operations there are two even more important fields in commercial vehicle engine development: the improvement of the economy of the operation, so enhancing the efficiency and seeking for alternative fuel production ways (demand from customers) [3], and the fulfilling of the future emission standards (demands from legislatives). Compression-ignition engines achieve load control in qualitative way and the combustion process is non-premixed so the thermodynamic states and the composition of the inducted gas into the cylinders has fundamental impact on the torque production, efficiency and the exhaust gas composition. Therefore the appropriate control of the air path parameters is an effective way to satisfy recent requirements to modern diesel engines.

Next generation emission standards (Euro 6 and US EPA 10) include significant limitations from which the reductions of the soot and nitrogen oxides level are the most challenging for developers. Basically there are two opportunities of the intervention: the exhaust gas aftertreatment (SCR, DPF) and the restriction of pollutant formation during the combustion i.e. raw emission limitation. The equipment of exhaust gas aftertreatment are costly, reach their nominal efficiency only in a limited exhaust gas temperature and composition range and most of them must be cyclically regenerated. It is pursued in nowadays diesel engine development to avoid aftertreatment systems or at least reduce the number equipment [10] due to the mentioned disadvantages.

A widely used method to reduce the formation of nitric oxides during the combustion is to recirculate a certain amount of exhaust gases into the intake side of the engine (EGR). This exhaust gas backflow have to be precisely adjusted depending on the engine operation due to inadequate amounts can negatively influence soot formation and indicated efficiency.

As a consequence improved air path control (fresh air and EGR) methods are effective ways to reduce engine raw emission levels under the legislative limits and ensure fuel economy and driveability.

2 Modeling aim and system description

In the last century the investigation and development of diesel engines was focused on steady-state operation conditions, however in usual applications the engines operate in unsteady (acceleration, deceleration), so called transient modes. Developers turn firstly to study transient operation to improve the dynamic torque build up and driveability of the engines. Nowadays the legislatives give the motivation.

Present and forthcoming commercial vehicle emission standards evaluate engine out pollutants during transient dynamometer cycles such as European Transient Cycle (ETC) in Europe, Federal Test Procedure (FTP) in the USA and Word Harmonized Transient Cycle (WHTC) as an international cycle.

Investigating circumstances of pollutant formation it is clearly seen that it is concentrated in the load steps (and their values are significant more than the steady-state results at equivalent speed and load) and remarkable part of the total emission forms during transients. The main reason is that the air-fuel ratio and amount of the recirculated exhaust gas differ from their static values. Detailed description about this complex phenomenon can be found in [15].

As presented, in the combustion chamber of a diesel engine a non-premixed combustion take place and the ignition begin shortly after the start of fuel injection into the cylinders. Therefore the time for mixture building is limited and as a consequence locally low air-fuel ratio values occur. Soot is formed where the local air-fuel ratio is lower than approximately 0.6. Besides the rich mixture the locally high pressures and temperatures also enhance the particle formation [8]. As a conclusion it can be seen that the soot emission can be influenced by the air-fuel ratio which is the function of the fresh air mass flow rate into the cylinders. With the compressed air booster system the injected fresh air mass into the intake manifold can be controlled arbitrarily during transients so that the particle emission can be reduced when it is most occurs, during accelerations.

Besides soot the NO_x emission is the other most critical limitation in the new directives. There are five main ways of NO_x production during the combustion: the thermal or Zeldovich-mechanism, prompt or Fenimore-mechanism, NO_x via N_2O, NO_x via NNH and by the fuel-bounded nitrogen. The largest amount of nitrogen oxides in diesel engines is produced along the Zeldovich mechanism, which needs high temperature ($T>1900$ K) due to the activation energy of its first reaction is very high [17]. So arrangements for lower cylinder (local) temperature are effective in NO_x concentration reduction. For instance these can be lower compression ratio, retarded fuel injection, boost pressure reduction and exhaust gas recirculation (EGR) etc. The first three modifications have deteriorating effect on the efficiency so in the commercial vehicle sector (where the fuel economy so important is) the exhaust gas recirculation seems to be the most favourable solution. EGR replace or add to the fresh air amount (supplied by the compressor or the compressed air booster system) and the recirculated CO_2 and water vapour will increase the heat capacity of the cylinder charge. Thanks to the dilution effect of the EGR the oxygen concentration in the cylinder charge also will decrease. The ignition in a non-premixed flame occur where the local equivalence ratio is stoichiometric or slightly above it. The fuel spray must penetrate more and occupy larger volume to achieve ignition conditions in a diluted (by EGR) cylinder charge compared with fresh air alone. Due to the larger heat capacity (larger volume) of the flame region the local temperature will be lower. More detailed explanation and investigation of the EGR effect can be found in [11]. Based on the described burning procedure the NO_x formation during combustion depends mainly on the oxygen concentration of the inlet charge and the emitted amount can estimated appropriately from it [4, 8].

Detrimental effect can be observed at high EGR rates on soot emission. To achieve optimal recirculated amounts an appropriate adjustment to the fresh air is needed. As a conclusion in [11] an additional EGR is suggested, rather than displacing fresh air. To recirculate additional amounts of exhaust gases a higher boost pressure is needed to keep the fresh air mass flow rate at constant level (detailed description can be found in [8].

As the result of the above described investigation and literature review the following engine setup has been built to effectively reduce the transient emissions. For the investigation the engine was installed on a dynamometer and pressure and temperature sensors were mounted in the air path to support the model building and validation. The review of the test cell can be read in [2].

The engine is a direct-injected common-rail with turbocharger and intercooler. To achieve fast dynamic in transients a high pressure EGR loop were designed with EGR cooler and electromechanically actuated butterfly valve on the hot side. The compressed air booster consists of a butterfly valve between the intercooler and the intake manifold, and a compressed air supply valve connected to the air tank. The butterfly valve is to avoid the air backflow to the intercooler during air injection and it is fully closed during activation and fully open in not activated cases. The exhaust gases flow back to the intake side as the result of the positive pressure difference between the exhaust- and the intake manifold. When compressed air is injected the pressure rises quickly in the intake manifold. The exhaust manifold pressure increases slower so during the activation no EGR flows back and this leads to increased NO_x emission. To avoid the negative pressure difference between the exhaust- and the intake manifold, downstream to the turbine an exhaust flap takes place. With the close of this valve the pressure of the exhaust manifold can be increased and the recirculated exhaust gas amount can be controlled. The system schematic is depicted below.

The main parameters of the modelled engine are summarized in the table below:

Tab. 1. Engine parameters

Bore [mm]	102
Stroke [mm]	120
Number of cylinders	4
Engine displacement [cm3]	3922
Number of valves	4/cyl.
Compression ratio	17,3:1
Rated effective torque [Nm]	600 (1200 to 1600 RPM)
Rated effective power [kW]	123 at 2500 RPM

The aim is to construct a dynamic, mean value model of the presented engine that describes the thermodynamical and mechanical processes in the air path with differential- and algebraic equations. The general aim is to reduce the NO_x and particle matter emission in transients while at least maintaining even

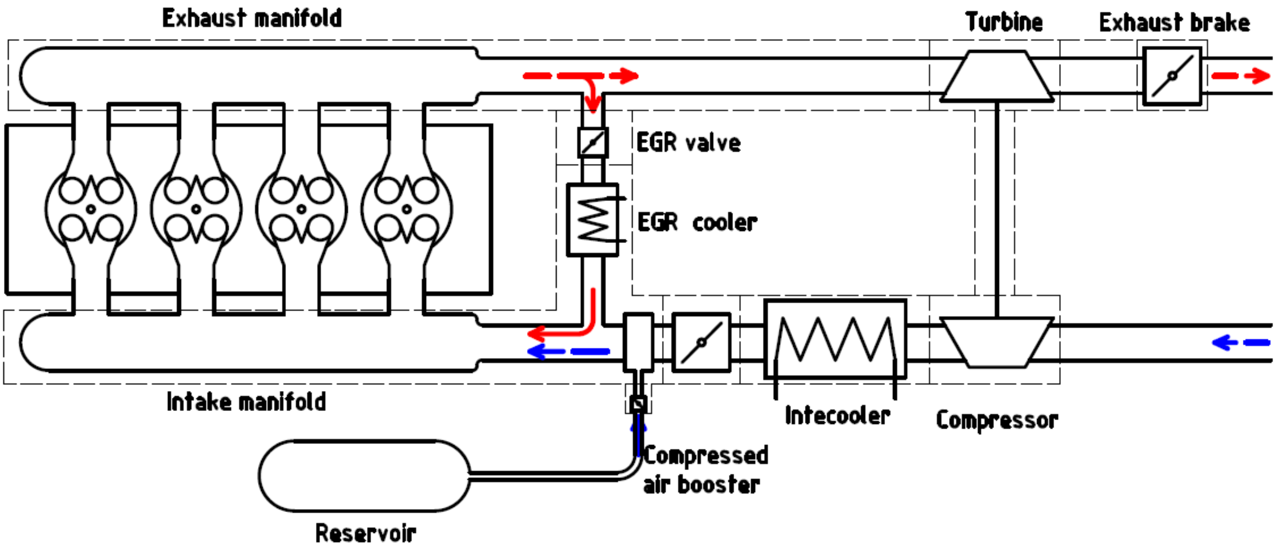

Fig. 1. Schematic overview of the modeled turbocharged and compressed air boosted diesel engine

more improving fuel economy and driveability. Another purpose is to define the model in the simplest form to serve the easy handling.

3 Detailed engine model

For preliminary studies, development of the control oriented air path model and for controller tuning and testing a detailed engine model were constructed in GT-Suite environment [2]. It helps the parameter estimation of the simplified model with offering comparison possibilities to unmeasured or immeasurable air path parameter signals (i.e.: compressor power) and give the opportunity to adjust simplified turbine and compressor model to its detailed maps. In this section the detailed model is described.

The intake and exhaust system were modeled in one-dimensional wave action form so the parts were discretized into numerous sub-volumes based on 3D CAD models. The turbine and compressor performance were specified as standard SAE datasets based on measurements. The heat transfer from gases to the pipes and flowsplits is simulated using the heat transfer coefficient which is calculated at every timestep from the fluid velocity and the thermo-physical properties and the wall surface finish. Pipe friction losses were also taken into account. Throttles (EGR valve, throttle valve of the compressed air booster, exhaust brake) were defined based on the measured dimensions and the discharge coefficients were tuned to the measurement data. Combustion process were imposed as heat release rate data based on indicating measurement in the whole engine operation map due to its accuracy (no emission prediction needed). In-cylinder heat transfer was calculated based on the Woschni model. The engine friction also was defined as a lookup based on indicated friction mean effective pressure measurements [7].

The detailed model were validated in stationary cases in the whole engine operation range (engine speed and load) in respect of the effective power and the main air path parameters (intercooler pressure, intake manifold pressure, exhaust manifold pressure, fresh air mass flow rate and air-fuel ratio). The deviations could be hold below 10%. The simulation model has been presented in more details in [2]. Fig. 2 shows the deviation of the predicted and measured intake manifold pressure data can be seen in percentages.

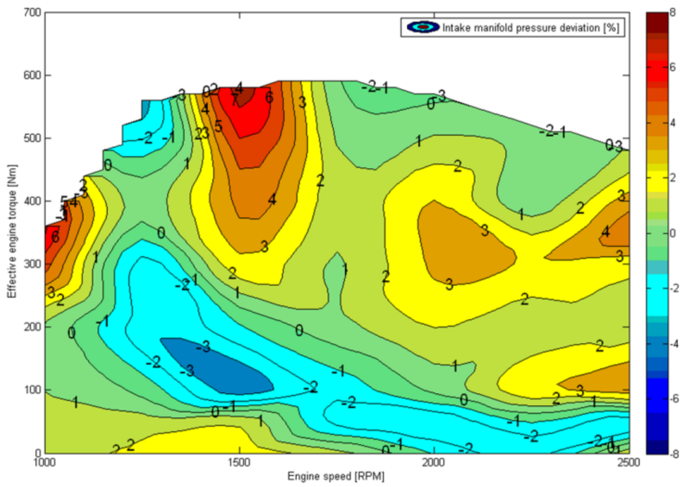

Fig. 2. Intake manifold pressure differences between measured and calculated signal by the detailed model in percentages

The simplified model was compared and verified to the detailed model results.

4 Simplified model equations

Preliminary modeling assumptions were taken to obtain the simplest model form according our goals:

1 the potential energy is neglected,

2 constant physical and chemical properties are assumed over the volume in each main part of the model, such as specific heat, specific gas constant and adiabatic exponent,

3 the adiabatic exponent and the specific gas constant (consequently the specific heats) of the air and the exhaust gas is equal, (Diesel engines operates with lean mixture.),

4 there are no mass and energy storage effects in the combustion chamber,

5 the fluids can be modeled as ideal gas,

6 the temperature of the outflowing gas from the receiver is equal to the receiver's temperature,

7 the inlet temperature and pressure of the compressor is equal to the ambient temperature and pressure $T_1 = T_{amb}, p_1 = p_{amb}$,

8 the outlet temperature and pressure of the turbine is equal to the ambient temperature and pressure $T_4 = T_{amb}, p_4 = p_{amb}$,

9 the mass of the fuel was neglected.

The intended use is control system design, so the model should be written in state-space form. The systematic modeling procedure is based on first engineering principles and follows the modeling procedure published in [14]. The hierarchical structure of a dynamic model can be separated into set of the following model elements:

- balance volumes over which conservation balances are constructed (the highest level),
- balance equations,
- terms in balance equations corresponding to mechanisms,
- constitutive equations,
- variables and parameters (the lowest level).

The modeled engine can be separated into the following four balance volumes: the intercooler, the intake manifold, the exhaust manifold and the volume between the turbine and the exhaust brake. The listed volumes are denoted with dashed lines in Fig. 1. These balance volumes can be modeled as receivers with mass and energy flows as inputs and outputs and for which the thermodynamic states assumed to be the same over the entire volume. The model equations can be derived from the following conservation laws as balance equations and additional equations that give the connections between the conservation differential equations and the thermodynamic parameters:

The energy conservation law:

$$\frac{dU(t)}{dt} = \dot{H}_{in}(t) - \dot{H}_{out}(t) + \dot{Q}(t) \tag{1}$$

where the caloric relations:

– the internal energy

$$U(t) = c_v m(t) T(t) \tag{2}$$

– the enthalpy flow into the receiver

$$\dot{H}_{in}(t) = c_p \sigma_{in}(t) T_{in}(t) \tag{3}$$

– the enthalpy flow out of the receiver

$$\dot{H}_{out}(t) = c_p \sigma_{out}(t) T(t) \tag{4}$$

– the heat transfer flow to the environment

$$\dot{Q}(t)$$

The temperature of the out-flowing gas is assumed to be equal to the gas temperature in the receiver.

The mass conservation law:

$$\frac{dm}{dt}(t) = \sigma_{in}(t) - \sigma_{out}(t) \tag{5}$$

where σ_{in} and $\sigma_{out}$the in- and outflowing massflow respectively.

To reach the differential equation of the pressure the ideal gas law can be used:

$$p(t)V = m(t)RT(t) \tag{6}$$

For balance volumes for which the temperature of the inlet gas and the temperature of the gas in the receiver are nearly equal, the isothermal assumption is a good approximation. The differential equation for the pressure can be obtained by differentiating (6) and substituting (5) in it in the following form:

$$\frac{dp(t)}{dt} = \frac{RT}{V}\left[\sigma_{in}(t) - \sigma_{out}(t)\right] \tag{7}$$

For balance volumes where the inflowing gas temperatures differ from the receiver's temperature the politropic form is a better approximation. The differential equation for the pressure level can be reached by the derivation of (6) and substituting it into the energy conservation law (1). With the use of the relations between the adiabatic exponent, the specific heats and the gas constant

$$\kappa = \frac{c_p}{c_v} \tag{8}$$

and

$$R = c_p - c_v \tag{9}$$

and the following relation obtained:

$$\frac{dp(t)}{dt} = \frac{\kappa R}{V}\left[\sigma_{in}(t)T_{in}(t) - \sigma_{out}(t)T_{out}(t) + \frac{\dot{Q}(t)(\kappa-1)}{\kappa R}\right] \tag{10}$$

To ensure the model simplicity the differential equation for the temperature calculation was avoided and the temperature was predefined such cases. Using the derived differential equations ((7) and (10)) the pressure level can be defined in each balance volumes.

4.1 Intercooler

The intercooler has one mass flow inlet from the compressor. The outlet is the butterfly throttle valve of the compressed air booster. The heat loss across the walls can be neglected. As a simplification, the effect of heat exchange was model with the inflowing temperature which was assumed to be equal with the outlet temperature and known. As a consequence, the intercooler pressure is obtained from (7) in the following form:

$$\frac{dp_{ic}(t)}{dt} = \frac{R_a T_{ic}}{V_{ic}} \left[\sigma_c - \sigma_{th}\right] \tag{11}$$

The intercooler temperature is computed with a parameter:

$$T_{ic} = T_{amb} + \Delta T_{ic} \tag{12}$$

The compressor mass flow rate was computed based on the formula below which was suggested in [1]. The compressor efficiency treated as a constant proved to be a good approximation and the simplest solution, after investigating several turbocharger model in [13].

$$\sigma_c = \frac{\eta_c}{\frac{R_a \kappa_a}{\kappa_a - 1} T_1} \frac{P_c}{\left[\left(\frac{p_{ic}}{p_1}\right)^{\frac{\kappa_a - 1}{\kappa_a}} - 1\right]} \tag{13}$$

The gas mass flow rate leaving the tank is influenced by the butterfly valve of the PBS®, which effective area is a control input. It closes before the air injection and opens only if the intercooler pressure is higher than the intake manifold pressure. During compressed air boosting the effective area of the valve is zero. Summarizing the operating conditions the hybrid behavior of the model can be avoided. Seeking for the easiest form of the mass flow rate formula for subsonic flows it was calculated based on the recommendation of [6]:

$$\sigma_{th} = A_{eff,th} \frac{p_{ic}}{\sqrt{R_a T_{ic}}} \sqrt{\frac{2 p_{im}}{p_{ic}} \left(1 - \frac{p_{im}}{p_{ic}}\right)} \tag{14}$$

4.2 Compressor power

Besides of the four differential equation for the balance volume pressures the a fifth one have to be written as a model state to define the compressor behavior. With a widely used approximation in the literature the turbo shaft dynamic is modeled as a first order lag. The compressor power can be written in function of the turbine power with a time constant τ_{tc} [1]:

$$\frac{dP_c}{dt} = -\frac{P_c}{\tau_{tc}} + \frac{1}{\tau_{tc}} \eta_t \left[1 - \Pi_t^{\frac{\kappa_a - 1}{\kappa_a}}\right] \left(\frac{R_a \kappa_a T_{em}}{\kappa_a - 1}\right) \sigma_t \tag{15}$$

where the η_t =const. proved to be a good approximation.

4.3 Intake manifold

The temperatures of fluid flowing into the intake manifold differ so the pressure level can be formulated based on (10):

$$\frac{dp_{im}(t)}{dt} = \frac{\kappa_a R_a}{V_{im}} \left(\sigma_{th} T_{ic} + \sigma_{pbs} T_{amb} + \sigma_{egr} T_{egr} - \sigma_{ei} T_{im}\right) \tag{16}$$

Note that in the in the equation above the heat transfer were omitted which is good assumption in case of small temperature differences between the fluid and the receiver's wall, short dwell times or small surface-to-volume ratios.

The σpbs is the injected mass flow rate by the compressed air booster which is assumed to be a control input since it is always a sonic flow (the air tank pressure is approx. 10 bar) therefore independent from the intake manifold pressure and the supply valve has only fully opened or closed position. The intake manifold temperature is assumed to be known.

The gas temperature downstream to the EGR cooler with the cooler efficiency is calculated as:

$$T_{egr} = (1 - \eta_{egrc}) T_{em} + \eta_{egrc} T_{eng,cool} \tag{17}$$

The EGR valve flow is assumed to be subsonic due to lower pressure differences between the exhaust and intake manifold. A hybrid mode is generated by the checkvalve in EGR loop. If the exhaust gas flow is $p_{em} > p_{im}$, then:

$$\sigma_{egr} = A_{eff,egr} \frac{p_{em}}{\sqrt{R_a T_{em}}} \sqrt{\frac{2 p_{im}}{p_{em}} \left(1 - \frac{p_{im}}{p_{em}}\right)} \tag{18}$$

and if $p_{em} < p_{im}$, then:

$$\sigma_{egr} = 0 \tag{19}$$

The mass flow induced into the cylinders with the engine volumetric efficiency:

$$\sigma_{ei} = \eta_{vol}(n_e) \rho_{im} \frac{V_d n_e}{2 \cdot 60} \tag{20}$$

A polinomial formulation for η_{vol}:

$$\eta_{vol} = \eta_{vol,a} + \eta_{vol,b} n_e + \eta_{vol,c} n_e^2 \tag{21}$$

4.4 Exhaust manifold

The differential equation for the exhaust manifold pressure is:

$$\frac{dp_{em}(t)}{dt} = \frac{\kappa_a R_a}{V_{em}} \left[\frac{\dot{H}_{eo} - \dot{Q}_{em}}{(\kappa_a - 1)} - T_{em}(\sigma_t + \sigma_{egr})\right] \tag{22}$$

In case of the exhaust manifold the heat transfer to the walls cannot be neglected due to the big temperature difference between the walls and the hot exhaust gases. Its effect is modeled integrated in the engine out enthalpy, and can be adjusted with the exhaust gas fraction of the non-utilized fuel energy. The outflowing enthalpy and the extracted heat loss based on the work in [18] is:

$$\dot{H}_{eo} - \dot{Q}_{em} = \dot{H}_{ei} + B_t H_l (1 - \eta_{ind}) \eta_{egf} \tag{23}$$

where $\dot{H}_{ei}$ is the enthalpy flow into the engine:

$$\dot{H}_{ei} = \frac{R_a \kappa_a}{\kappa_a - 1} \sigma_{ei} T_{im} \tag{24}$$

The indicated efficiency depends mainly on the engine speed and air-fuel ratio, so the model was defined as a quadratic polynomial function of these two input signal in the following form [5]:

$$\begin{aligned}\eta_{ind} = {} & (\eta_{ind,a} + \eta_{ind,b} n_e + \eta_{ind,c} n_e^2) \\ & (\eta_{ind,d} + \eta_{ind,e} \lambda + \eta_{ind,f} \lambda^2)\end{aligned} \tag{25}$$

and

$$\eta_{egf} = \eta_{egf,a} + \frac{\eta_{egf,b}}{\lambda} + \frac{\eta_{egf,c}}{\lambda^2} \tag{26}$$

where the air-fuel ratio is calculated from the engine-in fresh air and from the fuel flow which is sent by the engine EDC.

$$\lambda = \frac{1}{K_{L0}} \frac{\sigma_{th} + \sigma_{pbs}}{B_t} \tag{27}$$

Fluid-dynamic turbines can be approximated quite well as orifices, so the reduced mass flow rate through the turbine is defined with two parameters [6]:

$$\sigma_{t,red} = c_t \sqrt{1 - \Pi_t^{k_t}} \tag{28}$$

From which one can obtain the actual mass flow rate:

$$\sigma_t = \frac{p_{em}}{\sqrt{T_{em}}} \sigma_{t,red} \tag{29}$$

The exhaust manifold temperature is assumed as a measured value so it was added as a measurable disturbance.

4.5 Balance volume between the turbine and the exhaust brake

The inflow- and the receiver temperature is the same, so the last model state equation is:

$$\frac{dp_{to}(t)}{dt} = \frac{R_a T_{to}}{V_{to}} \left[\sigma_t - \sigma_{eb}\right] \tag{30}$$

The turbine out temperature is based on [9]:

$$T_{to} = T_{em} \left[1 - \eta_t \left(1 - \left(\frac{p_{em}}{p_{to}}\right)^{\frac{1-\kappa_a}{\kappa_a}}\right)\right] \tag{31}$$

The mass flow rate through the exhaust brake can be sonic and subsonic therefore it adds to hybrid mode to the model. The simplified orifice equations:

$$\sigma_{eb} = A_{eff,eb} \frac{p_{to}}{\sqrt{R_a T_{to}}} \sqrt{\frac{2p_4}{p_{to}} \left(1 - \frac{p_4}{p_{to}}\right)} \tag{32}$$

if

$$\Pi_{crit} \le p_1$$

$$\sigma_{eb} = A_{eff,eb} \frac{p_{to}}{\sqrt{R_a T_{to}}} \frac{1}{\sqrt{2}} \tag{33}$$

if

$$\Pi_{crit} > p_1$$

where the critical pressure ratio is:

$$\Pi_{crit} = \left[\frac{2}{\kappa_a + 1}\right]^{\frac{\kappa_a - 1}{\kappa_a}} p_{to} \tag{34}$$

The operation of the exhaust brake is independent from the EGR valve so the model has in total four hybrid modes.

5 Conversion into state-space form

There are four receiver volumes in the nonlinear model and for all of them one differential equation was defined from first engineering principles for the pressure states. An additional differential equation was defined for the power of the turbocharger compressor. Therefore the state vector consist of the values of the intercooler pressure, intake manifold pressure, exhaust manifold pressure, pressure of the volume between the turbine and the exhaust brake and the compressor power:

$$\mathbf{x} = \begin{bmatrix} p_{ic} & p_{im} & p_{em} & p_{to} & P_c \end{bmatrix}^{\mathsf{T}} \tag{35}$$

The input vector contains the effective areas of the actuator valves and the value of the injected air flow by compressed air booster:

$$\mathbf{u} = \begin{bmatrix} A_{eff,egr} & A_{eff,eb} & A_{eff,th} & \sigma_{pbs} \end{bmatrix}^{\mathsf{T}} \tag{36}$$

The disturbance vector consists of the exhaust manifold and ambient temperature and the engine speed and the injected dosage:

$$\mathbf{d} = \begin{bmatrix} T_{im} & T_{em} & T_{amb} & n_e & B_t \end{bmatrix}^{\mathsf{T}} \tag{37}$$

Substituting the constitutive equations into the differential conservation balances the state space model can be formulated as defined below:

$$\frac{d\mathbf{x}}{dt} = f(\mathbf{x}, \mathbf{d}) + \mathbf{B}(\mathbf{x}, \mathbf{d})\,\mathbf{u} \tag{38}$$

In expanded form:

$$\begin{bmatrix} \dot{p}_{ic} \\ \dot{p}_{im} \\ \dot{p}_{em} \\ \dot{p}_{to} \\ \dot{P}_c \end{bmatrix} = \begin{bmatrix} f_1(\mathbf{x}, \mathbf{d}, r) \\ f_2(\mathbf{x}, \mathbf{d}, r) \\ f_3(\mathbf{x}, \mathbf{d}, r) \\ f_4(\mathbf{x}, \mathbf{d}, r) \\ f_5(\mathbf{x}, \mathbf{d}, r) \end{bmatrix} + \begin{bmatrix} 0 & 0 & \xi_1 & 0 \\ \xi_2 & 0 & \xi_3 & \frac{\kappa_a R_a T_{amb}}{V_{im}} \\ \xi_4 & 0 & 0 & 0 \\ 0 & \xi_5 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \mathbf{u} \tag{39}$$

where the nonlinear state functions are:

$$f_1(\mathbf{x}, \mathbf{d}, 1) = \frac{\eta_c(\kappa_a - 1)(T_{amb} + \Delta T_{ic})}{V_{ic}\kappa_a T_1} \frac{P_c}{\left[\left(\frac{p_{ic}}{p_1}\right)^{\frac{\kappa_a - 1}{\kappa_a}} - 1\right]} \tag{40}$$

$$f_2(\mathbf{x}, \mathbf{d}, 1) = -\frac{\kappa_a p_{im} V_d n_e}{2 \cdot 60 V_{im}} \times \left(\eta_{vol,a} + \eta_{vol,b} n_e + \eta_{vol,c} n_e^2\right) \tag{41}$$

$$f_3(\mathbf{x}, \mathbf{d}, 1) = \frac{\kappa_a R_a}{V_{em}} \left[\frac{\eta_{vol}\kappa_a p_{im} V_d n_e}{120(\kappa_a - 1)^2} + \frac{B_t H_l (1 - \eta_{ind})\eta_{egf}}{\kappa_a - 1} - p_{em}\sqrt{T_{em}} c_t \sqrt{1 - \left(\frac{p_{em}}{p_{to}}\right)^{k_t}}\right] \tag{42}$$

$$f_4(\mathbf{x},\mathbf{d},1) = \frac{R_a p_{em}\sqrt{T_{em}}}{V_{to}} c_t \sqrt{1-\left(\frac{p_{em}}{p_{to}}\right)^{k_t}} \times \left[1-\eta_t\left(1-\left(\frac{p_{em}}{p_{to}}\right)^{\frac{1-\kappa_a}{\kappa_a}}\right)\right] \tag{43}$$

$$f_5(\mathbf{x},\mathbf{d},1) = -\frac{P_c}{\tau_{tc}} + \frac{\eta_t R_a \kappa_a p_{em} c_t \sqrt{T_{em}}}{\tau_{tc}(\kappa_a-1)} \times \sqrt{1-\left(\frac{p_{em}}{p_{to}}\right)^{k_t}}\left[1-\frac{p_{em}}{p_{to}}^{\frac{\kappa_a-1}{\kappa_a}}\right] \tag{44}$$

The coefficients of the inputs are:

$$\xi_1 = -\frac{\sqrt{R_a(T_{amb}+\Delta T_{ic})}p_{ic}}{V_{ic}}\sqrt{\frac{2p_{im}}{p_{ic}}\left(1-\frac{p_{im}}{p_{ic}}\right)} \tag{45}$$

$$\xi_2 = \frac{\kappa_a}{V_{im}}\left[(1-\eta_{egrc})T_{em}+\eta_{egrc}T_{eng,cool}\right] \sqrt{\frac{2R_a p_{em} p_{im}}{T_{em}}\left(1-\frac{p_{im}}{p_{em}}\right)} \tag{46}$$

$$\xi_3 = \frac{\kappa_a}{V_{im}}\sqrt{2p_{im}p_{ic}R_a(T_{amb}+\Delta T_{ic})\left(1-\frac{p_{im}}{p_{ic}}\right)} \tag{47}$$

$$\xi_4 = \frac{\kappa_a}{V_{em}}\sqrt{\frac{2R_a p_{em} p_{im}}{T_{em}}\left(1-\frac{p_{im}}{p_{em}}\right)} \tag{48}$$

$$\xi_5 = -\frac{p_{to}T_{em}}{V_{to}}\sqrt{\frac{R_a}{2T_{to}}}\left[1-\eta_t\left(1-\left(\frac{p_{em}}{p_{to}}\right)^{\frac{1-\kappa_a}{\kappa_a}}\right)\right] \tag{49}$$

6 Model validation

The performance of the simplified model was evaluated in a 15 seconds long test case from the urban part of the European Transient Cycle. The defined engine speed and load is depicted below:

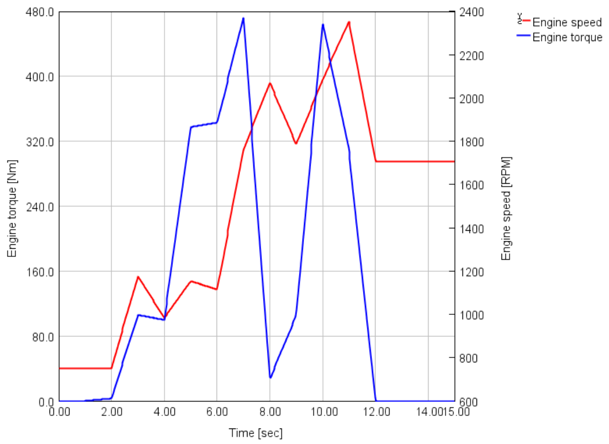

Fig. 3. Predefined engine speed and load in the test cycle

The compressed air booster was activated three times in the cycle from the fourth, sixth and the ninth seconds. The PBS® was controlled by its own control logic, implemented in MATLAB SIMULINK that runs coupled with GT Suite. The throttle of the compressed air booster was fully closed during activation and fully opened at not-activated modes. The exhaust brake operated based on the targeted concept, so it closed at the beginning of the air injection and opened gradually until the end of the PBS® activation following a predefined effective area function. The EGR valve was operated by the EGR controller of the GT Suite and followed a target EGR lookup map. The effective area of the EGR valve and the exhaust brake and the injected air mass flow during the test case can be seen in Fig. 4.

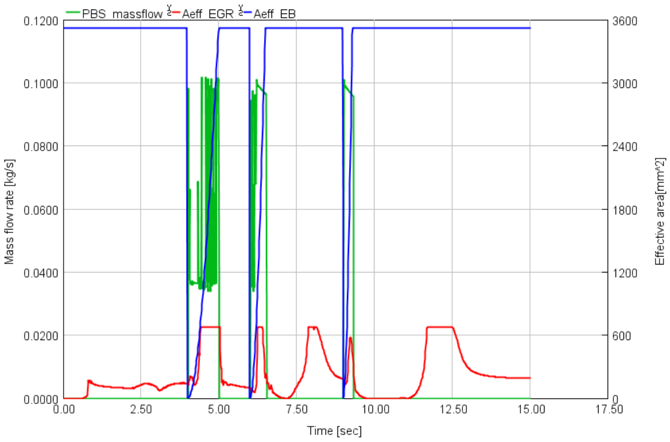

Fig. 4. Control inputs during the test cycle

The values of the model parameters are shown in Table 2.

The model was implemented in MATLAB SIMULINK and as a solver the stiff ODE15s (a variable order solver based on the numerical differentiation formulas) method was chosen. The comparison of the simplified and the detailed model is shown on Fig. 5 with red and blue lines respectively.

To reach the modeling aim the fit of the pressure state variables (these are the measurable quantities) is desired, therefore individual deviations for all of the pressure state variables were evaluated as root-mean square errors as follows for the entire cycle:

$$\varepsilon_{ic} = \sqrt{\frac{1}{T}\int_0^T\left(\frac{p_{ic,d}-p_{ic}}{p_{ic,d}}\right)^2 dt} = 0.0720 \tag{50}$$

$$\varepsilon_{p_{im}} = \sqrt{\frac{1}{T}\int_0^T\left(\frac{p_{im,d}-p_{im}}{p_{im,d}}\right)^2 dt} = 0.0708 \tag{51}$$

$$\varepsilon_{p_{em}} = \sqrt{\frac{1}{T}\int_0^T\left(\frac{p_{em,d}-p_{em}}{p_{em,d}}\right)^2 dt} = 0.1301 \tag{52}$$

$$\varepsilon_{p_{to}} = \sqrt{\frac{1}{T}\int_0^T\left(\frac{p_{to,d}-p_{to}}{p_{to,d}}\right)^2 dt} = 0.0491 \tag{53}$$

Where T is the length of the entire cycle namely 15 seconds and the suffix d denotes to the corresponding value of the detailed model. On Fig. 5 it is shown, that the simplified model gives a good fit to the pressure levels calculated by the detailed model and feasible for controller design. Even lower errors are achievable with identification methods in the future.

Tab. 2. Model parameters

Parameter name	Symbol	Unit	Value
Adiabatic exponent	κ_a	-	1.4
Ambient pressure	p_a	[Pa]	10^5
Compressor efficiency	η_c	[-]	0.68
EGR cooler efficiency	η_{egrc}	[-]	0.9
Engine coolant temperature	$T_{eng,cool}$	[K]	320
Engine displacement	V_d	[m^3]	0.003922
Engine volumetric efficiency par. 1	$\eta_{vol,a}$	[-]	0.7285
Engine volumetric efficiency par. 2	$\eta_{vol,b}$	$\left[RPM^{-1}\right]$	0.000153
Engine volumetric efficiency par. 3	$\eta_{vol,c}$	$\left[RPM^{-2}\right]$	-510^{-8}
Exhaust gas fraction par. 1	$\eta_{egf,a}$	[-]	0.85
Exhaust gas fraction par. 2	$\eta_{egf,b}$	[-]	0.0013
Exhaust gas fraction par. 3	$\eta_{egf,c}$	[-]	-0.2
Exhaust manifold volume	V_{em}	[m^3]	0.004
Indicated efficiency par. 1	$\eta_{ind,a}$	[-]	54.24
Indicated efficiency par. 2	$\eta_{ind,b}$	$\left[RPM^{-1}\right]$	-0.009479
Indicated efficiency par. 3	$\eta_{ind,c}$	$\left[RPM^{-2}\right]$	2.89810^{-6}
Indicated efficiency par. 4	$\eta_{ind,d}$	[-]	0.5656
Indicated efficiency par. 5	$\eta_{ind,e}$	[-]	1.217
Indicated efficiency par. 6	$\eta_{ind,f}$	[-]	-1.037
Intake manifold volume	V_{im}	[m^3]	0.005
Intercooler temp. deviation	ΔT_{ic}	[K]	320
Intercooler volume	V_{ic}	[m^3]	0.08
Turbine efficiency	η_t	[-]	0.65
Specific gas const.	R_a	[J/kgK]	287
Stoichiometric air-fuel ratio	K_{L0}	[-]	14.5
Turbine out volume	V_{to}	[m^3]	0.002
Turbine mass flow model parameter 1	c_t	$\left[\frac{kg\sqrt{K}}{sPa}\right]$	0.022
Turbine mass flow model parameter 2	k_t	[-]	-0.65

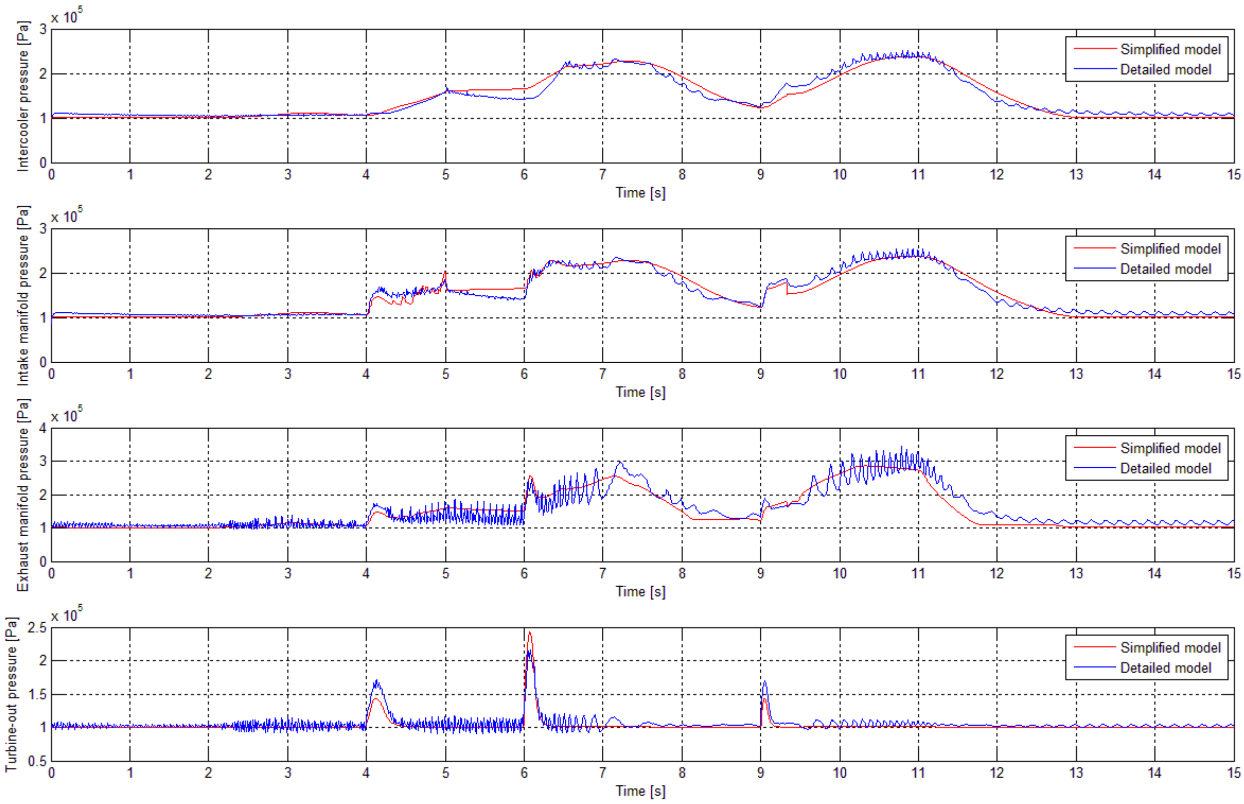

Fig. 5. Intercooler-, intake manifold-, exhaust manifold and turbine out pressure fit to the detailed model

7 Conclusion

The more and more rigorous emission standards, the increasing energy prices and the driver's demand for good

driveability give even more challenge to the engine developers. The most critical limitations are for the soot and nitrogen oxide emission. It was concluded, that the formation of the pollutants is concentrated to load steps due to rich mixture and decreased amount of recirculated exhaust gases. The lacking fresh air in transients have also detrimental effect on the fuel consumption and delays the torque buildup of the engine. As a consequence, to avoid these negative effects, a diesel engine was equipped with a compressed air booster system, with a high pressure EGR loop and with an exhaust flap to support the exhaust gas backflow. Measurements were carried out on an engine dynamometer. For further and easier investigation, parameter estimation for the simplified model and for the future controller tuning, a detailed model was designed and validated in GT Suite. To precisely adjust the fresh air and EGR flow into the engine, a model-based controller was targeted. As the first step of the development a simplified model was designed based on first engineering principles. For the pressure of the four balance volumes and for the compressor power differential equations were defined. The model was converted into state-space form which is the most convenient model representation for analysis and controller design. The simplified model was validated in a transient test case from the urban part of the European Transient Cycle compared to the detailed model which shows good fit of the pressure levels.

In next steps of the work with sensitivity analysis and some parameter identification methods simpler model structure and the improved fit is desired. With the obtained model the controller design can be started.

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