OPTIMIZATION BASED MEAN VALUE MODEL OF TURBOCHARGED DIESEL ENGINES

S. Puchner, B. Winkler-Ebner, D. Alberer, L. del Re
Johannes Kepler University Linz

Corresponding Author: D. Alberer, Johannes Kepler University Linz, Inst. f. Design und Regelung mechatronischer Systeme, Alternbergerstrasse 69, 4040 Linz, daniel.alberer<at>jku.at

Abstract. Model based control is generally seen as an indispensable tool to meet the coming challenges, but unfortunately models are difficult to determine. Available physical models are often too complex to be useful for control, and simplified models tend to be too imprecise. This paper proposes a mixed approach in which the model structure is determined by physical understanding, but the complexity of each block corresponds to its relevance for dynamical measurements and not to the physical complexity. Additionally, parameters are determined using standard measurements and optimization tools, and therefore do not rely on data from the producers of the components. This paper is focused on a nonlinear model for a turbo-charged Diesel engine with swirl valve and extends a previous work.

1 Introduction

While engine control is still widely based on heuristic approaches and maps, there is a wide consensus that model based control is an indispensable component to meet the coming challenges and exploit at best the increasing number of degrees of freedom offered by modern and coming engines [1], [2]. However, model based control requires models, whose determination is not a trivial task and usually the most demanding step in the control system development. Indeed, a good physical insight in engine dynamics and combustion properties allows deriving the main physical relationships between different components of the system, but the resulting models tend not to be sufficiently precise to enforce an optimal or almost optimal behavior of the engine as many "small" extremely hard to model effects - like friction nonlinearities, thermal properties etc. are critical for the precise behavior prediction. This is the main practical reason for the classical approach consisting in measuring almost all operating conditions and tuning the feedforward – basically the inverse model – to enforce optimal operation. Identification would be a suitable alternative, but unfortunately identification can be used efficiently only for few model classes, especially for linear ones, who do not easily capture the system behavior to the desired precision [4].

Against this background this paper proposes a mixed approach. The model structure is determined by physical understanding, the complexity of each block corresponding to its relevance for dynamical measurements and not to the physical complexity. The use of producer information is kept as limited as possible, and the parameters of each component are determined from standard overall measurements and optimization routines and not by component specific measurements, in such a way that a tracking of parameter changes during normal operation and wear could be possible along the same way. An important consequence of this is that the parameters eventually include also the non-modeled nonlinearities and special effects responsible to a large extent for the model precision. This paper presents a nonlinear model for a turbo-charged Diesel engine with swirl valve and is based on and extends a previous work by [3].

2 Problem statement

The interest in Diesel engine models is quite old [5] and has concerned both the engine itself and its used control loops. Engine related models have been very often quite complex models, with crank angle resolution and sometimes multidimensional combustion models [6], while engine related control models have been typically so called mean value models (MVEM), which resume all the behavior of the engine during a rotation as a single value. There are several mean value models for Diesel engines [7], one of the best known being the model by [5], which proposes a simple and logic structure of the airpath. Extending this basis [3] has designed a model which has a sufficiently simple structure to allow an optimization using QuickFit[1] and has shown a very good dynamic performance. However, also this model has limits. In particular, it contains no intake temperature dependency, the very simple structure necessary to allow the use of optimization tools is too simple for some operating conditions. Additionally, the model does not include a swirl valve model and some parts, in particular the exhaust pressure model and the volumetric efficiency, are not sufficiently precise. Similar limits are found also in other models in the available literature.

[1] http://www.iwr.uni-heidelberg.de/~Christian.Kirches/software_quickfit.html 10.1.2009

Against this background the work related to this paper was centered on designing a model whose structure follows the well known physical assumptions, but whose detail degrees and therefore its optimization possibility is defined not by the complexity of the components, but by their impact on the overall behavior of the system. For example, the turbocharger is precisely modeled in some works, but not here because the exact turbocharger maps are not critical for the dynamical behavior, while the time delays are.

The device under test for the mean value engine model (MVEM) was a BMW M47TÜ-OL Diesel engine with external exhaust gas recirculation and a variable geometry turbine turbocharger. The necessary measurement was performed on a dynamic engine test bench shown in figure 1.

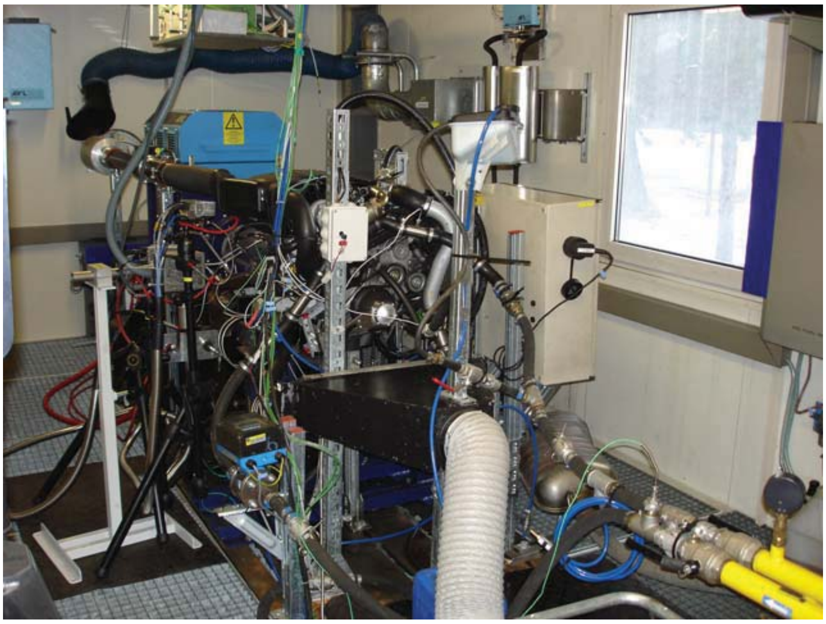

Figure 1. Experimental engine setup

Figure 2 shows the configuration of the engine's air path, which consists of the intake and exhaust manifold, a path for the EGR (valve and EGR-cooler), the turbocharger with the intercooler and the cylinder block with the swirl flaps. All the elements described in figure 2 were modeled by the MVEM.

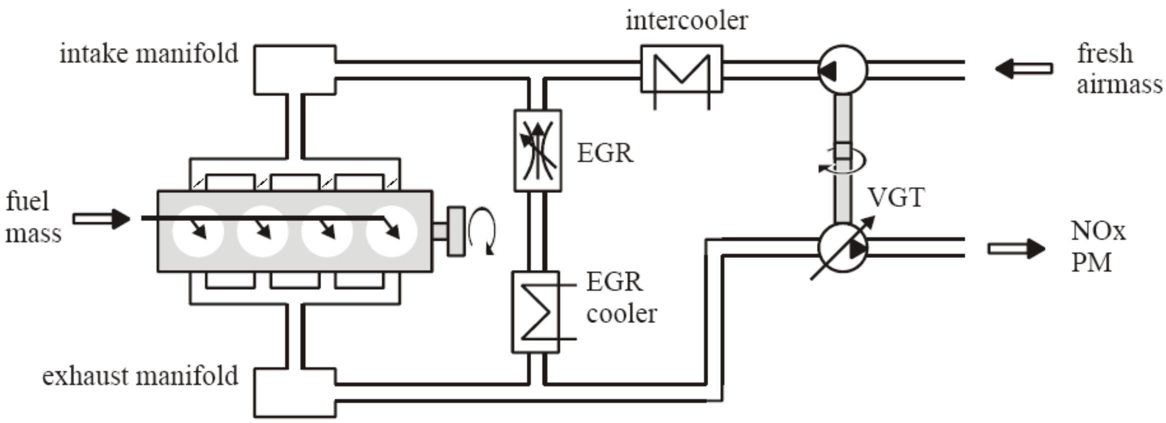

Figure 2 Scheme of the air path of a turbo charged Diesel engine

The MVE is characterized by the combination of physical equation with static maps. The static behavior of the engine is captured by maps and the dynamics is represented by physical equations. The physical model background is based on basic thermodynamics like the ideal gas equation and on equations for conservation of mass and energy. This approach is a very common one and can be found in many works on MVEs [3,5]. The model has 5 inputs (position of EGR and VGT valve, injected fuel mass, engine speed, swirl state) and 5 outputs (intake pressure MAP, fresh air flow MAF, oxygen concentration in the exhaust X_{O2ex} and oxygen concentration in the intake manifold X_{O2in},, crankshaft torque T).

3 Model Equations

As the engine structure is a standard turbocharger engine, the model consists essentially of the same structure as e.g. in [5], albeit with some modifications and different maps (which are not assumed to be polynomial). For completeness, we reproduce here also the equations from [5].

The turbocharger is driven by the mass flow

$$W_{xt} = f_{Wxt}^{map}\left(x_{VGT}, p_x\right) \tag{1}$$

with a corresponding mechanical power on the turbine

$$P_t = W_{xt} c_p T_x \left(1 - \left(\frac{p_x}{p_{40}}\right)^{\frac{\kappa-1}{\kappa}}\right), \tag{2}$$

which is translated in a low pass way to the compressor

$$\dot{P}_c = \frac{1}{\tau_{VGT}}\left(\eta_m P_t - P_c\right), \tag{3}$$

with the corresponding mass flow over the compressor and the temperature after compressor

$$\begin{aligned} W_{ci} &= f_{Wci}^{map}\left(P_c, p_i\right) \\ T_{ci} &= f_{Tci}^{map}\left(W_{ci}, p_i\right) \end{aligned}. \tag{4}$$

The main difference with respect to [5] is the model for the exhaust back pressure p_{40} obtained from the exhaust mass flow

$$p_{40} = f_{p40}^{map}\left(W_{xt}\right). \tag{5}$$

The exhaust gas recirculation follows the description

if $p_x \geq p_i$

$$W_{xi} = \frac{A_{EGR(X_{EGR})} p_x}{\sqrt{RT_x}} \sqrt{2 p_r (1 - p_r)} \tag{6}$$

if $p_x < p_i$

$$W_{xi} = \frac{A_{EGR(X_{EGR})} p_x}{\sqrt{RT_x}} \sqrt{2 \frac{1}{p_r}(1 - \frac{1}{p_r})} \tag{7}$$

where the pressure ratio p_r is defined as

$$p_r = \frac{p_i}{p_x}. \tag{8}$$

The steady state temperature drop over the recirculation path can be represented as a function of exhaust gas temperature and EGR flow

$$T_{xi} = f_{Txi}^{map}\left(T_x, W_{xi}\right) \tag{9}$$

and the dynamics is modeled as a first order lag

$$\dot{T}_{xif} = \frac{1}{\tau_{EGR}}\left(T_{xi} - T_{xif}\right). \tag{10}$$

The dynamics of the intake manifold is determined by the ideal gas law, the conservation of mass and the conservation of energy

$$\begin{aligned} \dot{m}_i &= W_{xi} + W_{ci} - W_{ie} \\ T_i &= \frac{p_i V_i}{R m_i} \\ \dot{p}_{i(t)} &= \frac{R\kappa}{V_i}\left(T_{ci(t)} W_{ci(t)} + T_{xi(t)} W_{xi(t)} - T_{ie(t)} W_{ie(t)}\right) \end{aligned} \quad . \tag{11}$$

Similarly, for the exhaust manifold it holds

$$\begin{aligned} \dot{m}_x &= W_{ex} - W_{xi} - W_{xt} \\ T_x &= \frac{p_x V_x}{R m_x} \\ \dot{p}_{x(t)} &= \frac{R\kappa}{V_x}\left(T_{ex(t)} W_{x(t)} - T_{xi(t)} W_{xi(t)} - T_{xt(t)} W_{xt(t)}\right) \end{aligned} \quad . \tag{12}$$

The volumetric efficiency of the cylinder block is given by

$$\eta_V = f_{\eta v}^{map}\left(n_{mot}, p_i\right) \tag{13}$$

and the total flow to the combustion chamber

$$W_{ie} = \eta_v \frac{m_i}{V_i} \frac{\omega}{2\pi} \frac{V_d}{2} . \tag{14}$$

The cylinder exhaust temperature is described by a function of the total flow into the combustion chamber and the injected fuel quantity

$$T_e = f_{T_e}{}^{map}\left(W_f, W_{ie}\right) \tag{15}$$

The oxygen values of the intake and the exhaust were modeled, too.

$$\begin{aligned} \dot{m}_{i,O2} &= \frac{m_{x,O2}}{m_x} W_{xi} + 0.21 W_{ci} - \frac{m_{i,O2}}{m_i} W_{ie} \\ W_{ex,O2} &= f_{Wex,O2}^{map}\left(W_{ie}, W_f\right) \\ \dot{m}_{x,O2} &= W_{ex,O2} - \frac{m_{x,O2}}{m_x}(W_{xi} + W_{xt}) \end{aligned} \tag{16}$$

A necessary extension here is the modeling of the swirl flap, which is represented as a function of engine speed and the injected fuel quantity:

$$S_{pos} = f_{Spos}^{map}\left(n, W_f\right) \tag{17}$$

S valve closed → $S_{pos} = 1$
S valve open → $S_{pos} = 0$

All of the required maps for the simulation are included for high swirl as well as for low swirl. In several blocks a linear interpolation is done as a function of the swirl valve position

$$x = x_{S_{closed}} S_{pos} + x_{S_{open}} (1 - S_{Pos}) . \tag{18}$$

The steady state engine torque is modeled as a function of the injected fuel and the engine speed

$$T_{mot} = f_{Tmot}^{map}\left(W_f, n\right), \tag{19}$$

whereas the torque dynamics is assumed to lie in the injection system. Therefore a second order system with a varying time constant was selected

$$\dot{x} = \begin{bmatrix} 0 & 1 \\ -\frac{1}{\tau_{IPS}^2} & -\frac{2}{\tau_{IPS}} \end{bmatrix} x + \begin{bmatrix} 0 \\ \frac{1}{\tau_{IPS}} \end{bmatrix} W_{fdem}$$

$$W_f = x_1 \qquad . \qquad (20)$$

$$\tau_{IPS} = \frac{60}{n}$$

Notation and symbols are listed in the annex.

4 Parameter optimization

To tune the dynamic behavior of the model the parameters have to be adapted in the appropriate way. In this work, the problem was especially important those for listed in table 5.1 as most parameters could be taken over from [3]. As already mentioned, due to the simple structure of the model these parameters do not correspond completely to their basic physical meaning, but must be considered rather as design parameters to achieve the best dynamic performance. To choose these parameters in an optimal way an optimization was done using *fmincon* in Matlab.

The optimization algorithm minimizes a cost function, in our case the weighted sum of squared modeling errors of MAF and MAP. The optimization variables are shown in table 1, the limits and the results in table 2. After a new parameter set is calculated by the algorithm the simulation of the model is run and the cost function is calculated again. This way the algorithm searches for the optimal parameter combination to represent the dynamics of the MVEM.

The input signal for this dynamic optimization is the FTP-driving cycle, which serves as a dynamic excitation for the model. Moreover table 2 shows the initial values of the parameters and the final values after the optimization and figure 3 shows the quality of improvement for MAP and MAF.

Parameter	Meaning
Vi	Intake manifold Volume in m^3
Vx	Exhaust manifold volume
$\boldsymbol{\eta_m}$	Efficiency of the turbocharger
$\boldsymbol{\tau_{vgt}}$	Time constant of the turbocharger dynamics
$\boldsymbol{\tau_{EGR}}$	Time constant of the EGR-dynamics

Table 1. Parameters

	initial value	**lower limit**	**upper limit**	**optimized**
Vi [mm^3]	0.0016	0.001	0.05	0.0109
Vx [mm^3]	0.006	0.0005	0.03	0.0249
$\boldsymbol{\eta_m}$ **[-]**	1	0.5	0.9	0.7090
$\boldsymbol{\tau_{vgt}}$ **[s]**	0.1	0.1	5	0.2124
$\boldsymbol{\tau_{EGR}}$ **[s]**	1.8	0.1	5	0.5

Table 2. Optimization limits and results

The results of the optimization can be seen in Figure 2

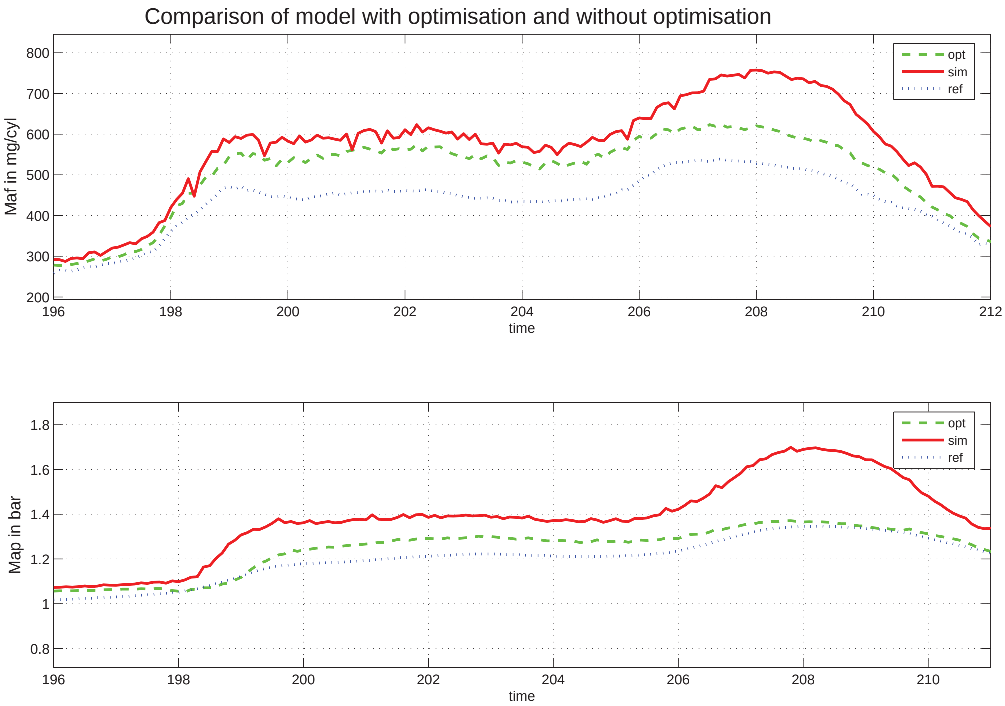

Figure 3.Effect of the I/O parameter optimization

5 Model Performance

To test the dynamic behavior of the model one part of our validation was to test it on a FTP driving cycle. Figure 3 shows the comparison of the simulated and the measured values for MAF and MAP. It can be seen easily that the model is able to describe the main dynamic tendencies of the air path in a quite good way. Table 3 shows the value of the Pearson correlation coefficient between the measured and computed values of MAF, MAP and T. The correlation coefficient proves that all the values can be modeled with enough accuracy although it can be seen that the results for MAP are a little bit worse. Other dynamic tests, for example a test on a NEDC-driving cycle gave essentially the same result.

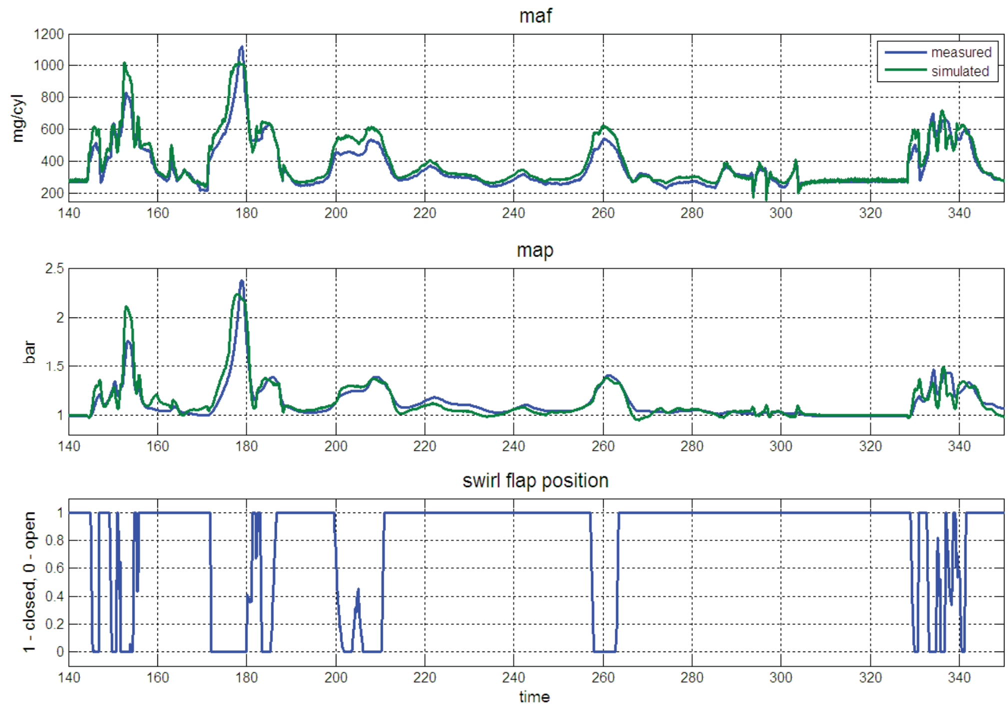

Figure 4.Overall model performance during swirl valve operation

Model output	*Correlation coefficient*
MAF	0.956
MAP	0.875
T	0.972

Table 3. Correlation coefficient for the most important model outputs

6 Conclusion

As the present paper shows, it was possible to achieve a model with a reasonable physical background without having to perform specific measurements or rely on manufacturer's data. On one side, this opens the door to fast model tuning and possibly adaptive schemes, on the other side it is impossible to separate different physical phenomena having very similar effects, so that at least part of the parameters lose the meaning of true physical parameters and become rather tuning parameters, with all implications known for this kind of approach. It must also be said that the optimization would have been impossible if no sensible initial values were available (although, as table 2 shows, the final values may be quite different). Also segmenting the model in parts which can be analyzed independently strongly simplifies the numerical problems. Also the static performance of the model first its turbocharger model, should be improved.

Still, as an overall result it can be said that the combination of a cautious physical approach and data based optimization seems a promising approach to reduce the modeling effort while keeping high precision standard.

7 Acknowledgement

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8 References

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9 Annex: notations

$(.)_a$	ambient	
$(.)_c$	compressor	
$(.)_{ci}$	compressor to intake manifold	
$(.)_e$	engine	
$(.)_f$	fuel	
$(.)_{EGR}$	exhaust gas recirculation	
$(.)_i$	intake	
$(.)_{ie}$	intake to engine	
$(.)_{Mot}$	engine	
$(.)_t$	turbine	
$(.)_{VGT}$	variable geometry turbine	
$(.)_{x\cdot}$	exhaust	
$(.)_{xi}$	exhaust to intake (EGR-path)	
$(.)_{xt}$	exhaust to turbine	
$(.)_{40}$	after turbine	
$A_{(.)}$	area	m^2
c_p	specific heat at const. pressure	J/kg/K
c_v	specific heat at const. volume	J/kg/K
$M_{(.)}$	torque	Nm
$m_{(.)}$	mass	kg
$n_{(.)}$	engine speed	rpm
$p_{(.)}$	pressure	Pa
$P_{(.)}$	power	W
$T_{(.)}$	temperature	K
$V_{(.)}$	volume	m^3
$W_{(.)}$	mass flow	kg/s
v	speed	m/s
$x_{(.)}$	actuator position	%
$\eta_{(.)}$	efficiency	%
ρ	density	kg/m^3
κ	specific heat ratio cp/cv	
λ	raw lambda value	
$\tau_{(.)}$	time constant	s
$m_{i,O2}$	O_2-mass intake	kg
$m_{x,O2}$	O_2-mass exhaust	kg
$XO2_{in}$	O_2-concentration intake manifold	%
$XO2_{ex}$	O_2-concentration exhaust manifold	%
S_{pos}	swirl position	%