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Modeling of a Large Marine Two-Stroke Diesel Engine with Cylinder Bypass Valve and EGR System

Guillem Alegret * Xavier Llamas **
Morten Vejlgaard-Laursen * Lars Eriksson **

* *MAN Diesel & Turbo, Copenhagen, Denmark*
** *Vehicular Systems, Dept. of Electrical Engineering Linköping University, Sweden, xavier.llamas.comellas@liu.se*

Abstract: A nonlinear mean value engine model (MVEM) of a two-stroke turbocharged marine diesel engine is developed, parameterized and validated against measurement data. The goal is to have a computationally fast and accurate engine model that captures the main dynamics and can be used in the development of control systems for the newly introduced EGR system. The tuning procedure used is explained, and the result is a six-state MVEM with seven control inputs that capture the main system dynamics.

Keywords: Engine modeling, diesel engines, parametrization, validation, nonlinear systems

1. INTRODUCTION

The upcoming Tier III regulation (International Maritime Organization, 2013) is the next milestone for EGR technology in large two-stroke engines. The EGR system is used to reduce NO_x emissions by recirculating a fraction of the exhaust gas into the scavenging manifold. This results in a lower combustion peak temperature and consequently a reduction in NO_x formation. Due to the high financial costs of performing tests on a real engine, a reliable and fast dynamic engine model is an important tool for the development of new EGR control systems.

A lot of research can be found in literature about Mean Value Engine Models (MVEM) with EGR systems for automotive engines, e.g., Wahlström and Eriksson (2011) and Nieuwstadt et al. (2000). However, much less research has been done in the same area with large marine two-stroke diesel engines. A few examples are Blanke and Anderson (1985), Theotokatos (2010) where an MVEM of a marine engine was developed, and Hansen et al. (2013) where a similar model of the engine used here was proposed.

In this study the proposed MVEM is based on the 4T50ME-X test engine from MAN Diesel & Turbo, which is a turbocharged two-stroke diesel engine with direct injection, uniflow scavenging and variable valve timing. It can provide a maximum rated power of 7080 kW at 123 RPM. It is equipped with an EGR system and a Cylinder Bypass Valve (CBV). The purpose of the valve is to keep the desired turbocharger speed when the engine operates under high EGR rates. In those situations less energy is transferred through the turbine, thus part of the compressor air mass flow is bypassed to boost the turbine.

2. MODELING

The MVEM consists of six states and seven control inputs. The states are scavenging manifold pressure and oxygen mass fraction, p_{scav} and $X_{O,scav}$, compressor outlet pressure, $p_{c,out}$, exhaust manifold pressure and oxygen mass fraction, p_{exh} and $X_{O,exh}$ and turbocharger speed, ω_{tc}. The control inputs are fuel mass flow, $\dot{m}_{fuel}$, EGR blower speed, ω_{blow}, fuel injection time, t_{inj}, fuel injection angle α_{inj}, exhaust valve closing angle, α_{EVC}, cut-out valve (COV) position, u_{cov}, and CBV position, u_{cbv}. Figure 1 gives an overview of the model. The engine model consists of several interconnected submodels which are introduced in the following subsections.

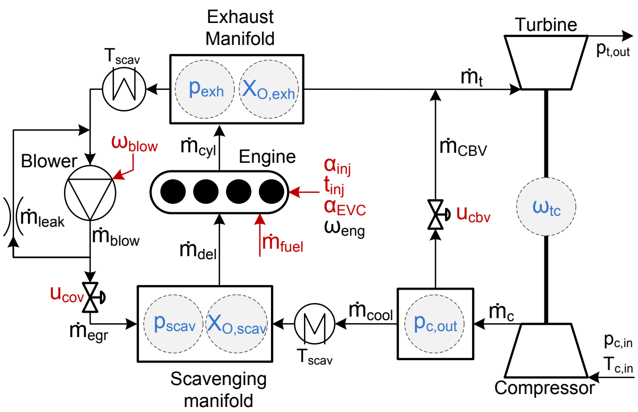

Fig. 1. Structure of system with state variables (blue) and control inputs (red)

2.1 Turbocharger

The turbocharger model includes submodels for the compressor, the turbine and the connecting shaft.

Compressor
The mass flow and efficiency models of the compressor are based on the parameterization of the performance maps in SAE format. The turbocharger speed and the compressor mass flow in the performance map are corrected in order to take into account changes in ambient conditions. The com-

pressor mass flow is modeled using super-ellipses centred at the origin. A similar approach is found in Leufvén and Eriksson (2013). The explicit expression of a super-ellipse is

$$\dot{m}_{c,corr} = a\left(1-\left(\frac{\Pi_c}{b}\right)^n\right)^{\frac{1}{n}} \tag{1}$$

where Π_c is the pressure ratio over the compressor, $p_{c,out}/p_{c,in}$. The variables a, b and n are described by third order polynomials of the corrected turbocharger speed, so the model has 12 tuning parameters.

The compressor efficiency is modeled by parameterizing the manufacturer performance map with rotated and translated ellipses. The implicit expression of an ellipse rotated α and translated from the origin to (a_0, b_0) is as follows

$$\left(\frac{(x-a_0)\,\cos\alpha-(y-b_0)\,\sin\alpha}{a}\right)^2+ \left(\frac{(x-a_0)\,\sin\alpha+(y-b_0)\cos\alpha}{b}\right)^2=1 \tag{2}$$

where in this case x corresponds to $\dot{m}_c$ and y corresponds to η_c. The coefficients a_0, b_0, a, b, and α are described using second order polynomials of Π_c so the model consists of 15 parameters to estimate.

Turbine
The turbine corrected mass flow is described as in Eriksson and Nielsen (2014)

$$\dot{m}_{t,corr} = C_t\sqrt{1-\Pi_t^{k_t}} \tag{3}$$

where Π_{turb} is the pressure ratio over the turbine, $p_{t,out}/p_{exh}$. Moreover, k_t and C_t are parameters to be estimated.

The turbine efficiency is commonly modeled using the Blade Speed Ratio (BSR), e.g. Wahlström and Eriksson (2011) and Eriksson and Nielsen (2014)

$$BSR = \frac{R_t\,\omega_t}{\sqrt{2\,c_{p,e}\,T_{t,in}\left(1-\Pi_t^{1-\frac{1}{\gamma_e}}\right)}} \tag{4}$$

where R_t is the turbine blade radius. The turbine efficiency is again modeled with rotated and translated ellipses using (2). In this case x corresponds to the BSR and y corresponds to the η_t. The coefficients a_0, b_0, a, b, and α are described as second order polynomials of the corrected turbocharger speed, thus 15 parameters need to be determined.

Connecting Shaft
The turbocharger shaft speed is described by Newton's second law using the power recovered from the exhaust gas by the turbine and transferred to the compressor

$$\frac{d}{dt}\omega_{tc} = \frac{P_t-P_c}{J_t\ \omega_{tc}} \tag{5}$$

where the parameter J_t corresponds to the overall turbocharger inertia. P_t and P_c are the turbine and compressor powers, respectively. Note that the mechanical efficiency is not included in (5), it is already included in the turbine efficiency of the SAE map.

The power generated by the turbine and the power consumed by the compressor are defined as in Dixon (1998)

$$P_t = \eta_t\,\dot{m}_t\,c_{p,e}\,T_{t,in}\left(1-(\Pi_t)^{\frac{\gamma_e-1}{\gamma_e}}\right) \tag{6}$$

$$P_c = \frac{\dot{m}_c c_{p,a} T_{c,in}}{\eta_c}\left((\Pi_c)^{\frac{\gamma_a-1}{\gamma_a}}-1\right) \tag{7}$$

2.2 Control Volumes

The model consists of three control volumes. The compressor outlet and the two manifolds, they are all modeled with standard isothermal models as proposed in Heywood (1988) and Eriksson and Nielsen (2014).

The pressure at the compressor outlet is described by

$$\frac{d}{dt}p_{c,out} = \frac{R_a\,T_{c,out}}{V_{c,out}}\left(\dot{m}_c-\dot{m}_{cool}-\dot{m}_{cbv}\right) \tag{8}$$

where $V_{c,out}$ is the control volume size, and it has to be estimated and $T_{c,out}$ is described in (14).

At the scavenging manifold, the temperature is assumed to be constant since the cooler is considered to be ideal and capable of maintaining a constant scavenging temperature. Two states are needed to fully characterize the manifold, the pressure and the oxygen mass fraction. The pressure is governed by the following differential equation

$$\frac{d}{dt}p_{scav} = \frac{R_a T_{scav}}{V_{scav}}\left(\dot{m}_{cool}+\dot{m}_{egr}-\dot{m}_{del}\right) \tag{9}$$

where V_{scav} is the volume of the manifold and has to be estimated. The oxygen mass fraction is described as in Wahlström and Eriksson (2011)

$$\frac{d}{dt}X_{O,scav} = \frac{R_a T_{scav}}{p_{scav}V_{scav}}\left(X_{O,exh}-X_{O,scav}\right)\dot{m}_{egr} + \frac{R_a T_{scav}}{p_{scav}V_{scav}}\left(X_{O,a}-X_{O,scav}\right)\dot{m}_{cool} \tag{10}$$

where $X_{O,a}$ is the mass fraction of oxygen in dry air.

As in the previous manifold, two states characterize the exhaust manifold, the pressure and the oxygen mass fraction. The exhaust pressure is driven by the following differential equation

$$\frac{d}{dt}p_{exh} = \frac{R_e\,T_{exh}}{V_{exh}}\left(\dot{m}_{cyl}-\dot{m}_{egr}-\dot{m}_{exh,out}\right) \tag{11}$$

with

$$\dot{m}_{exh,out} = \dot{m}_t-\dot{m}_{cbv} \tag{12}$$

and where V_{exh} is the exhaust manifold volume and a tuning parameter, and $\dot{m}_{cyl} = \dot{m}_{del}+\dot{m}_{fuel}$. The oxygen mass fraction is defined in a similar manner as in the scavenging manifold

$$\frac{d}{dt}X_{O,exh} = \frac{R_e T_{exh}}{p_{exh}V_{exh}}\left(X_{O,cyl}-X_{O,exh}\right)\dot{m}_{cyl} \tag{13}$$

where $X_{O,cyl}$ is the oxygen mass fraction coming out from the cylinders. Since the injected fuel combustion is assumed to be ideal and complete, $X_{O,cyl}$ is calculated as equation (16) in Wahlström and Eriksson (2011).

2.3 CBV

The CBV model consists of a submodel for the compressor outlet temperature, a submodel for the flow through the CBV valve and a submodel for the flow through the cooler.

The temperature at the compressor outlet is calculated using the definition of the adiabatic efficiency of the compressor from Dixon (1998)

$$T_{c,out} = T_{c,in}\left(1 + \frac{(\Pi_c)^{\frac{\gamma_a - 1}{\gamma_a}} - 1}{\eta_c}\right) \tag{14}$$

The mass flow through the CBV is modeled as a compressible turbulent restriction. A generic formulation of the model is presented as follows

$$\dot{m} = \frac{A_{eff}\,p_{in}}{\sqrt{R_i T_{in}}}\sqrt{\frac{2\,\gamma_i}{\gamma_i - 1}\left(\Pi^{\frac{2}{\gamma_i}} - \Pi^{\frac{\gamma_i+1}{\gamma_i}}\right)} \tag{15}$$

for this case, $\dot{m}$ is the mass flow through the CBV, Π is the pressure ratio $p_{exh}/p_{c,out}$, γ_i and R_i are the heat capacity ratio and the specific gas constant of air, respectively. A_{eff} corresponds to the CBV effective area A_{cbv}, which in this case is variable depending on the control input u_{cbv}, and it is defined as follows

$$A_{cbv} = A_{max}(1 - \cos(u_{cbv}\,\frac{\pi}{2})) \tag{16}$$

where A_{max} is a tuning parameter that corresponds to the maximum area of the restriction.

The mass flow through the cooler is described by an incompressible turbulent restriction, described in Eriksson and Nielsen (2014)

$$\dot{m}_{cool} = k_{cool}\sqrt{\frac{p_{c,out}\,(p_{c,out} - p_{scav})}{T_{c,out}}} \tag{17}$$

where k_{cool} is a parameter to be estimated.

In situations where the CBV is open, the turbine inlet temperature cannot be assumed to be equal to the exhaust temperature. To consider the temperature drop caused by the CBV flow, the perfect mixing model described in Eriksson and Nielsen (2014) is used

$$T_{t,in} = \frac{T_{exh}\,c_{p,e}\dot{m}_{exh,out} + T_{c,out}\,c_{p,a}\,\dot{m}_{cbv}}{c_{p,e}\,\dot{m}_{exh,out} + c_{p,a}\,\dot{m}_{cbv}} \tag{18}$$

With this formulation, an algebraic loop is encountered between the $T_{t,in}$ and the $\dot{m}_t$ calculations. In order to break the algebraic loop, it is assumed that $\dot{m}_{exh,out}$ in (18) can be approximated by its steady state value $\dot{m}_{exh,out} = \dot{m}_{cyl} - \dot{m}_{egr}$.

The exhaust oxygen measurement equipment is installed downstream of the turbine. When the CBV is open, it affects the measurement. Therefore, a new oxygen mass fraction is calculated in (19) for validation purposes.

$$X_{O,t} = \frac{X_{O,exh}\,\dot{m}_{exh,out} + X_{O,a}\,\dot{m}_{cbv}}{\dot{m}_t} \tag{19}$$

In this expression, the $\dot{m}_{exh,out}$ used is described by equation (12).

2.4 Cylinders

The mass flow through four-stroke engines is commonly modeled with the volumetric efficiency as in Wahlström and Eriksson (2011) and Heywood (1988). For two-stroke engines, the mass flow through all cylinders can be approximated with the flow through a compressible turbulent restriction. The continuous flow represents the average flow through all cylinders. The same approach is found in Hansen et al. (2013) and Theotokatos (2010). The same generic equation (15) is used, and in this case the $\dot{m}$ is the delivered mass flow $\dot{m}_{del}$ through the cylinders, Π is the pressure ratio over the cylinders p_{exh}/p_{scav}, γ_i and R_i correspond to the heat capacity ratio and the specific gas constant of air. A_{eff} is the effective area of the restriction, and has to be estimated.

It is common to characterize the scavenging process in two-stroke engines with the scavenging efficiency η_{scav} and the trapping efficiency η_{trap}. Their definitions can be found in Heywood (1988). The delivery ratio (DR) is defined as the ratio between the delivered flow and the ideal flow at the scavenging manifold density

$$DR = \frac{2\pi\,\dot{m}_{del}}{n_{cyl}\,\omega_{eng}\,V_1}\left(\frac{R_a\,T_{scav}}{p_{scav}}\right) \tag{20}$$

The model proposed here is a combination of the two limited ideal models introduced in Heywood (1988), the perfect displacement and the complete mixing. The perfect displacement assumes that the burned gases are displaced by the fresh gases without mixing, on the other hand, the complete mixing model assumes instantaneous mixing of the gases when fresh mixture enters the combustion chamber. By introducing the tuning parameters K_{se1} and K_{se2} in the complete mixing model (21) and (22), an intermediate formulation is obtained, with the purpose of taking into account the late exhaust valve closing.

$$\eta_{scav} = 1 - e^{-K_{se1}\,DR} \tag{21}$$

$$\eta_{trap} = \frac{1 - e^{-K_{se2}\,DR}}{DR} \tag{22}$$

Limited pressure diesel cycle
As an overview, six changes to the cycle presented in Wahlström and Eriksson (2011) have been incorporated.

(i) The constant volume burned ratio x_{cv} is considered variable. The maximum pressure rise in the cylinders is regulated by the control system as a safety measure. The regulation is accomplished by delaying the injection. To be able to model late injection, the x_{cv} is considered a linear function of the start crank angle and duration of the injection. The model is shown in (23). A similar model for x_{cv} is shown in Lee et al. (2010).

$$x_{cv} = c_1 + c_2\,\alpha_{inj} + c_3\,t_{inj} \tag{23}$$

where the three parameters c_i have to be estimated.

(ii) The compression process is considered to start when the exhaust valve closes. In that instant the crank angle is given by α_{EVC}. The volume of the combustion chamber based on the crank angle is used in the limited pressure cycle calculations, and it is defined as equation (4.3) from Eriksson and Nielsen (2014). Also, the expansion process is assumed to last until the bottom dead center.

(iii) Both the compression and the expansion processes are considered polytropic Jiang et al. (2009) in order to consider heat exchange with the cylinder walls, both polytropic exponents of the compression and expansion are tuning parameters.

(iv) The delivered mass flow is assumed to be heated by a tuning factor dT_{cyl} before the cycle starts. The heating affects both the trapped and the short-circuited flows (25). The pressure of the trapped gas when the combustion chamber is sealed is assumed to be the scavenging pressure, while the temperature is described by

$$T_1 = T_{cyl}\,\left(1-\eta_{scav}\right)+\eta_{scav}\left(T_{scav}+dT_{cyl}\right) \tag{24}$$

The algebraic loop between the initial cycle temperature, T_1 and the cylinder out temperature, T_{cyl}, is solved using the previous sample value for T_{cyl} similar to what is done in Wahlström and Eriksson (2011).

(v) The $c_{v,a}$ before the constant volume combustion starts and the $c_{p,a}$ at the beginning of the constant pressure combustion are calculated based on the temperatures at the respective crank angles. To perform such calculation, the NASA polynomials are used to describe these parameters in terms of temperature. The same polynomials found in Goodwin et al. (2014) are used here.

(vi) To determine the exhaust temperature T_{exh}, characterized by the mixture of the short-circuited flow and the trapped flow in the cylinder at their respective temperatures, the perfect mixing model is used again in the same manner as (18). The short-circuited flow is defined as

$$\dot{m}_{sh} = \dot{m}_{del} - \dot{m}_{trap} \tag{25}$$

Using the pressures and the volumes of each process in the thermodynamic cycle, the indicated power of the cycle is computed using equations (2.14) and (2.15) in Heywood (1988). To sum up, the limited pressure cycle has eight parameters to determine.

2.5 EGR loop

The EGR loop model consists of a blower to overcome the pressure difference between exhaust and scavenge manifolds, a recirculation valve and a cut-out valve (COV) to manage the start-up of the EGR system. The flow is considered ideally cooled to scavenging temperature.

EGR Blower
The performance map is expressed in a non-dimensional space described by the Head Coefficient (Ψ) and the Flow Coefficient (Φ), their definitions are shown in (26) and (28) respectively

$$\Psi = \frac{2\,T_{scav}\,c_{p,e}\left(\Pi_{blow}^{\frac{\gamma-1}{\gamma}}-1\right)}{\left(\omega_{blow}\,R_{blow}\right)^2} \tag{26}$$

where ω_{blow} is the blower angular speed, R_{blow} is the blower blade radius and Π_{blow} is the pressure ratio over the blower p_{scav}/p_{exh}.

The non-dimensional performance map is modeled with the same approach as the compressor mass flow, but here only one speed line is parameterized. Therefore, the parameters a, b, and n are constants and need to be estimated.

$$\Phi = a\left(1-\left(\frac{\Psi}{b}\right)^n\right)^{\frac{1}{n}} \tag{27}$$

Rearranging the definition of Φ, the mass flow through the blower $\dot{m}_{blow}$ is obtained

$$\dot{m}_{blow} = \frac{p_{exh}}{R_e\,T_{scav}}\left(\Phi\,\omega_{blow}\,\pi\,R_{blow}^3\right) \tag{28}$$

The existence of a leak in the recirculation valve is known, however, its magnitude is unknown. The leak mass flow $\dot{m}_{leak}$ is modeled as a compressible turbulent restriction, like in (15). But in this case, the Π is the pressure ratio over the recirculation valve p_{exh}/p_{scav}, γ_i, and R_i corresponds to the heat capacity ratio and the specific gas constant of exhaust gas respectively. A_{eff} corresponds to the leak effective area. The resulting mass flow through the EGR system is

$$\dot{m}_{egr} = \left(\dot{m}_{blow}-\dot{m}_{leak}\right)\,f\left(u_{cov}\right) \tag{29}$$

where $f\left(u_{cov}\right)$ describes the valve dynamics as

$$f\left(u_{cov}\right) = \left(1-e^{-\frac{1}{\tau_{cov}}u_{cov}}\right) \tag{30}$$

u_{cov} only regulates the flow during start-up of the system, then the flow is controlled using the blower speed.

3. EXPERIMENTAL DATA AND TUNING PROCEDURE

The parameters in the submodels are estimated using engine measurements. The measured signals: $p_{c,in}$, $p_{t,out}$, $T_{c,in}$ and ω_{eng} are used in the estimation and in the validation of the model in the same manner as if they were inputs to the model.

Unfortunately, the oxygen sensors were not properly calibrated before the measurements. Thus the stationary values cannot be trusted and will not be used for estimation purposes. More information (types, starts and stops and number of steps) about each of the dynamic datasets can be found in the top part of Table 2.

The following relative error is used to quantify the difference between the modeled signals, y_{mod}, and the measured signals, y_{meas}

$$e_{rel}[k] = \frac{y_{mod}[k]-y_{meas}[k]}{1/N\sum_{j=1}^{N}y_{meas}[j]} \tag{31}$$

the euclidean norm of this relative error is used as the objective function to minimize in the tuning procedure.

3.1 Submodels initialization

Some of the submodels are initialized using the maps provided by the component manufacturer. Table 1 presents the stationary errors of the submodels that are initialized.

Table 1. Relative errors of the initialized submodels

Model	$\dot{m}_c$	η_c	$\dot{m}_t$	η_t	$\dot{m}_{blow}$
Mean rel. error [%]	3.11	0.69	0.19	0.31	0.47
Max rel. error [%]	14.5	3.24	0.45	1.03	0.87

To get an initial guess and to avoid overparametrization in the pressure limited cycle submodels, e.g. (23), a few extra stationary measurements are used, which are not used later in the model simulation, e.g., the maximum cylinder pressure and ordered cylinder compression pressure. This initialization is based on a least-squares optimization with T_{exh} and the indicated power of the cycle as objectives. Since some of the submodel inputs are not measured, e.g., $\dot{m}_{del}$, those submodels have to be used in this initialization.

3.2 *Overall stationary estimation*

Since there are no mass flow measurements apart from $\dot{m}_{egr}$, some submodels cannot be properly initialized, e.g., the $\dot{m}_{eng}$ or the $\dot{m}_{cbv}$. Therefore all the parameters have to be estimated together, since the optimization problem cannot be separated. Another reason for estimating all parameters at the same time is that it is difficult to attain the same stationary levels for the modeled and the measured signals by fixing the previously estimated parameters. The overall estimation is performed with 27 different stationary points extracted from the estimation datasets. A point is considered stationary when the pressure and temperature signals are stabilized.

The measured states are used as inputs in the optimization since they cannot be integrated for isolated stationary points. To ensure that the model outputs are stationary at the stationary points, the derivative terms of (5), (8), (9) and (11) are added into the objective function weighted by the mean of the measured state to provide fair comparison. The objective function is defined as

$$V_{stat}(\theta) = \frac{1}{NM}\sum_{i=1}^{M}\sum_{n=1}^{N}\frac{(\dot{x}^i[n])^2}{1/N\sum_{j=1}^{N}x^i_{meas}[j]} + \frac{1}{NS}\sum_{i=1}^{S}\sum_{n=1}^{N}(e^i_{rel}[n])^2 \tag{32}$$

where the first row minimizes the residuals of the dynamic models. With $x^1 = p_{scav}$, $x^2 = p_{exh}$, $x^3 = p_{c,out}$ and $x^4 = \omega_t$, the second row minimizes the relative error of the EGR mass flow, the exhaust temperature and the engine indicated power. N is the number of stationary points available. The vector θ represents the parameters to be estimated, which in this case are all the static parameters, except for the compressor parameters and the turbine efficiency parameters. This selection has proven to be a good trade-off between objective function complexity and model accuracy.

3.3 *Dynamic estimation*

Keeping the static parameters already estimated fixed, the next step is to tune the parameters of the dynamic models (5), (8), (9), (11) and (30). From the available datasets, 13 step responses were extracted and used in the estimation. These steps consist of EGR blower speed steps, fuel flow steps and CBV steps. In the same manner as it is done in Wahlström and Eriksson (2011), the measurements and the model outputs are normalized so the stationary errors have no effect on this estimation. The objective function used is

$$V_{dyn}(\theta) = \sum_{i=1}^{J}\sum_{z=1}^{D}\frac{1}{L_z}\sum_{l=1}^{L_z}(x^i_{meas,n}[l] - x^i_{mod,n}[l])^2 \tag{33}$$

where x^i are the control volume pressures and the turbocharger speed, J, is the number of states (excluding oxygen mass fractions), D is the number of steps used, and L_z is the length of each step. The parameter vector is thus $\theta = [J_t, V_{scav}, V_{exh}, V_{c,out}, \tau_{cov}]^T$.

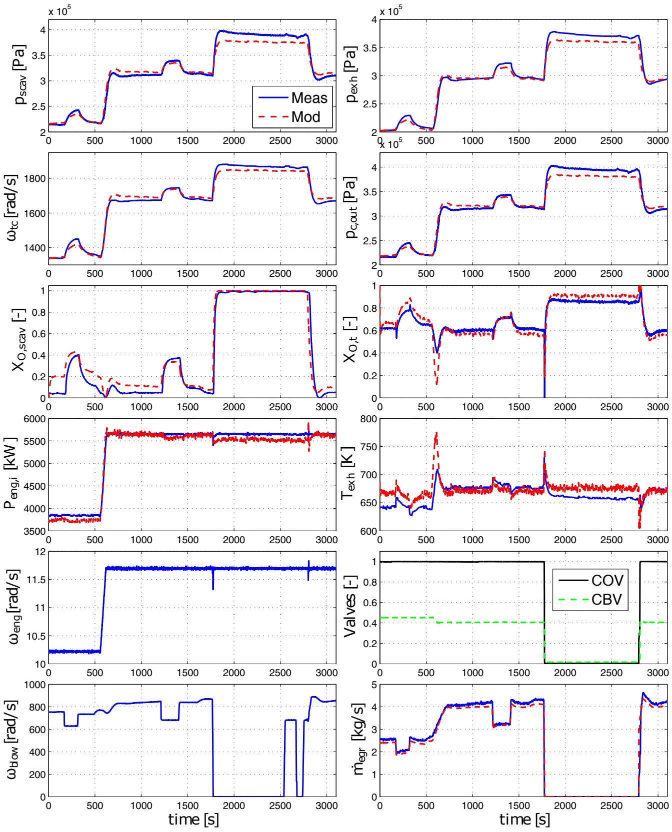

Fig. 2. Model simulation vs measurements of dataset 11.

4. MODEL VALIDATION

Table 2 presents the mean relative errors in percentage for all dynamic datasets. The 27 extracted stationary points are used to compute the mean required in the denominator of (31). This is done to provide a fair comparison between them, so all errors for different datasets are weighted with the same mean value. Excluding the EGR mass flow, the model errors are below 6.28% and in general below 3%. A higher error is observed for the EGR mass flow, where the mean for all datasets is 7.34%.

Figure 2 shows the states of the model compared to the measurements for dataset 11. Since the oxygen measurements are not calibrated, the modeled and the measured signals are normalized to compare only the dynamic behavior. This dataset has a load step, several EGR blower speed steps and a start and stop of the EGR system which is coupled to the CBV operation. It can be observed that the model captures the dynamics of the system.

5. CONCLUSION

An MVEM for a large marine two-stroke engine is proposed and validated. The estimation is done with part of the datasets available while the validation against measurements is done for another set of datasets. The overall agreement of the states is good, and the model is able to capture the general state dynamics.

Nevertheless this model is the first step towards a more general model to be used for development of control strategies. The next step is low load modeling, where new components need to be introduced.

Table 2. Top: number and type of steps contained in each dataset. Bottom: mean relative errors in % of the tuned model for the absolute measured signals in the estimation and validation datasets.

	Estimation Datasets										**Validation Datasets**				
	DS1	DS2	DS3	DS4	DS5	DS6	DS7	DS8	DS9	DS10	DS11	DS12	DS13	DS14	DS15
$\dot{m}_{fuel}$ steps	1	1	5	1	1	0	1	4	3	6	1	0	0	0	7
ω_{blow} steps	9	1	*fixed*	5	7	1	8	9	11	*fixed*	9	9	5	7	*fixed*
u_{CBV} steps	0	0	0	0	0	0	2	1	3	2	3	4	3	4	5
EGR start/stop	1	1	0	0	0	1	3	0	0	0	3	4	3	5	0
CBV start/stop	0	0	0	0	0	0	2	1	3	0	3	4	3	4	0
p_{scav}	1.34	3.03	1.71	2.25	2.71	1.27	2.36	2.32	1.92	2.47	2.67	3.12	3.22	6.23	2.94
$p_{c,out}$	1.32	2.94	1.71	2.24	2.70	1.24	2.31	2.39	1.96	2.51	2.49	2.88	2.98	6.28	2.96
p_{exh}	1.72	3.07	2.36	1.80	3.10	1.60	2.16	2.78	2.36	3.11	2.18	2.52	2.69	6.13	3.68
ω_{tc}	0.92	4.21	0.91	0.96	2.03	0.83	1.44	1.13	1.04	1.27	1.14	1.23	1.23	4.61	1.96
T_{exh}	1.36	4.22	1.39	1.48	1.10	0.77	2.66	1.39	1.96	1.91	1.93	1.60	1.61	2.37	2.83
$P_{eng,i}$	1.57	1.41	1.76	2.11	2.52	2.70	1.66	1.44	2.01	1.73	1.85	1.95	2.24	1.58	2.29
$\dot{m}_{egr}$	7.23	8.35	10.15	6.54	4.77	6.07	5.62	6.40	8.88	7.82	6.58	7.87	9.34	7.31	7.14

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Appendix A. NOMENCLATURE

Table A.1. List of symbols

A	Area	$[m^2]$
B	Bore	$[m]$
c	Connecting rod length	$[m]$
c_p	Specific heat at constant pressure	$[J/(kgK)]$
c_v	Specific heat at constant volume	$[J/(kgK)]$
J	Inertia	$[kg\, m^2]$
$\dot{m}$	Mass flow	$[kg/s]$
n_{cyl}	number of cylinders	[ ]
p	Pressure	$[Pa]$
P	Power	$[kW]$
R	Gas constant	$[J/(kgK)]$
s	Stroke	$[m]$
T	Temperature	$[K]$
V	Volume	$[m^3]$
X_O	Oxygen mass fraction	[ ]
α	angle	$[rad]$
γ	Specific heat capacity ratio	[ ]
η	Efficiency	[ ]
Π	Pressure ratio	[ ]
Φ	Flow Coefficient	[ ]
Ψ	Head Coefficient	[ ]
ω	Rotational speed	$[rad/s]$

Table A.2. Subscripts

a	air	inj	injection
$blow$	blower	$meas$	measured
c	compressor	mod	modeled
$cool$	cooler	$scav$	scavenging manifold
cyl	cylinder	t	turbine
del	delivered	$trap$	trapped
e	exhaust gas	$x, corr$	corrected quantity
egr	EGR gas	x, in	inlet of x
eng	engine	x, out	outlet of x
exh	exhaust manifold	x, n	normalized x