# Modeling the Operating Behavior of an Industrial Diesel Engine used as an Electrical Power Generator

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**Abstract.** Within the research project SIDYN the operating behavior of an electric power generator in an isolated, dynamically loaded grid is to be described and predicted by using simulation approaches. The stage of development shown in this paper includes a coupled thermodynamic/mechanic engine model, which has to be extended by a mechanic/electrical generator-grid-model in the course of the project. For verification of the engine simulation, model measurements were carried out at the engine test bed at Wismar University. One of the key measures to enable simulations of different load scenarios and its impact on the system is the precise definition of subsystem interfaces. To do so, an approach described in [2] is implemented into MATLAB Simulink®. To characterize the controlling behavior of the industrial engine the interfaces between the transient torque and angular velocity of the drive train were observed. Considering the existing experimental setup, it can be concluded that good results could be achieved with the present model. The calculated and simulated cylinder pressure curves show a high level of agreement at various stationary operating points. However, to strive for the calculation of transient operation behaviors, the model as well as the engine test bed require further extension.

## Introduction

To produce electrical energy in isolated grids GENSETS (combination of internal combustion engine and electric generator) are commonly used.

By doing so, the high energy density of fuels is converted into electrical energy. Already during the design process of the GENSET the main focus is put on the quantification of the interaction between system components and the identification of optimized system configuration and operational modes. The impact of dynamic grid load on the electric network and the reaction of the internal combustion engine has a direct impact on network frequency, when using GENSETS in isolated operating mode. Highly dynamic electric loads in isolated grids as they can be found in steel- and cement factories or on ships with hybrid drive trains will affect grid stability inducing the risk of negative repercussions on other connected electric components. Thus, a reliable and accurate prediction of the system behavior and occurring interactions are of great importance in the system design and configuration process. So far calculations like this were made by hand or using simulations that do not cover the whole system. Uncertainties may lead to oversized system components causing high procurement costs.

Basically, conventional simulations are based on engine speed-based models [7]. But the approach of this research project includes the assumption, that the development of an angle of rotation resolved calculation offers a more precise declaration of the whole system behavior. Due to that the plant can be optimized by many more parameters. As a result of this optimization process the systems economic efficiency can be raised while ensuring a high degree of stability and a minimum of network fluctuations.

Apart from a safe and good system performance, economic and ecologic aspects, such as efficiency and emissions can be defined as target values within the simulation model.

The simulation approach consists of two system models connected with an interface.

One of the systems is the thermodynamic/mechanic engine model and the second system covers up the mechanic/electrical generator-grid-model. As one of the first steps a 2.9-liter turbocharged diesel industrial engine and a generator working as an asynchronous machine are modelled, as they are installed at the engine laboratory at Wismar University. The interface is characterized by using the torque curve resolved versus time and angle of rotation in order to accurately describe the highly transient load pattern between the two machines. Both machines are connected with a damped, flexible coupling that is also part of the model.

In this document the current state of the internal combustion engine simulation model is presented. Furthermore, obtained simulation results are compared to engine measurements.

## 1 Simulation Environment

To simulate the diesel engine, state-of-the-art zero-dimensional models were considered. Although there are many commercial tools available, containing ready-to-run models of internal combustion engines, the decision was made to develop a stand-alone simulation tool to achieve the objectives of the research project. The reasons to do so are listed below:

- exact knowledge of the model structure
- specific adaptation of the models to the conditions of large engines and power generators in single and multi-engine system set-ups with mutual interaction
- free choice of model depth and application complexity of the submodels and the overall system
- interfaces can be easily added into the model e.g. implementation in a heating circuit using engines waste heat

Therefore, MATLAB® Simulink® was chosen as a flexible and performant simulation tool for the project. This platform allows for adjustable structures and existing modules can also be used as well as in-house developed program codes. By doing so, physical correlations can be programmed in a comprehensive and adaptable manner.

## 2 Modell Description

To simulate the internal combustion engine a zero-dimensional model is used, which is based on the first principle of thermodynamics. Thus, the combustion chamber is considered as an unsteady and open system [1]. The energy balance equation 1 derived by crank angle $\varphi$ in general is described by:

$$\frac{dU}{d\varphi} = -p\frac{dV}{d\varphi} + \frac{dQ_B}{d\varphi} - \frac{dQ_W}{d\varphi} + h_E\frac{dm_E}{d\varphi} - h_A\frac{dm_A}{d\varphi} \quad (1)$$

To solve this balance equation one variable has to be given by measurements, such as cylinder pressure over crank angle or an empirically determined combustion process (heat input due to combustion).

The inner energy inside the combustion chamber is calculated by using the caloric equation of state.

$$dU = m c_v dT \quad (2)$$

Based on [2], equation 2 is derived by time during the simulation which results in:

$$\Delta\dot{U} = m c_v \dot{T} \quad (3)$$

With this approach [2] the first principle of thermodynamics can be phrased as a differential equation of first order as shown below.

$$\dot{T} = \frac{-p\frac{dV(\varphi)}{dt} + \dot{Q}_B - \dot{Q}_W + \dot{H}_E - \dot{H}_A}{m c_v} \quad (4)$$

Polynomic equations depending on the calculated temperature for different air fuel ratios account for a realistic specific heat capacity $c_v$ in order to obtain improved calculation accuracy. The computation of the cylinder pressure results from the ideal gas equation.

$$p = \frac{mRT}{V} \quad (5)$$

Aside from the determination of pressure and temperature it is important to calculate the heat flux $\dot{Q}_B$ induced by the fuel combustion process. This is done by applying a Vibe function [8] [4]. To model a typical diesel-engine combustion process, a superposition of three vibe-functions is used.

With this approach three typical phases named as pre-mixed combustion, main combustion and post combustion can be depicted [4]. Figure 1 exemplary shows the three Vibe functions and the resulting superposition representing the rate of heat release.

As an input parameter for the simulation the combustion process (heat release rate) is generated separately and saved to a 1D Lookup-Table in Simulink®.

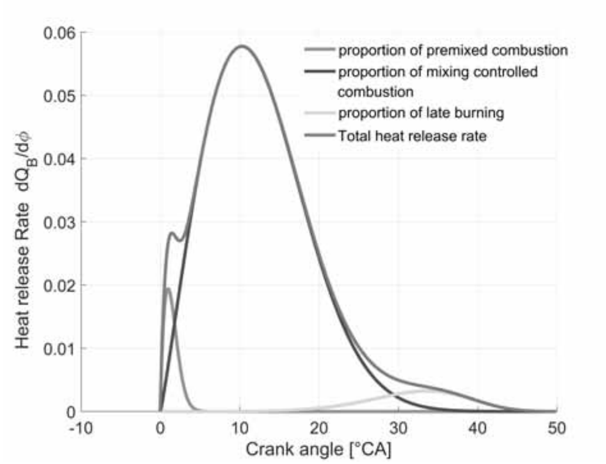


**Figure 1**: Modeling heat release rate.

By applying Newton's approach, the heat transfer $\dot{Q}_W$ from the hot cylinder gas to the cylinder wall and piston can be estimated.

$$\dot{Q}_W = \alpha A(\varphi)\,(T_{Gas} - T_{Wall}) \tag{6}$$

In equation 6 A($\varphi$) refers to the heat transmitting area which is composed by the cylinder head area, piston head area and the crank angle dependent shell surface of the combustion chamber.

To compute the heat transfer coefficient $\alpha$ equation 7 according to Woschni-Huber [3] was used. This equation does not only account for the high-pressure phase (compression and expansion stroke) but also to the low-pressure part (exhaust and intake stroke).

$$\alpha = 130\, d^{-0,2}\, p^{0,8}\, T^{-0,53}\, (C_1\, v)^{0,8} \tag{7}$$

The dimensionless constant $C_1$ is specified by the following parameters:
$C_1 = 2,28 + 0,308 \frac{v_u}{v_{KM}}$ during high-pressure phase
$C_1 = 6,18 + 0,417 \frac{v_u}{v_{KM}}$ during gas exchange
The velocity term $v$ is described in equation 8 according to [3].

$$v = v_{KM}\left[1 + 2\left(\frac{V_C}{V}\right)^2 {p_{mi}}^{-0,2}\right] \tag{8}$$

$p_{mi}$ refers to the mean indicated pressure, representing a specific engine load derived from the work performed from a single cycle related to cylinder displacement.

The enthalpy flux $\dot{H}_E$ and $\dot{H}_A$ linked to the gas mass flow through the intake and exhaust valves are calculated by [4]:

$$\dot{H}_E = \dot{m}_E\left(u_E + R_E\, T_E\right) \tag{9}$$

$$\dot{H}_A = \dot{m}_A\left(u_A + R_A\, T_A\right) \tag{10}$$

The index E indicates the condition just in front of the intake valve, whereas the index $A$ characterizes the condition right behind the exhaust valve. The calculation of the specific inner energy is based on substance data out of [3], which are expressed by polynomic equations as a function of temperature for a particular air-fuel ratio. To simulate the mass flow through the valves the equations 11 and 12 are used [3].

$$\dot{m}_E = \mu A_{VE(\varphi)} \frac{p_E}{\sqrt{R T_E}} \sqrt{\frac{2\kappa}{\kappa - 1}\left[\left(\frac{p_Z}{p_E}\right)^{\frac{2}{\kappa}} - \left(\frac{p_Z}{p_E}\right)^{\frac{\kappa+1}{\kappa}}\right]} \tag{11}$$

$$\dot{m}_A = \mu A_{VA(\varphi)} \frac{p_Z}{\sqrt{R T_Z}} \sqrt{\frac{2\kappa}{\kappa - 1}\left[\left(\frac{p_A}{p_Z}\right)^{\frac{2}{\kappa}} - \left(\frac{p_A}{p_Z}\right)^{\frac{\kappa+1}{\kappa}}\right]} \tag{12}$$

The indication $Z$ stands for conditions inside the cylinder whereas E stands for intake and $A$ indicates the exhaust regarding to the conditions right in front of the in-take valve or right after the exhaust valve.

$A_v$ describes the gap area of the intake or the exhaust valve which changes by time depending on valve lift curves. The flow coefficient $\mu$ accounts for the flow resistance inside the valve ducts. It is determined by experiments at cylinder head flow test benches or by CFD-analyses [4]. For the simulation presented in this paper the flow coefficients were adopted depending on relative valve lift from Maurer [1] and Merker & Schwarz [4]. Figure 2 shows the schematic Simulink® code to calculate the energy balance inside the combustion chamber. The simulation starts by specifying initial temperature, engine speed as well as air mass inside the combustion chamber. The piston starting position is at 0°CA (top death center = TDC). The block named "calculation cylinder pressure" calculates the cylinder pressure using equation 5. The current values for temperature and pressure are transferred to the particular subsystems. Integration of the rotational engine speed results in crank angle. Based hereon cylinder volume, valve lift and the gas mass flow are calculated in the subsystems.

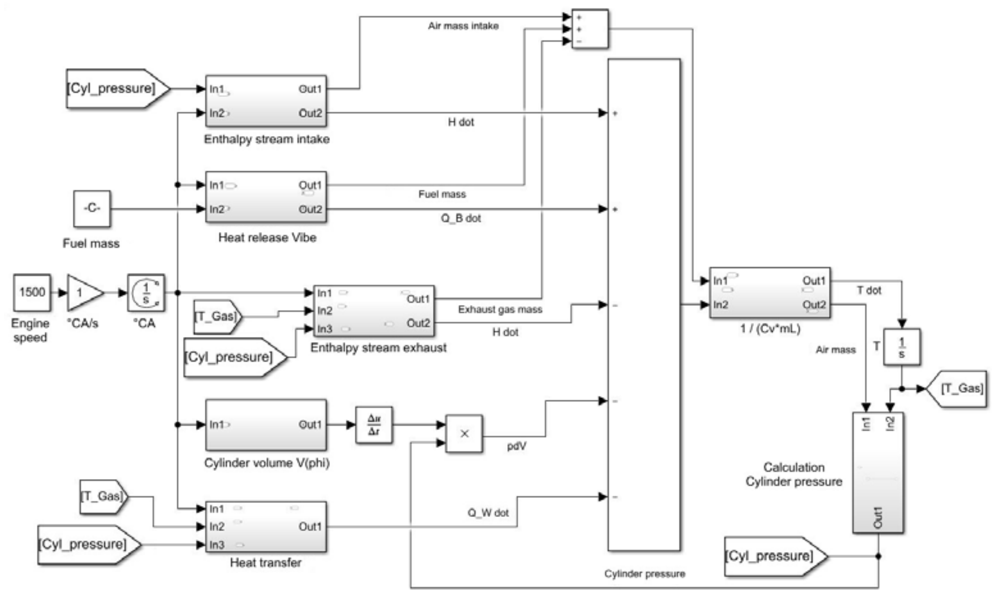


**Figure 2**: Model scheme in Simulink®.

An exemplarily calculated gas mass flow at the intake valve versus crank angle is shown in Figure 3.

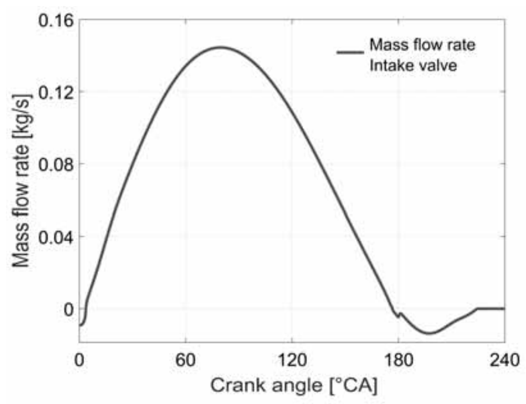


**Figure 3**: Mass flow rate at intake valve.

In the considered load point it can be recognized that there is a backflow starting at 180°CA, which stands for the end of the intake stroke. In this phase the piston is already moving back towards the TDC and a specific amount of air mass goes back through the intake valve that is still not fully closed. This phenomenon depends on engine speed and can lead to an increase in air mass inside the combustion chamber at higher engine speed. It is initially influenced by the lay out of the valve lift curves. With the closing of the intake valves the system can be treated as a closed system, e.g. as long as leakages are ignored.

Figure 4 shows the simulated cylinder pressure and the time dependent cylinder volume during one cycle (720°CA). With the piston moving upwards the cylinder volume decreases and the compression stroke starts, which leads to a rise in temperature and pressure. At around 360°CA the combustion starts and the pressure rises further. The downwards movement of the piston during the expansion stroke and the associated increase in cylinder volume leads to decreasing pressure and temperature. Shortly before 480°CA the outlet valve opens and exhaust gases are purged. When modelling a four-cylinder engine the energy balance is individually calculated for each single cylinder. While integrating the crank angle out of engine speed an offset has to be considered which corresponds to the firing intervals. For an inline four-cylinder engine the firing interval is 180°CA.

To simulate the engine speed variation during on cycle a crank angle resolved energy balance is calculated considering the work performed at the piston $(-p\frac{dV}{d\varphi})$, shaft power, friction losses and power required for the oscillating piston movement of each cylinder. Friction losses are determined with the help of a polynomial approach by Chen & Flynn [5] which allows for an easy adaptation to the boundary and operating conditions. Figure 5 displays the block scheme for the calculation of rotational speed variations in Simulink®.

By using the so called "Memory Block" a value for an engine start speed has to be defined to make the calculation of the energy balance possible.

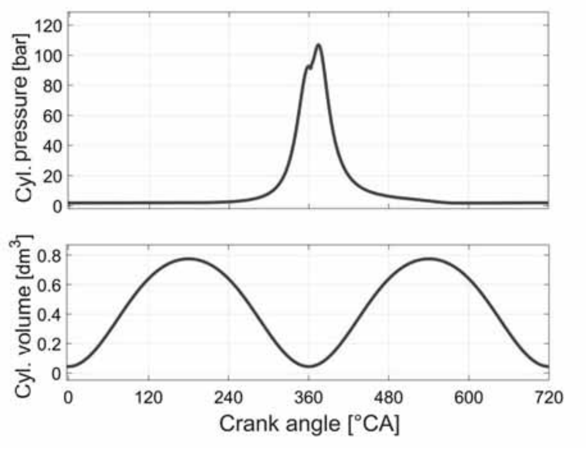


**Figure 4**: Top: Cylinder pressure curve over one cycle
Bottom:Cylinder volumen over one cycle.

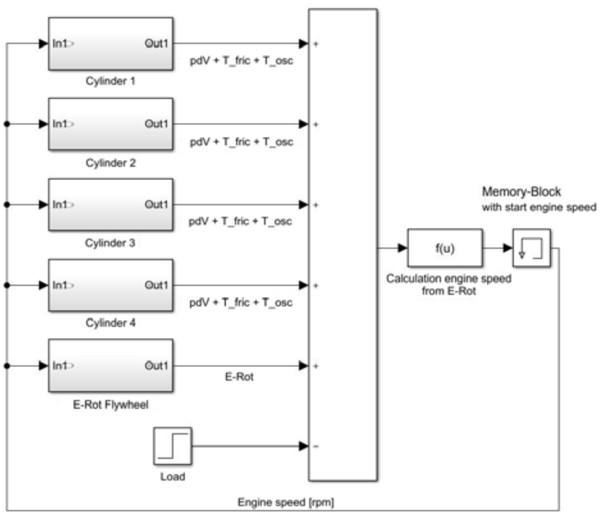


**Figure 5**: Schematic representation of four-cylinder model.

Subsystems "cylinder 1" to "cylinder 4" calculate the individual torque of each cylinder, considering the oscillating mass and friction losses. The load request (shaft power) of the system is defined by a variable, that can be applied as demanded. The energetic condition of the flywheel is described by using equation 13.

$$E_{Rot} = \frac{1}{2} J \omega^2 \tag{13}$$

Figure 6 shows the simulated torque sequence of a single cylinder during one cycle.

By merging all energy fluxes the rotational energy is attained for each time step. Rearranging equation 13

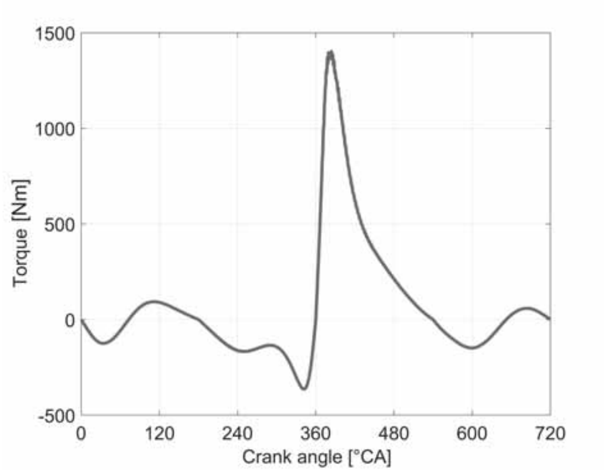


**Figure 6**: Simulated torque sequence.

allows for the calculation of the corresponding engine speed variation.

$$n = \sqrt{\frac{E_{Rot\ new}}{2 J \pi^2}} \tag{14}$$

Figure 7 shows the simulated engine speed for a single cycle according to equation 14 for a rated engine speed of 1500 $\text{min}^{-1}$. The engine speed variation of an internal combustion engine may have an impact on the behavior of a coupled electric machine. Due to that the characteristic of the coupling element has to be considered in the simulation. Two equations result from the basic laws of dynamics for rotational motion for each rotating mass (internal combustion engine and electric machine) [6].

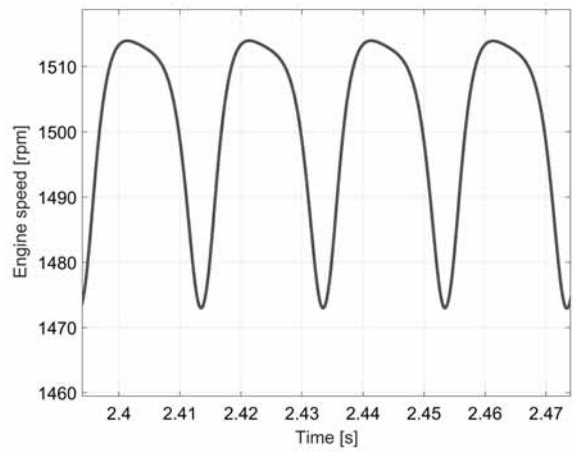


**Figure 7**: Simulated engine speed sequence of the
four-cylinder setup during two engine revolutions.

$$-k_D(\varphi_M-\varphi_G)-d_D(\dot\varphi_M-\dot\varphi_G)+M=J_M\,\ddot\varphi_M \quad (15)$$

$$k_D(\varphi_M-\varphi_G)+d_D(\dot\varphi_M-\dot\varphi_G)=J_G\,\ddot\varphi_G \quad (16)$$

The damping torque is expressed by $d_D(\dot\varphi_M-\dot\varphi_G)$ and $k_D(\varphi_M-\varphi_G)$ represents the torque resulting from a twist of both rotating masses to each other. M stands for the excitation torque which is introduced by the internal combustion engine. Rearranged to the highest derivative, equation 17 is derived.

$$-k_D\left(\frac{1}{J_M}+\frac{1}{J_G}\right)(\varphi_M-\varphi_G)-d_D\left(\frac{1}{J_M}+\frac{1}{J_G}\right) \quad (17)$$

$$(\dot\varphi_M-\dot\varphi_G)+\frac{M}{J_M}=(\ddot\varphi_M-\ddot\varphi_G)$$

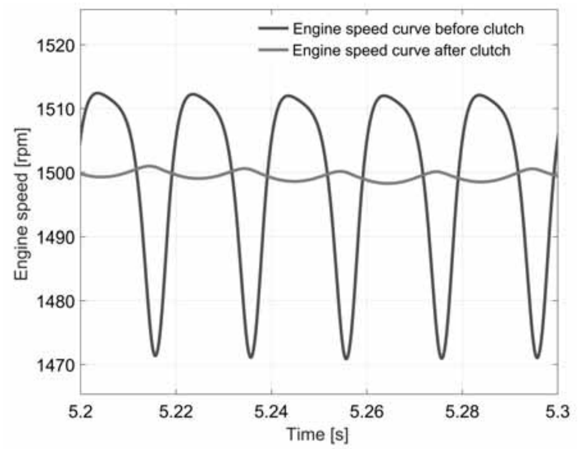


**Figure 8**: Simulated speed variation of engine and electric machine.

The simulated engine speed curves are shown in figure 8. The blue curve represents the rotational speed that goes into the elastic coupling element and the red curve shows the rotational speed of the electric machine. It can be seen that the engine speed variation of the internal combustion engine is reduced by the elastic coupling element, leading to considerably decreased amplitudes at the electric machine.

# 3 Model Validation

To verify the simulation model, the calculation results were compared to measurements obtained from the engine laboratory at Wismar University.

A comparison of the cylinder pressures over the course of one cycle at an engine speed of 2000 $\text{min}^{-1}$ and a nominal torque of 200 Nm is presented in figure 9. The rate of heat release derived from measurements was used as an input parameter for the simulation. Comparing both pressure traces it can be seen that simulation and measurement agree very well. This fact is an important requirement for an exact description of the torque at the interface of the internal combustion engine and the connected electric generator. A comparison between the mean pressure as an integral value to quantify engine load shows a discrepancy between simulation and measurement below 1%. A similar quality of the simulation results was recognized also at other load points. Therefore, it can be stated that the applicability of the simulation approach is confirmed for stationary operating conditions.

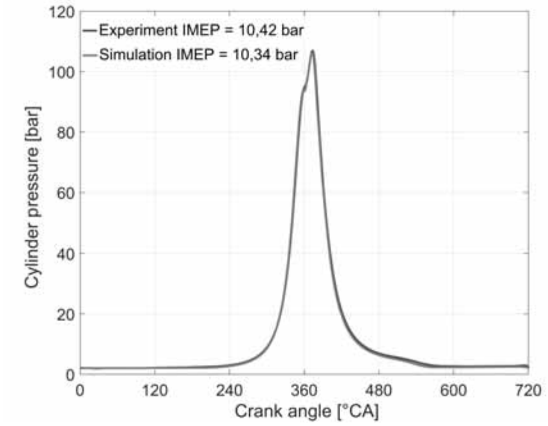


**Figure 9**: Comparison of measured and simulated cylinder pressure trace of one cylinder.

# 4 Conclusion and Outlook

In the course of the research project SIDYN the operating behavior of an electric power generator in isolated grid mode with dynamic electrical loads is to be predicted based on a modular simulation model. The model should enable the possibility to vary engine and network configurations as well as different load scenarios.

Based on thermodynamic and mechanical relations and specifically developed component submodels for internal combustion engines, a complete model of a four-cylinder four-stroke diesel industrial engine was developed in Simulink®. A comparison between simulation and experiments in steady state load points indicate a very good correlation, which supports the applicability of the chosen simulation approach.

Due to the fact that highly dynamic operating conditions are in the focus of the project a further development of the models and the engine test bed is required. This includes the modelling and experimental observation of the transient behavior of the turbo charger. Furthermore, the influence of heat transfer processes in the intake and exhaust system, as well as the impact of the exhaust gas recirculation on the overall system are to be examined.

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## Nomenclature

$\ddot{\varphi}_G$ - angular acceleration, generator sided [rad/s$^2$]
$\ddot{\varphi}_M$ - angular acceleration, engine sided [rad/s$^2$]
$\dot{\varphi}_G$ - angular velocity, generator sided [rad/s]
$\dot{\varphi}_M$ - angular velocity, engine sided [rad/s]
$\dot{H}_{E,A}$ - exhaust and intake enthalpy fluxes [J/s]
$\dot{m}$ - mass flux [kg/s]
$\dot{Q}_B$ - heat release rate of the combusted fuel [J/s]
$\dot{Q}_W$ - wall heat flux [W]
$\kappa$ - isentropic exponent [-]
$\mu$ - flow coefficient [-]
$\omega$ - angular velocity [s$^{-1}$]
$\varphi_G$ - twist angle, generator sided [rad]
$\varphi_M$ - twist angle, engine sided [rad]
$A$ - heat transferring area [m$^2$]
$A_v$ - gap area [m$^2$]
$c_v$ - specific heat capacity [J/(kg K)]
$d$ - bore diameter [m]
$d_D$ - torsional spring stiffness [Nms/rad]
$h$ - specific enthalpy [J/kg]
$J$ - inertia moment [kg m$^2$]
$J_G$ - generator inertia moment [kg m$^2$]
$J_M$ - engine inertia moment [kg m$^2$]
$k_D$ - torsional spring stiffness [Nm/rad]
$m$ - mass [kg]
$p$ - pressure [Pa]
$p_{mi}$ - indicated medium pressure [bar]
$R$ - gas constant [J/(kg K)]
$T$ - temperature [K]
$U$ - inner energy [J]
$u$ - specific inner energy [J/kg]
$V$ - volume [m$^3$]
$v_{KM}$ - mean piston velocity [m/s]
$v_u$ - swirl velocity [m/s]

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