Parameter Based Combustion Model for Large Prechamber Gas Engines

Jianguo Zhu
e-mail: jianguo.zhu@lec.tugraz.at

Andreas Wimmer

Eduard Schneßl

Hubert Winter

Franz Chmela

LEC-Large Engines Competence Center,
Inffeldgasse 21A,
A-8010 Graz, Austria

Challenging requirements for modern large engines regarding power output, fuel consumption, and emissions can only be achieved with carefully adapted combustion systems. With the improvement of simulation methods simulation work is playing a more and more important role for the engine development. Due to their simplicity and short computing time, one-dimensional and zero-dimensional calculation methods are widely applied for the engine cycle simulation and optimization. While the gas dynamic processes in the intake and exhaust systems can already be simulated with sufficient precision, it still represents a considerable difficulty to predict the combustion process exactly. In this contribution, an empirical combustion model for large prechamber gas engines is presented, which was evolved based on measurements on a single cylinder research engine using the design of experiment method. The combustion process in prechamber gas engines is investigated and reproduced successfully by means of a double-vibe function. The mathematical relationship between the engine operating parameters and the parameters of the double-vibe function was determined as a transfer model on the base of comprehensive measurements. The effects of engine operating parameters, e.g., boost pressure, charge temperature, ignition timing, and air/fuel ratio on the combustion process are taken into account in the transfer model. After adding modification functions, the model can be applied to gas engines operated with various gas fuels taking into account the actual air humidity. Comprehensive verifications were conducted on a single-cylinder engine as well as on full-scale engines. With the combination of the combustion model and a gas exchange simulation model the engine performance has been predicted satisfactorily. Due to the simple phenomenological structure of the model, a user-friendly model application and a short computing time is achieved. [DOI: 10.1115/1.4000295]

1 Introduction

Having experienced a renaissance since 1980, large gas engines are currently widely applied in the stationary engine field. The challenging requirements regarding power output, efficiency, and emissions can only be achieved with a carefully adapted combustion system. The experiment on the test bench is taken as a good method for engine development, but it is always associated with high expenditure and a long development time, particularly for large engines.

With the improvement of simulation methods and computing capability, the simulation work is playing a more and more important role during the process of engine development and optimization. Its application helps to investigate the process in the engines closely and to study further opportunities for engine development and optimization. With its help many experiments on the test bench can be saved. Therefore the expenditure and development time is reduced a great deal.

The one-dimensional engine cycle calculation represents the main instrument for the simulation and optimization of the thermodynamic working cycle because of its simplicity and short computing time. Nowadays the gas dynamic processes in the inlet and outlet systems can be simulated with sufficient precision. But to perform predictive calculation of the combustion process, emission formulation, and knocking behavior in large gas engines as correctly as possible represents, still, considerable difficulties. Provided these models are available, a preliminary optimization of operating parameters is possible by simulation. In Fig. 1 the engine cycle simulation tool together with the relevant input parameters is displayed.

Parallel to the work on physics based models [1–3] research work has been conducted at LEC to develop an empirical combustion model. This model should be able to predict the combustion process of gas engines at arbitrary operating conditions. Additionally, the impact of the air humidity and methane number (MN) on the combustion should also be taken into account.

2 Combustion Concept of Prechamber Gas Engines

Nowadays, the prechamber concept is widely applied for gas engines with a lean mixture, particularly for engines with a large cylinder bore size [4]. As shown in Fig. 2, the main components of this kind of engine are the prechamber accommodating the spark plug and the gas valve controlling the gas supply to the prechamber.

During the intake stroke the lean mixture of air and gas will flow through the intake valves into the cylinder. At the same time pure gas will flow into the prechamber via a pressure-controlled valve. Therefore, a significantly richer and easily ignitable mixture is available in the prechamber. At ignition timing the mixture in the prechamber will be ignited by the spark plug. Then the flame will propagate through the prechamber and via the channels between the two chambers into the main combustion chamber. There the lean mixture is ignited and the main combustion will take place.

3 Test Engines

In order to reduce the expenditure and development time, the investigations for the model development were carried out on a single-cylinder research engine (SCRE) at LEC [5]. After the

Contributed by the International Gas Turbine Institute (IGTI) of ASME for publication in the Journal of Engineering for Gas Turbines and Power. Manuscript received May 21, 2009; final manuscript received May 26, 2009; published online May 28, 2010. Editor: Dilip R. Ballal.

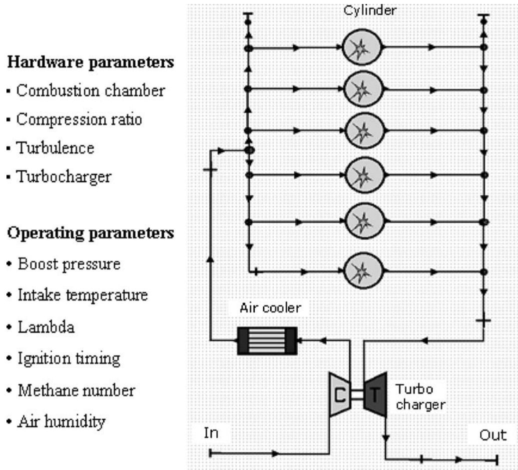

Fig. 1 Engine cycle simulation tool with relevant input parameters

model design comprehensive verifications were conducted, also based on measurements on a full-scale engine (FSE).

The SCRE at LEC shown in Fig. 3 was derived from a full-scale engine, which incorporates the same combustion system as the full-scale engine.

There is no turbocharger on the SCRE as on full-scale engines. Air and gaseous fuel are compressed individually by screw-type compressors and mixed together ahead of the intake pipe. The

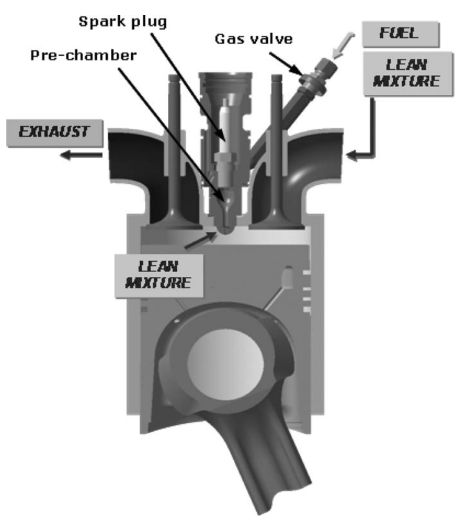

Fig. 2 Combustion system of gas engines with a gas fuelled prechamber [4]

Fig. 3 Single-cylinder research engine

Table 1 Engine data of the SCRE

	SCRE	FSE
Number of cylinders	1	12
Bore/stroke (mm)	190/220	
Displacement (dm^3/cylinder)	6.2	
Engine type	4-stroke	
Engine speed (min^{-1})	1500	
Number of valves	4	
Prechamber volume (% of clearance volume)	≈1–2	

charge pressure can be varied within a large range. With the air conditioning system on the intake side the temperature and the air humidity can be adjusted according to requirement. The intake and exhaust systems are designed by means of one-dimensional cycle simulation, so that the SCRE shows a comparable gas dynamic process in the intake and exhaust systems as well as the residual gas content as in the full-scale engine. Various gaseous fuels can be supplied to the engine for combustion.

In Table 1 the most important data of the SCRE and the corresponding FSE are summarized.

4 Rate of Heat Release in Prechamber Gas Engines

In Fig. 4 the typical heat release rate curve of gas engines with a gas fuelled prechamber is depicted. The figure shows the combustion characteristics in the main chamber, e.g., the ignition delay, the two phases of combustion, and the combustion duration. The combustion will take place after an ignition delay measured from the spark timing. The combustion in the main chamber can be divided into two phases. First, the combustion of the not yet completely burned gas jets out of the prechamber enhanced by the combustion of parts of the lean mixture in the main chamber entrained by the burning jets and, second, the combustion of the remaining bulk of the lean mixture in the main chamber. The first phase of combustion causes the first peak in the heat release rate profile. The second phase of combustion resembles the combustion mode in open chamber gas engines where a flame front propagates through the cylinder space.

In this figure as well as in Figs. 5 and 9 the heat release rate was normalized by the total energy input.

5 Heat Release Rate Simulation With a Double-Vibe Function

In order to describe the particular shape of the heat release rate curve in prechamber gas engines with two peaks, the double-vibe function was selected for its mathematical representation.

The double-vibe function comprises two single vibe functions as shown in Eqs. (1)–(3).

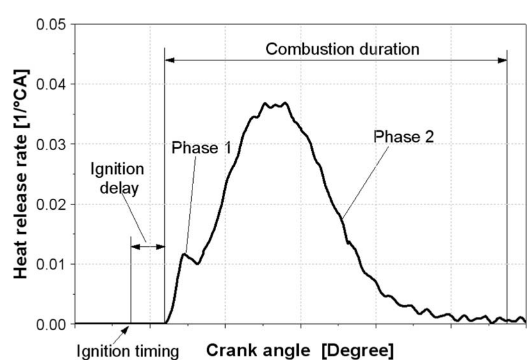

Fig. 4 Typical heat release rate in the main chamber of gas engines with a gas fuelled prechamber

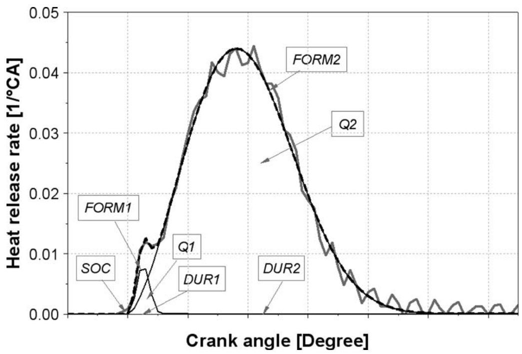

Fig. 5 Heat release rate simulation with double-vibe function

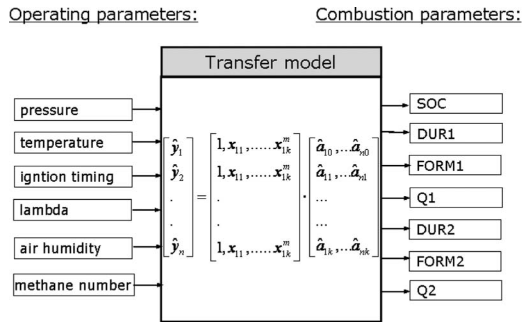

Fig. 6 Input and output parameters of the transfer model

$$\frac{dQ_1}{d\varphi} = \frac{Q_1}{DUR1} \cdot 6.908 \cdot (FORM1 + 1) \cdot \left(\frac{\varphi - SOC}{DUR1}\right)^{FORM1} \cdot e^{-6.908 * (\varphi - SOC/DUR1)^{FORM1+1}} \tag{1}$$

$$\frac{dQ_2}{d\varphi} = \frac{Q_2}{DUR2} \cdot 6.908 \cdot (FORM2 + 1) \cdot \left(\frac{\varphi - SOC}{DUR2}\right)^{FORM2} \cdot e^{-6.908 * (\varphi - SOC/DUR2)^{FORM2+1}} \tag{2}$$

$$\frac{dQ}{d\varphi} = \frac{dQ_1}{d\varphi} + \frac{dQ_2}{d\varphi} \tag{3}$$

In the above equations, *FORM*1 and *FORM*2 represent the form factors, *DUR*1 and *DUR*2 represent the durations in crank angle, and *SOC* represents the start of combustion. The amounts of fuel energy used up by the individual vibe functions are designated as Q_1 and Q_2.

In Fig. 5 the analyzed heat release rate curve of the SCRE (at 20 bar indicated mean effective pressure (IMEP) and 500 mg/Nm^3 NO_x) is presented together with the simulated curve using a double-vibe function. The full line shows the analyzed heat release rate and the dashed line indicates the simulated trace. Additionally, the two individual simulated heat release rate curves of the two combustion phases are displayed as thin full lines.

It can be seen that the character of the analyzed heat release rate curve is reproduced satisfactorily by the double-vibe function. This good agreement between analysis and simulation was achieved by adjusting the seven parameters of the double-vibe function by trial and error, which turned out as a rather time consuming exercise. In order to reduce the adaptation time of the vibe parameters, an optimization software developed at LEC has been utilized further, by which the parameter values for a given heat release rate curve can be determined automatically [6].

As the parameter values are only valid for one set of operating parameters, the prediction of a heat release rate curve for arbitrary operating parameters was not yet possible. Therefore, the next step was to develop a transfer model, i.e., a mathematical relationship between the primary operating parameters such as boost pressure, charge temperature, air/fuel ratio, and ignition timing, and the seven double-vibe parameters (see Fig. 6).

Besides of the primary operating parameters mentioned the influence of the gas fuel proportion in the prechamber, the methane number of the fuel gas, and the air humidity also have to be considered in the model.

The combustion process is much influenced by the methane number of gaseous fuels. The fuels with lower methane numbers show higher laminar flame velocities at given conditions of the cylinder charge. Consequently, the flame propagates faster than with fuels with higher methane numbers.

The air humidity is also an important parameter influencing combustion. The thermal capacity of the mixture will be higher with increased air humidity, therefore the flame temperature will be lower and the combustion process will be slowed down. This can be recognized from the heat release rates of test points with different air humidities [7].

6 Database for the Transfer Model

In order to establish the relationship between operating parameters and combustion parameters, a large amount of measurement data would have been required. To save cost and time, the design of experiment (DOE) method was utilized for setting up a test plan.

The test plan was designed for the engines described previously operating with natural gas with a standard MN of 94 at standard air humidity of 8 g/kg. The primary engine operating parameters such as charge pressure, charge temperature, ignition timing, and lambda were changed in a large area within the operating range. The variation ranges of the primary operating parameters are listed in Table 2.

The designed test points are evenly distributed within the operating range of the engines. Figure 7 provides an overview about the distribution of the planned test points.

In this figure, the area inside the dashed lines represents the engine operating range versus lambda and charge pressure. It can be seen that with reduced lambda the operating point of engines will move from the misfire limit to the detonation limit, with increased charge pressure the operating range will be narrower.

The points displayed as squares are the designed operating points for the model development, which are evenly distributed in the whole operating range. In addition the triangular test points

Table 2 Variation in operating parameters in the test plan

Operating parameter	Variation range
Charge pressure (bar)	1.2–4.5
Charge temperature (°C)	40, 50, 60, and 70
Lambda	1.6–2.1
Ignition timing (°CA BTDC[a])	26–16

[a]BTDC means before top dead center.

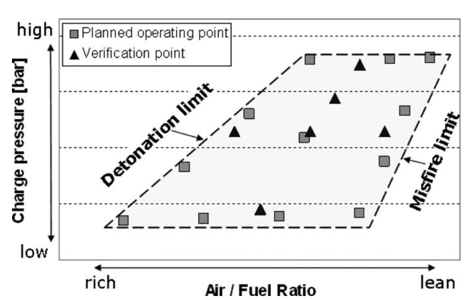

Fig. 7 Test plan for the transfer model development

are designed especially for model verification. From the picture it can be seen that the planned measurement points are covering the operating ranges with different loads from misfire to the detonation limit.

In addition to natural gas with a MN of 94, gas mixtures of natural gas and propane with methane numbers of 70 and 50 were used. In order to capture the influence of intake air humidity, ten operating points were selected for measurements with varying humidities.

With the application of the DOE method, the expenditure regarding time and cost on the SCRE was reduced considerably. The number of operating points on the test bench decreased for each methane number investigated from more than 300 points to 40 points only.

7 Transfer Model Development

The measured cylinder pressure traces together with the steady state data at each operating point were analyzed for the heat release rates. In order to eliminate the influence of cycle-to-cycle fluctuations the average pressure trace over 60 cycles was taken for the heat release rate analysis. The analyzed heat release rates were then reproduced by a double-vibe function through the determination of the seven model parameters using the optimizing tool mentioned earlier.

The correlation between the operating parameters and the double-vibe parameters was determined and expressed by means of polynomial functions, which are formulated in the following system of Eq. (4):

$$\begin{bmatrix} \hat{y}_1 \\ \hat{y}_2 \\ \cdots \\ \cdots \\ \hat{y}_n \end{bmatrix} = \begin{bmatrix} 1, x_{11}, \ldots, x_{1k}^m \\ 1, x_{11}, \ldots, x_{1k}^m \\ \cdots \\ \cdots \\ 1, x_{11}, \ldots, x_{1k}^m \end{bmatrix} \cdot \begin{bmatrix} \hat{a}_{10}, \ldots, \hat{a}_{n0} \\ \hat{a}_{11}, \ldots, \hat{a}_{n1} \\ \cdots \\ \cdots \\ \hat{a}_{1k}, \ldots, \hat{a}_{nk} \end{bmatrix} \tag{4}$$

In the above equation, $\hat{y}_1$ to $\hat{y}_n$ represent the seven parameters of the double-vibe function, i.e., SOC, $DUR1$, $FORM1$, Q_1, $DUR2$, $FORM2$, and Q_2. x_{11} to x_{1k} indicate the operating parameters of the engine, e.g., charge pressure, temperature, ignition timing, and lambda. $\hat{a}_{10}$ to $\hat{a}_{nk}$ are the coefficients in the polynomial functions returned by the DOE software.

The effects of methane number and air humidity have been considered in the model through the application of modification functions in which the calculated double-vibe parameters at standard humidity and MN of 94 will be modified according to the actual air humidity and methane number. The modification functions were selected and determined based on the measurement results on the SCRE and added to the transfer model.

8 Verification of the Transfer Model

In order to evaluate the model quality, comprehensive verifications were carried out based on the measurements on the SCRE and the FSE.

8.1 Verification of Combustion Parameter Simulation. For verification, the parameters of the double-vibe function were first predicted by the transfer model using the actual operating parameters. The simulation results of important combustion parameters such as start of combustion, main combustion duration, and form factor of the heat release rate were compared with the measured values and are displayed in Fig. 8.

All the test points selected for model development and verification have been simulated and represented in this figure. The points with different representations (circle, triangle, and square) indicate the operating points of engines with different gaseous fuels with methane numbers of 94, 70, and 50. The top picture shows the start of combustion, the middle one presents the burning duration, and the bottom one shows the main form factor of

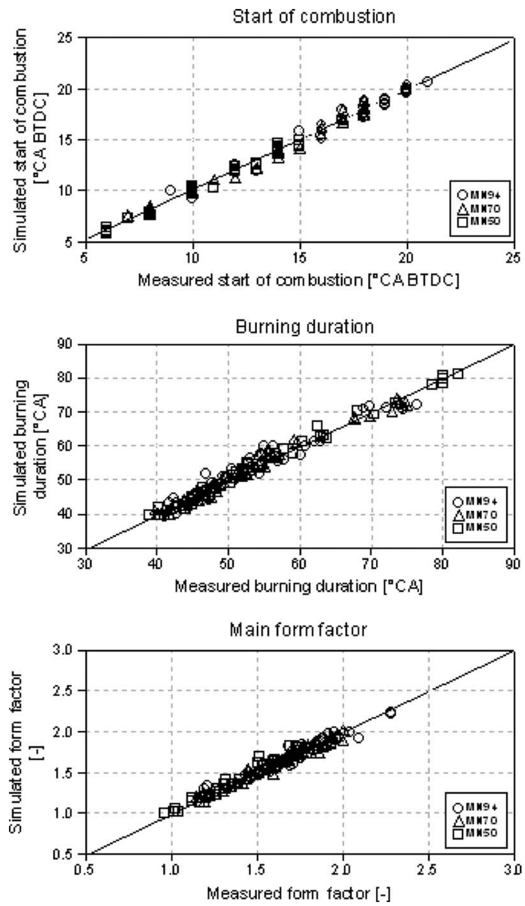

Fig. 8 Simulation of the main vibe parameters

the double-vibe function. It can be seen that the parameters have been predicted with the transfer model with a rather high accuracy. The simulated value of the start of combustion deviates from the measurement less than 1 °CA. The difference between simulation and measurement for the burning duration is not more than 5 °CA. The form factor also has been predicted very accurately. Altogether, a satisfactory agreement between simulation and measurement has been achieved.

8.2 Verification of Heat Release Rate Simulation. Finally, the entire simulation model for the heat release rate curve in prechamber gas engines was set up as a combination of the transfer model and the double-vibe function.

Figure 9 shows the heat release rates at the SCRE at various operating conditions. The full lines in the figure represent the analyzed heat release rates and the dashed ones are the simulated results.

The operating points displayed refer to various values for ignition timing, lambda, charge pressure, air humidity, and to fuels with three different methane numbers. It can be seen, generally, that the heat release rate curves have been predicted satisfactorily.

9 Simulation of Engine Performance

After verification, the model was implemented in the working cycle simulation software, so that it could be applied together with the gas exchange simulation model for the simulation of engine performance.

9.1 Simulation for the Single-Cylinder Research Engine. During the calculation process the operating parameters of the working cycle model for the SCRE will be transferred to the combustion model. The resulting heat release rate history will be

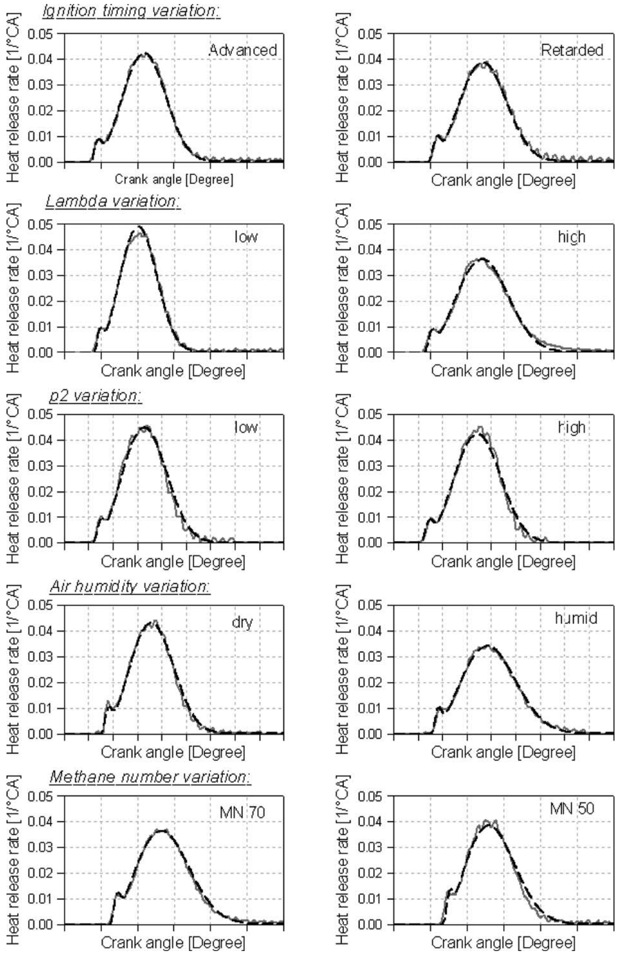

Fig. 9 Simulation of the heat release rate

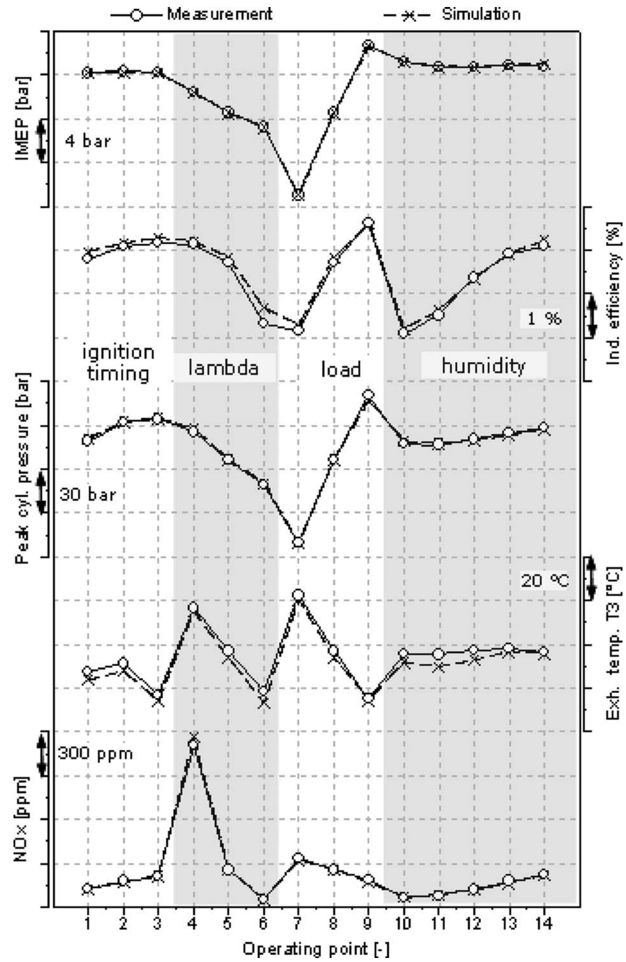

Fig. 10 Performance data simulation for the SCRE

transferred back to the working cycle calculation, so that the engine performance can be predicted. Additionally, the NO_x formation has also been simulated by means of the LEC NO_x model with the boundary conditions obtained from the performance calculation.

The engine performance data including IMEP, indicated efficiency, peak cylinder pressure, exhaust temperature, and NO_x under different operating conditions simulated at various operating points are shown in Fig. 10. Points 1–3 indicate the ignition timing variation, from point 4 to point 6 the lambda was changed, points 7–9 refer to a load increase from part load to full load, and the last points that are from 10 to 14 indicate a change in air humidity from 25 g/kg to 5 g/kg. The simulated results are displayed with dashed lines, while the measured values are shown as full lines.

It can be seen that the simulated exhaust gas temperature T_3 deviates from the measurement not more than $10^\circ C$ and the efficiency deviation is not more than 0.5%. These differences between simulation and measurement contain not only the inaccuracy of the combustion model but also the inaccuracy of the gas exchange simulation model. The deviation purely caused by the combustion model is even smaller.

NO_x emissions have been predicted at different operating points. The influences of ignition timing, lambda, charge pressure, and air humidity on the NO_x formation are correctly represented by the model. With the air humidity increase from 5 g/kg to 25 g/kg (points 14–10) under constant load the NO_x emissions are considerably reduced from 200 ppm to 75 ppm, the simulated results being in good agreement with the measurement results.

9.2 Simulation for the Full-Scale Gas Engine. In the same way as on the SCRE the combustion model has been also applied to the FSE.

In Fig. 11, the predicted and measured engine performances for the 12 cylinder gas engine is shown as a function of the engine load with a constant NO_x emission of $500\ \mathrm{mg/N\,m^3}$. In addition to the engine performance parameters such as thermal efficiency, mass flow, and T_3, also the state parameters of the turbocharger like efficiency and speed have been displayed in the figure. The dashed lines represent the simulated results, while the full lines indicate the measurement data.

At the operating points with higher loads a leaner mixture and earlier ignition timing have been used in order to achieve constant NO_x emission and better engine efficiency. It can be seen that the simulated and measured values for engine efficiency differ by only about 0.5%.

Exhaust temperature T_3 before turbine, which is an important parameter for the turbocharger operation, has also been predicted with the model. The reduction in the exhaust temperature at higher load due to the leaner mixture is reproduced by the simulation. The deviation between simulation and measurement is less than $10^\circ C$.

In the bottom of the figure the efficiency and speed of the turbocharger at different engine loads are displayed. Also the influence of the engine load has been simulated with good agreement.

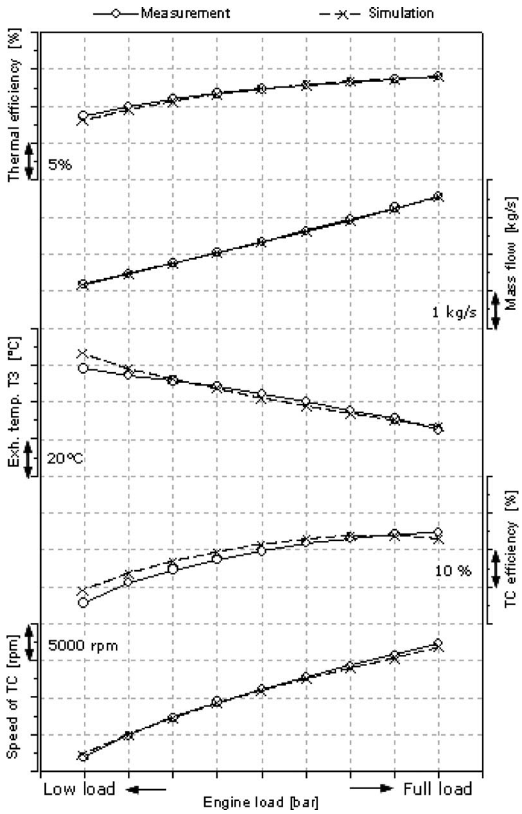

Fig. 11 Performance data simulation for the FSE

10 Conclusions

An empirical transfer model for the determination of the parameters of a double-vibe function for the heat release rate in large prechamber gas engines from the engine operating parameters was developed successfully. The model is based on comprehensive measurements on the SCRE at LEC. The influences of different operating parameters, e.g., charge pressure, charge temperature, ignition timing, and lambda, on the combustion process are taken into account by the model. With added modifications it can also be applied to engines with nonstandard air humidity and fuel methane number.

The verification process of the combined transfer and heat release rate model showed that both the vibe parameters and the heat release rate can be predicted satisfactorily for a considerable range of operating parameters.

After the implementation of this combustion model in the working cycle simulation software it can be utilized together with the gas exchange model for engine performance calculation. Its application on the SCRE and the FSE show that the engine performance can be predicted satisfactorily.

The combined burn rate and gas exchange simulation tool can be applied for the optimization of system components such as camshaft and turbocharger.

Nomenclature

NO_x = nitrogen oxides
p_3 = exhaust pressure
p_2 = boost pressure
T_3 = exhaust gas temperature
°CA = degree crank angle
SOC = start of combustion
$DUR1$ = duration of the first double-vibe function
$DUR2$ = duration of the second double-vibe function
$FORM1$ = form factor of the first double-vibe function
$FORM2$ = form factor of the second double-vibe function
Q_1 = energy fraction for the first double-vibe function
Q_2 = energy fraction for the second double-vibe function
Q = normalized burnt fuel energy
φ = crank angle (°CA)

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