APPLICATION OF 1D SIMULATION TO OPTIMIZE PERFORMANCE AND EMISSIONS OF LARGE GAS ENGINES WITH EXHAUST GAS RECIRCULATION

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Abstract: *Future emission legislation will be increasingly stringent. The current German TA Luft limit for nitric oxide (NOx) emissions from large gas engines is 500 mg/mn^3 @ 5% O_2 and there is a clear trend toward further reductions. One possible strategy to meet these limits for gas engines is exhaust gas recirculation (EGR). This paper focuses on the application of 1D simulation to a variety of different tasks in gas engine development. First, the basic effects of EGR in gas engines are explained by discussing several 1D simulation results from two 1D simulation models for a large stationary gas engine. A detailed single cylinder engine model is used to study the interaction between the pre-chamber and the main combustion chamber. The boundary conditions are provided by a multicylinder engine model that includes a turbocharger. Based on simulation calculations with both models as well as measurements from a single cylinder research engine, the thermodynamic conditions in the combustion chamber and the gas dynamics are analyzed.*

Keywords: INTERNAL COMBUSTION ENGINES, 1D SIMULATION, EXHAUST GAS RECIRCULATION

1. Introduction

The trend towards high power output, higher efficiencies as well as stringent emission legislation are greatly pushing the development of large gas engines.

When a new gas engine is developed, many design parameters have to be defined and optimized in advance. The number of degrees of freedom is even higher when exhaust gas recirculation (EGR) is applied. With pre-chamber gas engines, the thermodynamic states in the pre-chamber and in the main combustion chamber have to be investigated in detail in order to ensure stable ignition and combustion. 1D simulation is a good choice that allows the demands of this kind of investigation to be met. This paper explains how 1D simulation is applied to a variety of different tasks in gas engine development with EGR.

2. Emission limits

The current trend is towards lowering emission limits for large combustion engines. As Figure 1 shows, pollutant emissions from large engines and nitric oxide emissions (NOx) in particular are strictly regulated [1]. Whereas the current German TA Luft legislation [2] for gas engines defines the maximum nitric oxide emissions as 500 mg/m3 (norm) @ 5% O2, the trend is towards a further reduction to 250 mg/m3 (norm) @ 5% O2 (Gothenburg Protocol [3]) or 75 mg/m3 (norm) @ 15% O2 (EU Directive 2010/75/EU [4]), whereas 75 mg/m3 (norm) @ 15% O2 is equivalent to 200 mg/m3 (norm) @ 5% O2. In certain European regions, emissions below 100 mg/m3 (norm) @ 5% O2 are even required [5].

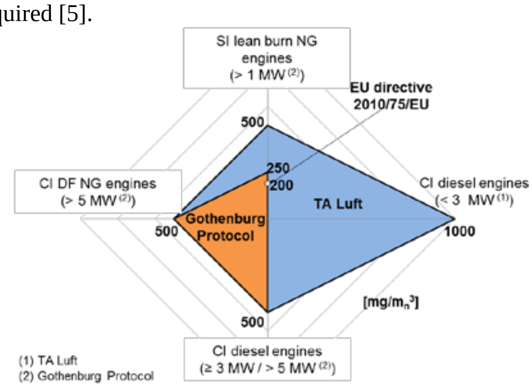

Fig. 1 *NOx emission limits for stationary gas engines in the EU.*

3. Engine concepts for EGR operation

The formation rate of nitrogen oxides is controlled by the temperature in the reaction zone and the equilibrium concentrations of the relevant species. One possible strategy to reduce NOx emissions is exhaust gas recirculation (EGR). The mechanism of temperature reduction with EGR is caused by the increased specific heat capacity determined by the content of carbon dioxide and water in the burned gas (caloric effect) [6].

There are three main combustion concepts for gas engines used with EGR: a lean burn concept with moderate EGR (< 15%), stoichiometric combustion with EGR and a three-way catalyst, and HCCI combustion. Furthermore, the ignition system is very important to the EGR combustion concept. It must provide good inflammation and stable combustion in the cylinder. The ignition systems that are most commonly used are: open chamber with direct spark ignition, pre-chamber with spark ignition and diesel pilot ignition of the mixture. The pre-chamber variant can be a small unscavenged pre-chamber spark plug, an unscavenged pre-chamber or a scavenged pre-chamber.

The investigations in this paper focus on the stoichiometric combustion concept with pre-chamber ignition using EGR. The EGR dilutes the mixture in the cylinder so that the engine can be operated at higher loads with reduced knock tendency and low NOx formation.

4. Application of 1D simulation to develop a new engine with EGR

The following section provides three examples of applications that use 1D simulation in the process of developing a new engine with an EGR system. First, the role of 1D simulation in the design of a single cylinder engine test bed and the determination of boundary conditions for testing is explained. Furthermore, basic investigations on the influence of pre-chamber scavenging and the influence of the scavenging pressure gradient on combustion in the pre-chamber are discussed in detail.

4.1. Design of a single cylinder test bed and determination of boundary conditions for testing

While the piping system of the single cylinder engine (SCE) test bed is being designed, it is important to adapt the gas dynamics in the intake and exhaust piping system. The objective is to achieve similar conditions in the cylinder of the SCE and the corresponding multicylinder engine (MCE). During the design phase, the measured pressure curves upstream and downstream of the cylinder head are compared using a 1D model of the SCE. The piping system of the SCE is adjusted according to these results; the engine setup and the adapted piping system on the test bed can be seen in Figure 2. To this end, satisfactory agreement between the pressure traces resulting from the gas dynamics of the 1D SCE model and the measured pressure curves from the MCE at different loads and engine speeds is found.

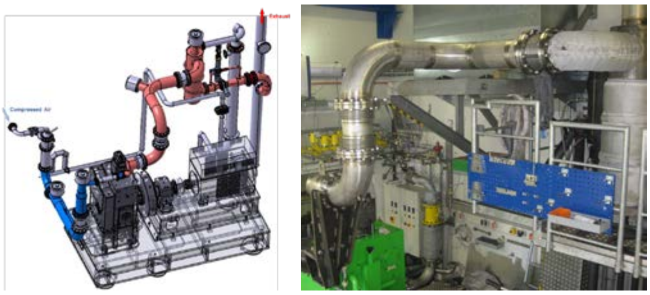

Fig. 2 Piping system of the single cylinder research engine on the LEC test bed.

For a detailed analysis of the combustion system, a 1D simulation model of the multicylinder gas engine has been used to generate the boundary conditions required for calculations with the 1D single cylinder model. The multicylinder engine is equipped with one-stage turbocharging. Its system layout is a low pressure EGR system with an EGR cooler and a back pressure flap on the exhaust side.

The 1D SCE model corresponds to the engine integrated on the test bed. Figure 3 shows the topology of the model in the commercial software package GT Power including the design of the EGR system on the test bed.

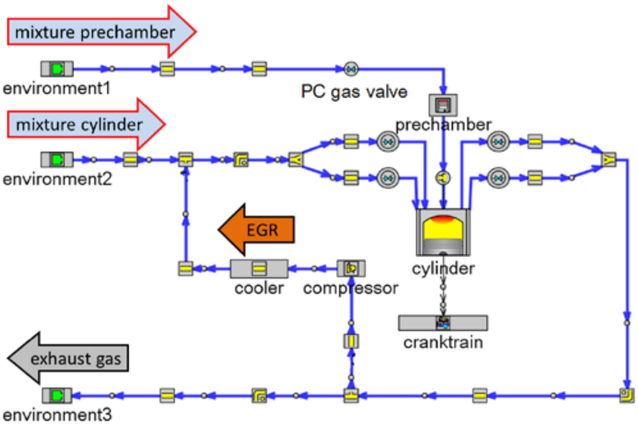

Fig. 3 *Topology of the 1D model of the single cylinder research engine including the EGR system.*

The challenge in developing the model is to determine the conditions in the pre-chamber as precisely as possible, therefore also the pre-chamber is depicted in the 1D model. Air and gaseous fuel is mixed in the intake system. Exhaust gas is taken from the exhaust pipe, pumped via a compressor to the desired pressure level and cooled in the heat exchanger. The recirculated exhaust gas is mixed with the fresh charge in the intake pipe upstream of the cylinder head. On the real test bed, water condensing in the EGR cooler on the test bench is removed. This part of the EGR system is not displayed in the 1D SCE model. The pre-chamber mixture flows through the gas rail and the pre-chamber gas valve.

4.2. Influence of pre-chamber scavenging

In an unscavenged pre-chamber, the mixture from the main combustion chamber is pushed into the pre-chamber during the compression stroke. Depending on the concept, a certain amount of residual gas from the previous cycle remains in the pre-chamber, thereby increasing the residual gas content in the pre-chamber at ignition timing.

With a stoichiometric combustion concept, a purely gas scavenged pre-chamber concept inevitably leads to an understoichiometric mixture in the pre-chamber in combination with EGR from the main combustion chamber [7]. Ideally, the pre-chamber is scavenged until shortly before ignition timing in order to keep the amount of residual gas in the pre-chamber low. An alternative procedure is scavenging either with pure air or λ=1 mixture without EGR at a lower pressure level. With comparatively low system complexity, it is thus possible to lower the residual gas content in the pre-chamber at ignition timing. To demonstrate the influence of the scavenging gas and different moderate rail pressures, simulations were carried out with the 1D SCE model.

Figure 3 shows the single cylinder model used for the calculations with all the concepts. The baseline was a measuring point with a pre-chamber spark plug and about 30% EGR. A gas injector was used as a pre-chamber gas valve. The boundary conditions for pre-chamber scavenging were baseline rail pressure, baseline rail pressure +2.5bar and baseline rail pressure +7.5bar.

Figure 4 provides examples of pre-chamber gas feed flow curves with λ=1 scavenging mixture. The start of injection is derived from the calculation with baseline rail pressure and is kept constant for all variations. It is the earliest crank angle at which a positive mass flow is possible because of the pressure scavenging gradient between the rail and the pre-chamber. The end of injection can be shifted in the direction of ignition timing by increasing the rail pressure.

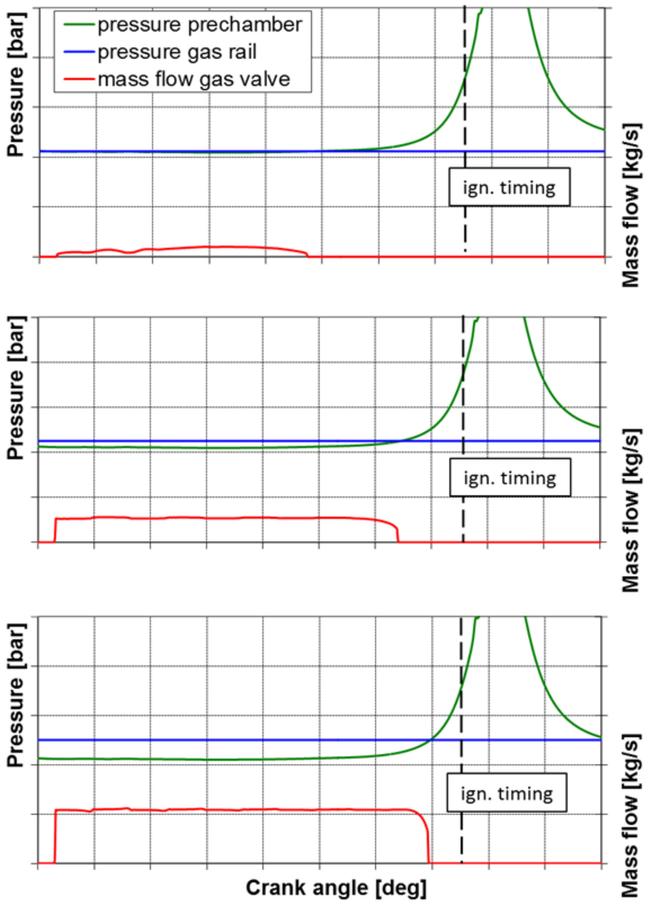

Fig. 4 *Pre-chamber gas feed mass flow at different gas rail pressures.*

In the unscavenged pre-chamber, the residual gas content decreases during the compression phase because a fresh mixture flows from the cylinder into the pre-chamber. In the pre-chamber scavenged with a λ=1 mixture, the residual gas fraction increases during the compression phase because the fresh mixture in the main combustion chamber contains recirculated exhaust gas. Figure 5 to Figure 7 show the simulation results with 30% EGR and pre-chamber scavenging with a stoichiometric mixture. Figure 5 contains a graph of the amount of residual gas in the pre-chamber versus crank angle. In contrast to the unscavenged baseline variant, the residual gas content can be reduced by approximately 20% points at the ignition timing with the highest rail pressure (gas rail variant 1). In comparison to the unscavenged pre-chamber, the reduction of the residual gas at the ignition timing for the scavenging variant with λ=1 and baseline rail pressure (variant 3) is only 8%.

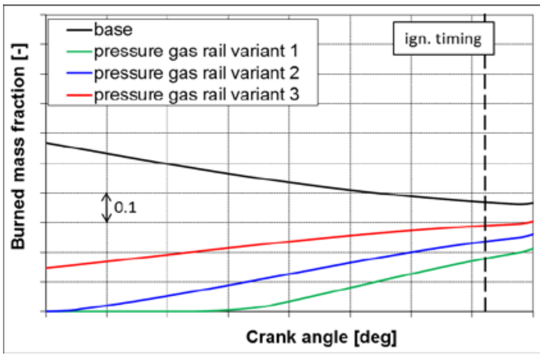

Fig. 5 Scavenging with a stoichiometric mixture: burned mass fraction in the pre-chamber at different gas rail pressures.

With the λ=1 scavenged pre-chamber, the fuel mass and thus the energy in the pre-chamber at the ignition timing can be increased, see Figure 6.

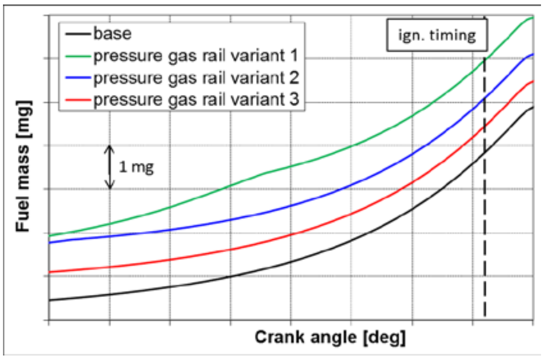

Fig. 6 Scavenging with a stoichiometric mixture: fuel mass in the pre-chamber at different gas rail pressures.

However, the temperature in the pre-chamber is lower with all the scavenging variants, see Figure 7. The temperature reduction is about 150K with the scavenging variant with λ=1 and the highest rail pressure (variant 1) in comparison to the unscavenged pre-chamber. Although there is a higher fuel mass and lower residual gas content at ignition timing, the effect of temperature reduction can impair the ignition conditions in the pre-chamber.

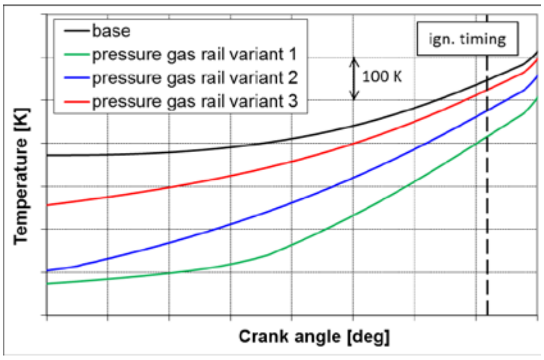

Fig. 7 Scavenging with a stoichiometric mixture: temperature in the pre-chamber at different gas rail pressures.

The effect of different scavenging gases on the conditions in the pre-chamber was also investigated. Figure 8 to Figure 11 compare the simulation results with 30% EGR and pre-chamber scavenging with methane, air and a stoichiometric mixture to those with the unscavenged baseline. The rail pressure is the same for all simulated variants and is equal to the boost pressure.

Due to the different densities of the scavenging gases, the mass flows through the pre-chamber gas rail vary, resulting in different residual gas fractions in the pre-chamber at the ignition timing. The burned mass fraction can be up to 8% points lower than that of the unscavenged pre-chamber variant, see Figure 8.

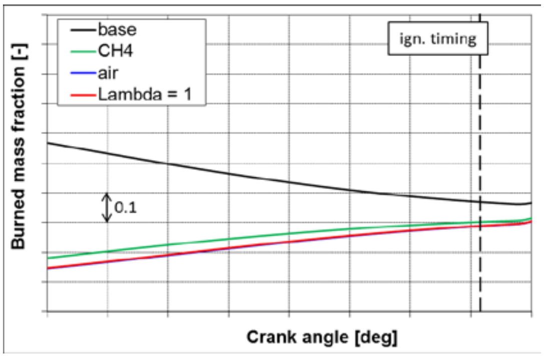

Fig. 8 Scavenging with different gases and baseline gas rail pressure: burned mass fraction in the pre-chamber.

The fuel mass and hence the energy in the pre-chamber at ignition timing increases when it is scavenged with methane, see Figure 9. A slight increase occurs even with a stoichiometric mixture. Scavenging with air leaves the fuel mass in the pre-chamber unchanged compared to the baseline variant.

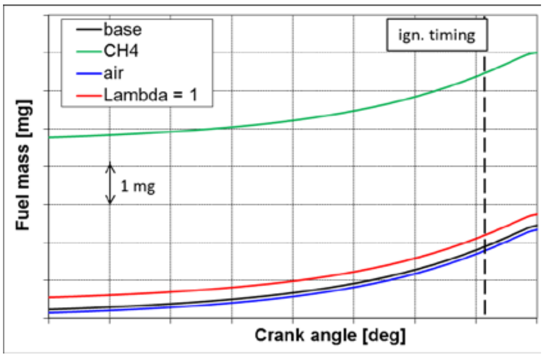

Fig. 9 Scavenging with different gases and baseline gas rail pressure: fuel mass in the pre-chamber.

Slight temperature differences in the pre-chamber at ignition timing are due to the different specific heat capacities of methane, air and λ=1 mixture. The higher specific heat capacity of methane causes the greatest reduction in temperature (50K) compared to the baseline variant, see Figure 10.

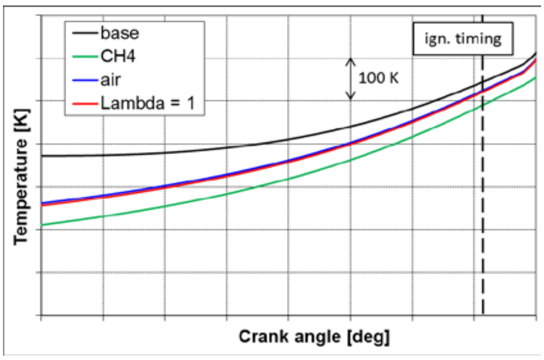

Fig. 10 Scavenging with different gases and baseline gas rail pressure: temperature in the pre-chamber.

As expected, pre-chamber scavenging with a stoichiometric mixture and the unscavenged baseline have a λ=1 mixture at ignition timing, (see Figure 11). Scavenging with air results in a higher λ value in the pre-chamber while scavenging with methane leads to a very rich mixture.

Scavenging with a stoichiometric mixture as well as scavenging with air yield similar temperatures and residual gas contents in the pre-chamber. The λ=1 mixture has a higher energy content at ignition timing.

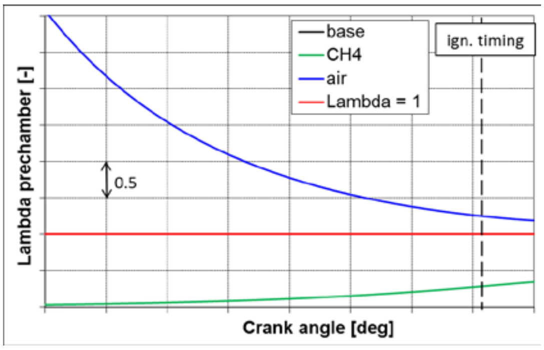

Fig. 11 *Scavenging with different gases and baseline gas rail pressure: lambda in the pre-chamber.*

4.3. Influence of the scavenging pressure gradient on combustion in the pre-chamber

The scavenging pressure gradient across the engine was investigated using a 1D SCE simulation model calibrated using measurements from the single cylinder research engine. These SCE measurements have shown that the scavenging pressure gradient has a clear influence on the exhaust temperature and combustion phasing. As shown in Figure 12, operating points with a variation in scavenging pressure gradient were simulated. The input parameters (burn rate, lambda, etc.) required for the simulation model were taken from the measurements. The measurements show that a high scavenging pressure gradient causes earlier combustion phasing as a result of the shorter ignition delay. The difference in ignition delay is mainly due to the states in the pre-chamber at ignition timing.

Figure 12 shows the residual gas content in the pre-chamber depending on scavenging pressure gradient.

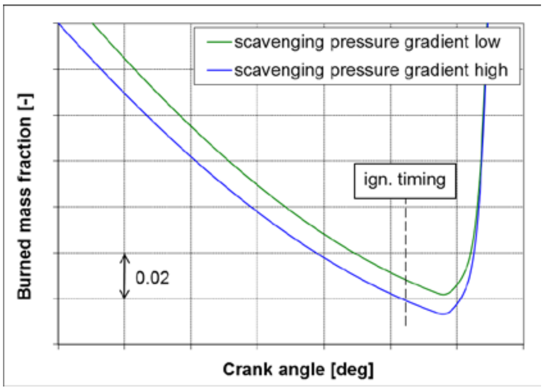

Fig. 12 *Variation in the scavenging pressure gradient, burned mass fraction in the pre-chamber.*

When the scavenging pressure gradient is high, the residual gas content in the pre-chamber is around 1% point lower than with a low scavenging pressure gradient. The difference between the low pressure gradient and the high pressure gradient is about 300 mbar. Since there is less residual gas mass in the pre-chamber, the fuel mass increases in the relevant crank angle range. The measurements from the SCE have shown that the scavenging pressure gradient influences the exhaust gas temperature. A higher scavenging pressure gradient results in a lower exhaust gas temperature. 1D simulation with a constant burn rate yields a reduction in exhaust gas temperature of about 12K with a high scavenging pressure gradient. The different pre-chamber scavenging concepts (including the unscavenged concept) were tested on the SCE test bench at LEC in order to evaluate them in terms of operating behavior and engine map limits. The results from the 1D simulation were confirmed. For example, the scavenged pre-chamber with a stoichiometric mixture and EGR operation leads to a greater pressure increase in the pre-chamber during combustion due to the higher fuel mass at ignition timing. As a consequence, the mixture in the main chamber burns faster.

5. Summary and Conclusions

In this paper several examples of the application of 1D simulation in the development of combustion systems for large gas engines with special focus on ignition concepts with EGR and λ=1 were shown.

The unscavenged pre-chamber ignition concept and the scavenged ignition concept were evaluated in terms of their suitability for operation with λ=1 and EGR. While experimental research was mainly conducted on a single cylinder research engine, 1D simulation was used to pre-design the concepts and to support the interpretation of the results.

Scavenged pre-chamber concepts were investigated using a 1D simulation model with a pre-chamber. The focus was on the composition of the charge in the pre-chamber at ignition timing. It turned out that the amount of residual gas in the pre-chamber is reduced when the pre-chamber is scavenged with pure air or a clean mixture (λ=1, without EGR).

The exhaust gas back pressure was reduced as a measure to reduce the exhaust gas temperature. Based on measurements on the SCE, the influence of back pressure was investigated with 1D simulation. The measurements and the 1D simulation reveal a reduction in exhaust gas temperature when the scavenging pressure gradient is increased. Measurements on the SCE test bench and results from 1D simulation show that combustion is highly sensitive to the conditions (residual gas content) in the pre-chamber. A reduction in the residual gas content in the pre-chamber of 1% point results in an earlier combustion phasing and a lower exhaust gas temperature (the scavenging pressure gradient is increased by 300mbar).

When operated with EGR, the pre-chamber scavenging concepts can be realized easily and offer benefits compared to the unscavenged variant.

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