Development of A Clean Diesel Combustion System by Engine Testing and CFD-Simulation

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Future European market trends favor system solutions with low fuel consumption and low raw emissions to reduce the amount of exhaust gas aftertreatment. On this market, the challenge is to deliver a system concept and demonstrate its technical advantage in the competition. The optimization of the combustion system within the engine boundaries of engine friction, turbo-charger and gas exchange for low emissions, noise and fuel consumption with a target of high power density is complex. Hence the engine testing becomes time- and cost intensive even though state-of-the art tools as Design of Experiments and Model Based Calibration methods are applied. Therefore the optimization of the piston bowl design and selection of nozzle parameters e.g. spray cone angle, no. of holes is evaluated by the usage of CFD simulation. Although this combined approach of simulation and testing has limited prediction, the new combustion system achieves the emission targets with the given fuel consumption penalty.

Introduction

The market demands for the next legislation limit of EU6 are quite challenging from system point of view. Low fuel consumption and low raw emissions are a necessity to get customer acceptance from environmental and system cost point of view. The key to control the combustion process is the injection system to phase the combustion in time and space and the air-path management for intake temperature and oxygen content control [1].

In this study, an existing 2.0l, 4 cylinder EU4 engine, CR=16, is used to demonstrate the capabilities of DENSO's Engine Management System. The engine configuration can be viewed in Fig. 1.

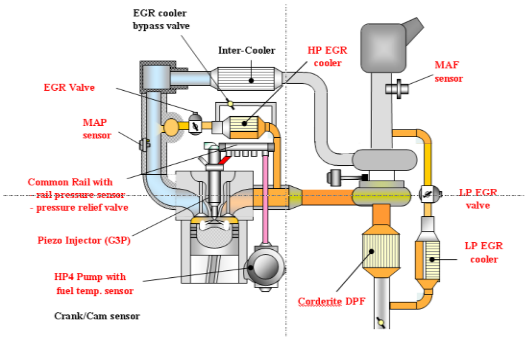

Fig. 1 Test-engine with replaced Engine Management System (EMS)

The objective of this study is to demonstrate EU6 emissions and to increase the power density from 55 to >60 kW/l by downsizing: two engine versions A and B are existing. The higher boost pressure of version B compared to A is beneficial to increase the engine power density [2]. Three steps have been applied to this base engine:

1. Change of the baseline series 1600 bar Piezo to 3rd generation 2000 bar common rail pressure system with DENSO's Piezo injector G3P.
2. Adaptation of the air path by an additional Low-Pressure-Loop (LPL) EGR system to demonstrate EU6 emission levels
3. Change to a high performance turbo-charger with adaptation of the bowl change to achieve EU6 emissions with an increased power density.

The design of the combustion chamber is the major focus in this study. LIEF measurements indicate that the spray of the G3P injector has a leaner distribution than the baseline injector as seen in Fig. 2. Moreover, the spray penetrates deeper into the piston bowl due to an increased rail pressure in comparison to the baseline. Both features have to be addressed by the design of the combustion bowl chamber. Thus a re-design of the bowl-chamber is necessary and will be supported by CFD simulations. The CFD code FIRE from AVL was used in this study. The spray model is the well known Discrete Droplet Model (DDM). Baseline engine data was used to calibrate the spray model parameters due to the limitations of this approach [3]. With regard to combustion, the ECFM-3Z model [4] is applied. In the following, a new piston bowl was developed by a combination of CFD and engine testing.

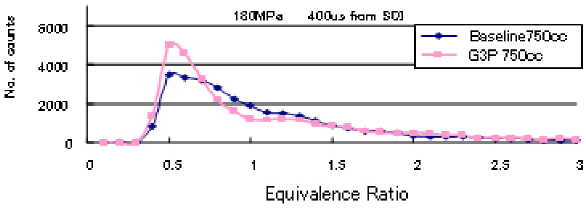

Fig. 2 Equivalence ratio distribution between baseline series engine injector and G3P.

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Development of a New Piston Bowl

An initial bowl design denoted as piston 1 was proposed based on CFD calculations as shown in ***Fig. 3***. The potential for this bowl is indicated by a better soot oxidation among the baseline and other bowl proposals.

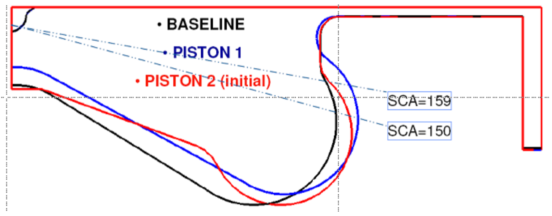

Fig. 3 Piston bowl designs

In a second step, nozzle parameters as spray cone angle and no. of holes were optimised by CFD to define a nozzle matrix since the spray-bowl interaction is one parameter to control the soot formation [5]. The CFD simulation predicts a better emissions performance of the new piston 1 bowl with an increased no. of holes from 6 to 8 and an increased spray cone angle from 150° to 159° as it is seen in ***Fig. 4*** and ***Fig. 5*** at a higher part-load emission Mode point (engine speed of 2250 rpm, BMEP of 8 bar).

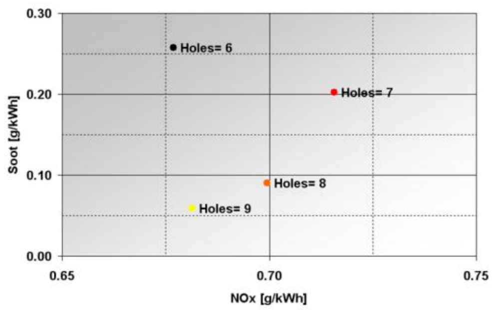

Fig. 4 Variation of no. of holes

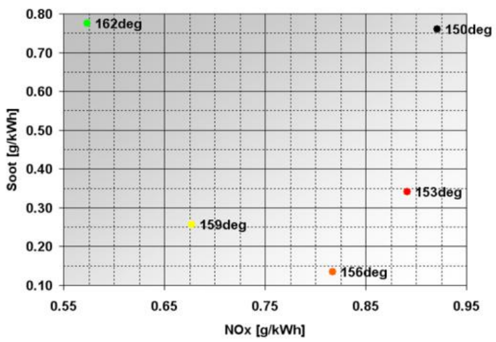

Fig. 5 Variation of spray cone angle

Engine Testing of Piston Bowl 1

The piston 1 design and nozzle samples were manufactured and evaluated by engine testing. The testing procedure includes a calibration procedure in all emission mode points by Design of Experiments (DoE) and Model Based Calibration (MBC) methods as well as manual calibration at full load. ***Fig. 6*** shows the Filter Smoke Number (FSN) at rated engine conditions for various nozzle tip protrusions, hydraulic Flow Rates (HFR) and no. of holes. The rated power is only limited by the turbine temperature. Furthermore, the nozzle tip protrusion (NTP) was fixed to 2.9 mm, the no. of holes to 8 and the HFR to 750 cm³/min which results in a hole size diameter of 121µm. Overall, a power density of 63 kW/l can be achieved.

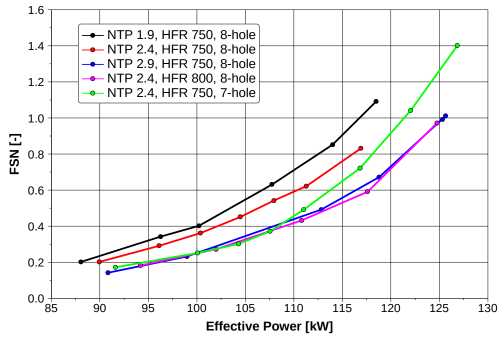

Fig. 6 Selection of NTP, HFR and no. of holes

For the high load emission points e.g. at engine speed of 2250 rpm and BMEP of 8 bar, only LPL-EGR was used. The NOx-soot trade-off is influenced by the cooling efficiency (***Fig. 7***). Increasing the efficiency from 54% to 85% reduces t_{ic} from 65°C to 40°C.

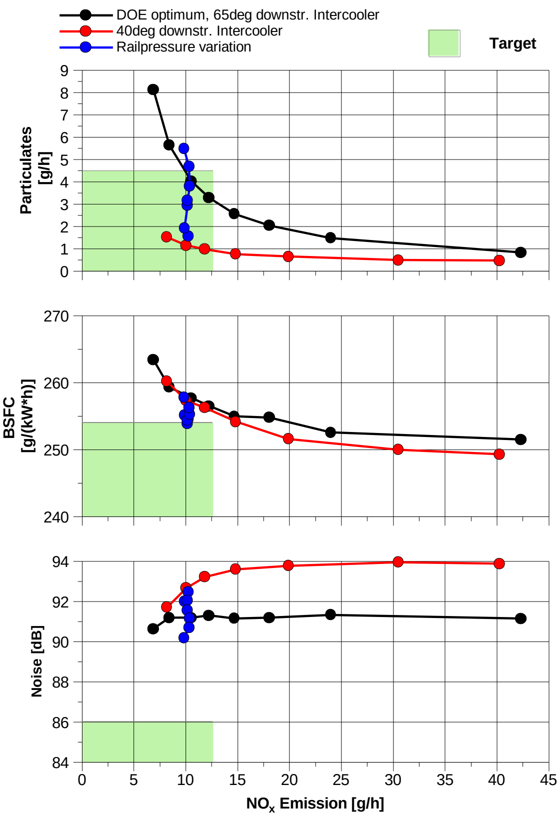

Fig. 7 Effects of intake charge cooling and rail pressure variation on performance

The effect of intake charge cooling can be viewed in the combustion analysis from ***Fig. 8***. The heat release by the early double pilot injection is not changed but the ignition of the late main injection is retarded. The premixed combustion is increased as the higher peak in ROHR indicates and less diffusion controlled combustion of rich areas occurs so that soot emissions are reduced.

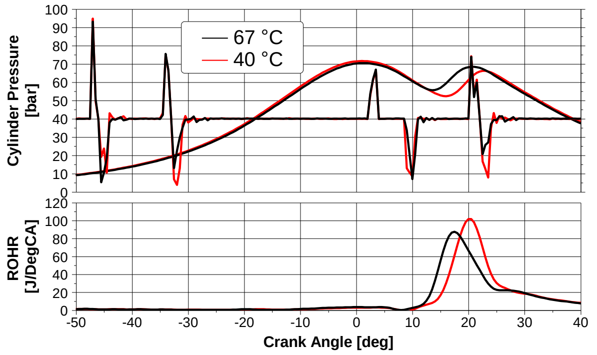

Fig. 8 Effect of intake charge cooling

A major challenge is to reduce the NOx-soot trade-off under a penalty in BSFC and noise. The retarded combustion shows a higher noise and lower soot level. If the rail pressure is additionally reduced, the soot emission benefit from the cooled intake charge is converted into a combustion noise benefit.

Evaluation of Piston 2 Bowl Design

In a second step, the piston 1 design was slightly changed to address the robustness sensitivity on injector production tolerances on the spray-bowl intersection and as shown in ***Fig. 3*** and to improve the thermal robustness.

The evaluation of piston 2 design included both, simulation and engine testing in a simultaneous process. The CFD simulation of piston 2 bowl design shows that the fuel vapor is pushed from the piston bowl into the squish area (***Fig. 9***). The mixture in the bowl becomes leaner (***Fig. 10***) but air-excess is still available in the squish area.

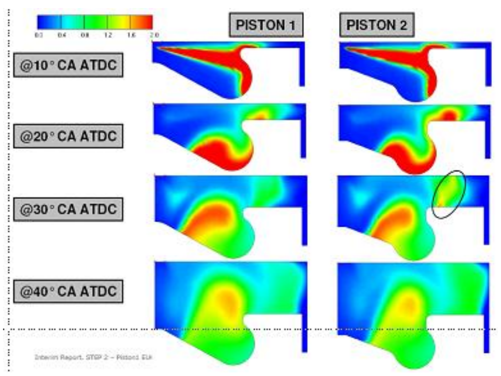

Fig. 9 Comparison of equivalence ratio distribution between piston 1 and 2 design

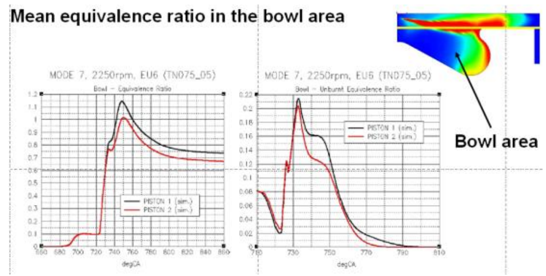

Fig. 10 Post-processed distribution of fuel vapor between bowl and squish area

The engine testing of piston 2 bowl design indicated that the SCA of 159° has to be decreased to 155° due to an increase in soot emissions. In order to address the penalty in noise caused by decreased intake charge temperature a second option is to advance the pilot injections closer to the main (***Fig. 11***) which follows a more effective pilot combustion.

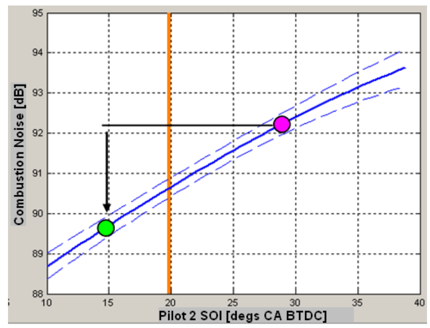

Fig. 11 Pilot timing effect on the noise model

A more effective pilot injection will shorten the ignition delay of the main injection. Therefore less time is available to homogenise the mixture and less premixed combustion will decrease the noise but vice versa more diffusive burning of rich mixture increases the soot emissions as it can be observed off-line from the MBC in ***Fig. 12***. The soot advantage of the decreased intake charge temperature can be changed into a noise benefit at a constant rail pressure level.

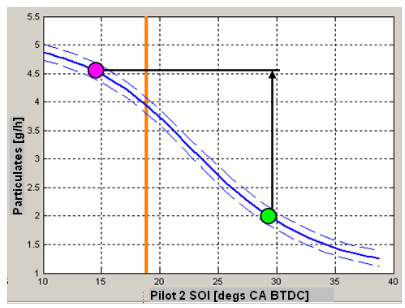

Fig. 12 Pilot timing effect on the soot model

The final engine performance is demonstrated in ***Fig. 13***. The emissions as well as the BSFC target can be achieved. An additional measurement showed that any further noise reduction would violate the BSFC penalty. This limitation is inherent to the system boundaries of the engine configuration. The high performance turbo-charger of engine B requires a higher back-pressure at the end of the expansion stroke compared to engine A and increases the pumping losses.

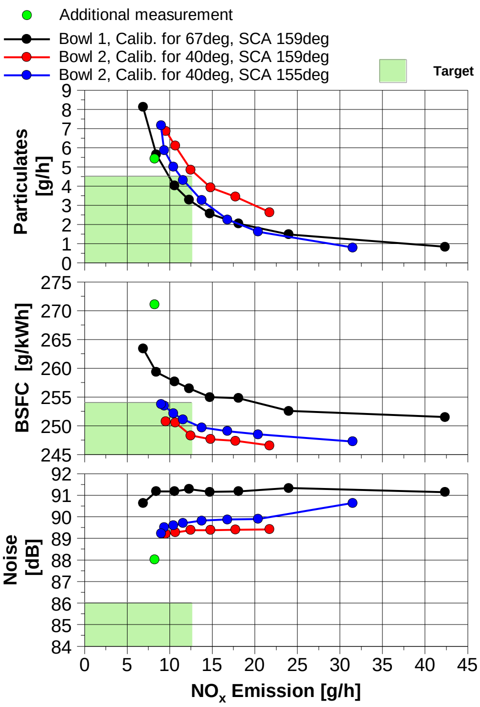

Fig. 13 Engine performance of piston 2 bowl design

Vehicle emissions are estimated for a NEDC in ***Fig. 14*** from four emission mode points. Overall, EU6 emissions are achieved with the current configuration. If noise and BSFC shall be furthermore reduced, the system configuration has to be changed. Either the low performance turbo-charger can be used if a lower power density is accepted or a two-stage turbo-charger with a better performance at part-load conditions could be considered but increase the system costs.

Summary and Conclusion

Engine development to meet new legislation limits is to be considered as a system optimization process within the given boundaries. This process was accomplished on a series production engine to demonstrate EU6 emissions with a high power density target including the FIS and air-management system.

The combined usage of CFD and engine testing enables a pre-selection of nozzle parameters and definition of bowl shape geometry which must be adapted to the individual spray characteristics.

Real engine testing is still mandatory and cannot be omitted. The final calibration of the engine testing by DoE and MBC is including high rates of cooled EGR to shift the combustion towards lower temperatures and better homogenisation of the spray. The BSFC depends on the air-management system. Especially the performance of the turbo-charger has to be selected carefully. The higher specific power at rated conditions requires a higher boost pressure but will violate the constraint in BSFC on part-load conditions which is transferred into a violation of the constraint in noise level. A higher fun-to-drive pays back immediately by an acceptance of a higher noise level or by usage of a two-stage turbo-charger and increased system cost.

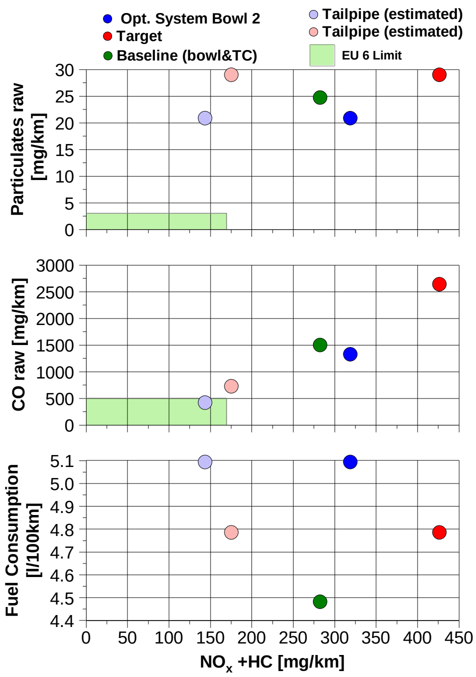

Fig. 14 Vehicle estimation

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