

CFD Analysis of the Strategies Aimed to Modernize an Industrial Diesel Engine

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Diesel engines occupy an important share of the commercial sector. Despite environmental trends, the transition to new powertrains will be gradual. Therefore, it is rational to make existing engines as powerful, efficient, and clean as possible.

In the main part of the paper, a series of diesel engine simulations were performed in which an increase in boost pressure and combustion air cooling were assumed. Further, the application of exhaust gas recirculation and a modification in the piston crown profile was simulated. It was analyzed how the aforementioned changes are reflected in the emissions of harmful gases.

Cylinder pressure rise was obtained. Nitrogen oxide emissions were significantly reduced by exhaust gas recirculation, although this strategy negatively influenced soot and hydrocarbon emissions. Finally, it was confirmed that the geometry of the piston crown can have a beneficial effect in reducing harmful gas emissions. The analysis shows that existing engines can be improved with relatively limited interventions.

Keywords: diesel engine modernization, computational fluid dynamics (CFD) simulation, boost air, exhaust gas recirculation (EGR), piston bowl profile

1. INTRODUCTION

The global energy sector is undergoing a profound transformation driven by the need to mitigate climate change and reduce greenhouse gas emissions. International agreements, national decarbonization strategies, and increasingly stringent environmental regulations have accelerated the transition toward sustainable energy systems. While renewable energy technologies continue to expand, the transportation sector remains one of the most challenging areas for decarbonization. Transport accounts for a significant share of global carbon dioxide emissions, making the improvement of vehicle energy efficiency and emission reduction a central objective of contemporary engineering research [1].

Despite the rapid development of alternative propulsion systems, diesel engines continue to play a crucial role in global transportation and industrial applications. Heavy-duty trucks, buses, construction machinery, agricultural equipment, marine vessels, and stationary power generation systems rely extensively on compression ignition engines due to their superior thermal efficiency, durability, and high torque characteristics. In many sectors, particularly long-haul freight transport and off-road machinery, diesel engines remain difficult to replace because current battery technologies face limitations related to energy density, charging infrastructure, cost, and operational flexibility [2-5].

The perception and development trajectory of diesel technology changed significantly following the Diesel-

gate scandal in 2015 [6,7]. As a consequence, public confidence in diesel-powered vehicles declined, while regulatory authorities introduced more rigorous testing procedures and tighter emission standards. These developments accelerated research into cleaner combustion strategies, advanced aftertreatment systems, and alternative propulsion concepts.

European emission legislation has evolved continuously to address environmental concerns associated with internal combustion engines. The introduction of successive emission standards, culminating in the Euro 6 regulation and its subsequent revisions, has imposed strict limits on nitrogen oxides (NO_x), particulate matter (PM), particle number (PN), hydrocarbons (HC), and carbon monoxide (CO) emissions [8]. Compliance with these regulations requires the integration of advanced fuel injection systems, exhaust gas recirculation (EGR), selective catalytic reduction (SCR), diesel particulate filters (DPF), and sophisticated engine control strategies.

Simultaneously, significant changes are occurring in energy production and transportation technologies. Hybrid powertrains, battery electric vehicles (BEVs), fuel-cell electric vehicles (FCEVs), and renewable fuels are increasingly considered essential components of future mobility solutions. Nevertheless, projections from numerous energy and transportation studies indicate that internal combustion engines will continue to contribute substantially to the global vehicle fleet for decades, particularly in commercial transportation and regions where electrification infrastructure remains underdeveloped [2-5]. Therefore, improving the efficiency and environmental performance of existing engine technologies by retrofitting solutions and modifications remains an important pathway toward reducing overall emissions during the transition period.

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Recent research trends in internal combustion engine development focus on maximizing thermal efficiency while simultaneously minimizing pollutant formation. Some advanced combustion concepts demonstrated significant potential for reducing emissions. Further, improvements in turbocharging systems, high-pressure common-rail fuel injection, variable valve actuation, waste heat recovery, and digital engine control have enabled substantial gains in engine performance and fuel economy. The integration of renewable and synthetic fuels, including biodiesel, hydrotreated vegetable oil (HVO), e-fuels, and hydrogen-assisted combustion, has also emerged as an active area of investigation aimed at reducing lifecycle carbon emissions [9-12].

The increasing complexity of modern engine systems has made computational tools indispensable in the development process. Computational Fluid Dynamics (CFD) has become one of the most powerful methodologies for analyzing in-cylinder processes, enabling detailed investigation of air motion, fuel atomization, spray dynamics, evaporation, mixture formation, combustion, heat transfer, and pollutant generation [13].

There are many studies on the influence of boost pressure, EGR, piston bowl shape, and other parameters on engine performance.

Söyler [14] tested how increased boost pressure influences engine performance and emission characteristics with various fuels. He concluded that increased boost generally improves engine power and reduces CO, HC, and particulate emissions. However, NO_x emissions increase with the boost level. Vargün et al. [15] analyzed the influence of fuel injection parameters and boost pressure on combustion characteristics in a diesel engine using alcohol/diesel blends. Ju et al. [16] conceptualized a small diesel engine featuring high boost pressure and EGR values and advanced injection timing. They performed a numerical simulation, and both the combustion performance and brake-specific fuel consumption were improved; moreover, the NO_x emission met the Euro 6 emission regulation. Hunicz et al. [17] optimized the single-cylinder engine operation on HVO. They used split injection, varied boost and EGR level. Better efficiency and lower emissions were obtained.

There were several authors [18-22] who focused on diesel engine piston bowl optimization. CFD tools and optimization techniques are used to decrease emissions, often in combination with other strategies and alternative tools. Sreedharan et al. [23] made an attempt to develop a tool to design the bowl profile, based on spray parameters obtained by a numerical simulation software. Hatz et al. [24] optimized the operation of an industrial engine by means of compression ratio, piston-bowl shape, and injection strategy. The performances were improved while emissions could be reduced.

Despite considerable research into advanced boosting, EGR, injection strategies, and piston-bowl optimization, relatively little attention has been given to their combined application as a modernization strategy for established industrial diesel engines. A considerable number of diesel engines currently in operation were designed before the implementation of many modern

technologies. There exists substantial potential for performance enhancement through the application of advanced engineering approaches, including optimized combustion chamber geometries, improved air management strategies, modern turbocharging concepts, and refined fuel injection characteristics. The present paper is exploring some of the methods that can improve performance and emission characteristics of these already installed engines.

The MAN D 0826 diesel engine represents a typical example of a durable and widely utilized engine platform whose original design predates many recent technological advancements. By employing modern CFD simulation techniques and leveraging contemporary understanding of combustion and fluid flow phenomena, it is possible to identify opportunities for increasing engine power while maintaining or even reducing pollutant emissions. Therefore, this study investigates the potential for enhancing the performance of the MAN D 0826 engine through computational analysis and optimization, demonstrating how modern engineering tools can extend the operational relevance and environmental compatibility of established diesel engine designs in the ongoing transition toward a lower-carbon transportation sector.

2. MATERIALS AND METHODS

2.1 The analyzed engine

The MAN D 0826 LOH15 engine, mounted on a brake in the Laboratory for Thermal Piston Engines of the Faculty of Mechanical Engineering, University of Ljubljana, was used for the analysis, Figure 1. The basic characteristics of the engine are given in Table 1. The engine is equipped with a mechanical, in-line injection pump. The injector was Bosch DSLA 154 P 492. It features 7 holes of the diameter of 0.23 mm and a 154° spray angle. Injection pressure varies depending on the regime and equipment state, but it can be estimated around 250-270 bar. In order to maximize the flow area of the 2 valves (intake and exhaust), the injector is placed eccentrically, and therefore the recess in the piston is also placed this way. This is important when generating a mesh for 3D numerical simulation, because when simulating only one sector that includes 1 jet to save processor time, the sectors are not equal. However, it showed that the eccentricity influence of the injector is negligible [25].

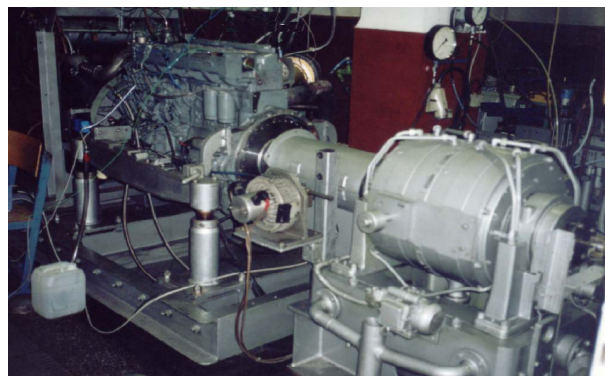


Figure 1. The engine MAN D 0826 LOH 15 at the test bed

Table 1. Main engine characteristics

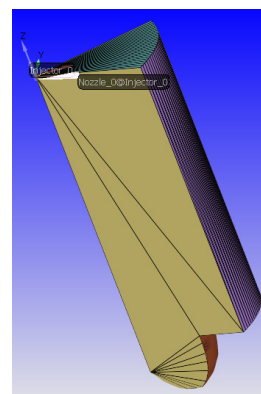
Displacement	6.87 l
Number of cylinders	6
Power	160 kW
Bore	0.108 m
Stroke	0.125 m
Compression ratio	18
Connecting rod length	0.1872 m

2.2 Numerical simulation settings

The commercial software Converge CFD [26] was used for engine simulations. The simulations are based on the finite volume method and Reynolds Averaged Navier Stokes (RANS) approach. Turbulence modeling was done with the renormalization group (RNG) k- ϵ model, which predicts, due to its additional terms, the turbulent dissipation and, therefore, the flow field more accurately in engine flows than the standard k- ϵ model. The RNG model was developed using renormalisation group (RNG) methods to account for the effects of smaller scales of motion [27]. The liquid phase was modelled using a Lagrangian approach and the built-in surrogate fuel DIESEL2 available in CONVERGE CFD. This surrogate represents a pure diesel liquid phase and is employed solely for spray breakup and evaporation modelling. Upon evaporation, the fuel is converted to n-heptane (C_7H_{16}), which serves as the species for the chemical reaction mechanism. To reduce computational cost, droplets with identical properties are grouped into parcels. Fuel injection was modelled using the Blob Injection Model, in which all injected droplets are initially assigned a diameter equal to that of the injector nozzle. Aerodynamic drag was calculated using a dynamic drag model that accounts for droplet deformation predicted by the Taylor Analogy Breakup (TAB) model. The primary spray breakup of a high-pressure liquid jet close to the injector is predicted with the Kelvin–Helmholtz (KH) model, whereas the secondary spray breakup of so-called child parcels is additionally modeled with the Rayleigh–Taylor (RT) model [28]. The CTC/Shell model was used to simulate combustion. It was developed in [29] and can be used to simulate autoignition in diesel engines. It was modified by [30] to account for mass conservation. As for NOx emissions, the extended Zeldovich mechanism is used [31]. Soot formation and oxidation were modeled by a standard Hiroyasu-NSC model [32], [33].

A great feature of Converge CFD software is its grid control. The mesh does not have to be defined by the user in advance, as in traditional CFD simulations. Only the boundaries of the domain have to be defined, for instance, with surfaces in the standard “.stl” format. The domain is then automatically meshed with a uniform Cartesian, orthogonal mesh of predefined cell size during the simulation. The grid can be refined in critical regions (technique called “embedding”) or based on fluctuating and moving conditions such as temperature or velocity, a technique called “Adaptive Mesh Refinement” (AMR) [26], [34], [35]. This frees the user of the very time-consuming task of mesh preparation. The simulated injector has 7 holes; hence, a 51.43° sector was simulated to save time. The basic domain

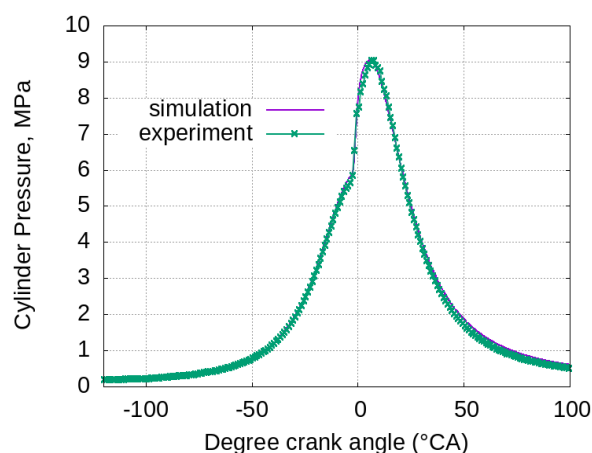
outline of the simulated engine sector was generated with the Converge utility “Make engine sector surface”, as shown in Figure 2.

**Figure 2. Domain outline of the cylinder sector**

3. SIMULATIONS

Validation

In order to check the influence of different operation parameters, the model has to be validated against experimentally obtained results. Cylinder pressure was measured at various operation points in the laboratory, Figure 1. The comparisons of the simulated and experimental cylinder pressure for the operation points of 1400 rpm and mean effective pressure of $p_e=7$ bar and 1800 rpm and $p_e=7$ bar are shown in Figures 3 and 4. It can be seen that the agreement of the simulation and experimental pressure is very good. As for emissions, the simulations show a concentration of 550 ppm for the 1400 rpm validation case and 225 ppm for the 1800 rpm case. The experimentally measured concentration of NOx in the exhaust gases for the presented validation cases is 1080 and 900 ppm, respectively. This means that the order of magnitude of NOx emissions is well captured by the simulation, but the values are notably underpredicted. This qualifies the engine to be Euro 3 compliant (about 5 g/kWh). Soot emissions are measured with the Bosch Smoke Meter, in the unit Bosch Smoke Number (BSN). The experimentally obtained values range around BSN 0.4 to 0.6 for the reported cases.

**Figure 3. Comparison of simulated and experimental cylinder pressure for the load case 1400 rpm and $p_e=7$ bar**

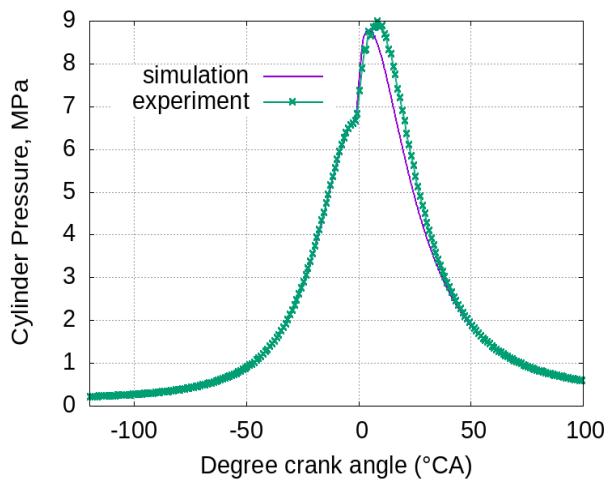


Figure 4. Comparison of simulated and experimental cylinder pressure for the load case 1800 rpm and $p_e=7$ bar

However, BSN cannot be converted exactly to soot concentration in g/m^3 obtained by the simulations because they measure different quantities: BSN is an optical measure of the blackening of a filter paper after a fixed volume of exhaust is drawn through it, while g/m^3 is a mass concentration of particulate matter in the exhaust gas. Hence, in the further analysis, only pollutant concentration trends will be observed for the different simulation cases.

Mesh sensitivity

CFD simulation results are often sensitive to mesh density or basic cell size. Hence, a mesh sensitivity test is usually performed to select a correct mesh density. In this analysis for the preliminary and validation cases, a base grid size of 3.2 mm was used. This is pretty coarse, but Converge CFD software allows for embedding in the region of the injector and adaptive mesh refinement (AMR), which is a functionality that divides the cells in regions where high gradients are met. This strategy allows us to run the basic simulation with a coarser mesh, which requires lower computer resources and less computing time, while the mesh is made finer just at the moment and in the area where it is needed. The mesh sensitivity simulations are based on the regime of 1400 rpm and mean effective pressure of $p_e=7$ bar. Besides the 3.2 mm grid size, a case with a finer (2.8 mm) and a case with a coarser (3.6 mm) grid size are run. The results for various grid sizes are shown in Figures 5 to 8. It can be seen that a finer mesh led to a negligibly lower maximum cylinder pressure and temperature. The influence on NOx formation is more important, Figure 7. The finer mesh results in NOx concentration of 500 ppm, the standard 3.2 mm cell mesh results in 550 ppm, and the coarser mesh results in 580 ppm. The influence of grid size on soot formation is negligible, figure 8. In conclusion, the influence of grid size on the main results is very low. From Figure 7 it results that a coarser mesh even improves the precision of NOx prediction by pushing it towards the experimental 1080 ppm. However, the standard 3.2 mm grid size mesh is already relatively coarse. Therefore, the further simulations are performed with the same 3.2 mm grid size mesh.

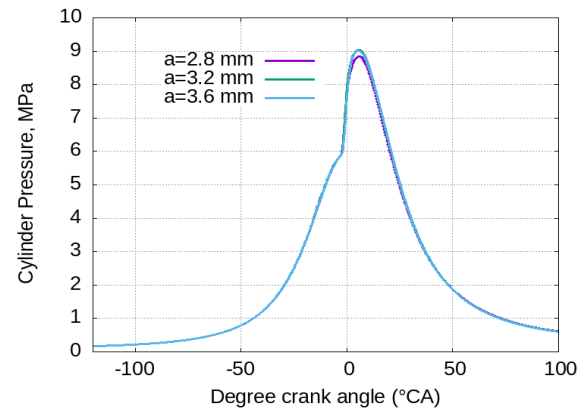


Figure 5. Comparison of simulated cylinder pressure for different grid sizes

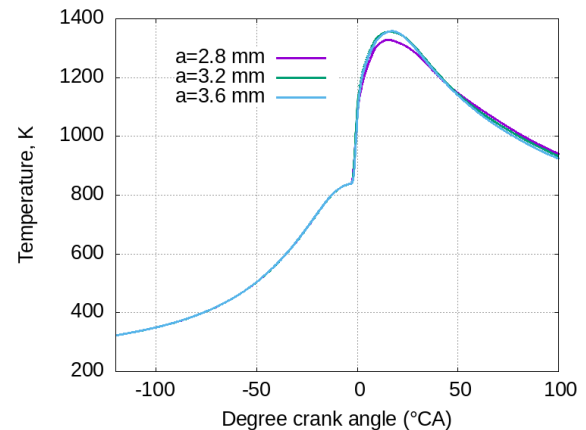


Figure 6. Comparison of simulated cylinder temperature for different grid sizes

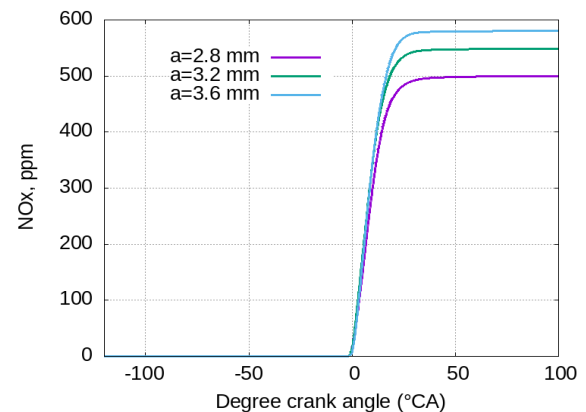


Figure 7. Comparison of simulated NOx concentration for different grid sizes

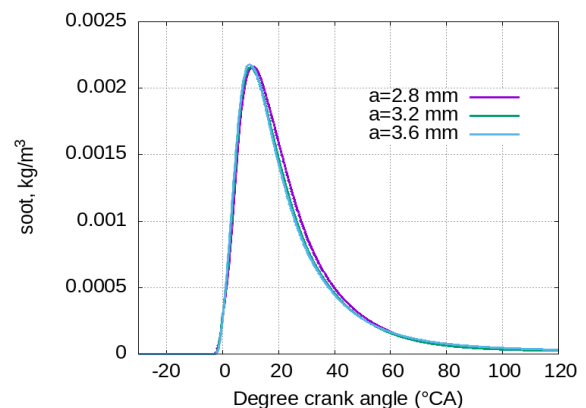


Figure 8. Comparison of simulated soot concentration for different grid sizes

Boost pressure and intercooling influence

The goal of the present work is to obtain higher power, without severe emission deterioration. Hence, an intermediate rpm regime is simulated, but with a load on the upper level of the engine torque. For the base case, the 1600 rpm and 15 bar mean effective pressure was selected. Experimental measurements for this regime were not available, so the model validation was performed on other regimes, as it was mentioned in the previous sections. For this regime, the boost pressure (after the charger) was 1.2 bar, which yields an absolute pressure at the beginning of engine compression of 2.2 bar and a boost air temperature of 403 K. A further case with increased boost pressure to 2.8 bar was simulated. Boost air temperature accordingly increased to 431.7 K. In order to take advantage of the increased air mass trapped in the cylinder, an increased fuel mass was injected. A final simulation case featured the same boost pressure of 2.8 bar and enhanced boost air cooling. The air temperature dropped from 431.7 to 333 K at the beginning of the compression phase. The varied initial conditions are summarized in Table 2.

Table 2. Initial conditions of the simulated cases

	p_{in} (bar)	T_{in} (K)	m_{fuel} (kg)
base	2.2	403	$1.272 \cdot 10^{-5}$
boost	2.8	431.7	$1.513 \cdot 10^{-5}$
boost+interc.	2.8	333	$1.513 \cdot 10^{-5}$

The influence of different boost and intercooling configurations is presented in Figures 9 to 14. Heat release rate is presented in Figure 9. The case with increased boost pressure (boost) shows an increased rate of combustion due to higher fuel quantity for the same injection period. The case with boost and intercooler shows an increased combustion delay and an evident premixed combustion peak. The resulting rate is the highest among all cases and focused around TDC. Figure 10 shows cylinder pressure. The most noticeable effect is the increase in peak cylinder pressure: approximately 11.5 MPa for the base case, 14.6 MPa for the boost case, and 16.3 MPa for the boost + intercooler case. This trend is expected because increasing the boost pressure increases the mass of air trapped in the cylinder. Intercooling further increases air density by reducing the intake temperature, allowing even more air to enter the cylinder. The additional oxygen promotes more rapid combustion and a higher heat-release rate, resulting in the highest peak pressure. The peak pressure occurs approximately 8–12° after TDC, which is typical for a direct-injection diesel engine operating near maximum torque. This suggests that the injection timing and combustion phasing remain appropriate despite the increased boost. Figure 11 shows the temperature trend. The temperature graphs run approximately parallel for all three cases, with the highest temperature for the boost case and the lowest for the boost+intercooler case. Temperature is the main factor for NOx production, which is reflected in Figure 12, which presents NOx emissions. It can be seen that NOx production starts shortly after the start of combustion, when high-temperature zones develop around the flame. After a certain

amount of piston expansion and after the end of intensive combustion, the temperatures drop under a certain threshold and NOx production ends. It can be seen that higher temperatures for the boost case give origin to an increase in NOx emissions of approximately 30% in comparison to the base case (1669 ppm versus 1265 ppm). Further, the application of an intercooling system can reduce NOx emissions by 50% (829 ppm for the boost+intercooler versus 1669 ppm of the boost case). Figure 13 shows the influence on soot emissions. The boost case gives the highest soot level due to greater fuel injected, but towards the end of combustion most of the soot is oxidized, and the final soot concentration in the cylinder is equal to the base case.

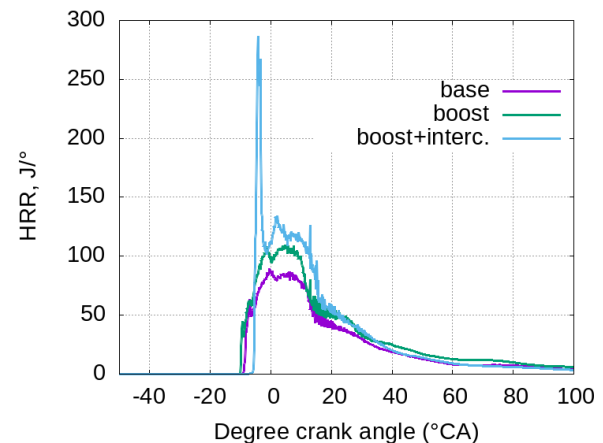


Figure 9. Comparison of simulated heat release rate

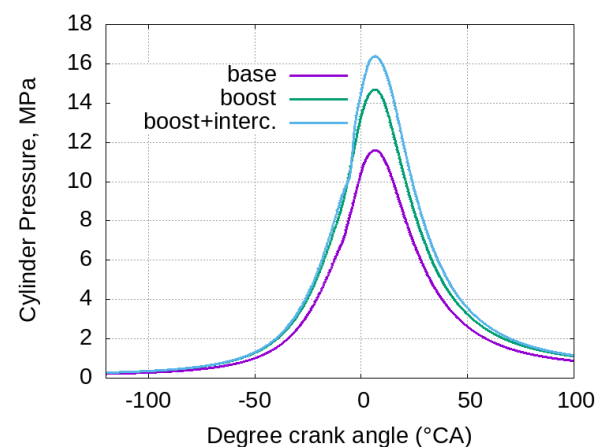


Figure 10. Comparison of simulated cylinder pressure

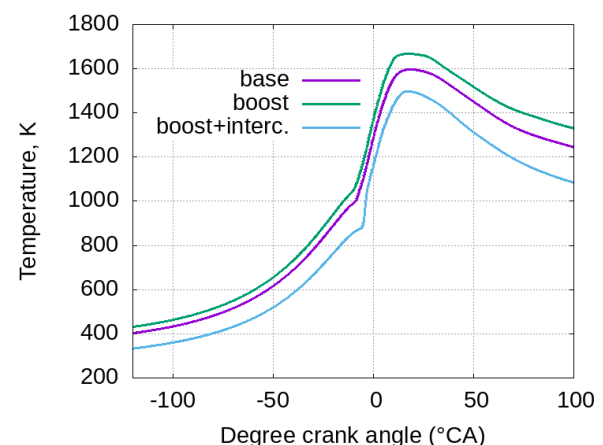


Figure 11. Comparison of simulated cylinder temperature

On the other hand, the boost+intercooler case results in slower soot pro-duction due to lower temperatures and also intensive oxidation. Therefore, the final soot concentration is approximately 25% lower for the boost+interc. case than for the other cases. Figure 14 shows the trend of hydrocarbon concentration in the cylinder. The highest values (final hydrocarbon concentration) are recorded for the base case (189 ppm), while the lowest for the boost+interc. case (50 ppm).

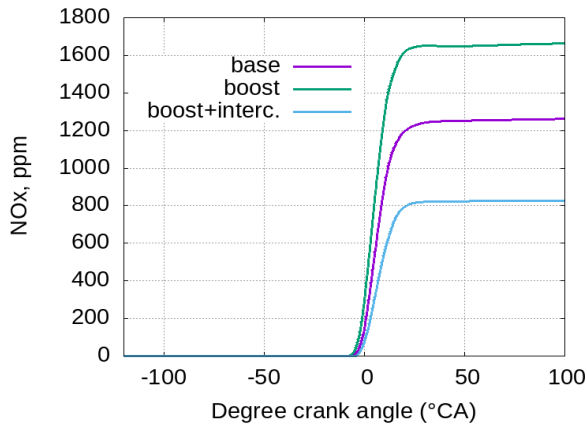


Figure 12. Comparison of simulated NOx concentration

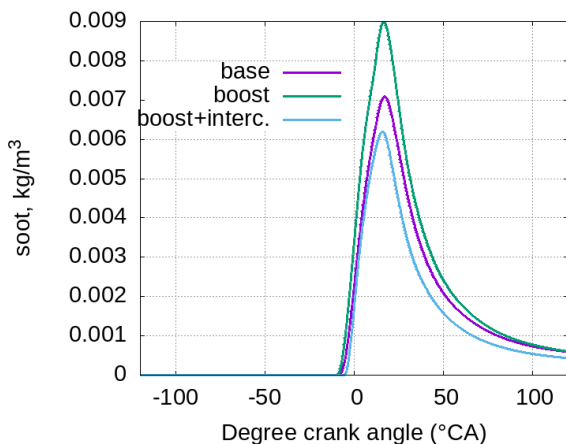


Figure 13. Comparison of simulated soot concentration

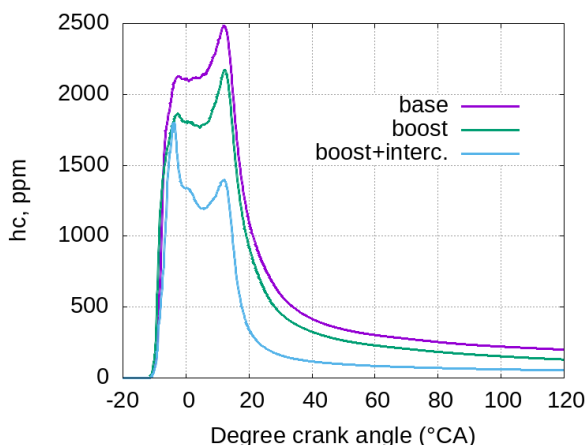


Figure 14. Comparison of simulated hydrocarbon concentration

3.4 Influence of exhaust gas recirculation

The following simulated technique is exhaust gas recirculation. It is used primarily for NOx reduction by

lowering cylinder temperature and especially the concentration of nitrogen in the combustion area. The species concentration set at the beginning of comp-ression is presented in Table 3.

Table 3. Species concentrations for the simulated levels of exhaust gas recirculation

	0% EGR	15% EGR	30% EGR
O ₂	0.232	0.21055	0.19278
N ₂	0.768	0.76355	0.75986
H ₂ O	0	0.00783	0.01431
CO ₂	0	0.01808	0.03305

The influence of EGR on the combustion and emissions is presented in Figures 15 to 20. The increase of EGR level reduces slightly the heat release rate, while the ignition delay is increased, Figure 15. It is reflected in a small decrease of maximal cylinder pressure, Figure 16. This will be reflected as decreased power and effi-ciency. The maximum pressure is reduced from 16.3 to 15.7 MPa for 30% EGR, which is a small and accep-table margin. A similar trend is repeated for cylinder temperatures, Figure 17. Maximal cylinder temperature is reduced from 1500 to 1420 K. The real benefit of EGR can be clearly seen in Figure 18 that shows the influence on NOx concentration. The production of NOx is reduced drastically for the 30% EGR level. The final concentrations are 829, 340, and 103 ppm, res-pectively, for the cases of 0% EGR, 15% EGR and 30% EGR, which represents a reduction of 87%. Figure 19 shows the influence on soot concentration. The cases with more EGR show a smaller rate of soot production, so the maximum level is lower. However, also the soot oxidation process is dampened, so the final concen-trations are somewhat higher for higher EGR levels. For 30% EGR the concentration is $5.19 \cdot 10^{-4}$ kg/m³, while for the 0% EGR it was $3.85 \cdot 10^{-4}$ kg/m³, which is an increase of about 35%. A similar effect can be seen for hydrocarbon emissions, Figure 20. A higher hydro-carbon concentration is found during the whole com-bustion period. It can be explained by the lower concen-tration of oxygen, which slows down hydrocarbon consumption. At the end of the process, the concen-trations are as follows: 50 ppm, 70 ppm, and 85 ppm for the cases 0, 15, and 30% EGR.

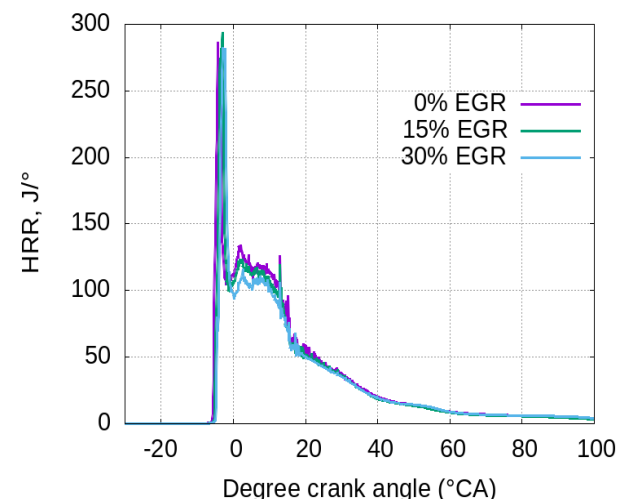


Figure 15. Influence of EGR level on heat release rate

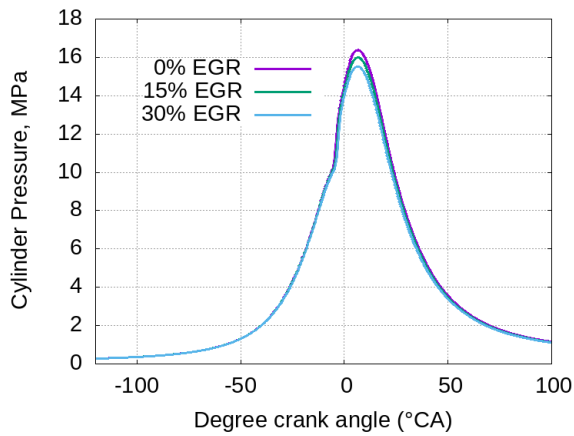


Figure 16. Influence of EGR level on cylinder pressure

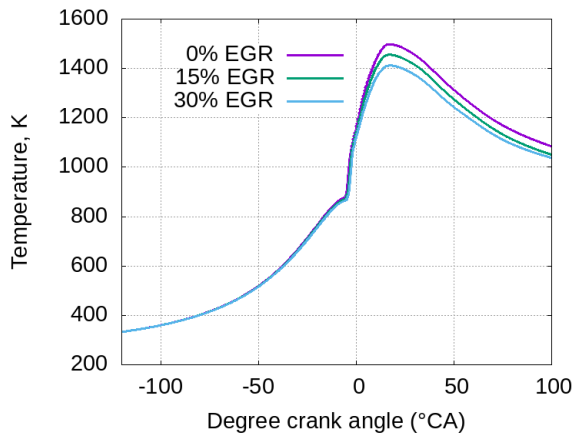


Figure 17. Influence of EGR level on cylinder temperature

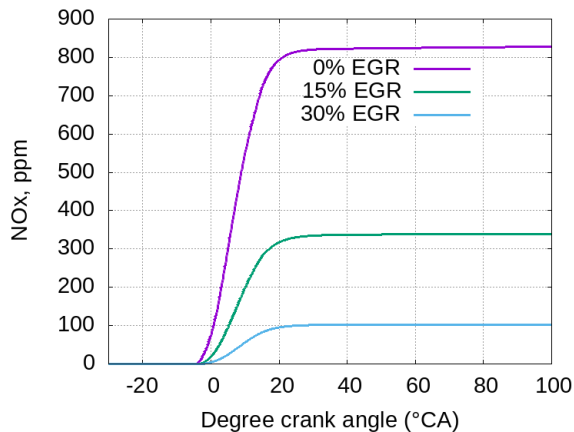


Figure 18. Influence of EGR level on NOx concentration

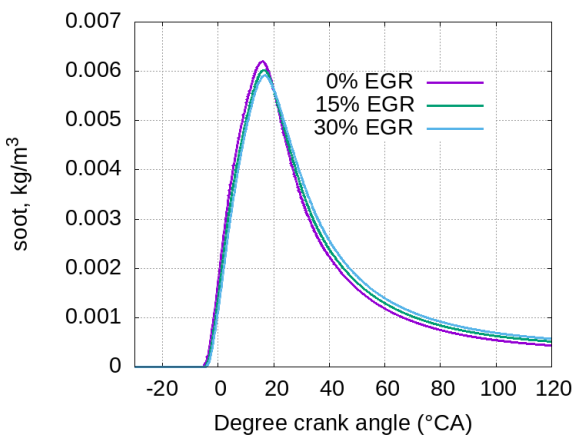


Figure 19. Influence of EGR level on soot concentration

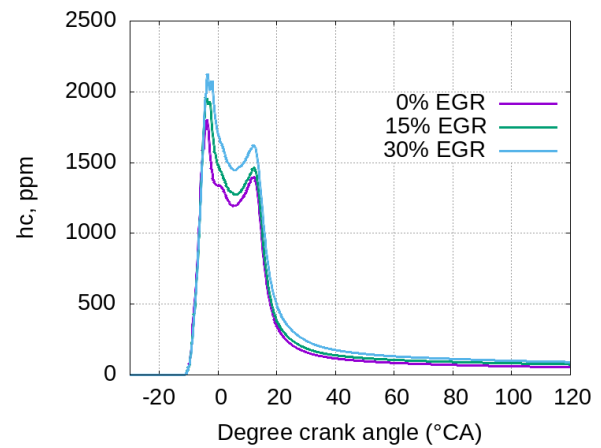


Figure 20. Influence of EGR level on hydrocarbon concentration

Influence of the piston bowl profile

In the final simulations group, different piston bowl shapes were simulated to check their influence on engine performance and emissions. In the literature, various shapes can be found [18-24]. Here, beside the original, deep "Mexican hat" piston bowl, a shallow and wider piston bowl was simulated. Additionally, a re-entrant type of piston bowl is simulated. In Figure 21 the outlines of the mentioned and simulated piston bowls are presented.

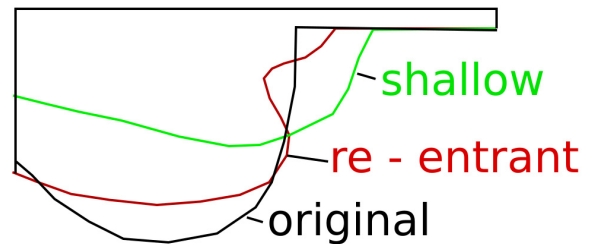


Figure 21. Outline of the piston bowl profiles for the simulated geometries

In Figure 22, the influence of piston bowl shape on heat release rate can be seen. The original deep bowl gives origin to a high premixed combustion peak and early and intensive diffusion combustion. The shallow bowl results in a smaller premixed combustion peak and somewhat later and less intensive diffusion combustion. The re-entrant type of bowl results in an intermediate premixed peak with a less intensive diffusion combustion that decreases more gradually than for other bowl shapes. In Figure 23, the influence of piston bowl shape on cylinder pressure is presented. It can be seen that early and intensive combustion around TDC seen in the previous figure results in higher peak cylinder pressure for the original piston bowl shape. This could result in a power and efficiency advantage, but sometimes late-stage heat release can affect the expansion stroke more directly than the pressure peak. In Figure 24, cylinder temperature is presented. The combustion dynamic from Figure 22 is reflected here. The highest temperature is found for the original bowl shape. The peak is found early, after which the temperature drops fast. For the shallow piston bowl, the temperature peak is less evident, and the temperature

remains high for a longer period. For the re-entrant bowl type, the maximal temperature is lower, and the temperature curve is more constant without a bump. Figure 25 shows how it is reflected on NOx emissions. The already reduced NOx emissions are further reduced for the re-entrant and shallow piston bowl type by more than 20%. For the re-entrant bowl type, NOx concentration drops to 80 ppm and to 75 ppm for the shallow bowl. Soot concentration is presented in Figure 26. For the modified piston bowl profiles, the soot concentration rises sharply and reaches about 25% higher maximal values than for the original profile. But after that, soot oxidation is also more intensive for these cases, and final soot concentrations are lower in comparison to the original bowl. Soot concentrations at the end of the expansion are $5.19 \cdot 10^{-4} \text{ kg/m}^3$, $3.65 \cdot 10^{-4} \text{ kg/m}^3$ and $3.35 \cdot 10^{-4} \text{ kg/m}^3$, respectively, for the original, shallow, and re-entrant piston bowl profile. Similar behavior is seen for hydrocarbon concentration, Figure 27. Initially the faster combustion for the original profile does not allow for hydrocarbon concentration to rise in the cylinder, and the alternative bowl profiles result in higher hydrocarbon concentrations until about 60 °CA. After that moment the combustion of the remaining hydrocarbons proceeds for the alternative bowl profiles, and the concentration decreases, while it remains almost constant (and higher) for the original profile. The final hydrocarbon concentration is 33 ppm for the re-entrant bowl versus the 85 ppm of the original profile, which is a reduction of more than 60%.

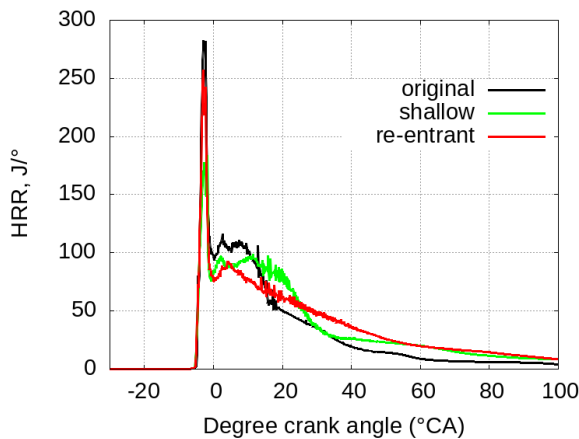


Figure 22. Influence of piston bowl shape on heat release rate

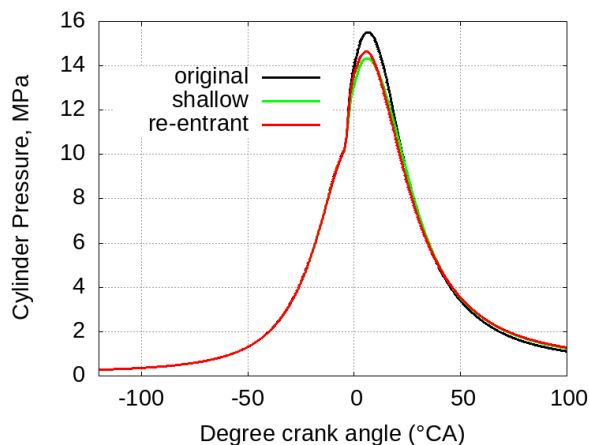


Figure 23. Influence of piston bowl shape on cylinder pressure

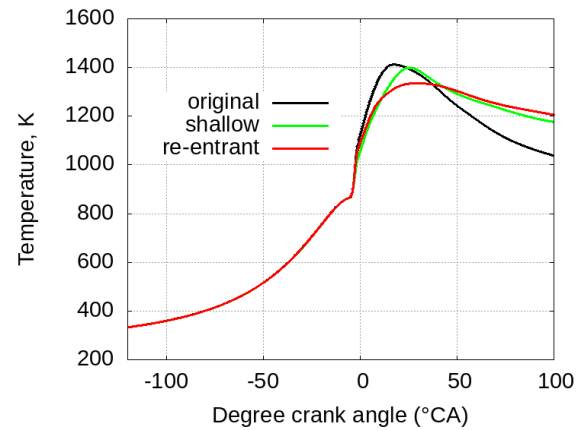


Figure 24. Influence of piston bowl shape on cylinder temperature

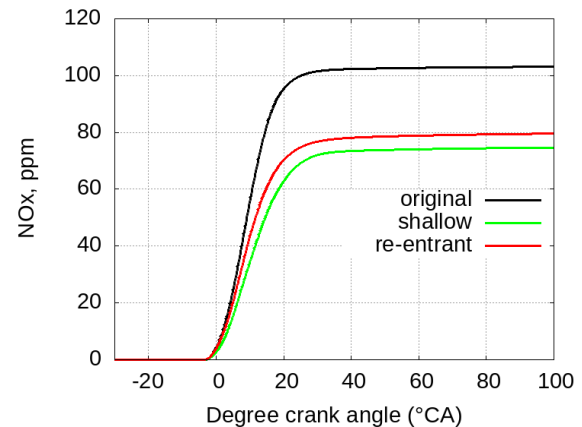


Figure 25. Influence of piston bowl shape on NOx concentration

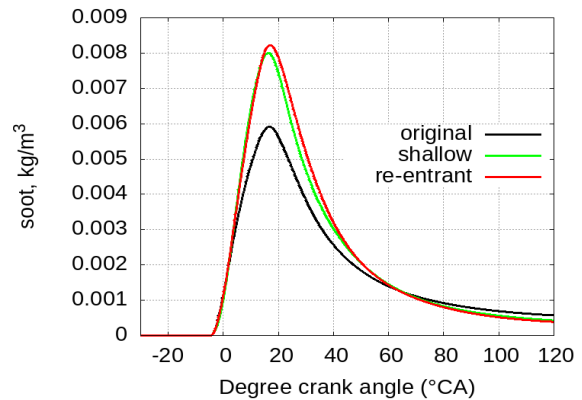


Figure 26. Influence of piston bowl shape on soot concentration

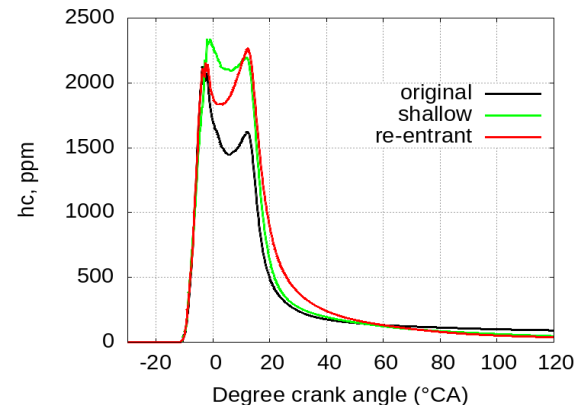


Figure 27. Influence of piston bowl shape on hydrocarbon concentration

The reasons for the results in Figures 22 to 27 can be explained from Figures 28 and 29 that show the spatial evolution of temperature and air excess ratio for the three different bowl profiles. The results clearly demonstrate that the bowl profile has a significant influence on flame development, mixing, and the spatial distribution of the combustion temperature. Figure 28 presents the temperature distribution at crank angle 20°.

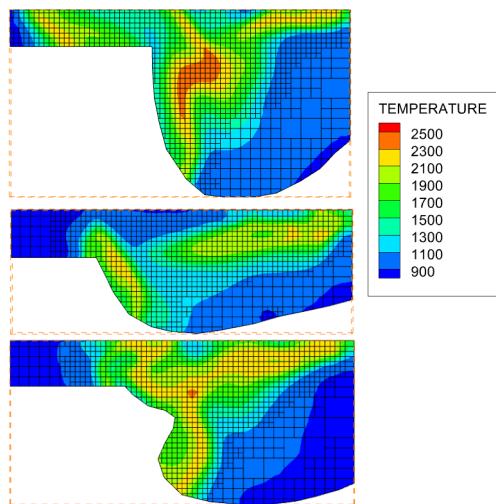


Figure 28. Temperature distribution for the moment of 20 °CA for the original, shallow and re-entrant bowl shape

The original geometry (top) exhibits a highly localized high-temperature region, with peak temperatures approaching 2500 K from 10 to 20 °CA. The flame is concentrated near the bowl lip and upper sidewall, producing a relatively small but intense combustion zone. Such localized temperature peaks may promote increased thermal NO_x formation, while the limited spatial extent of the high-temperature region suggests less uniform utilization of the available air. The shallow geometry (middle) produces a more evenly distributed temperature field throughout the combustion chamber. Peak temperatures are lower than in the upper case, and the combustion products occupy a larger fraction of the bowl volume. This indicates improved air-fuel mixing and a more gradual heat release. The absence of pronounced hot spots suggests the potential for lower NO_x emissions while maintaining efficient combustion. The re-entrant geometry (bottom) generates the largest high-temperature region, extending over much of the bowl and into the squish region. Although localized temperature peaks are less pronounced than in the original case, the elevated temperature occupies a considerably larger volume, indicating enhanced entrainment of surrounding air into the fuel spray. This behavior is characteristic of stronger recirculation generated by the re-entrant bowl geometry. Such improved mixing is generally associated with more complete combustion and reduced soot formation. These results confirm that piston bowl geometry plays a crucial role in controlling combustion development and the resulting emission characteristics by influencing both the intensity and spatial distribution of the heat release.

Figure 29 compares the distribution of the air excess ratio (λ) for the three piston bowl geometries. Significant differences in mixture formation are observed,

demonstrating the influence of piston bowl design on in-cylinder air motion and spray-air interaction. The original bowl profile (top) produces a compact low-lambda region surrounded by extensive lean zones, indicating limited fuel-air mixing. The shallow geometry (middle) generates a more uniform distribution of intermediate lambda values, reflecting improved air entrainment and mixture homogenization. The re-entrant bowl (bottom) exhibits the broadest region of combustible mixtures, suggesting enhanced turbulence and stronger recirculation within the combustion chamber. The lambda distributions closely correspond to the temperature fields, confirming that improved air-fuel mixing leads to a more spatially distributed combustion process. Consequently, piston bowl geometry plays a decisive role in determining combustion efficiency and the balance between soot and NO_x formation.

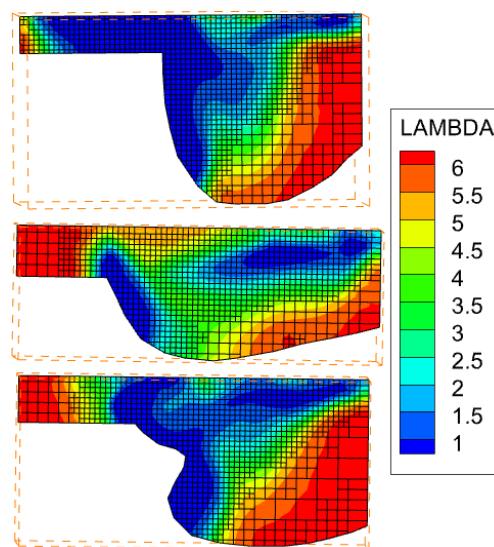


Figure 29. Air excess ratio for the moment of 20 °CA for the original, shallow and re-entrant bowl shape

4. CONCLUSION

A series of CFD simulations was performed to evaluate the possibility of modernization of older generations of industrial diesel engines. The simulation results were validated against experimental measurements for different load conditions, and the agreement was very good as far as cylinder pressure is concerned. As for emissions, NO_x, soot, and unburned hydrocarbons were evaluated. The validation cases showed that the order of magnitude for NO_x concentration was captured, while soot emissions were measured in BSN, for which there is no direct conversion factor to the concentration obtained by the simulations. There were no measurements for unburned hydrocarbons available. It was concluded that trends in emissions can be trusted, while simulations are not sufficient to get precise absolute values.

The increase in air boost pressure, with the appropriate fuel quantity increase, results in a substantial power increase but also emissions levels increase. The enhancement of boost air cooling results in further power increase and also emissions decrease. Exhaust gas recirculation is another strategy that allowed substantial NO_x emissions reduction for a small cylinder pressure penalty. However, soot and unburned hydro-

carbon levels were also increased. Finally, different profiles of the piston bowl were tested. Instead of the original, narrow and deep "Mexican hat" bowl, a wider "shallow" profile and a "re-entrant type" of bowl were tested. It is proved that the alternative bowl shapes result in a more intensive air entrainment into the fuel jets and a less intensive combustion that spreads over a wider volume. This results in somewhat lower peak cylinder pressures and temperatures and especially lower local temperatures. This leads to lower NO_x and soot emissions. The influence of the bowl shape on combustion characteristics is a part that should be further analyzed in the following research.

To summarize, there is a substantial margin to increase the power and reduce emissions on commercial engines of older generations. For the original simulation, it yielded 1265 ppm, and after boost, intercooling, 30% EGR, and a different bowl profile, it was reduced to 75 ppm before the exhaust. It means that NO_x emissions can be reduced with the combinations of different strategies for a whole order of magnitude.

For further work the influence of some further parameters could be investigated, like injection timing, including split injection patterns, injection pressure, and spray orientation. Further, today alternative fuels and their blends get increasing attention. Some of the alternative fuels are natural gas, biodiesel, dimethyl ether, methanol, ammonia, and hydrogen. Then, there are also some very efficient aftertreatment methods. The combination of all these strategies can allow the diesel engine easier compliance with environmental standards.

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CFD АНАЛИЗА СТРАТЕГИЈА УСМЕРЕНИХ НА МОДЕРНИЗАЦИЈУ ИНДУСТРИЈСКОГ ДИЗЕЛ-МОТОРА

Т. Сенчић, П. Блечић, В. Мрзљак, О. Буковац

Дизел-мотори заузимају значајан удео у комерцијалном сектору. Упркос еколошким трендовима, прелазак на нове погонске системе биће постепен. Стога је рационално учинити постојеће моторе што снажнијим, ефикаснијим и еколошки прихватљивијим.

У главном делу рада спроведен је низ симулација рада дизел-мотора, при чему су разматрани повећање притиска пуњења и хлађење ваздуха за сагоревање. Такође је симулирана примена рецикулације издувних гасова, као и модификација профила чела клипа. Анализирано је како наведене измене утичу на емисију штетних гасова.

Забележен је пораст притиска у цилиндру. Емисија азотних оксида значајно је смањена применом рецикулације издувних гасова, иако је ова стратегија негативно утицала на емисију чађи и угљоводоника. Коначно, потврђено је да геометрија чела клипа може имати повољан утицај на смањење емисије штетних гасова. Анализа показује да се постојећи мотори могу унапредити уз релативно ограничене интервенције.