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Impact of Fuel Properties on the Performance of a Direct Injection Diesel Engine under Part Homogeneous Operating Conditions

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ABSTRACT

Tightening of emission norms necessitate intensified research in the field of emissions reduction. Fuel research opens up a vast area of potential improvement, since combustion behavior and the nature of the combustion products can be heavily influenced by fuel composition. In this paper, the effects of fuel properties on combustion and emissions shall be discussed, based on the study of standard diesel fuel, two types of diesel-like fuels and a kerosene fuel. Investigations were conducted on a single cylinder heavy duty direct-injected diesel engine operating under part-homogeneous combustion in the part-load operating range. For this purpose, a statistical design of experiments method (DOE) was utilized in order to evaluate the influence of each fuel property and, thus, develop a model for all selected fuels. Variation in EGR rates, injection and air patterns have significant effects on the combustion in the fuels under investigation. Therefore, common DOE plans with the same engine DOE parameters and ranges have been considered for all investigated fuels. On the other hand, the centroid of combustion was maintained constant for each operating point for all the fuels, to have the same evaluation basis. This investigation contains the experimental results obtained at the test cell, followed up with heat release calculations, to analyze combustion rates. Based on these investigations, the impact of the different fuels on the efficiency and raw engine emissions shall be discussed. Results show the potential of each fuel, based on its physical and chemical properties. Kerosene, with its high volatility and zero aromatic content appears desirable for application in heavy duty diesel engines. Further, part homogeneous combustion offers a possibility to reduce the amount of exhaust after-treatment.

INTRODUCTION

Factors like global warming, increasing pollutant gases pumped into the environment from engine exhaust emissions, rising petroleum prices and the need to conserve petroleum sources have created interest in the research of alternative combustion and fuels for engines.

Emission regulations and fuel economy targets are being stringently revised, amid growing concern about air pollution and limited fuel resources in worldwide. Through the enforcement of stringent emission regulations, research institutes and industries are encouraged to find better ways to limit emissions and reduce fuel consumption in vehicles, while ensuring continued mobility of passengers and goods. Especially in the heavy duty sector, the new generation of vehicles is equipped with accurate control strategies and advanced hardware for better combustion, as well as after-treatment systems. These systems involve not only additional technical complexity and financial expense, but they also demand monitoring and precise control strategies. On the other hand, new combustion concepts have received considerable attention in the recent past because of their potential for reducing engine-out emissions from diesel engines. These concepts mainly concentrate on alternative combustion strategies, like homogeneous combustion or HCCI in diesel engines. The HCCI concept has been studied by many researchers, also for different fuel categories [1, 2, 3, 4, 5]. In the HCCI concept the proportion of premixed combustion has to be extended by preparation of a well-mixed combustible mixture ready for combustion. This can be realized by different methodologies, based on the type of engine, load and speed. The combustion is mostly controlled by the ignition quality and chemical behavior of the combustion mixture, which are governed largely by the fuel composition and properties. Although HCCI engines offer substantial benefits in terms of providing high efficiency with low NOx and particulate emissions due to the lower combustion temperatures, control of auto ignition timing and extending the high-load limit can be challenging [6].

There are two concerns in HCCI operation; one of which is the definition of the injection pattern. This includes setting up of the number of injections, period of injection and the separation between the injection events. Besides, the shape of the injection profile also has an influence on HCCI combustion.

Another concern is the air path control, where the introduction of EGR (Exhaust Gas Recirculation) plays a role in controlling the ignition behavior. Using EGR in engines also affects thermal efficiency and emissions. Changes in thermal efficiency with EGR can result from changes in both combustion efficiency as well as thermodynamic efficiency. Through the intensive reduction of NOx in the HCCI concept, it is possible to eliminate a NOx after-treatment system, which is one of the main targets of the current work. Investigations on the HCCI concept included testing of different fuels to determine the best compromise between the part homogeneous combustion concept and fuel properties. Part homogeneous combustion has a higher degree of pre-mixing than a conventionally placed single injection. This is achieved through multiple pilot injections, in combination with a main injection. The start of main injection determines the start of hot flame combustion.

Conventional combustion (heterogeneous) in diesel engines has been considered in this study as a reference. This was followed by a series of tests with diesel fuels and kerosene under part homogeneous combustion conditions. The influence of fuel composition and EGR on mixture formation, ignition and heat release in the heterogeneous and part homogeneous combustion processes is explained in this paper.

DESCRIPTION OF THE TEST FUELS

For the purpose of the investigations, three diesel fuels and one kerosene-type fuel were selected. Table 1 presents the important properties of the fuels studied.

Fuel Properties	Unit	Std-Diesel	Kerosene	Diesel A	Diesel B
Density @15 °C	kg/m^3	839.6	769.5	799.7	817.3
Viscosity @ 40°C	mm^2/s	2.7	1.067	2.468	3.117
Cetane number (CFR)		52.5	50.3	50.34	60.44
T10	°C	207.1	157.1	200.6	247.4
T50	°C	266.9	165.1	244	275
T95	°C	348.5	308.3	317.6	321.6
LHV	MJ/kg	42.9	43.87	43.98	43.49
Mono Aromatics	M %	22.3	0.0	2.7	4.5
Di Aromatics	M %	3.9	0.0	0.1	0.1
Poly Aromatics	mass %	4.5	0.0	0.1	0.1
Total Aromatic	mass %	26.8	0.0	2.8	4.5
Mol. weight	g/mol	185	155.4	202.07	220.81

Table 1: Fuel matrix

Diesel A has the same cetane number as Kerosene but a different molecule structure and molecular weight. All the diesel fuels have aromatics, with Diesel A and B having a low aromatic content. Kerosene is free of aromatic content. On the other hand the density, viscosity and molecular weights of these fuels are quite scattered. A few of the fuel properties shown in Table 1 are briefly explained below.

CETANE NUMBER

Cetane number is an ignition property of diesel fuels. A fuel with a high cetane number ignites shortly after it is injected into the cylinder; therefore, it has a short ignition delay period [7]. Conversely, a fuel with a low cetane number resists auto ignition and has a longer ignition delay period. A shorter ignition delay leads to a correspondingly lower amount of pre-mixed combustion, resulting in lowered NOx [7]. A high cetane number also reduces white smoke on startup [8]. In the current investigation, a variation of 10 units in cetane number exists between the fuels (refer Figure 1).

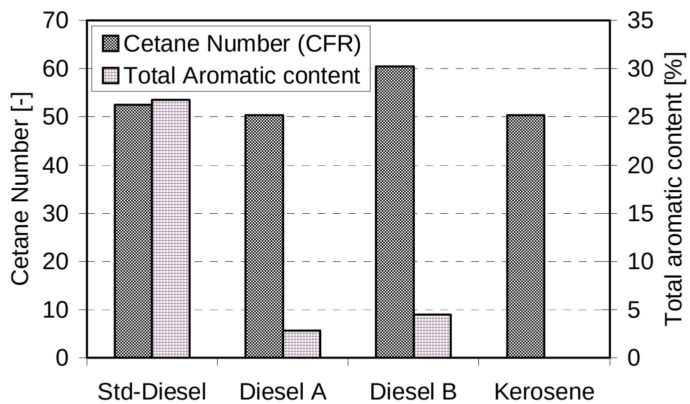

Figure 1: Cetane number and aromatic content of the fuels

AROMATICS

Aromatic hydrocarbons are HC compounds containing one or more “benzene-like” ring structures. They can be distinguished from paraffins and naphthenes, the other major HC constituents of diesel fuel, which lack such structures. The fuel variants have got a lower aromatic content, when compared to standard diesel (refer Figure 1).

DENSITY AND VISCOSITY

Density is a physical and intensive property of fuel. Changes in fuel density affect the energy content of the fuel brought into the engine, given a fixed injection setting. It is also of relevance to the mixture generation, and the extent to which the fuel is influenced by charge motion in the engine.

Viscosity is also a physical and intensive property of the diesel fuel and mainly has an impact on the fuel injection system performance. It affects fuel spray atomization, fuel system lubrication and leakage. On the other hand, an interrelation between viscosity and density exists as shown in Figure 2. In addition, it has an influence on boiling characteristic, thereby influencing engine performance too.

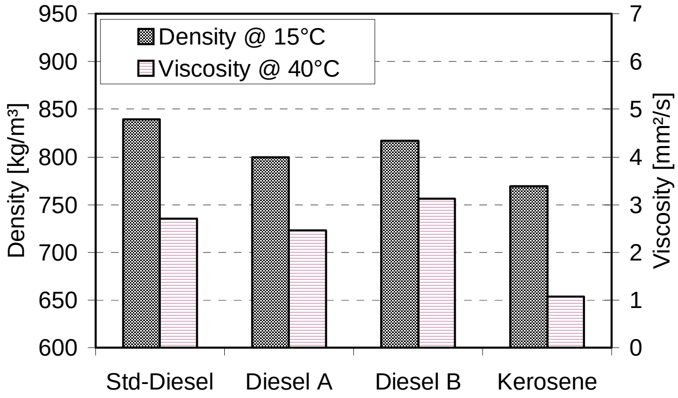

Figure 2: Density and viscosity

DISTILLATION

Diesel fuel consists of a mixture of hydrocarbons with different molecular weights and boiling points. As a result, as some of it boils away upon heating, the boiling point of the remainder increases. This fact is used to characterize the range of hydrocarbons in fuel in the form of a distillation curve, specifying the temperature at which 10%, 20%, etc. of the hydrocarbons have boiled away.

Figure 3 shows the distillation curves for these fuels. The curves of the diesel fuels are closely bunched, though Diesel B lies significantly higher up to the 50% boiling point. For Kerosene, it is a completely different behavior, with a very low distillation curve from 0 to 80 percent, distillation occurring at a relatively constant temperature. There is, then, a sudden increase of 140°C from 80 to 100% distillation.

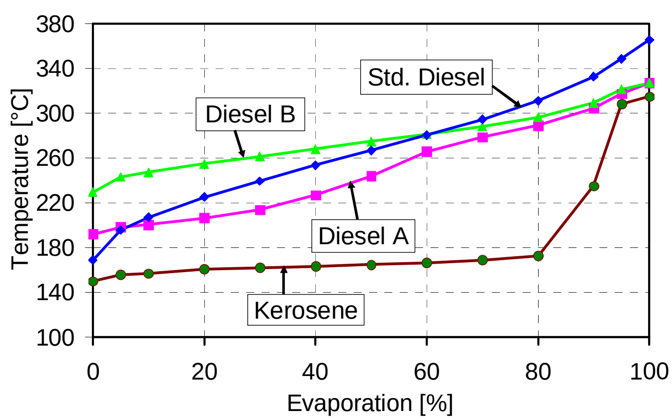

Figure 3: Distillation curves for selected fuels

Kerosene has a low density and viscosity, as also a lower molecular weight compared to others. The molecular weight shown in Figure 4 is determined by the distribution of molecular sizes of the individual fuel components. Such a distribution is also shown in Figure 5.

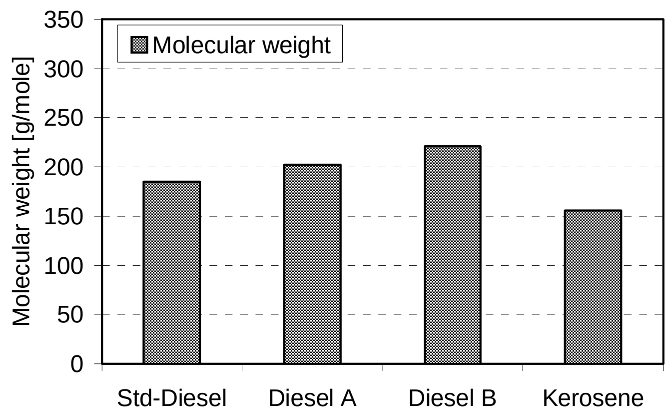

Figure 4: Molecular weight of different fuels

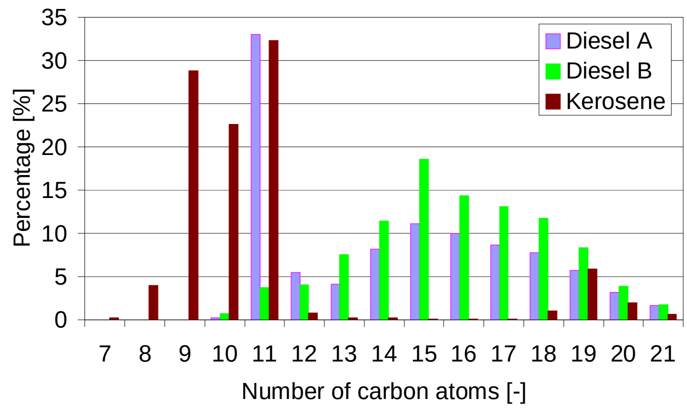

Figure 5: Molecular size distribution in the fuels

TEST BENCH APPARATUS

The experimental setup used for the investigations is a single cylinder heavy duty diesel engine. The specifications of the test engine are given in Table 2. The research engine, built at FEV Motorentechnik GmbH, was based on the MAN D2876. The single cylinder engine is representative of a typical cylinder displacement for truck applications. For the purpose of testing under part homogeneous conditions, the compression ratio was reduced from the standard 17.5 to 15.2 by changing the piston bowl geometry.

Bore	128 mm
Stroke	166 mm
Displacement	2.136 l
Compression ratio	15.2
Con rod length	256 mm
Number of valves	4
Boost system	External (3.6 bar)
Injection System	CRIN2 (2000 bar)
Nozzle hole number	6 holes
Nozzle cone angle	80°
Nozzle hole diameter	0.206 mm
Nozzle HFR	1400 cm^3/min

Table 2: Specifications of the single cylinder engine

The engine can be operated over the entire engine map and is equipped with mass balancing of the first and second orders to ensure relatively vibration-free operation. The boost pressure is obtained through an external charging system and can, hence, be set independent of engine operation. A peak boost pressure of 3.6 bar can be achieved, which is state-of-the-art for conventional truck applications. The boost temperature was held constant for each load point. The temperature for each load point was determined based on that of a typical turbocharger. It ranged from 33°C at low load and speed to 52°C at full load and high speed.

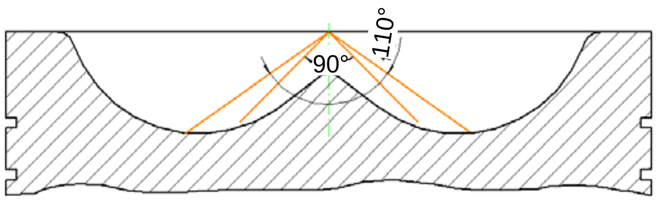

Figure 6: Piston bowl geometry

Since increased homogenization was realized through early pilot injections, the design of appropriate bowl geometry was necessary. Figure 6 shows the design of the piston bowl used for the investigations. The layout is such that a large air volume is available closer to the cylinder axis than in a typical w-bowl or omega-bowl. The cone in the centre of the bowl is short and steep to avoid interference with the spray jets, which could occur with narrow cone angles.

This bowl design, along with a narrow nozzle cone angle, ensures low interaction between fuel and the cylinder liner, upon application of early pilot injections. Also, with a narrow cone angle, the piston geometry plays a bigger role in mixture generation. Therefore, a 6-hole nozzle, with a cone angle of 80° was chosen. Although, a small hole diameter is beneficial for quick mixture generation, the hole diameter requires to be large enough for a good fuel consumption at full load. A hydraulic flow rate of 1400 cm^3/min for the nozzle was selected.

The rail pump, a Bosch CP4 pump, is driven by a separate electric motor, allowing free choice of rail pressure. Peak pressures of 2000 bar can be achieved. In order to reduce the pressure fluctuations in the fuel line, which typically result in single cylinder engine operation, a larger second rail is used, in addition to the regular rail that controls the rail pressure. This second rail acts as a high pressure surge tank. The injection system permits high flexibility, in that 6 injections per working cycle can be performed. The start and duration of each injection event can be selected to influence the extent of homogenization.

The engine load point is set by adjusting the injection duration of the main injection to achieve the required brake torque. The operation is monitored by temperature and pressure sensors. This data is captured at a frequency of 5 Hz and stored for further analysis. The placement of the sensors is shown in Figure 7. Further, engine performance data, such as speed, torque and fuel consumption are also monitored and stored with the same resolution.

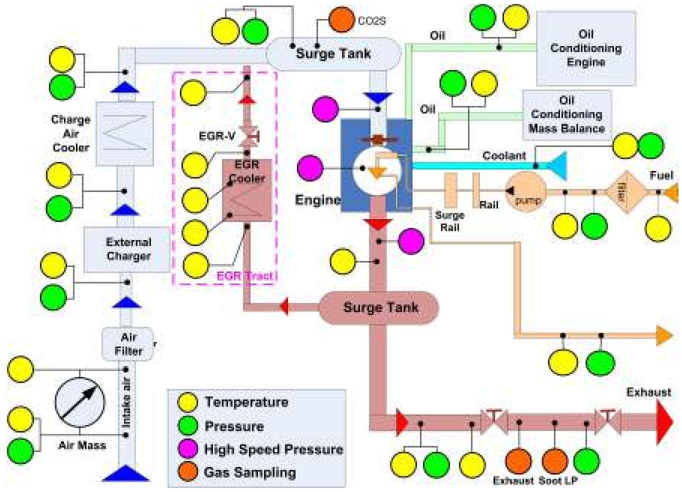

Figure 7: Test cell set-up [9]

High frequency data capture of cylinder pressure, as well as pressures in the intake and exhaust manifolds is used to acquire data with a resolution of up to 0.1 °CA. The indices based on indicated cylinder pressure can thus be determined. This data also forms a basis for heat release analysis of the measured operating point.

Untreated emissions are measured using a chemiluminescence detector for NOx emissions and a Flame Ionization Detector for hydrocarbon emissions. A heated AVL smoke meter (415s) is employed to determine the particulate matter emissions, based on the AVL correlation.

A closed loop control was realized to automatically set the desired EGR rate. The EGR rate was adjusted by varying the engine back pressure, thereby changing the pressure drop across the EGR valve. The increased back pressure also increases the internal EGR rate. In this paper, the influence of internal EGR is not analyzed since the same boundary conditions apply to all the fuels tested. A further control for setting the required CA50 was developed by application of an online computation of the burning function, based on which the start of main was adjusted. A burning function shows the fraction of total heat release up to any particular crank position.

DESIGN OF EXPERIMENTS (DOE)

Design of experiments, DOE, is nowadays used in many research and industrial sectors. For instance, it is applied in the development and optimization of calibration processes in diesel engines with a certain number of engine parameters as variable tools. By using DOE methods, it is possible to predict the engine response to a number of parameters or factors without having test results for every possible combination [10]. It is impossible to run a lot of combinations of the engine parameters at each operating point to find out the best compromise. In this case, through testing some combinations of parameters like EGR rate, at the test bench with a certain DOE plan, a desktop model can be generated. This engineering model can be used to predict the best trade offs. The advantage of the DOE method is that a defined mathematical model is available which enables the prediction of engine responses, due to change in one or more of the investigated parameters simultaneously. Of importance with such modeling is the quality of the model, which should be taken into account.

A typical DOE optimization path is shown in Figure 8. In this approach, the design space of the DOE to be performed is drawn up. The design space consists of the variation range of each parameter (dimension) to be varied. The multi-dimensional space can also be trimmed to limit the specific combination of parameters. Engine testing is done based on the test plan generated using the design space. Then, one part of the experimental data is utilized to build a DOE model, which is validated against experimental data. Once satisfactory models are set up, the optimization is done, for a given set of constraints and parameter weightings.

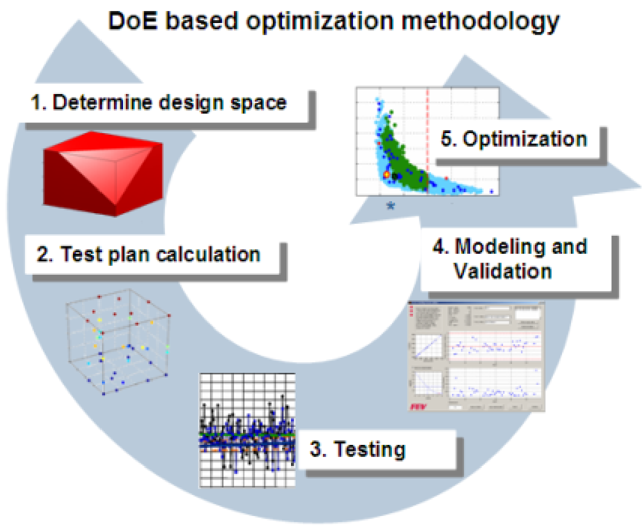

Figure 8: Biography of implementing Design of Experiments (DOE)

DOE plans for each operating point have been developed according to the engine response in that point, under homogeneous combustion conditions. No DOE modeling was done for the heterogeneous operating mode. Therefore, all results relating to heterogeneous operation in this work are measured data, while the homogeneous results are based on DOE models.

TEST PLANNING AND EXECUTION

In order to establish a transition from conventional combustion to part homogeneous combustion, the fuels were tested in both operating modes. The testing plan is shown in Figure 9. For both modes, the A (1220 rpm) and C (1780 rpm) speeds of the European Stationary Cycle were chosen. While the part homogeneous operation was limited to the 25% and 75% load points, testing in the heterogeneous mode also included full load operation. At the A speed, the maximum load was reduced from 75% to 70%, while at the C speed it was reduced from 75% to 60% for homogeneous investigations to ensure stable engine operation.

Multiple injections were applied, with varying injection strategies for the load points. Double pilot injections were employed, along with a conventionally placed main injection, for the 25% load points. At the A70 point, a single pilot and a post injection were applied with the main injection. At the C60 load point, a double pilot injection and a post injection were applied, in addition to the main injection. The injector activation duration of the pilot and post injections were varied for each load point. For example, the injector activation duration of the first pilot injection at the A25 load point was 200 μs, while that of the second pilot was 300 μs. This corresponded to an injection quantity of 3 mg/stroke and 12 mg/stroke respectively.

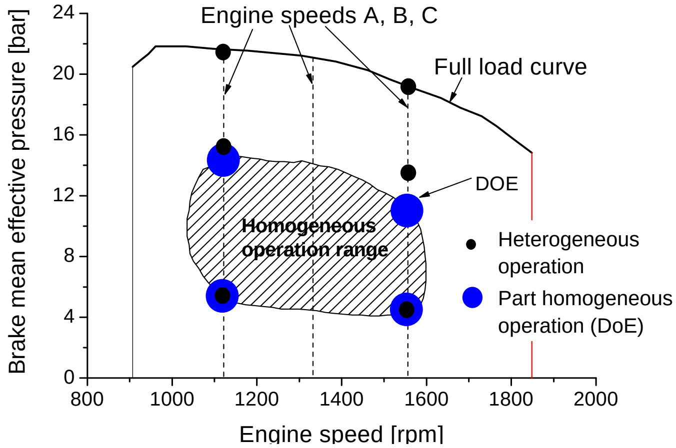

Figure 9: Engine operating points for fuel testing

Under heterogeneous operation, EGR variations were performed at part load points, with a constant CA50, i.e. simultaneous variation of EGR rate and start of injection. Full load testing was done by a variation of start of injection and a constant EGR rate.

Fuel testing at the part homogeneous operating mode was carried out using a DOE approach. A four dimensional test plan with space filling was developed, in which the following parameters were varied:

- Begin of pilot injection
- Rail pressure
- Boost pressure
- EGR rate

In cases where multiple pilot injections were applied, the start of injection of all the pilots was shifted together, keeping the electrical separation between them constant. This is seen in Figure 10, which represents the injection strategy for the C25 load point. The DOE tests were done with a constant CA50, which, as mentioned earlier, was maintained by adjusting the begin of main injection.

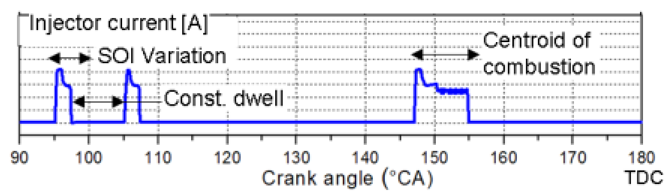

Figure 10: Injection strategy for the C25 point

RESULTS AND ANALYSIS

In this study, two factors were investigated, namely, fuel behavior and the combustion mode (heterogeneous or part homogeneous). In order to separate the influence of each factor, the fuels were first tested under heterogeneous combustion. Thereupon, the testing of the fuels was done under part homogeneous conditions.

HETEROGENOUS COMBUSTION

Figure 11 shows the NOx-PM and NOx-BSFC trade-offs of the fuels at the C100 load point. At first glance, it is clear that the fuels behave very similarly with regard to NOx-PM trade-off above a NOx emission of 3 g/kWh. Below this value, they begin to diverge, highlighting the relevance of fuel composition with regard to achieving future emission norms with minimal after-treatment effort. For achieving Euro-6 or Tier 4f NOx limits, a NOx value below 0.4 g/kWh is necessary, which requires optimization of the hardware configuration (nozzle parameters) and a higher boost pressure and EGR rate [11]. Amongst the points selected for comparison, the start of injection for all fuels except Diesel A was 7 °CA BTDC. Diesel A was, however, tested only up to 9 °CA BTDC. Thus, this point was chosen for comparison.

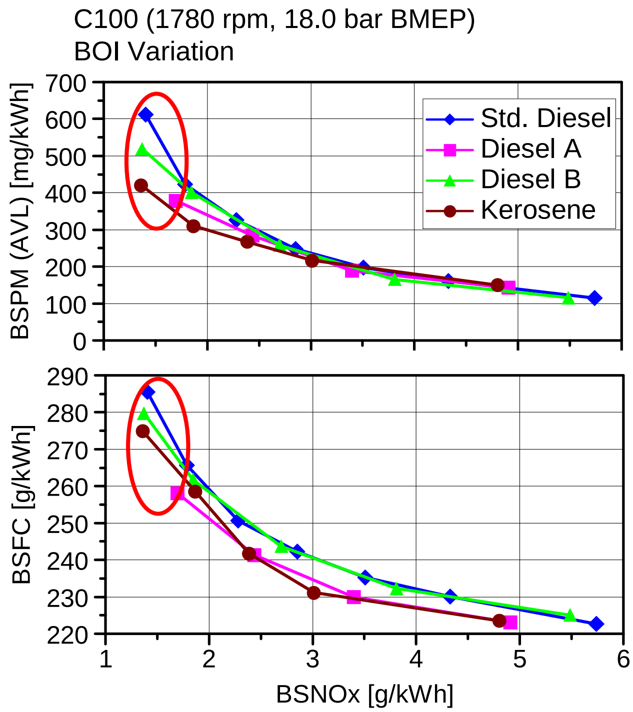

Figure 11: Performance of fuels at full load

The cetane number does not play a significant role at full load, however differences in ignition delay are seen. In Figure 12, the measurement points with least NOx emission for each fuel were analyzed for heat release.

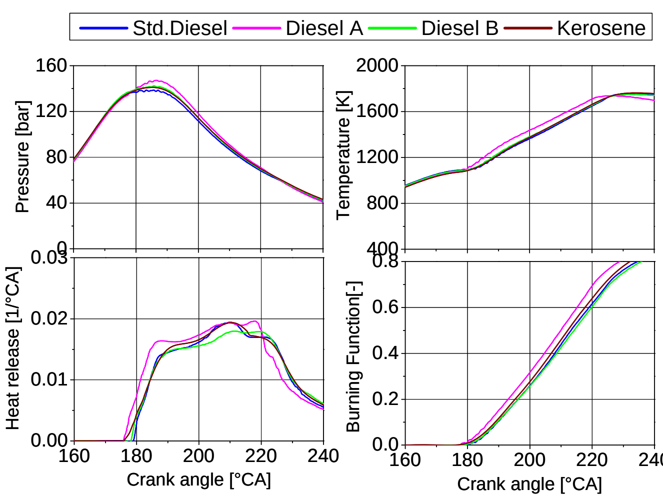

Figure 12: Heat release analysis of the full load point (C100) under heterogeneous operation

The Kerosene fuel had a significantly shorter ignition delay, which is attributable to its higher volatility and lower viscosity, enabling rapid generation of a combustible mixture. Kerosene, therefore, ignites at approximately the same time as Diesel A, which had a 2° CA advanced start of injection. The smaller rate of heat release at the start of Kerosene combustion also contributes to a lower NOx level, for a given start of injection (when compared to Std. Diesel and Diesel B). The combustion duration of Kerosene is the longest since the injection duration had to be adjusted to accommodate for its lower density. The higher temperatures in the latter stages of diffusion combustion aid the oxidation of soot. A 33% reduction in particulate matter, in comparison to standard diesel, is obtained with the Kerosene fuel. The long combustion duration begins to have a negative effect on BSFC below a NOx level of 3 g/kWh. In this range, the rise in BSFC is sharper for Kerosene than for the other fuels.

Amongst the diesel fuels, the emissions performance shows a correlation with the boiling points of the fuels. Standard diesel, with the highest PM value, also has the highest FBP. Also, it has a significant aromatic component, which further deteriorates the PM performance. Diesel B has a relatively high PM due its elevated distillation curve, though it has approximately the same FBP as Diesel A. The most volatile of the diesels, Diesel A, has the least PM emissions. A similar trend is observed in the BSFC.

PART HOMOGENEOUS COMBUSTION

The fuels were studied under part homogeneous conditions at both the A and C engine speeds. However, the change in engine load represents a more significant change in engine performance than a change in engine speed, particularly for heavy duty engines, which have a narrow speed range. Therefore, the discussion shall be restricted to the low load and high load points at the A engine speed, i.e. at the A25 and A70 load points of ESC.

Analysis at the A25 (1220 rpm, 5.2 bar BMEP) load point

The results of the DOE optimization of the fuels at the A25 load point are shown in Figure 13. The optimization was performed by limiting the NOx emissions to below 0.4 g/kWh (Euro-6 / Tier 4f NOx limit). The BSFC and PM emissions were minimized with weightings of 60% and 40% respectively. Since a DPF would in any case be utilized, considering the overall PM level (at all load points), a greater focus was given to reduction of BSFC. All the 3 variants have a higher BSFC (5-6%) than Std. Diesel. On the other hand, both Diesel B and Kerosene have excellent PM behavior, although the overall PM level for all fuels is such that a DPF would not be required at this load point.

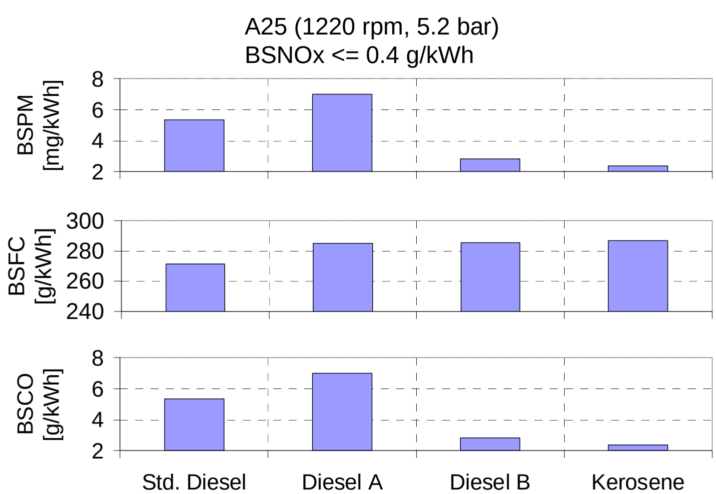

Figure 13: DOE optimized comparison of the A25 load point under part homogeneous operation

A heat release analysis of the fuels at this load point (Figure 14) shows that while standard diesel has an earlier ignition of the cool flame, the ignition of the hot flame occurs significantly after that of the other fuels. The fast combustion thereupon is reflected in the lower BSFC of Std. Diesel. The high aromatic content and their associated stability is the likely reason for the extended ignition delay at a low load point, like A25, where cylinder temperatures are correspondingly low. The terminal stages of combustion are significantly slower due to the very high FBP (refer Burning Function in Figure 14), resulting in incomplete combustion and high CO emissions.

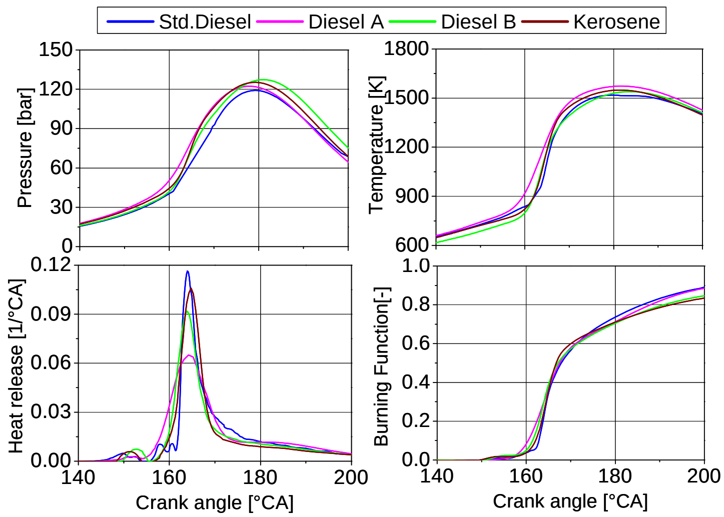

Figure 14: Heat release of the A25 load point under part homogeneous operation

Diesel A has the lowest boiling curve amongst the diesel fuels. Despite the aromatic content (evidently small), it has a significantly earlier ignition of the hot flame, followed by a lowered pre-mixed combustion peak. This produces a longer combustion (lower cylinder temperatures) which, along with the relatively cold cylinder conditions of the A25 load point, principally results in higher CO emissions.

Despite the higher cetane number, Diesel B displays a late cool flame combustion peak. This is, like with Std. Diesel, attributable to the aromatic content in the fuel. Although the boiling curve up to 60% evaporation point is higher than that of the base diesel, the FBP is lower, permitting complete combustion of the fuel.

Kerosene, on the other hand, combined with a higher volatility and very low viscosity delivers the lowest CO and PM. These properties enable the fuel to generate mixture in the bowl, even though the spray penetration is low due to lower density. At low part loads, the spray penetration, particularly in the ballistic range of the injection needle lift is not as crucial as at high loads.

Analysis at the A70 (1220 rpm, 14 bar BMEP) load point

The optimization of the DOE for the A70 load point was done with a NOx emission limit of 1.5 g/kWh. This NOx level was chosen despite the initial NOx target of 0.4 g/kWh because preliminary tests showed an unacceptably high PM emission at such a low NOx level. In Figure 15, the comparison of the fuels is shown. Std. Diesel, has even at this load point, the highest PM emissions, due to the high aromatic content.

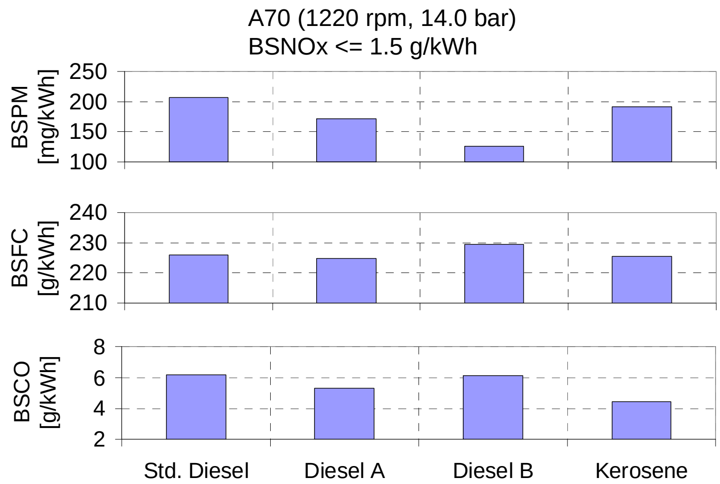

Figure 15: DOE optimized comparison of the A70 load point under part homogeneous operation

The heat release analysis (Figure 16) shows that the base diesel has higher peaks of heat release. All fuels display a double cool flame pattern. The burning functions (Figure 16) are very similar for the hot flame. Diesel A and the base diesel have a larger cool flame heat release peak, which is beneficial for BSFC.

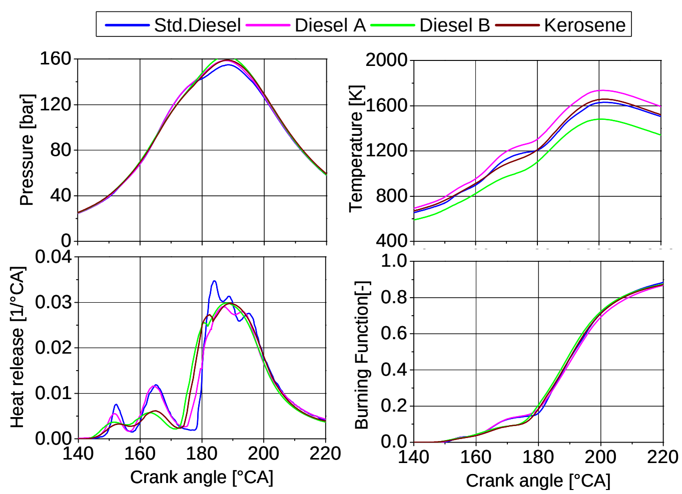

Figure 16: Heat release of the A70 load point under part homogeneous operation

However, due to the high volatility and lower density of Diesel A, the spray penetration of the pilot injections, in particular, is much lesser. This leads to mixture formation primarily in the bowl, with the air in the squish volume not sufficiently utilized, leading to high CO and PM emissions. This is even more severe for Kerosene, which also has a high PM emission level.

Diesel B, on the other hand, has the lowest PM emissions, while it has a small BSFC deterioration of 1%, in comparison to Std. Diesel. The PM emissions are lower due to the smaller fraction of aromatics, and importantly, a higher cetane number. The high cetane number, in combination with the elevated pressures and temperatures of the A70 load point, permits the fuel to ignite (hot flame) earlier than the others. Its point of ignition and its duration of combustion are comparable to those of Kerosene. However, the greater spray penetration, due its higher density, ensures very good air entrainment with Diesel B, resulting in good PM emissions.

COMPARISON OF HETEROGENEOUS AND PART HOMOGENEOUS COMBUSTION

A direct comparison of the emissions performance of the fuels, under both heterogeneous and part homogeneous operation at the A25 load point is shown in Figure 17. The results show that a significant improvement in both NOx and PM emissions can be achieved with increased homogenization of the fuel-air mixture. An additional factor is the increased compatibility to EGR exists with homogenization. The part homogeneous points were run with and EGR rate of 54%, while the heterogeneous tests were performed with 40% EGR. This can be interpreted as improvement in mixture formation and its uniform distribution inside the combustion chamber. Significantly, the trends in particulate matter at both combustion modes are similar for the fuels.

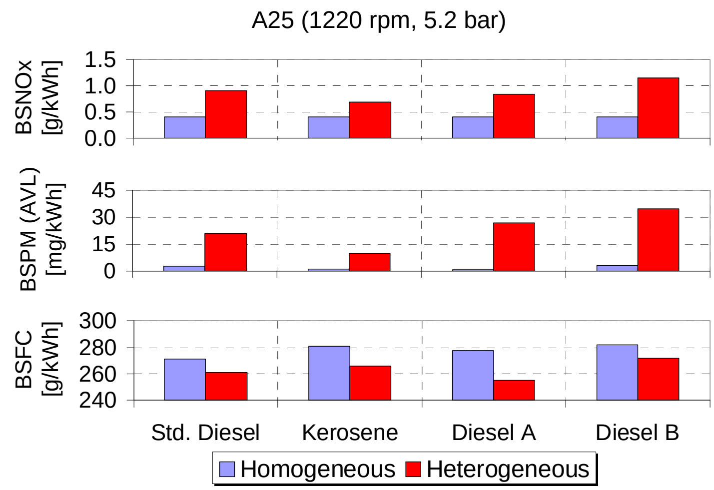

Figure 17: Emission comparison between heterogeneous and part homogeneous combustion

There is, however, a drawback with regard to BSFC. The disadvantage ranges between 3% -10%, depending on the fuel. This is due to the location of the CA50 much before ignition TDC, therefore, being thermodynamically inefficient.

SUMMARY AND CONCLUSIONS

Part homogenous combustion offers the potential for significant reduction in engine-out emissions, both in NOx and particulates. The reduction in PM emissions with higher homogenization is possibly due to lesser rich zones, thanks to the higher mixture formation time. Therefore, higher EGR rates can be permitted, in order to reduce NOx emissions. Further, since the hot flame ignition is achieved by a main injection, it largely minimizes the risk of uncontrolled ignition, which is a generally a challenge of HCCI operation.

It has been demonstrated that a large part of the engine map can be covered using this operation mode, which despite a fuel consumption disadvantage, can potentially offer an economically viable solution by eliminating the need for a complex after-treatment system i.e. SCR systems for NOx and diesel particulate filters (DPF’s). Even in the event of using exhaust after-treatment, the possibility of reducing the size of the SCR system or the use of an open DPF exists, implying savings on fuel and the SCR medium. Further potential for improved emissions behavior can be tapped with the optimization of the engine hardware, combined with the use of higher EGR rates and boost pressure.

In this study, it has been confirmed that aromatic content in fuels is detrimental for PM emissions, even under part homogeneous conditions. The distillation curve, density and viscosity, which influence the mixture formation, also appear to play an important role in the final result It has been shown in these experiments that a low distillation curve, accompanied by low fuel viscosity (as in

Kerosene) delivers good performance. At high loads, where the relative air-fuel ratio is lesser than at low loads, a good spray penetration is required. The higher cylinder pressures and hence charge density also affect the spray penetration. In such cases, the fuel with a high density (Diesel B) offers benefits. The final boiling point correlates to CO emissions in the results shown. Even though the cetane number was approximately the same for 3 of the fuels tested, strongly varying performance was obtained, confirming that the cetane number in itself is insufficient to predict the behavior of fuels. Amongst the fuels considered for this study, Kerosene, with its high volatility and zero aromatic content displays high potential for application in heavy duty diesel engines, where low particulates need to be achieved, in spite of EGR use throughout the engine map. Its performance at the medium load points, where pilot injections are applied, can be improved through a nozzle optimization to increase spray penetration.

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ACRONYMS, ABBREVIATIONS

BSPM: Brake specific particulate matter

BSNOx: Brake specific NOx

BSFC: Brake specific fuel consumption

BSCO: Brake specific CO

DPF: Diesel particulate filter

EGR: Exhaust gas recirculation

FBP: Final Boiling Point

FC: Fuel consumption

HCCI: Homogeneously charged compression ignition

HFR: Hydraulic Flow Rate

SCR: Selective catalytic reduction