

Vehicle Demonstration of Naphtha Fuel Achieving Both High Efficiency and Drivability with EURO6 Engine-Out NO_x Emission

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ABSTRACT

Demand for transport energy is growing but this growth is skewed heavily toward commercial transport, such as, heavy road, aviation, marine and rail which uses heavier fuels like diesel and kerosene. This is likely to lead to an abundance and easy availability of lighter fractions like naphtha, which is the product of the initial distillation of crude oil. Naphtha will also require lower energy to produce and hence will have a lower CO₂ impact compared to diesel or gasoline. It would be desirable to develop engine combustion systems that could run on naphtha. Many recent studies have shown that running compression ignition engines on very low Cetane fuels, which are very similar to naphtha in their auto-ignition behavior, offers the prospect of developing very efficient, clean, simple and cheap engine combustion systems. Significant development work would be required before such systems could power practical vehicles. Naphtha has much lower Cetane compared to a conventional European diesel fuel.

The aim of this study is to investigate if and how an existing modern diesel engine could be run on low Cetane fuels in general and naphtha in particular. Extensive single-cylinder engine tests using different low Cetane fuels showed that the engine could be run on narrow-cut naphtha with a derived Cetane (DCN) number of 38, at all relevant speed and load conditions, while meeting or exceeding the efficiency and emissions requirements. Based on these engine studies, a downsized four-cylinder 1.6 liter diesel engine adapted with piezoelectric fuel injectors, EGR cooling and two-stage air boosting system was installed in a demonstration vehicle equipped with an on-board combustion control system. The vehicle was successfully run under cold NEDC (New European Driving Cycle) operation at high efficiency with good transient operation and acceptable noise levels while achieving engine-out NO_x emission below EURO6 levels using naphtha. Moreover, engine-out PM (particulate matter) emissions were lower than those generated with conventional diesel fuel on the base engine. This work demonstrated that a modern diesel vehicle can run at high efficiency and low noise and emissions on a simple fuel that requires little processing in the refinery compared to the diesel fuel and will be easily available in the future.

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INTRODUCTION

Around 95% of global energy for transport is currently supplied by liquid fuels made from oil and oil will continue to be the primary source of energy for transport in the future. The demand for transport and hence transport fuels is

increasing rapidly, primarily driven by growth in non-OECD (Organization for Economic Co-operation and Development) countries. One projection [1] suggests that transport energy demand in 2040 will be around 40% higher than in 2010 - other studies have come to similar conclusions [2, 3, 4, 5]. The worldwide annual demand for transport fuels is large and

even though alternatives such as biofuels and electric cars are likely to grow, they have limited capacity to replace conventional petroleum-based fuels and even by 2040, around 90% of transportation energy is expected to come from oil [1]. Figure 1 shows that oil reserves - known quantities of oil that are economically recoverable - have consistently increased over the past three decades. This is because of new discoveries as well as improving recovery rates. Also plotted in Figure 1 is the ratio of the reserves to the annual production level which has hovered around 40 years for the last twenty years or so. Thus reserves have kept pace with increasing demand. The data are from the U.S. Energy Information Agency [2]. There are also vast resources of “unconventional” oil in the form of oil from oil sands and shale and heavy oil, which are more difficult to recover and are of lower quality compared to conventional oil [6]. If and when demand for oil increases and the price is high enough, more of these large unconventional resources will come into play. This is already the case, to some extent, with Canadian oil sands. Indeed, oil supply capacity is growing worldwide at an unprecedented level and there is an abundance of oil to meet future demands [7].

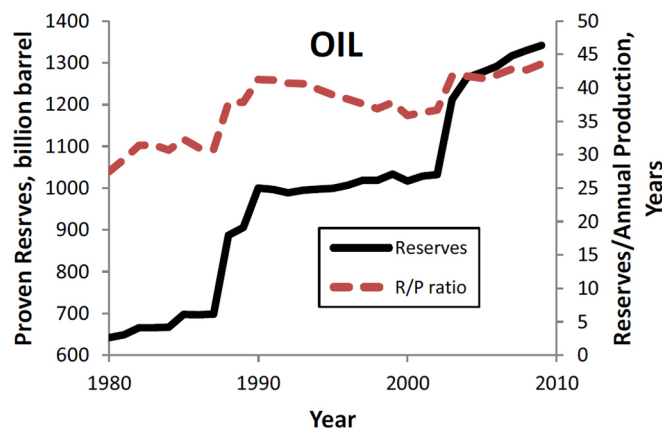


Figure 1. Proven oil reserves and reserves/annual production ratio, U.S.EIA [2]. In 2009, there were enough reserves to last for ~44 years.

The growth in fuel demand will be skewed very heavily towards commercial transport (trucks, planes, ships and rail) which uses heavier fuels like diesel and kerosene [1, 2, 3, 4, 5]. For example, the number of personal vehicles is expected to double to 1.6 billion units in 2040 (compared to 2010), but the fuel demand from this sector is not expected to increase [1]. This is because of the expected large improvement in efficiency, e.g., through hybridization and reduction in average passenger car size and the miles driven compared to today's average passenger car, especially in countries like China and India where vehicle numbers are expected to increase most. In contrast, the energy demand for the commercial transport sector is expected to increase by about 75% over the same period in spite of some gains in efficiency [1]. Thus in the future much more oil will have to be refined

to meet the increased demand for commercial transport. The first process in the refinery is distillation, to separate the crude oil into different boiling-range fractions. The light fraction resulting from distillation and boiling roughly in the gasoline range from $\sim 30^{\circ}\text{C}$ to $\sim 200^{\circ}\text{C}$ is termed “naphtha.” It consists of a complex mixture of hydrocarbon molecules generally having between 5 and 11 carbon atoms. It typically constitutes 15-30% of crude oil by weight, depending on the crude. Usually the RON of Naphtha is around 70. Light naphtha is the fraction of naphtha boiling between 30°C and 90°C and consists of molecules with 5-6 carbon atoms. Heavy naphtha boils between 90°C and 200°C and consists of molecules with 7-11 carbon atoms and is preferred for further upgrading in the catalytic reformer. Light naphtha has lower Cetane (higher Octane) compared to heavy naphtha and a range of Cetane numbers can be obtained by taking different cuts in different boiling ranges. Naphtha is the least processed product of a refinery and the energy required to produce naphtha and hence its CO_2 impact will be lower compared to conventional gasoline or diesel. Moreover, light fractions like naphtha will be readily available and could be in surplus as the demand shifts towards heavier fuels used in commercial transport.

Meanwhile great efforts are being made by engine manufacturers to improve internal combustion engines and reduce overall fuel consumption. To achieve this objective, the fuel and engine should be considered as one system and optimized synergistically. The primary development trend for spark-ignition (SI) engines is towards improving their efficiency. For the more efficient diesel engines, the goal is to reduce their soot and NO_x emissions without compromising their efficiency while controlling their cost. Recent experimental results [8, 9, 10, 11, 12, 13, 14] show that there is great potential for developing highly efficient, clean, simple and cheap engines that need less processed fuels, by properly matching the fuel and engine combustion systems. For instance, current diesel engines are much more efficient than gasoline engines but are significantly more expensive, primarily because of the great effort needed to reduce NO_x and soot emissions simultaneously. This is because they use current diesel fuels that ignite too easily when injected in the engine, before the fuel has a chance to mix well with oxygen. Hence they have to use expensive, high injection-pressure injection systems to promote more premixed combustion and even then will require expensive NO_x and soot control after treatment. The emission requirements could be met much more easily and the engine cost reduced (simpler injection and after-treatment systems) if diesel engines are run on fuels which are more like gasoline and do not ignite readily, i.e., have low cetane numbers. This will give more time for fuel and air to mix before combustion starts [8, 9, 10, 11, 12, 13, 14, 17, 18]. In [14] it was demonstrated that the fuel consumption of an SI engine with a compression ratio, CR, of 12 could be improved by about 19% on a Federal test Procedure (FTP) cycle by running it in compression ignition

Table 1. Summary of Naphtha properties used in different studies

Fuel / Paper Number	Naphtha for This Paper	Light Naphtha SAE 2012-01-0677 [14]	Heavy Naphtha SAE 2012-01-0677 [14]	Light Naphtha SAE 2008-01-1599 [15]	Heavy Naphtha SAE 2008-01-1599 [15]	Naphtha1 SAE 2010-01-0334 [16]	Naphtha2 SAE 2010-01-0334 [16]
Author	Saudi Aramco & FEV	Saudi Aramco	Saudi Aramco	Toyota & Exxon Mobil	Toyota & Exxon Mobil	CONCAWE and extra	CONCAWE and extra
DCN from IQT	38	34	41	34	31.2	33	35.1
RON	52	66	62	66	59	N/A	N/A
MON	54	62	58	65	60	N/A	N/A
T10 °C	89	46	94	43	90	88	54
T50 °C	91	52	125	55	123	112	96
T90 °C	92	68	157	64	168	145	179
Hydrogen/Carbon	2.27	N/A	N/A	2.29	1.94	2.00	2.08
Specific Gravity	0.70	0.66	0.73	0.66	0.76	0.75	0.72
Aromatics%v/v	0	1	12	1	17	13.3%/m/m	4.2%/m/m
Sulfur	<10	<20	<20	179	0	N/A	N/A
LHV MJ/kg	44.25	42.2	44.9	44.5	43.2	41.4	41.9
Compression Ratio	17-19	12		16		15	
Fuel System	Diesel	GDI (50-130bar)		Diesel (300-1000bar)		Diesel (up to 2000bar)	

(CI) mode using heavy naphtha with DCN of 38 compared to running it on SI mode using gasoline. The optimum solution could be a compression ignition engine with a compression ratio of 15-16 using an injection system with moderate injection pressures (maximum of 500 bar?) and injectors more like in a gasoline DI engine. The after-treatment focus would be shifted to HC and CO reduction rather than NO_x and soot control. Moreover the optimum fuel for such a system will have a RON of 70-85 [9,10,11] with no stringent requirements on volatility - a much "lower quality" gasoline compared to today's gasolines. The octane quality of such a fuel will be very much like that of naphtha, which is likely to be readily available in the future. Clearly, significant development work is needed before such compression ignition engines, using gasoline-like fuels, can power practical vehicles.

Previous work using naphtha in diesel engines includes [15] where both heavy and light naphtha, with DCNs of 34 and 31 respectively, were tested in a single-cylinder engine with a CR of 16. The engine could not be operated satisfactorily on these fuels at all the operating speed/load points required by a practical engine. In [16] two full boiling range naphthas of DCNs of 33 and 35 were tested along with three other fuels with DCNs in the mid-40s in a single cylinder engine with a compression ratio of 15 and compared with a European diesel fuel. In addition these six fuels were tested in a demonstrator vehicle equipped with a four-cylinder engine based on the single-cylinder version. The Naphtha fuels could not meet the Euro6 NO_x levels in this

case. The goal of the current work was to investigate if and how this demonstrator car could be run with high efficiency, with minor modifications, to meet EURO 6 emissions targets on low cetane (compared to conventional diesel fuel) fuels, e.g., naphtha cuts. As discussed above, such fuels are likely to be abundant in the future. The naphtha cut chosen had a narrow boiling point range and a DCN of 38. It was already hydro-treated so that the sulfur level is low enough to be comparable with a modern gasoline. Table 1 summarizes the properties of the naphtha cuts used in the current work as well as in previous studies.

The current work differs from previous studies using low Cetane fuels in diesel engines in demonstrating that a modern diesel car can run over a wide operating range (e.g., the NEDC cycle) at high efficiency and acceptable noise and transient operation with improved emissions (NO_x below EURO6 levels) on a simple fuel that requires little processing in the refinery and will be easily available in the future compared to the diesel fuel. Extensive single-cylinder engine tests using different low Cetane fuels showed that the engine could be run on narrow-cut naphtha with a derived Cetane number of 38, at all relevant speed and load conditions, while meeting or exceeding the efficiency and emissions requirements. Based on these engine studies, a downsized four-cylinder 1.6 liter diesel engine - adapted with piezoelectric fuel injectors, EGR cooling and two-stage air boosting system - was installed in a demonstration vehicle equipped with an on-board combustion control system.

2400 rpm, 14.8 bar IMEP Part load - - - KIVA simulation - Base Configuration: CR=15.1, HFR = 312 cm³/30sec
 — Heat release analysis of test data

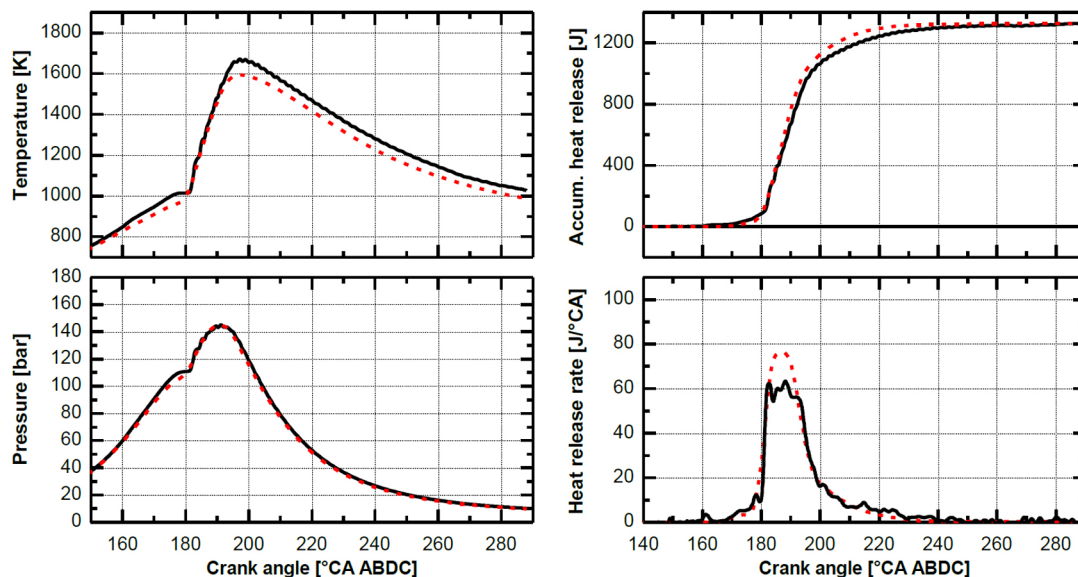


Figure 2. Matching of simulation and experiment for naphtha operation with CR 15 at 2400rpm, 14.8bar IMEP

CFD SIMULATION: PISTON BOWL LAYOUT

One of the key measures to improve low load operation with low CN fuels is the increase of the compression ratio (CR). Generally, increasing the compression ratio puts a challenge on the mixture formation at higher loads due to the significantly reduced bowl volume and free spray length available. This can lead to increased soot emissions due to poor soot oxidation boundary conditions. Therefore the piston bowl layout has to be carefully designed. This design has been supported by 3D CFD combustion simulation with KIVA 3V Release2 [22]. Simulations have been performed at 2400 RPM and 14.8 bar IMEP (indicated mean effective pressure) gross and 4000 RPM full load to assess the mixture formation at high loads. The model has been calibrated to available base engine test data with a compression ratio of 15 and operation with Naphtha fuel. To account for the high volatility of Naphtha, n-heptane has been used as surrogate fuel. The ignition delay behavior has been matched to the test data. Figure 2 shows the satisfactory matching of pressure, temperature and heat release traces. At this specific case, hydraulic flow rate (HFR) of 312 cm³ / 30 sec at 100 bar is used for standard flow rate injector.

Two different CR 17 piston bowl geometries have been investigated (Figure 3). To evaluate the mixture formation, air utilization characterization method has been used. The main idea of this procedure is to assess the quality of the mixture formation based on the air-fuel ratio (AFR) distribution in the combustion chamber and not to rely on sensitive emission models. For this purpose the combustion

chamber volume is divided by four characteristic air-fuel ratio limits for each degree crank angle (CA) as shown in the sample graph in. In this graph the blue area corresponds to “unused” air, which does not take part in the combustion process, the green area corresponds to very rich zones with normalized AFR, λ , below 0.5 and the yellow area to rich zones with $\lambda < 1.0$. This yellow area is of special importance for soot emissions, as no soot oxidation can take place with local λ below 1.0. Therefore, the goal of the optimization is to enlarge the green area with ideal λ s between 1.0 and 2.0 as much as possible. The simulation results in show that both investigated CR 17 bowl shapes achieve a mixture formation quality similar to the base bowl with CR 15, with a very slight advantage for bowl 1 compared to bowl 2. As the same trend is predicted by the full load simulations, bowl 1 has been chosen for the experimental investigations. In a later step, the CR has been increased further to CR 19 based on the same bowl shape.

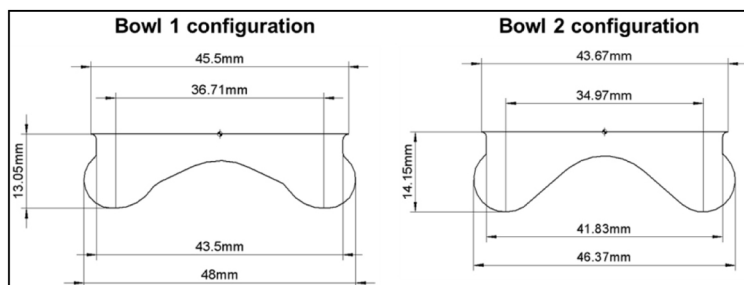


Figure 3. Investigated CR 17 bowl variants

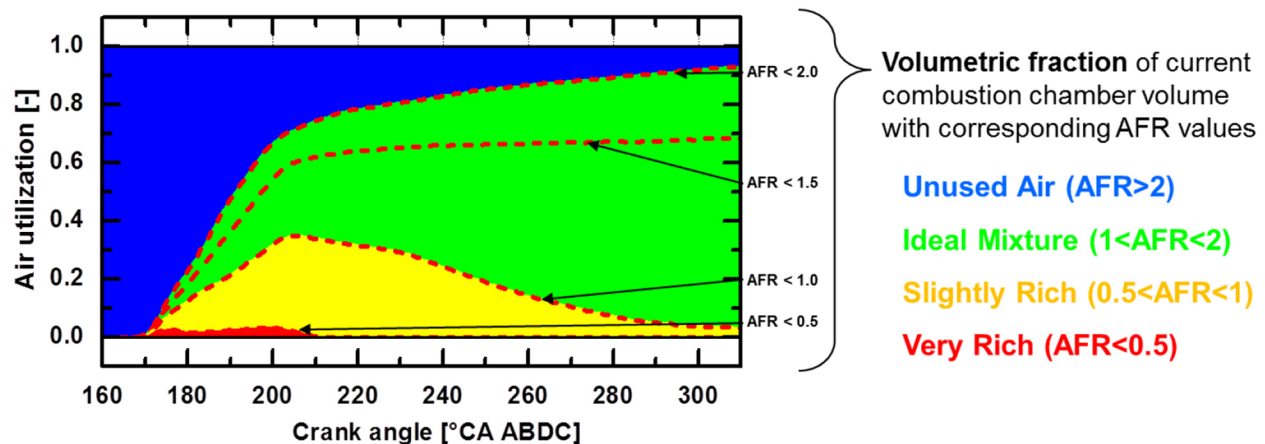


Figure 4. Sample air utilization characterization

2400 rpm, 14.8 bar IMEP Part load

- KIVA simulation - Base Configuration: CR=15.1, HFR = 312 cm³/30sec
- KIVA simulation - Bowl 1 Configuration: CR=17.0, HFR = 312 cm³/30sec
- KIVA simulation - Bowl 2 Configuration: CR=17.0, HFR = 312 cm³/30sec

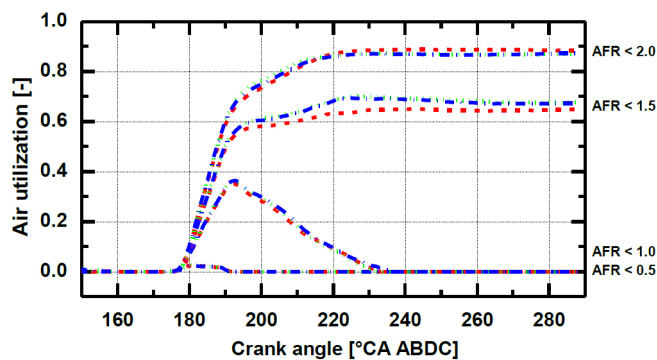


Figure 5. Air utilization characterization for CR 17 piston bowls compared to CR 15 piston

EXPERIMENTAL SETUP

Single Cylinder Diesel Engine

The single-cylinder Diesel research engine used for the tests is optimized to feature the best possible combustion characteristics with regard to low emission levels and high fuel efficiency. The enhancements include high fuel injection pressures, as well as stronger charge air and exhaust gas recirculation cooling, to achieve low intake manifold temperatures and thus a good cylinder charge even at the highest EGR rates. To optimize the flow characteristics, one intake port is designed as a filling port, the second one as a classic swirl port. Creating the charge movement is supported by seat swirl chamfers on both intake valves. The layout of the combustion chamber geometry shows a conventional recess shape, which was further optimized together with the nozzle geometry (8-hole) to achieve the best possible air

utilization. The recessed valves eliminate valve pockets in the piston and thus further improve the flow quality near the recess. For Diesel operation, a compression ratio (CR) of 15:1 was selected to represent acceptable peak pressures in spite of the increased charge density. With this compression ratio, the combustion system reaches a specific output of 80 kW/l at maximum peak firing pressures under 190 bar. An advanced common rail system is used as the injection system; the injector can quickly open and close the nozzle needle at a maximum injection pressure of 2000 bar.

Reducing the compression ratio, using a higher maximum cylinder and rail pressure, as well as improved EGR cooling increases ignition delay and makes low particulate emissions possible. As a result the research engine meets the Euro 6 standard with conventional Diesel. The performance of the single-cylinder Diesel engine thus exceeds that of currently available production engines. These technical enhancements will also be implemented in future Diesel engines. Due to the different chemical and physical properties - and in particular its reduced DCN for the compression ratio was increased in two steps to 17:1 and then to 19:1 for naphtha operation by adjusting the piston bowl geometry, as shown in Table 2. The injection system remained unchanged; thus a series production injection system was used for Naphtha operation. The only change required is due to the reduced boiling temperature of Naphtha. To prevent fuel boiling in the return line, a backpressure valve had to be installed. Compared to Diesel, Naphtha has a lower kinematic viscosity and a reduced lubricating ability. Since modern Diesel high-pressure pumps are lubricated by the fuel, Naphtha had to be treated with 200 ppm of a commercial lubricity additive. The baseline low-sulfur diesel fuel will also contain this additive, probably at a slightly different dose rate. Emissions comparisons between these fuels are not expected to be affected by these small differences. The injector nozzle had to be reviewed as well. For Diesel operation, a nozzle with a hydraulic flow rate (HFR) of 310 cm³/30 sec at 100 bar has proven to be best, but due to lower viscosity and higher

volatility compared to Diesel, a nozzle with a HFR of 520 cm³/30 sec was tested as well. The main difference between the tested nozzles is the nozzle hole diameter, that is increased for HFR 520 from 0.109 to 0.141 mm. Table 3 provides a summary of the details of the test engine used. Additional information on the single-cylinder research engine can be found in previous publications [19, 20]. Regulated emissions, CO, HC, NO_x and PM are measured at engine exhaust. Based on the measurement of the filtered smoke number (FSN) by an AVL 415 Smoke meter, indicated specific PM emissions could be calculated. The exhaust gas is sampled in the exhaust gas manifold. To avoid condensation, the measurement lines are heated up to 180 °C. The EGR-rate is calculated based on CO₂ concentration in the intake manifold. Fuel consumption is measured by a Coriolis type fuel flow meter. Air mass flow is calculated by the air volume flow measured by a rotary displacement meter with measurement of pressure, temperature and water content of the air. Boost pressure can be adjusted independently by external three-stage charging system, which can also provide low intake air temperatures via two charge air coolers. An electric, position controlled EGR-valve is used for adjusting the EGR-rate. Since the test bench is not equipped with exhaust gas after-treatment systems, the exhaust gas back pressure is controlled by two flaps in the exhaust gas track. For measuring the in-cylinder pressure, a water-cooled piezoelectric pressure transducer (Kistler 6041A) is utilized. A charge amplifier (Kistler 5007) converts the sensor signals to an electric voltage, which is proportional to the cylinder-pressure. This voltage is fed into the Combustion Analyzing System (CAS). This system is also used for determining combustion related parameters, e.g., center of combustion (CA50). All pressures are recorded with steps of 0.5 °CA, except the cylinder pressure, which is recorded each 0.1 °Crank Angle. Indicated values, e.g., IMEP, are recorded as averages of 50 cycles. The position of the crank angle and the differentiation between high pressure phase and gas exchange phase is done by a crank angle sensor and a sensor at the camshaft.

Table 2. Utilized piston bowl geometries for adjusting the compression ratio

	Unit	CR 17	CR 19
Maximum bowl width	[mm]	48	48
Maximum bowl depth	[mm]	13.8	11.4
With / Depth ratio	[-]	3.678	4.21

Table 3. Hardware specifications for the research engine
*(*ks: factor of a nozzle is defined as (ID- ED)/ 10, where ID is the nozzle inlet diameter and ED is the nozzle exit diameter in microns. The ks factor is applied for round inlets, whereas a k factor is defined to be applied for sharp inlets).*

	Unit	Specification
Swept volume	[cm ³]	390
Stroke	[mm]	88.3
Bore diameter	[mm]	75
Valves per cylinder	[-]	4
Compression ratio	[-]	17:1, 19:1
Max. cylinder pressure	[bar]	220
Boost pressure	[bar]	Max. 3.8 abs.
Injection system	[-]	Bosch Piezo Common Rail System
Max. injection pressure	[bar]	2000
Nozzle hole number	[-]	8
Nozzle diameter	[mm]	0.109 (HFR 310) 0.141 (HFR 520)
Cone angle	[°]	153
ks*	[-]	1.5 (HFR 310) 3.0 (HFR 520)

RESULTS AND DISCUSSION

Single-Cylinder Test Results

For defining the final hardware setup, several engine load points were tested with changing configurations. After choosing the hardware setup, the main focus was on reducing HC- and CO-emissions in particular at lowest engine loads. Further, cold start ability was tested as well. The engine was equipped with a glow plug but it did not seem to affect cold start very much. All investigated load points are depicted in Figure 6. Five of these part load points are within the NEDC cycle reference area while the highest part load point at 2400 rpm and 14.8 bar IMEP gross is outside this range. Engine tests were carried out with pilot injection and without pilot injection and simulated closed loop combustion control. This was done by adjusting the fuel injection timing so that the center of combustion (CA50) was always located at the same degree crank angle in the engine cycle, chosen to maximize the engine efficiency. The difference between the energizing time of the injector and start of injection was measured to be 160 μs. The intake manifold temperature was measured and controlled directly downstream of the intercooler and before the EGR mixing. The conditions at the six part load operating points and at full load operation were chosen with respect to today's achievable ranges and are listed in Table 4. The other calibration parameters such as intake manifold pressure, fuel