

Advantages and Challenges of Lean Operation of Two-Stroke Engines for Hand-Held Power Tools

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ABSTRACT

One of the most significant current discussions worldwide is the anthropogenic climate change accompanying fossil fuel consumption. Sustainable development in all fields of combustion engines is required with the principal objective to enhance efficiency. This certainly concerns the field of hand-held power tools as well. Today, two-stroke SI engines equipped with a carburetor are the most widely used propulsion technology in hand-held power tools like chain saws and grass trimmers.

To date, research tended to focus on two-stroke engines with rich mixture setting. In this paper the advantages and challenges of leaner and/or lean operation are discussed. Experimental investigations regarding the influence of equivalence ratio on emissions, fuel consumption and power have been performed. Accompanying 3D-CFD simulations support the experiments in order to gain insight into these complex processes. The investigations concentrate on two different mixture formation processes, i.e. CCI (crankcase injection) and LPDI (low pressure direct injection). Further focus is put onto combustion phenomena of lean mixtures. The main issue addressed in this paper is the impact of mixture setting on fuel consumption. Therefore, the losses from theoretical to effective efficiency of different settings calculated by thermodynamic analyses are explained. Savings potential of fuel consumption and HC emissions are demonstrated.

The results indicate that fulfillment of required durability and power density using a leaner mixture setting is a significant challenge. Furthermore, the increase of NO_x emissions is a fundamental drawback. Here, basic research regarding reduction of NO_x engine-out emissions is required.

CITATION: Trattner, A., Grassberger, H., Schoegl, O., Schmidt, S. et al., "Advantages and Challenges of Lean Operation of Two-Stroke Engines for Hand-Held Power Tools," *SAE Int. J. Engines* 8(1):2015, doi:10.4271/2014-32-0009.

INTRODUCTION

Today, hand-held power tools facilitate specific jobs in many application areas, e.g. garden/landscape maintenance, construction industry and forestry. This market is highly diversified and demands special tools which are optimized for each application purpose (brush cutters, chain saws, cut-off saws, blowers, etc.). Most of them are driven by two-stroke SI engines equipped with a carburetor. High specific power, simplicity, low weight, low costs and compact design are characteristics of these two-stroke engine powered tools. The use of piston control or reed valves, a carburetor for mixture formation, oil-in-gasoline lubrication and loop scavenging is well established. Inherent disadvantages are high fuel consumption, high HC engine-out emissions and changing operation behavior during varying ambient conditions.

Nevertheless, significant improvements regarding emission behavior have been achieved in recent years due to stratified scavenging systems which have become a standard.

The contribution of these small SI engines to today's air pollution inventory is not predominant, but a further reduction of emissions is nevertheless required, even more so as the user is often directly exposed to the exhaust gas. Furthermore, the shortage of resources demands sustainable development in all fields of combustion engines with the principal means of efficiency enhancement. In order to respond to these future challenges and more demanding customer needs, the present engine technology has to be further developed. Recently, several research areas are considered in order to improve the overall product, i.e. injection systems, ECU, engine control functionality, exhaust gas after-treatment and combustion

systems [1, 7, 8, 9, 10, 11, 12, 13]. To date, the research on combustion systems for hand-held power tools tended to focus on rich mixture setting. The customer demands for high power and quick engine response without additional costs. These demands are met by a rich mixture in PL and WOT operation. Moreover, the rich mixture reduces NO_x formation and additionally cools the engine.

In this paper, the leaner and/or lean (over-stoichiometric) operation of loop scavenged two-stroke SI engines is discussed. Studies of lean combustion are among the oldest in the combustion literature. In fact, the first useful estimations for the lean limit of methane/air mixtures were reported by Davy (1816) in his efforts to prevent explosions of methane gas (called "fire-damp") in coal mines [2, 3]. Lean combustion is employed in nearly all combustion technology sectors, including gas turbines, boilers, furnaces, and internal combustion engines [2]. For automotive applications, the lean combustion or lean burn of an SI engine has been recognized as one of the most promising methods for further improving fuel consumption. There has been, however, difficulty in extending the lean misfire limit far enough to realize NO_x emission levels below the mandatory level and still reach satisfactory drivability [4]. Furthermore, the challenges of lean combustion are low reaction rates, flame extinction, combustion instabilities, mild heat release and sensitivity to mixing [1, 2, 15]. Considerable effort is necessary in order to counteract these effects. Spray-guided stratified concepts represent an effective solution. Many automotive manufacturers have brought lean-burn SI engines (four-stroke) into series production and are performing research on homogenous [e.g. 5] and stratified [e.g. 6] combustion. Generally, the exhaust gas after-treatment of lean combustion systems poses a great challenge.

In contrast to the automotive SI engines sector, only few studies and applications deal with the use of lean combustion in two-stroke engines [1, 27]. Two-stroke engines for the propulsion of scooters in India are often operated with lean mixture. In India, the main requirement to engines is fuel consumption and, therefore, higher NO_x emissions due to lean combustion are actually accepted by the Indian legislative. Other known applications are high-performance two-stroke engines for outboards and snowmobiles which use a stratified lean mode in part load operation realized by a sophisticated injection system [14, 26]. The transfer of these strategies or technologies to the hand-held power tools sector is not possible without additional research effort on engine design and combustion systems. Much research work regarding the fundamentals of lean combustion and description of complex interactions of flame propagation, burning velocities, ignition delay, combustion instabilities and turbulence exists [1, 2, 15, 21].

Within this paper the particular physics are not treated in detail as the focus is put on general engine characteristics and results. The paper starts with a brief overview of requirements on hand-held power tools and a description of the investigated

engine concepts which serve as a starting point. Subsequently, parameters, such as lambda and the equivalence ratio, are defined. Following, the fundamentals of the theoretical working process regarding the impact of equivalence ratio on the efficiency and IMEP are presented. The next chapter shows the experimental results of lean two-stroke combustion and results of 3D-CFD simulations which should provide insight into the complex processes. The general influence of equivalence ratio on emissions, power and fuel consumption is explained. The investigations concentrate on a prototype two-stroke engine with CCI and stratified scavenging. CCI with stratified scavenging is a very good starting point for necessary future improvements. Furthermore, investigations of an LPDI prototype engine have been accomplished in order to show the impact of the mixture formation system on the lean misfire or ignition limit. The main issue addressed in this paper is the impact of the mixture setting on fuel consumption. Therefore, the thermodynamic losses from theoretical to effective efficiency of different settings are explained. Additionally, the 'Parameter Studies for a New Combustion System' chapter shows theoretical measures to further increase efficiency. Finally, a short summary is given.

REQUIREMENTS TO HAND-HELD POWER TOOLS

A combustion engine used for the propulsion of a hand-held power tool has to fulfill various opposed requirements. The engine has to be simple, while reliable operation under varying ambient conditions has to be ensured. The demand for robust and simple technology with high power density leads to air-cooled single-cylinder engines with engine speeds up to 15.000 min^{-1} . Thus, small displacements with ergonomic weight and packaging advantages are realizable. Most of the present applications have displacements between 20 and 80 cm^3 with a specific power ranging from 30 to 60 kW/l . Additionally, strict requirements regarding noise and vibrations are presupposed for health and safety which are likewise challenging [12]. Also, maximum surface temperatures and exhaust gas temperatures at prescribed distances are limited by legislature. Moreover, emission legislation has to be fulfilled which is primarily not customer-relevant. USA and EU have introduced strict emission limits, which other countries will probably use as a role model. The present emission test procedure is a steady-state two-mode test cycle. The first operating point is at maximum power (100% load) and produces 85% of the overall weighted emissions of the engine. The second point is at idle operation and affects the residual 15%. Future emission limits and their period of validity are still in discussion. Limited emissions by EPA [16] are HC and NO_x and CO emissions. The limits are shown in Figure 1.

Additionally, NO_x emissions are limited to 10 g/kWh in the EU [17]. Furthermore, the manufacturer has to guarantee the limits in compliance with the emission legislation for the classified product life cycle (EPA regulation: 50, 125 or 300 h).

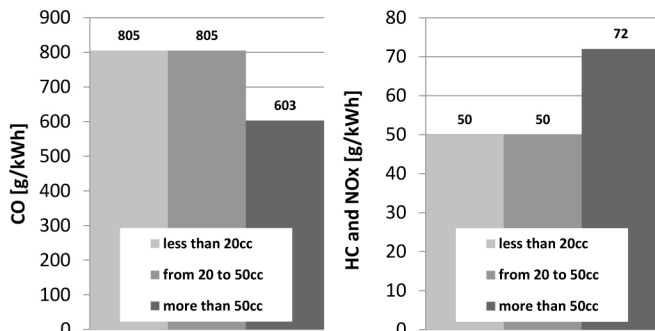


Figure 1. EPA emission limits for displacement categories [comp. 16]

STARTING POINT

A 47 cm³ series production two-stroke engine with stratified scavenging and a single-flow carburetor has been used as starting point for the investigations and as the reference for comparison of the results. Two different prototype engines, CCI and LPDI, have been investigated regarding lean combustion. The main difference between the series production and the prototype engines is the mixture formation concept and the ignition control as well as injection by an external ECU [comp. 12]. A standard series production MPFI (multi point fuel injection) injector is used for both prototypes. Here, compromises between injection duration at WOT and idle as well as between droplet size and flow rate are accepted [see 12]. Significant improvements are conceivable, if the injector layout can be adjusted to the special requirements. Figure 2 shows a section through the cylinder of the reference engine and the two different injection positions (LPDI, CCI) are marked. The port configuration, exhaust system, combustion chamber, etc. are unchanged. The engines are briefly described below. A detailed description of the LPDI and CCI systems as well as of emissions, power and fuel consumption behavior is shown in [12].

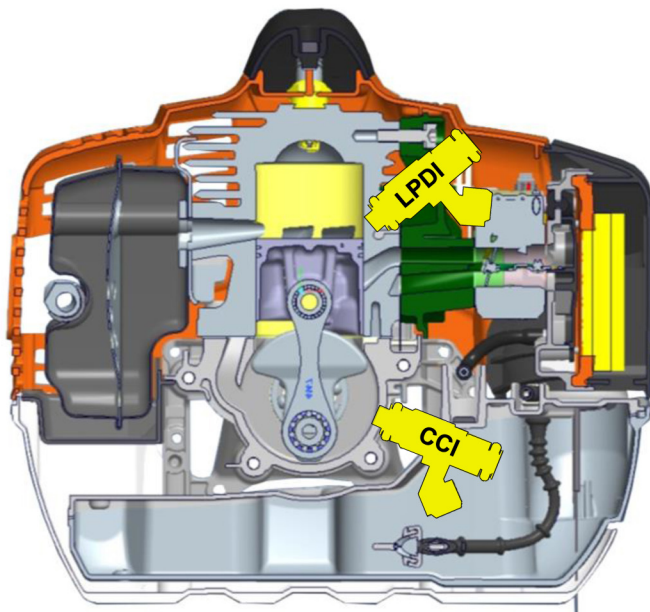


Figure 2. Different injection concepts

Series Production Engine

The series production two-stroke engine with stratified scavenging and carburetor serves as a starting point as well as a reference. In Table 1, the engine data is summarized.

Table 1. Engine data

Cylinder:	1-cylinder / air-cooled
Displacement:	47 cm ³
Compression ratio:	9.6 (geometric)
Bore / Stroke:	42 / 30 mm
Specific power	46 kW/l
Principle:	Loop and stratified scavenging
Ports:	4 scavenging ports / 1 exhaust port
Exhaust after-treatment:	Without

The reference engine uses crankcase compression and has a conventional air and fuel supply into the crankcase via a carburetor. A single-flow carburetor with a split-plate is used to separate the fuel supply from pure air intake. An auxiliary inlet port connects the lower area of the split carburetor (contains pure air) via a recess in the piston to the scavenging ports close to their entry point into the cylinder. During the induction stroke, the recess in the piston opens and pure air streams towards the crankcase, displacing the mixture of air and fuel ahead of it. When crankcase compression begins, the intake ports are ideally filled with air only and when scavenging starts, the air gets into the combustion chamber downstream of the rich air-fuel mixture. This reduces the scavenging losses and, consequently, fuel consumption. Both paths, the mixture as well as the pure air path, are controlled by piston movement. The principle is explained in detail in [12].

CCI - Crankcase Injection

The investigated CCI system applies fuel with oil into the crankcase (Figure 2) and uses the stratified scavenging method of the reference engine. Therefore, the port configuration of the series production engine is used. The spray is oriented to the crank arm, using the high relative velocity between fuel droplets and crank shaft to support mixture formation. Injection into the crankcase can be realized as long as the pressure in the crankcase is lower than the injection pressure. These investigations used fuel pressure of 5 bar and, therefore, injection phasing is unlimited. Engine lubrication is ensured, as oil-in-gasoline (1:50) is injected by the fuel injector.

LPDI - Low-Pressure Direct Injection

This LPDI system applies fuel without oil directly into the cylinder. The injection nozzle is located in the cylinder wall opposite to the exhaust port (Figure 2) and is connected to the cylinder via a small connection duct. For direct injection, the spray is oriented towards the piston surface and thus has to

penetrate the strong flow field of the loop scavenging process. Direct injection can be realized as long as the cylinder pressure is lower than the injection pressure (5 bar) and as long as the connecting duct is not covered by the piston. Generally, the top limit of the injector position is narrowed by the exhaust port timing as the used low-cost injector cannot resist high cylinder pressure and temperature. Again, the port configuration of the reference engine is used. Engine lubrication is ensured by an oil-dosing valve, which injects pure oil directly into the crankcase. The oil-fuel ratio (1:50) is controlled by the ECU.

DEFINING LEAN COMBUSTION

In order to theoretically achieve complete combustion, a minimum amount of air mass is required which can be calculated by the laws of stoichiometric combustion in terms of mass or number of moles of each fuel [15]. Since different fuels require different amounts of air for stoichiometric conditions, specified by the air-to-fuel ratio (AFR), a normalized value is more appropriate for comparison across fuel types. The lambda (λ) and its inverse, the equivalence ratio (ϕ), are two common quantities. In this circumstance, lean (over-stoichiometric) combustion occurs when $\phi < 1$ or $\lambda > 1$.

$$\lambda = \frac{m_{Air}}{m_{Fuel} \cdot L_{st}} \quad (1)$$

$$\phi = \frac{1}{\lambda} \quad (2)$$

In [equation \(1\)](#), lambda is defined by the air mass m_{Air} divided by the fuel mass m_{Fuel} which is multiplied with the stoichiometric constant (unit is $\text{kg}_{Air}/\text{kg}_{Fuel}$) of each fuel. The equivalence ratio is the inverse of lambda, [equation \(2\)](#). [Figure 3](#) shows the correlation between the air-to-fuel ratio, equivalence ratio and lambda. In the following, the equivalence ratio is used to describe the results.

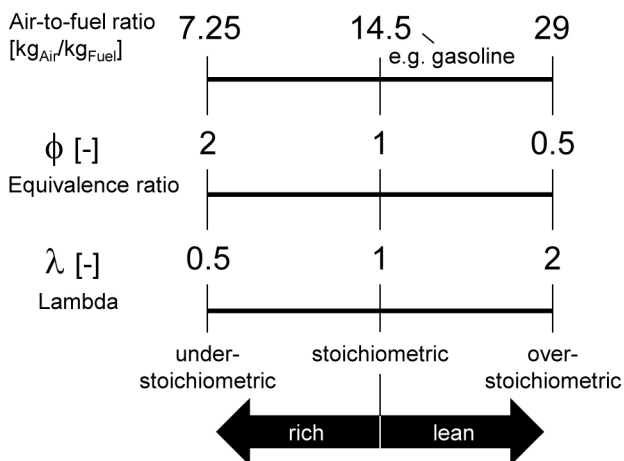


Figure 3. Correlation between AFR, equivalence ratio and lambda

FUNDAMENTALS

In this chapter the impact of the equivalence ratio on efficiency and IMEP of the theoretical SI working process is presented in order to show the efficiency enhancement potential of lean combustion. The theoretical working process uses isochoric heat release and adiabatic combustion chamber walls. The cylinder is completely filled with fresh charge mass (fuel and air mixture with defined equivalence ratio and ambient pressure as well as temperature) at BDC. The calculation of the working process also considers temperature-dependent heat capacities and incomplete combustion behavior up to the thermodynamic equilibrium. Further, the calculations have been performed for gasoline with a lower heating value H_u of 41000 kJ/kg and a stoichiometric constant L_{st} of 14.5 $\text{kg}_{Air}/\text{kg}_{Fuel}$.

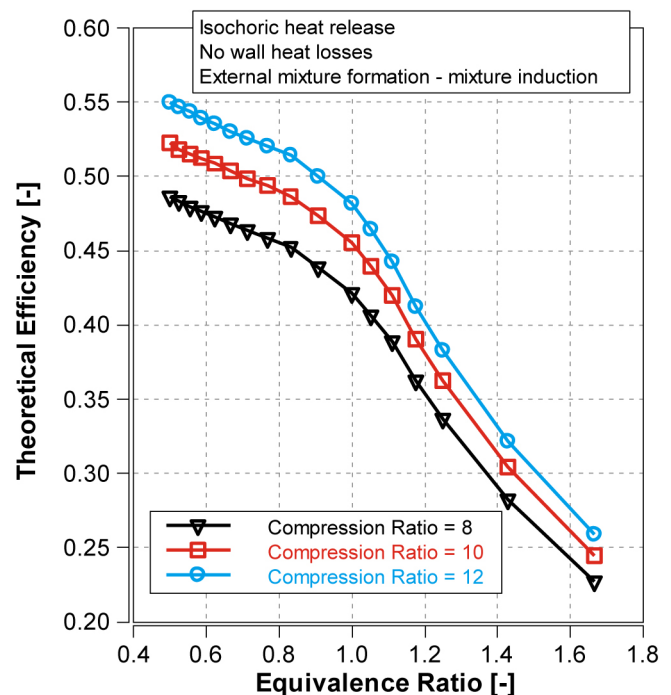


Figure 4. Theoretical efficiency (calculation result)

The efficiency of the theoretical working process mainly depends on the compression ratio and the equivalence ratio. Here, the relevant areas of both parameters for small two-stroke SI engines are considered. The compression ratio is varied from 8 to 12 and the equivalence ratio from 0.5 to 1.667. The efficiency results of different compression ratios over the equivalence ratio are shown in [Figure 4](#). Generally, efficiency increases with higher compression ratio but the choice of compression ratio is limited due to knocking. The graph shows that the absolute increase of efficiency due to a higher compression ratio is higher at lean mixtures and lower at very rich mixtures $\phi > 1.4$. Besides the compression ratio, the equivalence ratio has a significant influence on efficiency. Efficiency increases with decreasing equivalence ratio. Especially in the area of rich mixtures ($\phi > 1$), the theoretical efficiency increases drastically with lower equivalence ratios due to the influence of incomplete combustion (fuel cannot be completely oxidized). In the area of lean combustion $\phi < 1$ the efficiency increase is less steep.

Efficiency increases continuously below $\phi < 1$ due to increasing specific heat ratios of the gas. Today, two-strokes for hand-held power tools are operated with rich mixtures for several reasons. If a leaner adjustment can be realized, fuel consumption is significantly reduced.

The equivalence and compression ratio also influence the IMEP of the working process considerably. Generally, the IMEP (3) can be calculated based on the delivery ratio λ_a (4), the indicated efficiency η_i and the volumetric heat value H_G (5). The delivery ratio represents the quality of gas exchange and is defined as the quotient of the delivered mass to the theoretically delivered or reference mass. The delivery ratio is set to 1 for these theoretical investigations and the indicated efficiency is set to the values of the theoretical efficiency.

$$IMEP = \lambda_a \cdot \eta_i \cdot H_G \quad (3)$$

$$\lambda_a = \frac{m_E}{m_{th}} \quad (4)$$

$$H_G = \frac{H_u \cdot \rho_g}{\frac{1}{\phi} \cdot L_{st} + 1} \quad (5)$$

The volumetric heat value H_G in kJ/m³ represents the supplied energy with 1 m³ fresh charge mass into the cylinder with regard to ambient conditions. (5) shows the equation of the volumetric heat value for mixture induction. The volumetric heat value depends on the lower heating value of the fuel, the equivalence ratio and the density of the fuel-air mixture. In Figure 5 an almost linear dependency between the volumetric heat value and equivalence ratio can be seen. The volumetric heat value essentially determines the power output of an engine.

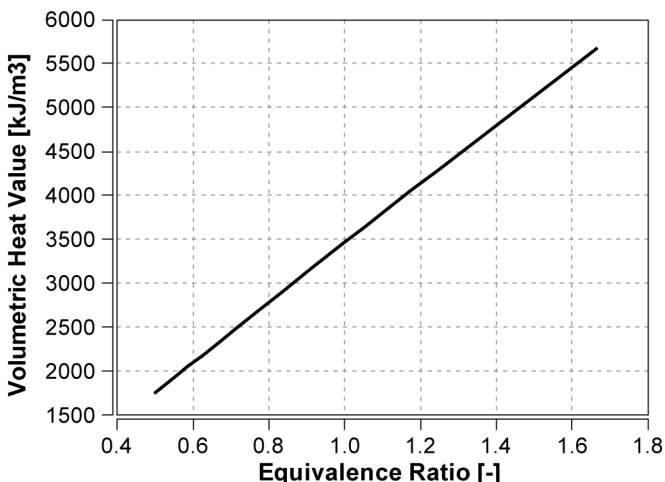


Figure 5. Volumetric heat value for mixture induction on ϕ (calculation results)

The power output and the IMEP of the theoretical working process on equivalence ratio, respectively, are shown in Figure 6.

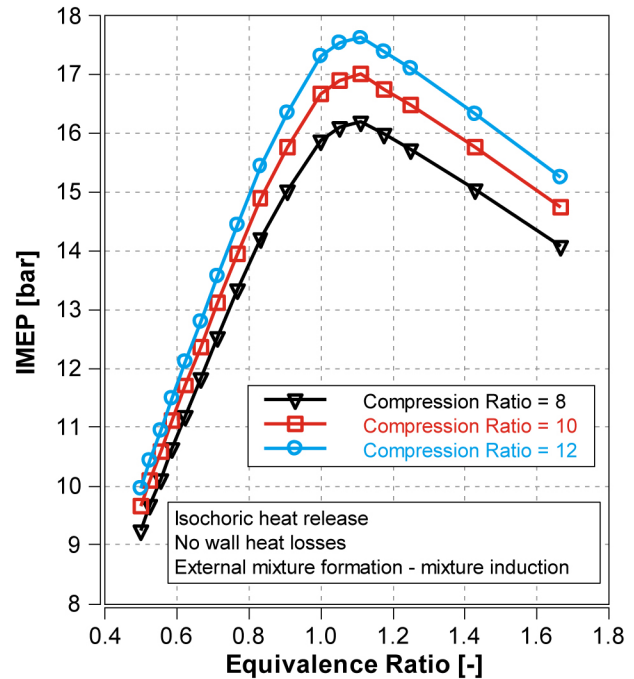


Figure 6. IMEP of the theoretical working process (calculation results)

The maximum is achieved at $\phi \sim 1.1$ (rich mixture), with increasing and decreasing equivalence ratios, the IMEP decreases. Starting from $\phi = 1$, efficiency decreases drastically with richer mixture setting (see Figure 4), but at the same time the volumetric heat value increases (see Figure 5). The product of both causes the maximum around $\phi \sim 1.1$ and this behavior remains approximately the same as in a real engine process. A further increasing equivalence ratio causes lower IMEP. With leaner mixture setting ($\phi < 1.1$) the IMEP drastically decreases. The volumetric heat value is lowered and although the efficiency concurrently increases the product of both becomes less and less.

RESULTS OF TWO-STROKE LEAN COMBUSTION

This chapter presents experimental as well as simulation results focusing on the CCI prototype engine. In the beginning the general behavior regarding the influence of equivalence ratio on fuel consumption, emissions, power and combustion process at WOT is explained. Regarding the question of durability at lean operation, this chapter also discusses abnormal combustion at high revolutions which often causes engine damage. However, additionally to the investigations with CCI, the results of the LPDI prototype engine should show the impact of the mixture formation system on the lean misfire or ignition limit.

General Behavior

In the operation range from 5000 min⁻¹ to approximately 10000 min⁻¹ WOT, the influence of the equivalence ratio remains largely the same. The operation point 5000 min⁻¹ WOT is chosen for exemplar description of the results, as it

represents the maximum power point and is, therefore, relevant for emission legislation. The results are based on oil-in-gasoline mixture which consists of 50 parts gasoline with a RON of 95 and one part oil (Castrol RS 2T). The used fuel (5% ethanol fraction) has a calorific value of 42310 kJ/kg, a stoichiometric constant of 14.2 kg_{Air}/kg_{Fuel} and a density of 0.75 kg/dm³. Investigations with alkylate-based petrol show similar results, except for higher knocking resistance of alkylate-based petrol due to lower fuel sensitivity (defined as the difference between RON and MON) [see 15, compare 24].

Fuel Consumption, Emissions and Power

During the investigations the ignition timing has been adjusted in order to achieve an optimum center of combustion regarding fuel consumption for each mixture setting point. The global equivalence ratio is determined via the Brettschneider calculation method, which is based on measurement of the exhaust gas composition [18]. As a result of the stratified scavenging process, the equivalence ratio in-cylinder and in-exhaust is different. The in-cylinder equivalence ratio can only be measured at very rich mixture conditions by exhaust gas analysis [21]. The achievable accuracies of both methods are shown in [18, 21]. E.g. at operation point 9500 min⁻¹ WOT with $\phi = 1.2$, the global value in exhaust is 0.04 points leaner compared to the equivalence ratio in cylinder. The gap between global and in-cylinder becomes a little larger at lean mixtures because of the different trends of trapping efficiencies of fuel and O₂ or air. Extrapolation of the trend at rich mixture setting results in a difference of 0.06 at $\phi = 0.65$. Detection of in-cylinder values at leaner conditions is not possible and, therefore, all experimental results are shown against the global and exhaust equivalence ratios, respectively. Evidently, the equivalence ratio difference considerably depends on the operation point as trapping efficiencies are different.

Figure 7 displays the BSFC, HC, NO_x and CO emissions on a global equivalence ratio. Generally, this engine has the ability to operate lean mixtures up to a global equivalence ratio of approximately 0.65, where misfiring already occurs sporadically. The emissions are significantly influenced by the equivalence ratio. The NO_x emissions increase from 1.5 g/kWh at an equivalence ratio of 1.2 to the maximum of approximately 9 g/kWh at $\phi = 0.92$. Here, the NO_x level is too close to the legislative limit in order to think about series production realization. Generally, the NO_x emission level of conventional four-stroke engines at lean mixtures is twice as high as that of two-strokes. Two-strokes have high internal EGR values and the EGR rate of this engine is approximately 25% at this operation point. With leaner mixtures $\phi < 0.92$, the NO_x emissions are reduced until $\phi = 0.7$ is reached. Surprisingly, NO_x emissions increase again from $\phi < 0.7$ until the engine reaches the lean misfire limit. This increase is caused by sporadic misfiring. If misfiring occurs in one cycle, the EGR concentration in the next cycle is lower, the mixture becomes richer and, therefore, higher NO_x emissions occur.

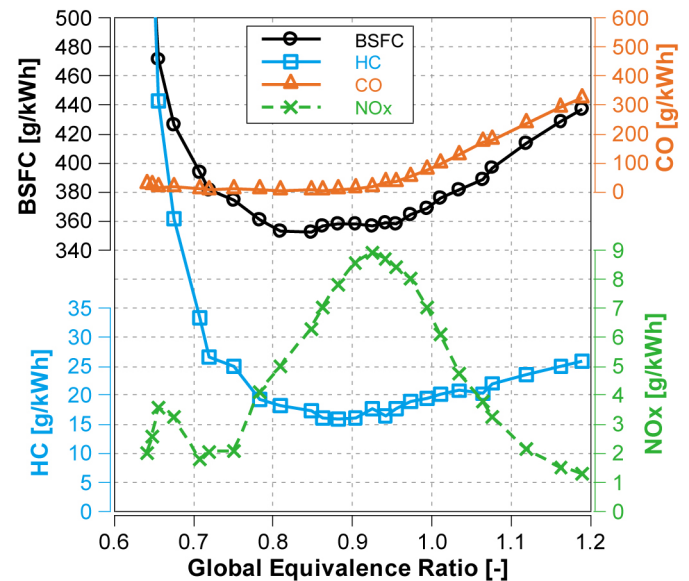


Figure 7. BSFC and emissions on an equivalence ratio at 9500 min⁻¹ WOT (experimental results)

HC and CO emissions as well as BSFC have an almost linear correlation in an equivalence ratio range from 1.2 to 0.95. A trapping efficiency of 94% is reached at $\phi = 1.2$, although HC emissions are further reduced by 35% with a leaner mixture setting. CO emissions decrease from 325 to 30 g/kWh and the BSFC is 18% lower. Within the equivalence ratio range of 0.95 to 0.8 the values of HC, CO and BSFC are almost unchanged. Beginning with $\phi < 0.8$, combustion instabilities and misfiring increase, which causes higher HC emissions. The same trend can be observed for fuel consumption; BSFC increases below an equivalence ratio of 0.8 because of the higher HC emissions, enhanced combustion instability (see COV in Figure 10) and/or misfiring and a lower mechanical efficiency. Figure 8 presents the IMEP, BMEP, FMEP (friction mean effective pressure) and the mechanical efficiency on the equivalence ratio. The mechanical efficiency, which is the quotient of the BMEP to the IMEP, decreases with lower equivalence ratio. The FMEP, which is the difference between IMEP and BMEP, is almost constant (PMEP is lumped into FMEP). The BMEP is essentially influenced by the equivalence ratio and gets very low using lean mixtures $\phi < 0.8$ and, therefore, this is not expedient for WOT operation. Sufficiently high BMEP is reached within an equivalence ratio range of 1.2 - 0.8. The highest BMEP is reached at $\phi \sim 1.08$. The primary reason for the lower BMEP and/or IMEP with leaner mixture is the decreasing volumetric heat value (see Figure 9).

As already shown before, the IMEP (3) can be calculated depending on the delivery ratio, indicated efficiency, and volumetric heat value. These variables plus the spark plug seat temperature are set out in Figure 9.