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ORIGINAL PAPER

Development of a method to investigate the influence of engine oil and its additives on combustion anomalies in hydrogen engines

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Abstract
To counteract the anticipated consequences of climate change, significant efforts are required across all industries. The use of hydrogen in internal combustion engines can make a valuable contribution as a carbon-neutral bridging technology. One challenge in developing a suitable combustion process is the high ignition propensity of hydrogen, impacting combustion stability due to combustion anomalies. Avoiding such combustion phenomena is of utmost relevance to ensure a long-term stable engine operation. Oil entry into the combustion chamber can impact the occurrence of combustion anomalies. Therefore, a specific method for controlled oil entry into the combustion chamber has been developed, and this method is detailed in the following article.

Keywords Hydrogen · Lube oil · Pre-ignition · Dosing methods

Abbreviations

BMEP	Brake mean effective pressure
C	Carbon
CO	Carbon monoxide
CO_2	Carbon dioxide
DI	Direct injection
Exh	Exhaust
HC	Hydrocarbon
HD	Heavy-duty
H_2	Hydrogen
IA	Intake air
LCV	Light commercial vehicle
LOI	Lube oil ignition
MFB05%	Mass fraction burned 5 %
MFB50%	Mass fraction burned 50 %
MFB90%	Mass fraction burned 90 %
PC	Passenger car
PFI	Port fuel injection
PI	Pre-ignition
RH	Relative humidity
rpm	Revolutions per minute
SOHC	Single overhead camshaft
VNT	Variable nozzle turbocharger

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1 Introduction

The (H2) engine can make a significant contribution to achieving global climate goals. Especially, in long-haul and (HD) transportation, as well as in mobile machinery, hydrogen engines may become an alternative in the future due to requirements for range, performance, and robustness. Research activities on this topic have significantly increased in recent years. In most cases, an existing platform is converted and adapted for hydrogen operation. Previously published concepts predominantly rely on spark-ignited combustion with external or internal mixture formation. Both configurations have inherent advantages and disadvantages regarding cylinder filling, mixture formation, raw emissions, H2-pressure requirements, and achievable power density. A crucial goal in mechanical development is the suitability of all components for use with hydrogen to ensure long-term stability and, above all, safe operation. Hydrogen-specific solutions for individual components such as spark plugs, injectors, pressure control and crankcase ventilation are part of intensive research and development. Furthermore, the wide ignition limits and the minimum ignition energy

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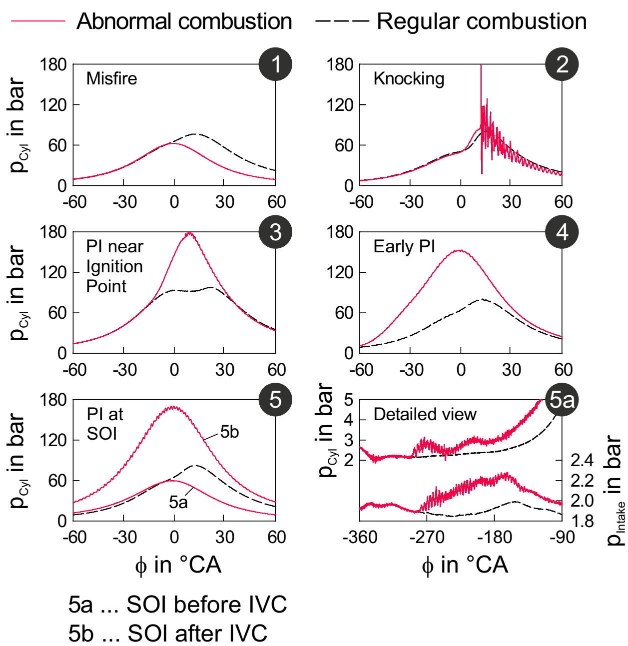

Fig. 1 Different types of abnormal combustion observed on a HD hydrogen engine [2]

of hydrogen are of great importance for use in internal combustion engines [1]. In this context, the combination of low minimum ignition energy and wide ignition limits poses a significant challenge in optimizing a stable combustion process due to combustion anomalies. This unwanted phenomenon can generally be classified into knocking, surface ignition, and pre-ignition [2].

For turbocharged hydrogen engines, the phenomenon of (PI) is primarily a concern and this issue has also become increasingly prominent in downsizing gasoline engines. For this reason, pre-ignitions and their development mechanisms are currently the subject of a wide range of studies [1, 3–8]. Since hot particles and oil droplets are sources of influence for the occurrence of pre-ignition, a new method for assessing oil quality is presented here.

2 Combustion anomalies

The term "combustion anomalies" refers to those combustion cycles that are not (or not exclusively) initiated by the defined spark, but rather arise through autoignition or other sources of ignition. These anomalies can be described as knocking combustion, surface ignition and pre-ignition. Figure 1 shows the pressure curves of the mentioned anomalies. These phenomena primarily differ in their mechanisms of occurrence, the timing within the working cycle and their effects on the combustion process and mechanical components.

Pre-ignition means that the air–hydrogen mixture ignites in the cylinder before the ignition timing without a initial spark ignition. Compared to regular combustion, anomalies are typically of a stochastic nature, characterised by high cylinder pressures and/or unfavourable combustion timings. They contribute to limitations in achieving full load and the calibration parameter space. Moreover, the stochastic occurrence makes it challenging to precisely identify potential mechanisms or triggers [3]. Both pre-ignition and knocking effects lead to a significant increase in the combustion chamber temperature and can thus mutually influence or amplify each other as primary initiators.

2.1 Formation mechanisms of pre-ignitions

In a hydrogen engine, various influencing factors can lead to pre-ignition. Research activities on the mechanisms of pre-ignition have already been conducted in turbocharged gasoline engines, where potential causes for pre-ignition have been described. Optical studies on a full-scale gasoline passenger car engine revealed that ignition occurred at different locations due to randomly drifting droplets (oil, fuel). While ignition at glowing particles could not be ruled out, the most plausible theory emerged as the increased detachment of oil droplets from the cylinder wall film [3, 9].

Diverse studies involving (PFI) and (DI) have already identified several potential initiators for combustion anomalies in hydrogen engines. Aside from the ignition system, previous experiments have illustrated the impact of oil temperature and deposits resulting from the oil [2, 9]. In Fig. 2, alongside the various causes of pre-ignition, the potential sources of engine oil introduction into the combustion chamber are illustrated. Engine oil can enter the intake air through the crankcase ventilation, as well as through the bearings of the turbocharger and the valve stem guides. On one hand, the entry of oil into the combustion chamber can lead to oil deposits [10] and on the other hand, there is a possibility that oil droplets may enter the combustion chamber, potentially causing pre-ignition. This formation hypothesis by (LOI) is shown schematically in Fig. 3.

If these oil droplets are in the temperature range between 200 *and* 300 °C [3], evaporization (a) (Fig. 3) of the more volatile components begins, so that with increasing temperature at the droplet surface, an oil–air mixture (λ_{oil}) gradually forms. Upon reaching the self-ignition conditions of the oil (local temperature and λ_{oil}), oxidation (b) (Fig. 3) and flame propagation occur. If the flame front is not quenched prematurely, the surrounding hydrogen–air mixture ($\lambda_{H_2, local}$) is ignited by the released energy from the oil droplet. However, it is also possible that the oil droplet may change completely to the gas phase (droplet lifetime) before the self-ignition temperature is reached, due to its boiling behaviour, size and/or cycle time .

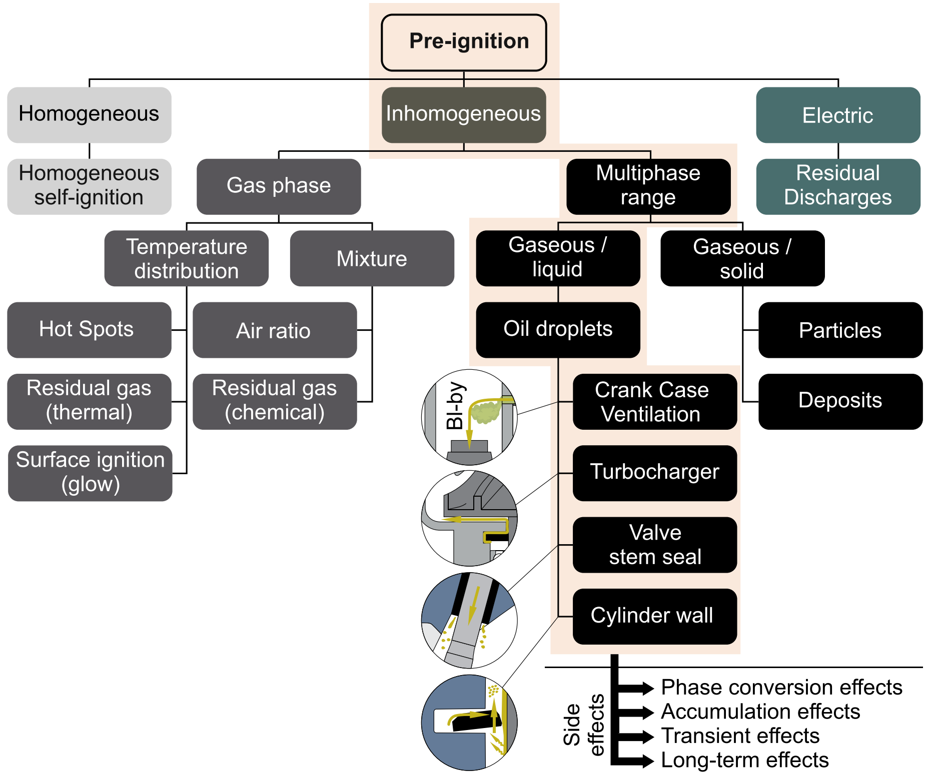

Fig. 2 Overview of irregular combustion phenomena and their possible causes [3, 9]

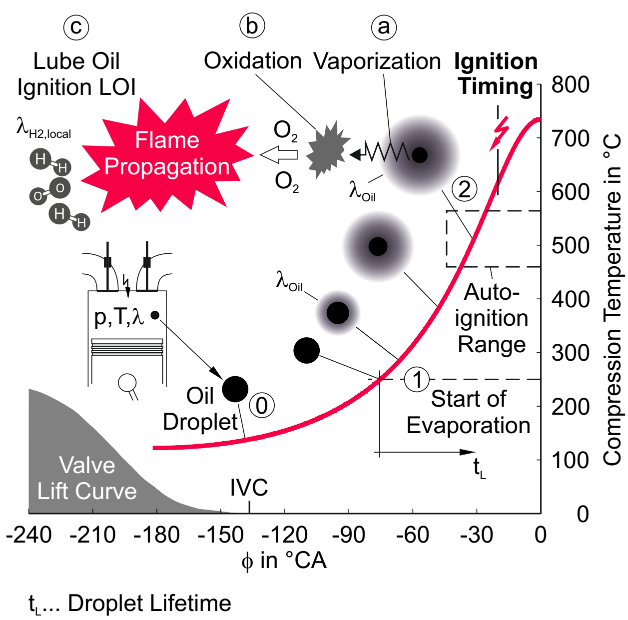

Fig. 3 Hypothesis lube oil ignition

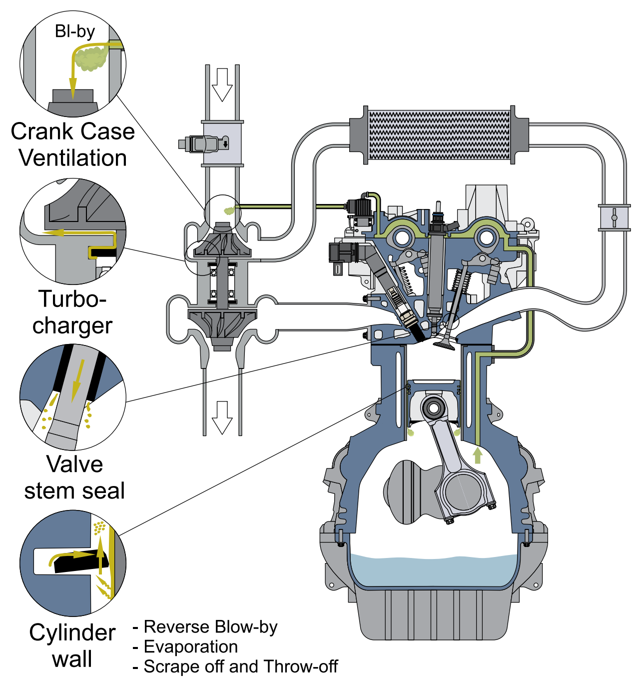

Fig. 4 Potential sources of oil consumption using the example of a turbocharged combustion engine

2.2 Oil consumption and sources

Multiple sources and mechanisms contribute to the loss of engine oil in an internal combustion engine. In Fig. 4, the potential sources of oil consumption in a turbocharged internal combustion engine are schematically depicted. These sources are divided into leakage losses directly escaping from the engine housing, oil losses through valve stem seals and the sealing of the turbocharger lubrication, oil transport through crankcase ventilation due to blow-by and oil consumption off the piston system, consisting of the piston, piston rings, and liner. The underlying mechanisms depend on a variety of engine-specific conditions such as engine design and configuration, load point, or wear condition. Therefore, the precise contribution of each source to oil consumption and the resulting absolute amount cannot be universally stated.

However, it can be reasonably assumed that the valve stem seals of modern engines and the direct escape through lubricant leaks, in a proper and well-maintained condition, contribute negligibly to the total oil consumption. Accordingly, the oil consumption shares are primarily caused by the lubricant supply of the piston system, as well as crankcase ventilation and sealing of the turbocharger lubrication to the exhaust or charge air side.

For a better understanding about the percentage of the individual oil consumption sources, Fig. 5 shows measurement results for three representative steady-state load points (urban, intercity, motorway) based on a passenger car diesel engine. The average load demand of the analysed load points increases in the order shown. The average intercity point has a comparable torque in relation to the urban point, but at a higher engine speed level. In addition to the percentage breakdown of the sources, the percentage change in the overall consumption and the qualitative increase or decrease (arrows) in the individual quantities between the load points are also plotted. The evaluation emphasises that the overall lube oil consumption (LOC) increases with increasing load demand, whereby the main proportion comes from the piston–ring-liner system. The blow-by oil increases at higher engine loads and speeds due to the enhanced blow-by mass flow. The proportion of lube oil consumption caused by the turbocharger lubrication is highest in the low-load range. A possible explanation for this is the unfavourable pressure conditions in the turbocharger at low-load operation.

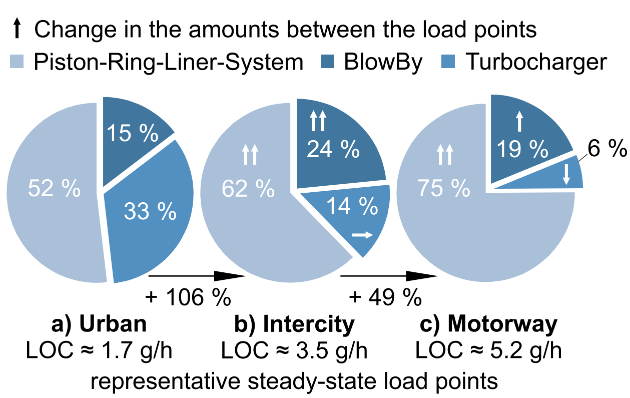

Fig. 5 Percentage breakdown and change in oil consumption shares for representative steady-state load points based on measurements on a passenger car diesel engine: **a** Urban; **b** Intercity; **c** Motorway [11]

2.2.1 Oil consumption through the piston–bore interface

The dominant share of oil consumption originates from the lubrication of the piston–ring package/cylinder liner (see Ref. [11]). The engine oil is introduced into the combustion chamber in gaseous or liquid form. The main responsible effects for transport are the evaporization of oil on the hot cylinder wall, the throw off of oil accumulations, scraping off the oil film from the cylinder wall, and gas backflows through the ring gap into the combustion chamber ("reverse blow-by") under appropriate pressure differences. The evaporization rate is significantly influenced by the liner temperature, with this effect increasing especially with rising load in relation to the total transport. In addition, the type of oil has a substantial impact on evaporization behaviour. At higher engine speeds, in addition to the evaporization rate, the acceleration forces acting on oil accumulations increase, reinforcing axial transport towards the combustion chamber. In contrast, in the lower load and speed range (idle range), unfavourable pressure conditions contribute to increased oil transport through enhanced backflows during the intake stroke [11–13]. Transient effects and throw off of larger oil accumulations can also lead to temporarily increased oil entry. The design of the piston system and its runtime-dependent wear are further influencing factors on oil transport in this area.

2.2.2 Oil loss through blow-by

Blow-by gases flow from the combustion chamber against the direct oil transport into the crankcase, accumulating in the piston–ring package with evaporized and liquid engine oil. In turbocharged engines, oil-containing gases enter the crankcase through the oil return of the turbocharger lubrication, further increasing the blow-by flow. In the crankcase, additional enrichment of blow-by gases with oil can occur due to splash oil, oil leaks from sliding bearings, or the throw off of engine oil on moving components. With increasing blow-by flow, the potential for increased entrainment of oil from various areas rises. The entire blow-by flow (cylinder + turbocharger) is mixed with fresh air through crankcase ventilation and returned to the combustion chambers. The size spectrum of the recirculated oil droplets extends into the submicron range. A passive or active separation system integrated into the ventilation path filters out a portion of the oil droplets from the blow-by raw gas.

2.2.3 Oil consumption through the turbocharger sealing

A frequently mentioned cause in the literature for oil leaks from the turbocharger lubrication is the pressure gradient between the oil space in the bearing housing and the fresh air or exhaust path. An existing external overpressure leads to oil discharge through the non-contact shaft seal, often equipped with labyrinths. In addition, pressure-based oil discharge can periodically intensify due to pressure fluctuations over an engine cycle. Gas transport can occur on the compressor and turbine sides, both as blow-by towards the crankcase and in the opposite direction. Another aspect concerns the pressure conditions in the wheel-side space, the gap between the compressor or turbine wheel and the housing. Due to gas rotation in this gap, the static pressure decreases from the outer diameter of the impeller to the shaft. Consequently, depending on the compressor speed and pressure levels in the fresh air or exhaust path, as well as in the crankcase, a vacuum at the sealing ring and an associated suction effect for the oil can occur [11, 13, 14]. In Ref. [14], another noteworthy transport mechanism for oil leaks is assumed. It is suggested that engine oil is pressed past the sealing ring (oil squeezing) due to vibration movements of the turbocharger shaft. Increased wear can favour the mentioned transport processes over the engine operational life [11, 14, 15].

3 Oil dosing method

The objective was to develop a methodology to systematically investigate the influence of oil droplets from different engine oils (base oils, additives, aged conditions) on their pre-ignition propensity in a comparable manner. Figure 6 shows possible influencing factors for oil-related pre ignitions.

With this setup, it should be possible to generate oil droplets and introduce them into the combustion chamber over a very short path. To make a qualitative assessment through a statistical analysis of the influence of oil droplets, it is necessary to administer a minimum quantity of oil. To determine the optimal approach for introducing oil droplets into the combustion chamber, four different variants were compared with each other (Fig. 7).

In the continuous dosing method (Fig. 7), two options for oil dosing were identified. The first option involves introducing oil droplets into the intake air using an atomizer aerosol generator and the second option is a dosing peristaltic pump. With the dosing peristaltic pump, it is possible to dose an oil droplet into the intake air at defined time intervals. The comparison of the variants revealed that employing a sequential intake manifold injection, as well as low-pressure and high-pressure direct injection of oil (Fig. 7), involves significant complexity [16, 17]. In addition to the cost-intensive aspects and potential thermal changes due to the influence on the water jacket, this variants also alters the combustion chamber geometry, potentially increasing the tendency for pre-ignition. For these reasons, the low-pressure and high-pressure direct oil injection (Fig. 7) was not considered for our investigations. Since the continuous method with the aerosol generator and dosing peristaltic pump allowed for the same dosing position and to determine the advantages and disadvantages of these methods, both were examined on the engine test bench using two different test engines.

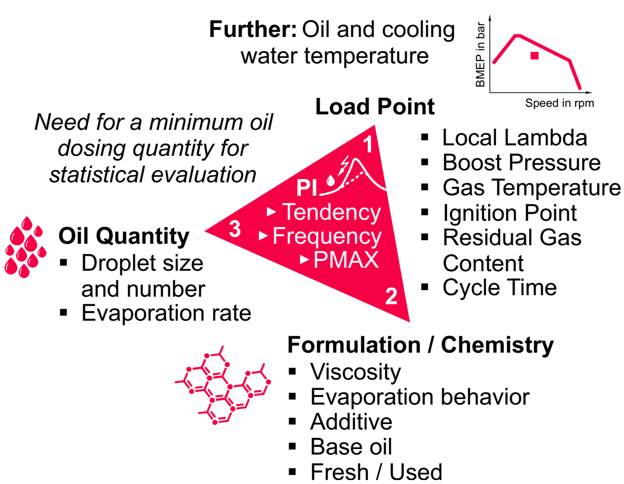

Fig. 6 Influences on oil-related pre ignitions

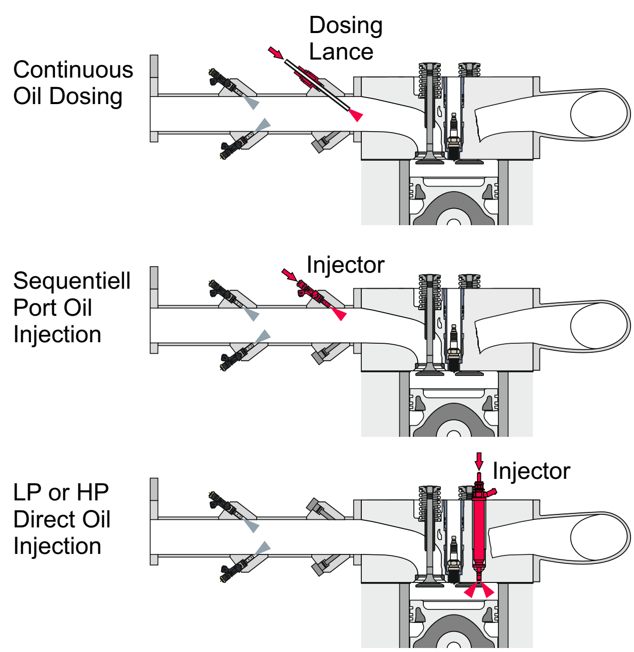

Fig. 7 Comparison of the different dosing options

The aerosol generator from Topas (Table 1), which was used, provides the capability to adjust the oil quantity via the dosing pressure and the temperature of the oil in the oil reservoir.

The dosing peristaltic pump allows for dosing in the range of 2–150 ml/min with a maximum back pressure of 2.5 bar. This method enables the precise definition of an oil quantity that remains unaffected by factors such as oil temperature and viscosity. In both options, variations in the load point additionally result in different changes in parameters, such as lambda, intake manifold pressure, residual gas content, compression end temperature, spark timing, and cycle time.

3.1 Experimental setup

The experimental testing was conducted on two different test rigs. First, a 2-L four-cylinder gasoline engine was used as a powertrain for passenger cars (PCs) and light commercial vehicle (LCV). Second, a 12.8-L 6-cylinder (HD) engine was employed. Both engines were modified for hydrogen operation through adjustments to the charging system, injection, and ignition system [19, 20] (Tables 2, 3).

A significant difference between the two test engines is the mixture formation. The PC engine features internal mixture formation, while the HD engine has external mixture formation. In addition, for both engines, a dosing lance was integrated into the intake manifold at cylinder 4 (Fig. 8) to introduce oil droplets as close to the valve as possible, minimizing the likelihood of wall deposits.

Table 1 Technical data TOPAS ATM 210 H [18]

TOPAS atomizer aerosol generator ATM 210 H	
Aerosol output	500–2500 l/h
Mass flow rate	20 g/h
Filling quantity	10–80 ml
Particle number concentration	> 108 /cm^3
Particle size (median value)	0.1–0.5 µm
Pressure drop atomizer nozzle	1.5–6 bar
Maximum back pressure	10 bar
Compressed air supply	Max. 15 bar

Table 3 Engine data for HD engine

Engine data	
Base engine	6 cylinder inline engine
Bore × stroke	130 mm × 161 mm
Displacement	12,822 cm^3
Compression ratio	10.2
Injection	Series gasoline injectors
Ignition	Modified ignition coil and modified racing spark plugs M14
Cam phasing	SOHC, 4 valves per cylinder
Charging system	VNT turbo charger
Engine control unit	AVL RPEMS
Additional	AVL VISIO

3.1.1 Dosing method using aerosol generator

In this dosing method (shown in Fig. 9), the dosing lance and the atomizer aerosol generator are connected to each other via a short hose. At the transition between the hose and the lance, the pressure and temperature of the aerosol are measured. A temperature sensor has been integrated into the oil vessel of the aerosol generator to regulate the oil temperature in the vessel. In addition, to heat the oil, a heating mat has been mounted around the oil vessel. The atomizer requires a supply of compressed air, which is pre-regulated to 6 bar using a pressure regulator. Between the pressure regulator and the aerosol generator, a mass flow meter and a valve for controlling the air supply are installed to release or stop the dosing.

3.1.2 Dosing method using dosing peristaltic pump

In contrast to the dosing method with the aerosol generator, the lance is connected directly to the dosing peristaltic

Table 2 Engine data for PC engine

Engine data	
Base engine	4 cylinder inline engine
Bore × stroke	83 mm × 92 mm
Displacement	1991 cm^3
Compression ratio	9.8
Injection	Series gasoline injectors HDEV4 (H2 DI with up to 175 bar)
Ignition	Modified series ignition coil and racing and prototype spark plugs
Cam phasing	Series cam phasers (intake and exhaust)
Charging system	Series diesel turbo charger (with VNT) optional: additional e-compressor
Engine control unit	Prototype (motorsports)

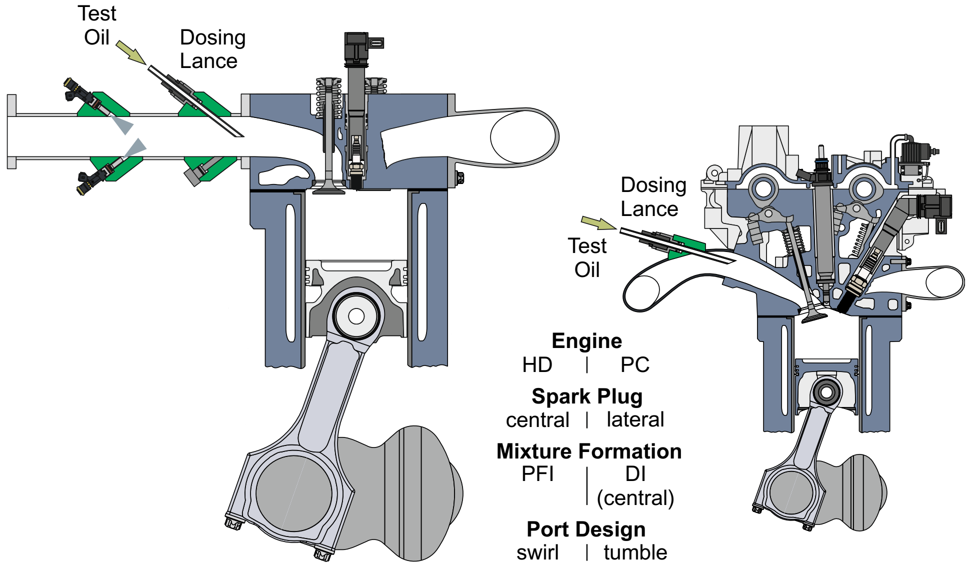

Fig. 8 Comparison between HD and PC engine

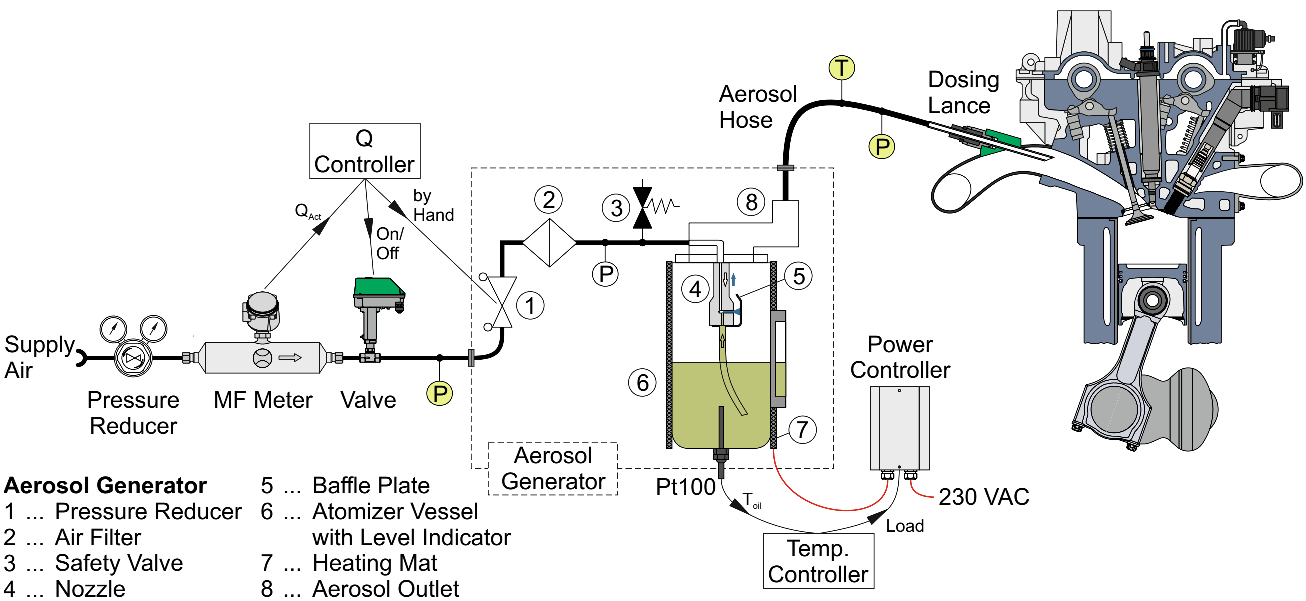

Fig. 9 Test setup for the dosing method using aerosol generator

pump via a short hose. In addition, there is an oil vessel from which the dosing peristaltic pump draws the oil. The pump can be controlled directly through the test bench automation system, allowing for precise oil dosing (Fig. 10).

3.2 Experimental procedure

The objective was to select an operating point where the hydrogen engine runs stably with minimal combustion anomalies, and the compression end temperature is within the range of oil autoignition temperature. In addition, the time for the evaporation of oil droplets should be short, favouring higher speed ranges. For this reason, experiments were initially conducted at different load points and speeds. It was found that on the PC engine the operating point of 3000 rpm and an BMEP of 16 bar is well-suited for oil dosing experiments. This is because the compression end temperature is in the range of 500–550 °C, aligning with the autoignition temperature window of oil droplets. Similarly, various operating points were investigated in advance on the HD engine to find an ideal operating point for oil dosing experiments. Through these investigations, the operating point of 1700 rpm and an BMEP of 12 bar were defined for the HD engine. This is because, in this case as well,

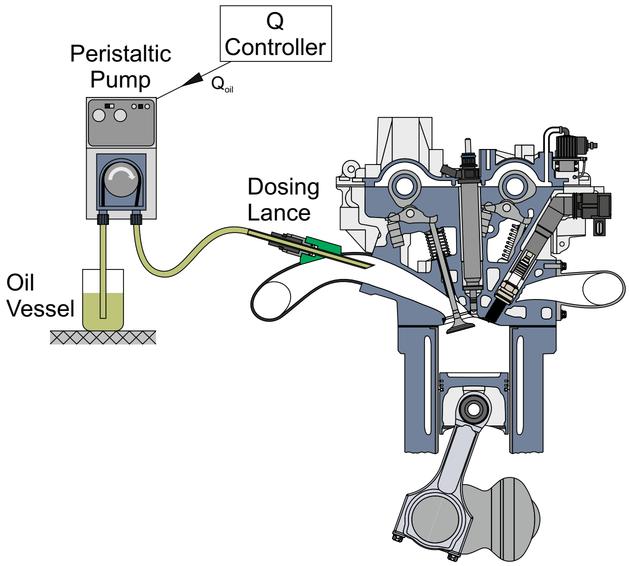

Fig. 10 Test setup for the dosing method using dosing peristaltic pump

Table 4 Defined parameters for dosing tests

	PC engine	HD engine
Oil temperature	25 °C	45 °C
Aerosol–air temperature	35 °C	37 °C
Intake air temperature	30 °C	25 °C
Dosing pressure	4.0 bar	3.2 bar

the compression end temperature is in the same range like the PC engine and it runs almost free of pre-ignition. To make a qualitative assessment of the number of combustion anomalies before and after oil dosing, a test run with three phases of 20 min each was defined. The first phase serves for preconditioning, the second 20 min involve oil dosing, and the third phase is for post-conditioning.

3.2.1 Test procedure dosing method using aerosol generator

It was additionally necessary to conduct experiments with different oil temperatures in the vessel and various dosing pressures. From these experiments, it became apparent that, for the PC engine, a dosing pressure of 4 bar and an oil temperature in the vessel of 30 °C are most suitable for the tests. For the HD engine, a dosing pressure of 3.2 bar and an oil temperature of 40 °C were found to be the conditions for the investigations. The defined parameters for both engines are shown in Table 4.

To ensure that no residues from the dosed oil remain in the dosing lance, the system is disassembled and thoroughly cleaned after each experiment. After cleaning, the oil vessel in the atomizer is filled and weighed. Subsequently, the setup is reassembled and a warm-up program is initiated. Once the engine reaches the defined test values, it is brought to the selected operating point, and the test run is started. Afterwards the test run, a motored measurement is performed at the chosen speed before the engine is shut down directly and the oil dosing unit is disassembled. Following this, the oil vessel is weighed again to determine the dosed oil quantity gravimetrically. This entire procedure is repeated for the next experiment.

3.2.2 Test procedure dosing method using dosing peristaltic pump

With this method, the exact dosing quantity for the tests had to be determined, which can be set with the speed of the dosing peristaltic pump. The dosing quantity for the PC engine was defined as approximately 0.3 g/kWh and 0.45 g/kWh for the HD engine. These values represent a realistic oil consumption for engines with increased wear . Before the test run, the lance, including the hose, is disassembled and filled or pumped through with the oil under investigation. Thus, the system is filled with the oil to be tested up to the top of the lance and, in this state, is installed in the intake manifold. The oil vessel is gravimetrically weighed. The same procedure is then carried out until the engine stops as for the dosing method with the aerosol generator. After the test, the oil vessel is gravimetrically weighed again, the lance is removed and the described process is repeated with the next test sample.

3.3 Test evaluation

In addition to defining the test run a measurement methodology focusing on the oil dosing tests was established and an evaluation tool was developed (Fig. 11).

The data are recorded throughout the entire test run at a rate of 10 Hz using the test bench automation program to monitor both, the experiment and the test engine, to ensure the plausibility of the test runs. From these measurement data, a temporal profile of oil consumption is also recorded and using the in Chapter 2 described carbon balance, for calculating the dosed oil quantity .

The indicating system continuously records fast data throughout the entire test run. This provides cyclic data for determining combustion anomalies from the entire cycles of a measurement sequence. In the subsequent analyses, the maximum cylinder pressure and the mass fraction burned 5 % (MFB05%) were used for each cycle, thus filtering out deviations caused by any anomalies. Therefore, it is possible in retrospect to present the exact number of pre-ignitions in each measurement phase and to validate the tests.

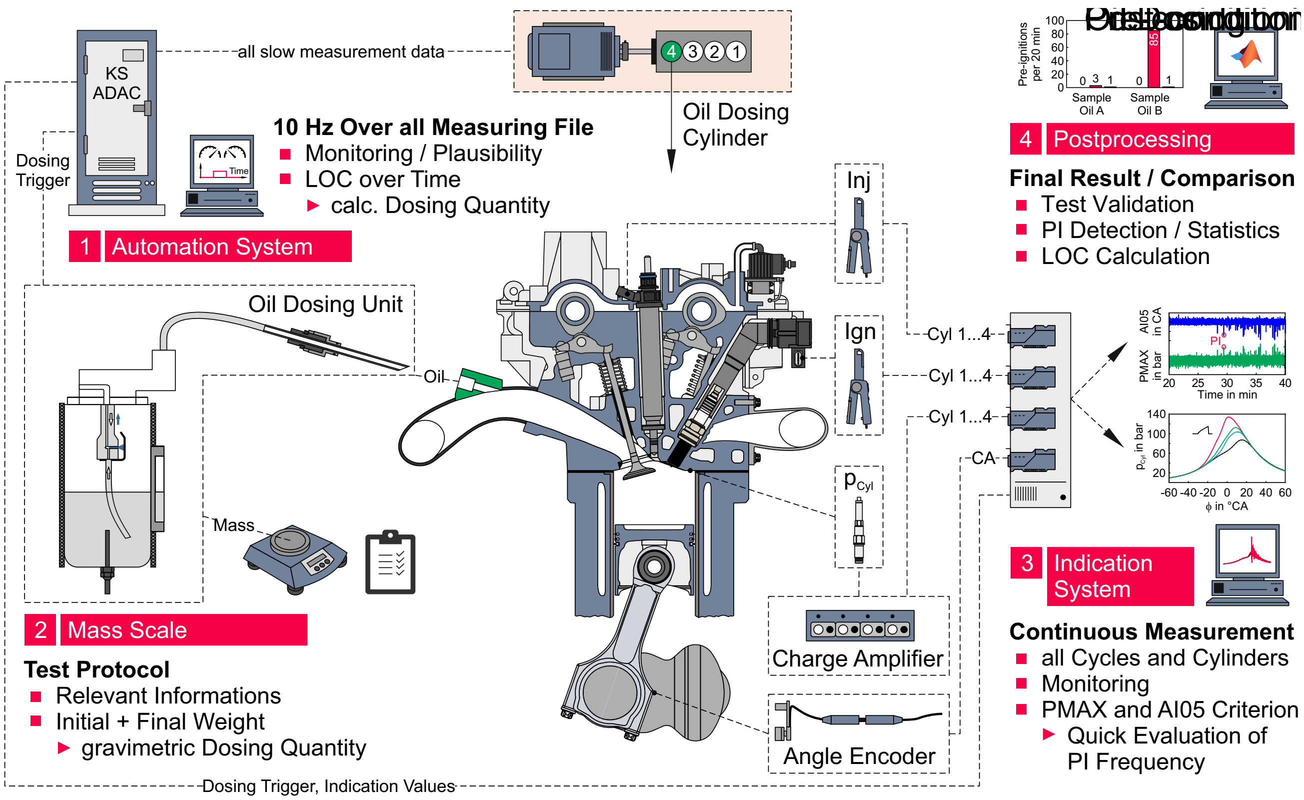

Fig. 11 Measuring methodology and evaluation focus on oil dosings tests

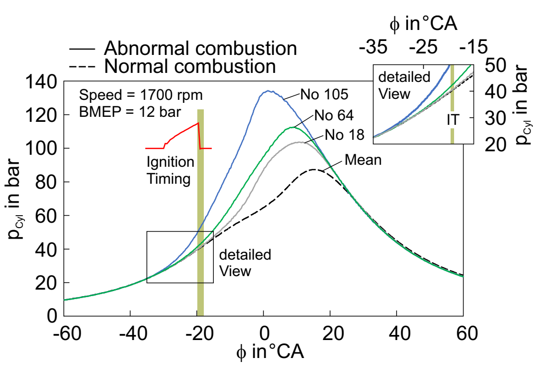

Fig. 12 Damage potential of different combustion anomalies

In addition to the propensity for oil-induced pre-ignitions, the developed analysis tool allows differentiation between pre-ignitions that exhibit higher and lower levels of damage potential. The damage potential is highly dependent on the timing of the ignition onset (peak pressure and temperature). To account for this factor, a qualitative classification into "stronger" and "more moderate" pre-ignitions is made based on the MFB05%, with the spark timing serving as the boundary for the classification. In Fig. 12, the damage potential of different combustion anomalies, the crank angle-resolved cylinder pressure traces for a regular combustion (mean) and three oil-induced pre-ignitions (No. 18, 64, 105) are plotted, along with the spark timing. For the "stronger" pre-ignition No. 105, the MFB05% occurs significantly before the spark timing, resulting in an increase in peak pressure by up to 45 bar compared to regular combustion. In contrast, the ignition onset for the two "more moderate" pre-ignitions (No. 18 and 64) is noticeably later, leading to a lower level of damage potential to the crank mechanism. Furthermore, these very late pre-ignitions, like No. 18, were exclusively detected under the influence of engine oil. Droplet ignition is often cited in the literature as a possible triggering mechanism for oil-induced pre-ignitions [21, 22]. This would explain the delayed ignition due to the absence of self-ignition conditions for the oil at an earlier point in time. The compression temperature plays a major role here. Pre-ignitions triggered by hot spots or hot particles are typically induced much earlier [23]. However, it can also occur that, due to the consequential effect of an initial oil-induced pre-ignition, such as the significant heating of the combustion chamber or flanking deposits, a subsequent series of secondary pre-ignitions can occur, complicating the separated evaluation of the actual triggering mechanisms and the statistical comparison of different test oils.

3.3.1 Calculation and normalization

To ensure the verification of oil consumption during dosing, in addition to the gravimetric oil quantity, the oil consumption is calculated using the carbon balance to confirm that the oil reaches the combustion chamber. In the case of the carbon-free fuel hydrogen, it is appropriate to establish a carbon balance across the engine to determine the engine's oil consumption. In addition, this provides the opportunity to calculate the dosed oil quantity in the experiments. The system boundary for the balance equation of the incoming and outgoing mass flows, as depicted in Fig. 13, has been drawn. This requires the introduced fuel quantity, (CO2) measurement of the (IA) and (Exh), as well as the non-fully oxidised carbon-containing components produced during incomplete combustion in the exhaust, (CO), and (HC). With these input parameters, it is now possible to determine both the oil consumption of the entire engine and the dosing quantity.

The conducted gravimetric measurement of the dosed oil allows the calculation of the dosed oil to be verified through the carbon balance. In general terms, the mass balance can be represented as shown in the following equation:

$$\dot{m}_{\mathrm{IA}} + \dot{m}_{\mathrm{Fuel}} + \dot{m}_{\mathrm{Oil}} = \dot{m}_{\mathrm{Exh}}, \tag{1}$$

which contains the mass flow of the intake air ($\dot{m}_{\mathrm{IA}}$), fuel ($\dot{m}_{\mathrm{Fuel}}$), lube oil consumption ($\dot{m}_{\mathrm{Oil}}$) and engine exhaust gas ($\dot{m}_{\mathrm{Exh}}$). In Eq. (2) the mass balance is extended with their mass concentration as expressed as

$$\begin{aligned}\dot{m}_{\mathrm{Exh}} \cdot \mu_{\mathrm{C}_{\mathrm{Exh,w}}} = \dot{m}_{\mathrm{IA}} \cdot \mu_{\mathrm{C}_{\mathrm{IA,w}}} + \dot{m}_{\mathrm{Fuel}} \cdot \mu_{\mathrm{C}_{\mathrm{Fuel}}} \\ + \dot{m}_{\mathrm{Oil}} \cdot \mu_{\mathrm{C}_{\mathrm{Oil}}},\end{aligned} \tag{2}$$

which can be used to determined the oil consumption. The use of carbon-free hydrogen eliminates the mass fraction Eq. (4) of the fuel fraction. In addition, substituting Eq. (1) for $\dot{m}_{\mathrm{Exh}}$ gives the oil consumption as

$$\dot{m}_{\mathrm{Oil}} = \frac{(\dot{m}_{\mathrm{IA}} + \dot{m}_{\mathrm{H}_2}) \cdot \mu_{\mathrm{C}_{\mathrm{Exh,w}}} - \dot{m}_{\mathrm{IA}} \cdot \mu_{\mathrm{C}_{\mathrm{IA,w}}}}{\mu_{\mathrm{C}_{\mathrm{Oil}}} - \mu_{\mathrm{C}_{\mathrm{Exh,w}}}} \tag{3}$$

with

$$\mu_{\mathrm{C}_{\mathrm{Fuel}}} = \mu_{\mathrm{C}_{\mathrm{H}_2}} = 0. \tag{4}$$

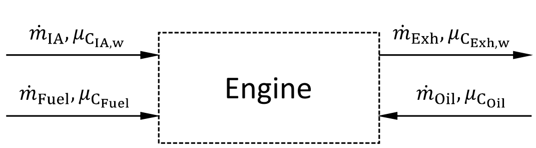

Fig. 13 Carbon balance equation

The oil quantity can now be calculated using the carbon-containing concentrations, which can be measured wet or dry depending on the analyzer. In order to draw up the balance, all concentrations are calculated on a wet basis (with indices w), for which the air–fuel equivalence ratio (λ), the relative humidity of the intake air ($\mathrm{RH}_{\mathrm{IA}}$) and the pressures and temperatures from the inlet and outlet path are required .

The input variables shown in Eqs. (5–8) are now used to determine the oil quantity ($\dot{m}_{\mathrm{Oil}}$) in g/h. To accomplish this, the mass fraction of CO2 is continuously recorded in both, the intake air and exhaust gas, along with the concentration of CO and HC in the exhaust gas. With the molar masses of the intake air ($M_{\mathrm{IA,w}}$), exhaust gas ($M_{\mathrm{Exh,w}}$) and carbon (M_{C}) the respective concentrations are defined as

$$\mu_{\mathrm{C}_{\mathrm{Oil}}} \approx 0.85, \tag{5}$$

$$\mu_{\mathrm{C}_{\mathrm{IA,w}}} = \frac{M_{\mathrm{C}}}{M_{\mathrm{IA,w}}} \cdot \nu_{\mathrm{CO}_{2_{\mathrm{IA,w}}}} \tag{6}$$

and

$$\mu_{\mathrm{C}_{\mathrm{Exh,w}}} = \frac{M_{\mathrm{C}}}{M_{\mathrm{Exh,w}}} \cdot \nu_{\mathrm{C}_{\mathrm{Exh,w}}}, \tag{7}$$

where

$$\nu_{\mathrm{C}_{\mathrm{Exh,w}}} = \nu_{\mathrm{CO}_{2_{\mathrm{Exh,w}}}} + \nu_{\mathrm{CO}_{\mathrm{Exh,w}}} + \nu_{\mathrm{HC}_{\mathrm{Exh,w}}}. \tag{8}$$

The amount of oil dosed in the test sequence is determined by comparing the relative change in engine oil consumption during phases with and without oil dosing.

In addition to calculating oil quantities using the carbon balance, the number of combustion anomalies in each of the three phases of each test run was analysed, where specific criteria (max. cylinder pressure, MFB05%) were met. This allows a quick assessment of the tendency of each test sample. To standardise the number of pre-ignitions, it is helpful to consider the dosed oil quantity and normalise to a defined number of engine cycles. In the following examples (Sect. 4) it was based on 200,000 engine cycles as the following expression describes:

$$\mathrm{PI}_{\text{oil-rate}}\left[\frac{1}{\mathrm{g}_{\mathrm{Oil}} \cdot \#\mathrm{cycles}}\right] = \frac{\mathrm{PI}}{m_{\mathrm{Oil}}} \cdot \frac{\frac{n_{\mathrm{engine}}}{2} \cdot t_{\mathrm{dosing}}}{\#\mathrm{cycles}_{\mathrm{dosing}}}. \tag{9}$$

Hence, it is possible to calculate the number of PIs per gram of dosed oil and make a qualitative statement about the different oils.

Table 5 Oil samples

Coding	Unit	Sample A	Sample B
Base oil group		4	5
Kinematic viscosity @ 40 °C	mm^2/s	66.2	30.4
Kinematic viscosity @ 100 °C	mm^2/s	10.4	6.3
Calcium	ppm	0.7	1330
Magnesium	ppm	1228	0.6
Density	g/cm^3	0.85	0.95
Flash Point	°C	237	239

4 Results

FUCHS LUBRICANTS GERMANY GmbH provided us with two significantly different oils, sample A and B, for the tests on both test engines, which we used to validate the two methods (Table 5).

The initial measurements were conducted on the PC engine using the aerosol generator dosing method. This approach revealed that minor fluctuations in oil temperature within the vessel and disparities in viscosity between the two samples had a notable impact on the dosed oil quantity. In addition, two consecutive test runs, utilizing the same sample, demonstrated a considerable difference in the time elapsed before a significant increase in oil quantity was recorded. The first test (depicted in red in Fig. 14) exhibited a prolonged period before measurable oil quantity elevation post-dosing initiation, contrasting with the immediate increase observed in the subsequent test (depicted in blue in Fig. 14). The reason for this difference is probably that the aerosol hose was not lubricated with oil during the first test. This required more time for the wall film to form, unlike the second test where residual oil from the initial test helped to lubricate the aerosol hose more quickly.

Figure 14 clearly demonstrates the significant influence of oil temperature on the dosing method of the aerosol generator. In the initial test, the aerosol temperature is slightly elevated, resulting in decreased oil viscosity. Consequently, a greater quantity of oil is introduced into the cylinder, complicating direct comparisons between the two measurements. In addition to temperature's influence, the oil tests involve the use of different oils with varying viscosities. To address this variability, standardization of anomalies with respect to the dosed oil quantity over a specified number of cycles, as outlined in Chapter 2, was implemented. This standardization facilitates comparisons between different oils when utilizing the aerosol generator dosing method. The base oil consumption of the engine is shown in Figs. 14 and 15, as depicted in the bottom diagram in brown.

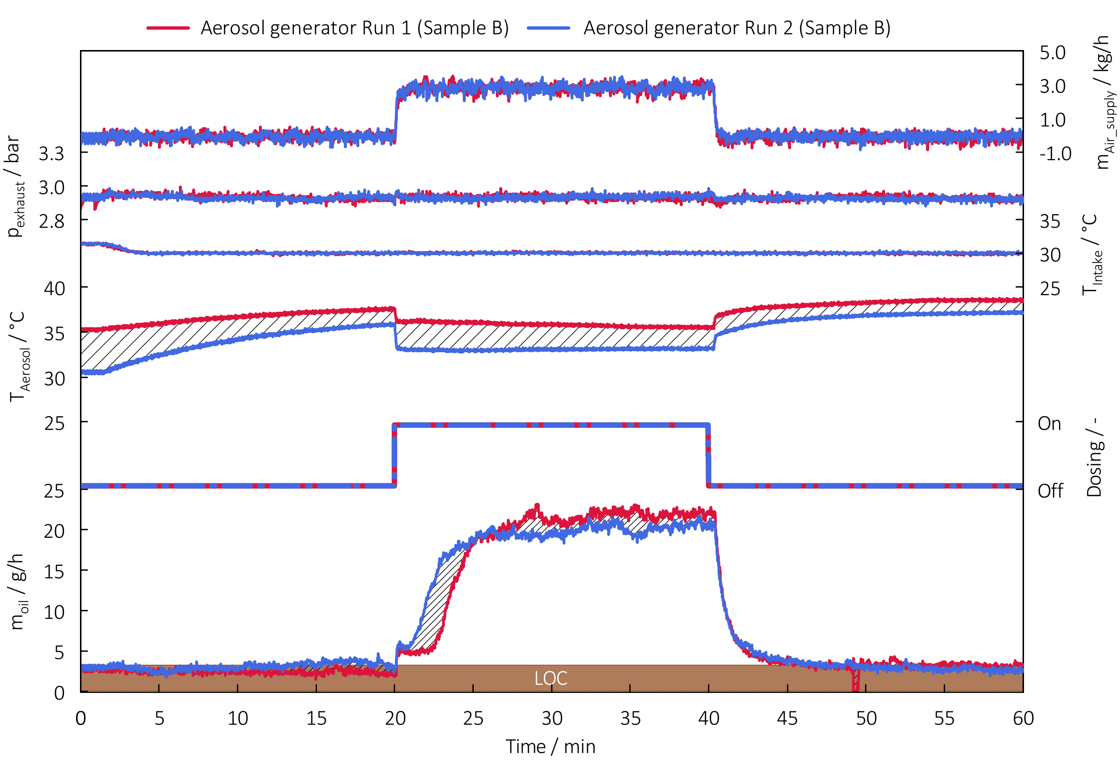

Fig. 14 Comparison of two test runs with the same oil sample and with the aerosol generator dosing method

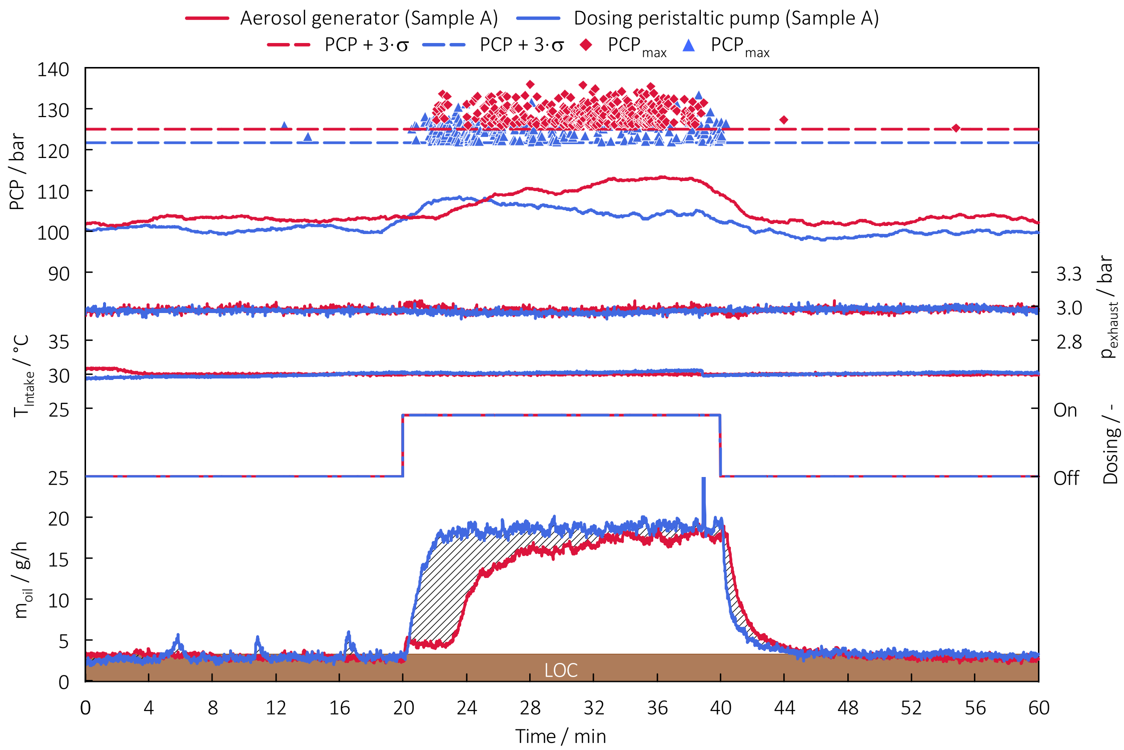

Fig. 15 Comparison of the two dosing methods with the same oil sample

When comparing the two dosing methods in terms of the amount of oil dosed (as shown in Fig. 15), it is evident that the peristaltic pump method (depicted in blue) results in an immediate increase in oil quantity compared to the aerosol generator dosing method (depicted in red). The streamlined area in the bottom of Fig. 15 illustrates the discrepancy in the dosed oil quantity between the two methods. Furthermore, it is evident that the peristaltic pump dosing method maintains a nearly constant oil quantity throughout the entire dosing period, unlike the aerosol generator dosing method.

By repeating the tests with the dosing peristaltic pump several times, an almost identical oil consumption could be determined gravimetrically and with the carbon balance each time. With this method, the oil quantity is not influenced by the viscosity or the intake air temperature.

Furthermore, in Fig. 15, the top graph illustrates the peak cylinder pressure of both test runs, as well as the respective threshold for recording anomalies, defined as the peak pressure plus three times the standard deviation, and the individual maximum peak pressures of cycles exceeding the defined threshold. The evaluation of all anomalies includes establishing boundaries for MFB05%, MFB50%, and MFB90%, using three times the standard deviation of each parameter. This enables not only the filtering of pre-ignitions, but also the identification of anomalies caused by additional ignition sources after the initial ignition, such as oil droplets [21]. Figure 15 shows that there are few anomalies with peak pressure above the defined threshold during the preconditioning and post-conditioning phases. However, as soon as oil dosing begins, the number of anomalies increases significantly. These evaluation methods allow for the comparison and assessment of different oils under constant operating conditions.

As the dosing peristaltic pump makes it possible to dose the same amount of oil at different intake air temperatures, three test runs were measured at different temperatures. During the measurements, the temperature of the oil in the dosing lance was indirectly increased. In the pre-conditioning phase, the dosing lance is filled with oil and protrudes into the intake manifold, causing it to reach the same temperature as the intake air.

Figure 16 shows how the intake air temperature affects the dosing method, which in turn is influenced by the oil temperature and viscosity. To ensure accurate oil tests, it is important to maintain constant conditions and minimize external influences. Therefore, the dosing peristaltic pump is a suitable method.

Several influencing factors are considered responsible for the increase in pre-ignition events with increasing intake manifold temperature. The rise in intake manifold temperature leads to an increase in the compression end temperature

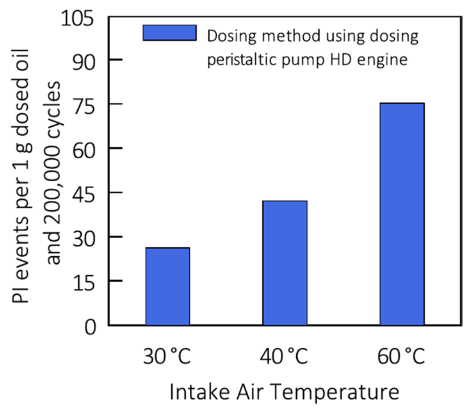

Fig. 16 Variation of the intake air temperature

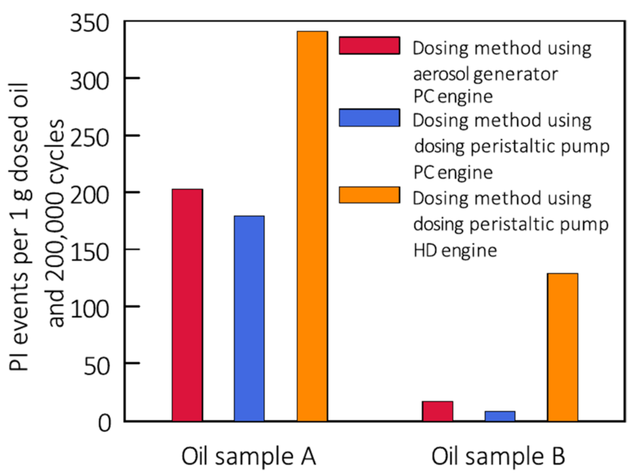

Fig. 17 Normalized results of the tested oils on PC and HD engine

and thus to improved autoignition conditions for the dosed engine oil. In addition, the oil droplets are heated to a higher temperature level before entering the combustion chamber, which subsequently affects the viscosity and later evaporation behaviour.

Figure 17 shows that, on one hand, the two different dosing methods did not result in the same number of combustion anomalies. However, it can be seen very clearly that the trend of the oils tested is similar for the two different variants. In other words, in the tests on the PC engine with sample A, a significantly higher number of events was recorded with both variants than with sample B.

As the tests on the PC engine showed that the aerosol generator dosing method requires considerably more effort and that it is not independent of viscosity to achieve almost the same amount of oil, we decided to investigate only the dosing method with the perestaltic pump on the HD engine for random tests.

A direct comparison of the dosing method with the dosing peristaltic pump between the HD and PC engines shows that the ratio between the oils tested changes, which in turn can be attributed to the absolute number of combustion anomalies. These differences in the absolute number are due to several differences between the two test engines , such as the engine geometry, the intake port geometry, the mixture formation, the position of the dosing, the exhaust back pressure, the engine speed, the compression and temperature, the charge air temperature, the flow velocity and possibly other influences in combination with the properties of the different oil samples. In addition, the definition of the limits for recording anomalies is not directly comparable between the PC and HD engines, which in turn can also contribute to the differences in the number of combustion anomalies. However, it can be seen from the comparison between the PC and HD engine that the trends for combustion anomalies are similar between the oils. Both methods can be used to make reliable statements about the tendency of an oil-induced combustion anomaly under certain boundary conditions between the two test engines. This information can be used to optimize engine oil .

5 Conclusion and outlook

With the presented dosing methods, it is possible to investigate the influence of oil-induced combustion anomalies on different highly charged hydrogen engines. Specifically, various engine oils with their specific compositions can be validated for their tendency to pre-ignition. Consequently, it is possible to infer their damage potential based on pre-ignitions and use the obtained results for optimizing engine oil. Comparable results were achieved with both methods, whether applied to the PC engine or the HD engine. The significantly more complex and elaborate aerosol generator dosing method revealed that slight differences in viscosity, dosing pressure, and oil temperature in the vessel have a substantial impact on the dosed oil quantity. Therefore, with this method, achieving the exact same oil quantity without altering these parameters is not possible. However, this was not desired in our studies, as comparability was only possible with the same parameters. Nevertheless, normalizing oil-induced pre-ignitions per gram of dosed oil over 200,000 cycles allowed for qualitative assessments despite varying oil quantities. In the much less complex dosing peristaltic pump method, the temperature and viscosity of the oil had no effect on the dosed oil quantity. Here, it was possible to dose the same oil quantity with all samples without parameter changes. From these studies, it can be concluded that the dosing method influences droplet size and, consequently, the LOI, leading to the recording of different numbers of oil-induced pre-ignitions. However, the tendency of the oils, both on the HD and PC engines, remained the same. This suggests that, despite the higher costs of studies on the commercial vehicle engine, it is indeed worthwhile to conduct oil investigations like these on the PC engine.

At the current stage, it can be stated that both methods are suitable for fundamental investigations of different oils regarding oil-induced pre-ignitions. However, from our point of view, the dosing peristaltic pump currently makes more sense for pure oil tests, as it is possible to dose the same amount of oil with different oils and their properties. This enables good comparability of the tests with regard to the combustion anomalies that occur. For the future, optical examinations are planned on a laboratory setup under similar operating conditions for both dosing methods to gain insights into the precise droplet introduction. Further optimizations of the dosing lance for both methods and the inlet section for the aerosol generator dosing method will be investigated in this test setup. In addition, experiments with different exhaust back pressures are planned for both variants to examine the influence on oil-induced pre-ignitions.

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Data availability No data sets were generated or analysed during the current study.

Declarations

Conflict of interest The authors declare that they have no Conflict of interest.

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