21PFL-0654

Thermal Management Concept for the Exhaust Aftertreatment of Commercial Vehicle Diesel Engines Using Variable Mixtures of Diesel Fuel and Rapeseed Oil

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Abstract

The use of vegetable oil as a fuel for agricultural and forestry vehicles allows a CO_2 reduction of up to 60 %. On the other hand, the availability of vegetable oil is limited, and price competitiveness depends heavily on the respective oil price. In order to reduce the dependence on the availability of specific fuels, the joint research project “MuSt5-Trak” (Multi-Fuel EU Stage 5 Tractor) aims at developing a prototype tractor capable of running on arbitrary mixtures of diesel and rapeseed oil.

Depending on the fuel mixture used, the engine parameters need to be adapted to the respective operating conditions. For this purpose, it is necessary to detect the composition of the fuel mixture and the fuel quality. Regardless of the available fuel mixture, all functions for regular engine operation must be maintained. A conventional active regeneration of the diesel particulate filter (DPF) cannot be carried out because rapeseed oil has a flash point of 230°C, compared to 80°C for diesel fuel. This leads to a condensation of rapeseed oil while using post-injection at low and medium part load operating points, which causes a dilution of the engine oil.

In this work, engine-internal measures for achieving DPF regeneration with rapeseed oil and mixtures of diesel fuel and rapeseed oil are investigated. In order to provide stationary operating conditions in real engine operation, a “high-idle” operating point is chosen. The fuel mixtures are examined with regard to compatibility concerning a reduction of the air-fuel ratio, late combustion phasing and multiple injections. The highest temperatures are expected from a combination of these control options. After the completion of a regeneration cycle, the fuel input into the engine oil is controlled. These investigations will serve as a basis for the subsequent development of more complex regeneration strategies for close-to-reality engine operating cycles with varying load conditions.

1 Introduction

Biofuels are already contributing to the reduction of greenhouse gas emissions of internal combustion engines. The production of biofuels is based on the use of renewable resources. This reduces the fossil CO_2 footprint of these fuels. Currently, the CO_2 footprint cannot be completely eliminated, as the provision of these fuels may include CO_2-generating processes during production, processing and transport. Nonetheless, biofuels will be an important factor in the evolution towards a CO_2 neutral society.

The goal of current efforts in research and development is therefore to investigate potential applications of environmentally friendly fuels and to optimize their use. During the introduction of new alternative fuels, availability may still be limited, which in turn means that engines for such fuels should still be capable to operate also on conventional diesel fuel or on blends of conventional and alternative fuels. In order to prepare the engine for the use of such arbitrary mixtures of different fuels, adjustments to the fuel system and the engine management as well as the development of new operating strategies are necessary. [1]

In this context, the regeneration of the particulate filter under all potential operating conditions represents a particular challenge. Therefore, this study investigates a thermal management concept with the aim of raising the exhaust gas temperature when using rapeseed oil fuel to a level at which the regeneration of the particulate filter becomes possible. The aim is to select engine operating modes which can be used for regeneration in regular engine operation. During the investigations, the input of rapeseed oil into the engine oil is evaluated, and basic mechanisms for oil dilution are investigated.

1.1 Benefits of Rapeseed Oil as a Fuel

For the production of biofuels such as biodiesel or HVO (Hydrogenated Vegetable Oils), natural vegetable oil is processed in further production steps. Natural vegetable oil consists of three fatty acid chains which are connected by a glycerin molecule. In order to produce biodiesel, vegetable oil is mixed with methanol and a catalyst. In a subsequent reaction at 50-80 °C, the vegetable oil molecule is split into the three fatty acid chains and the glycerin molecule, and methanol is bound to the fatty acid chains by esterification. Biodiesel is therefore also referred to as “vegetable oil methyl ester” or “fatty acid methyl ester” (FAME). When obtained from rapeseed oil, it is called “rapeseed oil methyl ester” (RME). For the production of HVO, the vegetable oil is hydrogenated, which means that alkanes are produced from the fatty acid chains by adding H_2. These fuels have properties which are comparable to those of fossil diesel fuel, which greatly facilitates the use in existing engines. However, the refining processes require the input of additional energy, which might negatively affect the CO_2 reduction potential of these bio-based fuels if the energy used is not derived from renewable sources. [1] When vegetable oils are used directly as a fuel (thus chemically unchanged), this might on the other hand improve the CO_2 balance compared to biodiesel or HVO. Figure 1 shows the CO_2 reduction potential of rapeseed oil and various other biofuels compared to fossil diesel fuel. It becomes obvious that the direct use of rapeseed oil as a fuel (compared to further processing into biodiesel) can reduce CO_2 emissions by an additional 16 % compared to fossil diesel fuel. In the same way, 8 % of CO_2 can be saved when compared to the further processing into HVO.

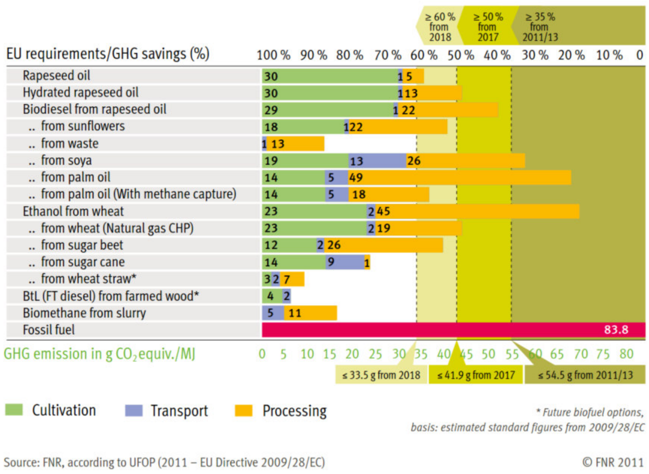

Figure 1: Potential for CO_2 reduction of rapeseed oil and other biofuels compared to fossil diesel [1]

In addition to the advantage in CO_2 reduction potential, vegetable oils are biodegradable. This makes them particularly suited for usage in environmentally sensitive areas such as agriculture, forestry and inland waterway transport. Vegetable oil is a by-product of the production of protein-rich animal feed. In rapeseed, between 33 % and 42 % of the yield mass is separated as oil. Today, rapeseed is the most important oil plant for cultivation in Western Europe. Due to this already existing availability, rapeseed oil can serve as a sustainable fuel alternative, especially in the agricultural sector. [1]

In addition, the greenhouse gas balance of vegetable oil can be further improved, considering the CO_2 content of feed production. [1] Based on the data displayed in Figure 1, rapeseed oil produces only 42 % of the CO_2 emissions per kWh compared to fossil diesel. Considering additionally that 58-67 % of the CO_2 emissions of rapeseed oil production can be attributed to livestock feed production, this results in an effective CO_2 emission per kWh of just 14-18 % compared to fossil diesel.

1.2 Context of the Current Study

1.2.1 Previous Investigations on Rapeseed Oil as a Fuel

The feasibility of using rapeseed oil as a fuel in non-road diesel engines has already been investigated in previous studies. It was shown that the use of rapeseed oil reduces HC, CO and particulate emissions compared to diesel. On the other hand, nitrogen oxide emissions tend to increase. This is mainly due to the increased ignition delay of rapeseed oil, which means that a larger quantity of fuel is injected into the combustion chamber until the onset of auto-ignition, and thus the proportion of thermal NO_x increases. [2]

Previous investigations were aimed at developing a concept for a rapeseed oil diesel tractor compliant with EU Stage IIIb and IV emission regulations. These studies focused on how to determine the fuel blend ratio by means of additional sensors, and how to generate maps for defined mixtures. [3] In the same study, the performance differences resulting from the use of vegetable oil compared to the use of diesel were investigated, including the effect on pollutant emissions. [4]

In [5], the soot emission characteristics of a diesel engine in rapeseed oil operation are described. The loading cycles of the diesel particulate filter with diesel and rapeseed oil are compared for this purpose. In addition, active particulate filter regeneration with rapeseed oil is investigated. The results show that engine-internal regeneration measures lead to a dilution of the engine oil with rapeseed oil. As an alternative, external catalytic burners are also investigated. However, with the EU Stage IV system used, the continuous regeneration in combination with the reduced particulate mass due to rapeseed oil proved sufficient for robust operation. [5]

1.2.2 Thermal Management of the Exhaust Aftertreatment of Diesel Engines

Thermal management of exhaust gas aftertreatment systems for diesel engines is needed to achieve sufficient conversion rates of the catalytic converters over the entire engine map. This implies both keeping the temperature within the operating window (above the minimum operating temperature and below the maximum permissible temperature) and reducing the light-off time required for heating up the aftertreatment system. Based on the example of an exhaust gas aftertreatment system consisting of a diesel oxidation catalyst (DOC), diesel particulate filter (DPF), urea dosing system and downstream selective catalytic reduction (SCR) catalyst, it is shown in the following which functions and reactions need to be controlled.

With the aid of the DOC, HC and CO emissions are oxidized to CO_2 and H_2O. The time required to reach the light-off temperature of the catalyst should be as short as possible during cold start. The light-off temperature is the temperature above which the catalyst can convert 50% of the corresponding component. Figure 2 shows the conversion of CO and HC emissions by a DOC as a function of temperature. The CO light-off temperature is also marked. [6]

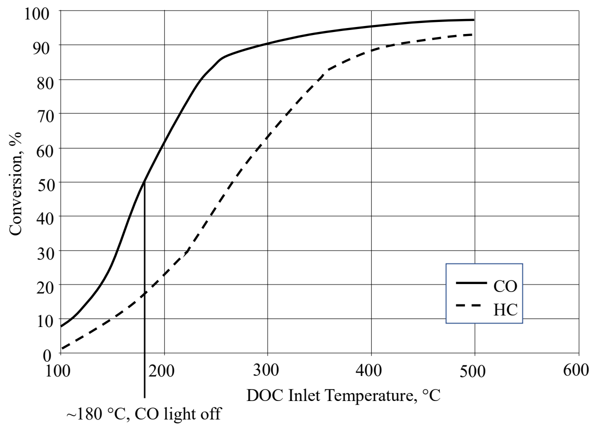

Figure 2: DOC Conversion of HC- and CO-Emission depending on Temperature (data taken from [7])

In addition, a certain share of the NO emission needs to be oxidized to NO_2 in the DOC to achieve a balanced ratio (1:1) of NO to NO_2. This ratio is needed in the SCR catalyst for the "fast SCR reaction", which is the dominant reaction in the SCR catalyst at low temperatures. Additionally, NO_2 is used in the DPF for the oxidation of soot. Exhaust gas temperature must be raised, especially at cold start and low load points, to allow sufficient conversion of the emission components.

In addition to reducing emissions, the DOC is also used to raise the exhaust gas temperature. In this process, hydrocarbons are deliberately introduced into the exhaust gas and catalytically oxidized in the DOC. The resulting heat is used to oxidize the soot in the particulate filter or for rapid heating of the SCR catalyst during cold start. [6][7]

The DPF is used to filter particles from the exhaust gas. These result from the inhomogeneous mixture formation with locally rich areas in the diesel engine. In the DPF, the exhaust gas temperature influences the oxidation rate of the soot. There are two types of regeneration - continuous and active regeneration - which differ in the oxidation rate and the associated heat release. The temperatures required for active regeneration represent the highest temperatures which occur in the exhaust gas aftertreatment of diesel engines. [7] The thermal management concept described in this paper refers to engine-internal measures for the regeneration of the DPF when using rapeseed oil as fuel. Therefore, the regeneration and the problems when using vegetable oil are explained in more detail in chapter 2.

For the SCR catalysis, ammonia (NH_3) is required. This is obtained from an aqueous urea solution within the exhaust gas aftertreatment. The formation of NH_3 takes place via thermolysis and hydrolysis reactions starting at approx. 180 °C. In addition to the conversion of urea into NH_3, heat is required to evaporate the water content. If the urea exhaust gas mixture is cooled below 220 °C, deposits of cyanuric acid are formed, which only decompose at temperatures > 370 °C. [6][7]

SCR catalysis takes place via various reaction pathways. The "fast SCR reaction" takes place from approx. 250 °C. Depending on the operating point and the raw emissions of the engine, this temperature must be exceeded for sufficient NOx reduction. The conversion rate of SCR catalysts increases with temperature. At temperatures in the range between 450 °C to 500 °C, conversion breaks down, so that this exhaust gas temperature range should be avoided. For SCR catalysts with vanadium coating, temperatures > 550 °C lead to permanent damage to the catalytic surface. The conversion rate of the SCR catalyst as a function of temperature is shown in Figure 3. [8][9]

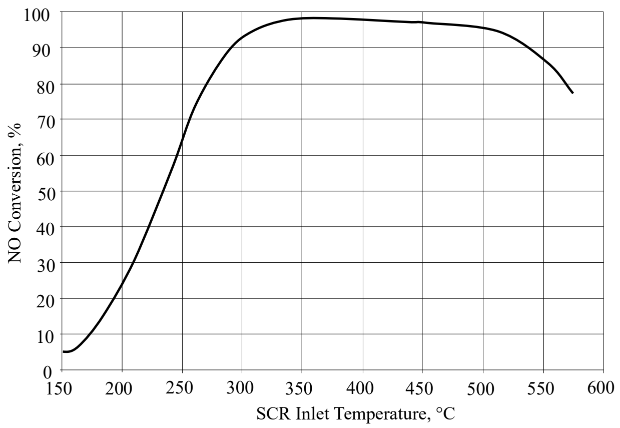

Figure 3: NO Conversion depending on SCR Inlet Temperature (data taken from [8])

1.2.3 Methods for Increasing Exhaust Gas Temperature

As chapter 1.2.2 shows, the major requirement for the function of exhaust gas aftertreatment is to increase the exhaust gas temperature, especially during light-off and at low engine load. Various measures for achieving this increase have already been studied and also implemented in series diesel engines. The most important methods are:

1. Post-Injection: Unburned hydrocarbons are supplied to the exhaust gas by post-injection of fuel. These hydrocarbons are oxidized in the DOC, and the heat produced by this process increases the exhaust gas temperature. For this measure, it is necessary that the DOC light-off temperature has already been exceeded. [6]
2. Throttling: By throttling either the exhaust or fresh air flow, the gas exchange work increases and the thermal mass of the exhaust gas decreases. The result is a deterioration of the effective efficiency and an increase in the exhaust gas temperature. [7]
3. Hot EGR: Rebreathing of uncooled exhaust gas (internal EGR) leads to an increase in process temperature and, as a result, also in exhaust gas temperature at low load points. This requires a second lift event of the exhaust valves which can be switched on and off, e.g. by a variable valve train. [10][11]
4. Cylinder Deactivation: When individual cylinders are shut down, the exhaust gas mass flow decreases and the mean effective pressure of the remaining cylinders increases (load point shift). This increases the combustion and exhaust gas temperature while improving the effective efficiency at low engine load. Either a fully variable valve train or switchable cams are required to implement this measure. [12][13]
5. Early Exhaust Opening: With early exhaust valve timing, the exhaust valve is opened at elevated cylinder pressure and gas temperature. As there is less time and stroke for the gas to expand, this measure reduces the conversion of heat into mechanical work. As a result, the exhaust gas temperature rises. In order to implement this measure, it is necessary to control the phasing of the exhaust camshaft. [14][10]

The potential of the measures described above has been the subject of numerous studies. Several authors have also compared different measures with respect to their effectiveness in increasing exhaust gas temperature and to their influence on fuel consumption. [15][12] Figure 4 shows the result of such a study, correlating the achievable increase in exhaust gas temperature (on the y axis) to the effect on brake specific fuel consumption (on the x axis) for post-injection, throttling, cylinder deactivation and early exhaust valve opening. From this, it was concluded that cylinder deactivation is the most powerful measure, as in contrast to all of the other measures, it may also improve fuel consumption. This is also in line with the findings in [15]. However, this measure also requires a substantial technological effort in order to be realized, both in terms of additional engine components and in terms of application effort. In a recent publication, several alternatives to cylinder deactivation for thermal management are studied, in which a part of the cylinders are skip-fired without completely deactivating the valve lift of the intake and/or exhaust valves [16]. In these cases, either the opening duration of the valves of the non-fired cylinders was significantly increased, or the valve spread was modified in order to open the valves around bottom dead center. It was found that such strategies have the potential to deliver exhaust temperatures and fuel economy benefits similar to complete cylinder deactivation. Still, all of the measures studied require extensive variabilities of the valvetrain, meaning that the technological effort is still significant.

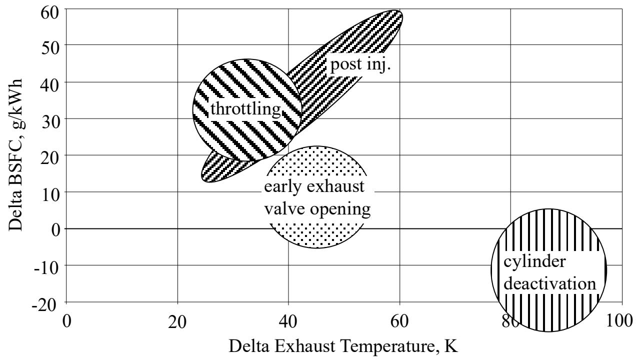

Figure 4: Change of exhaust gas temperature and brake specific fuel consumption for cylinder deactivation, throttling, early exhaust valve opening and post injection (data taken from [12])

1.2.4 Context of the "MuSt5-Trak" Project

Besides the additional cost and risks for failure due to the required modifications to the engine, potential users of a vegetable oil tractor would also have to face additional economic risks due to non-competitive vegetable oil prices or even machine downtime caused by shortages in supply. In order to mitigate these disadvantages, the project "MuSt5-Trak" (Multi-Fuel EU Stage 5 Tractor) aims at the realization of a prototype tractor allowing the fully flexible use of diesel fuel, rapeseed oil or arbitrary mixtures of these two fuels. At the same time, full compliance with the EU exhaust gas regulation stage 5 for non-road equipment (comparable to TIER 5 regulation in the US) is required. This intended outcome is to remove the major obstacle to a successful introduction of vegetable-oil-driven tractors into the market.

The realization of this prototype tractor has to face various challenges:

1. The fully flexible use of diesel and rapeseed oil means that any arbitrary mixture of these fuels may be produced when refueling. Since rapeseed oil differs from diesel in important characteristic values such as energy density, flash point, flammability limit, boiling point and temperature-dependent viscosity, it is necessary to determine the actual mixture ratio for regular engine operation.
2. Depending to the fuel mixture determined, the ECU output must be adjusted with respect to the actual amount of fuel injected and its energy density, so that the actual engine power corresponds to the modeled (required) engine power of the ECU. This is necessary to ensure that the correct maps or map range resulting from the operating point are used to control the engine (e.g. the EGR ratio), so that full compliance with the regulated exhaust emission limits is guaranteed at all times.
3. It has to be ensured that all relevant engine operations such as catalyst heating or diesel particulate filter regeneration remain possible.

In the context of another project, investigations into the potential of integrating a fully variable valve train into the same base engine have been carried out. [17] Such a modification would enable advanced thermal management concepts for exhaust gas aftertreatment such as early exhaust opening or cylinder deactivation. However, the associated increase in system complexity was considered to be contradictory to the core objectives of the current project "MuSt 5-Trak", as the associated costs for this modification might result in an additional obstacle to market introduction. Therefore, this study will focus exclusively on investigations of thermal management using the technological equipment which is already available on the engine.

2 Procedure of DPF Regeneration

2.1 Active DPF Regeneration

In conventional diesel operation, active DPF regeneration is realized by post-injection of fuel. Depending on the operating point, fuel is injected at a timing of > 30 °CA ATDC, in some cases also >180 °CA ATDC. Figure 5 shows common injection patterns for multiple injections in diesel engines, plotted as a function of crank angle position.

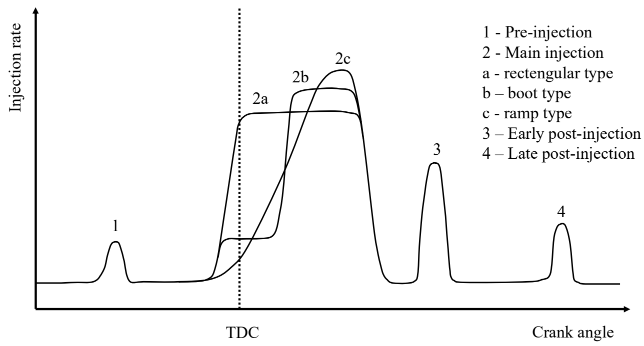

Figure 5: Injection patterns for multiple injections in modern common rail diesel engines [18]

The pre-injection, marked as "1" in Figure 5, takes place several °CA before TDC. The early heat release from this injection reduces the ignition delay of the main injection, which in turn reduces combustion noise. Also, the shorter ignition delay reduces local peak temperature, which helps to limit the formation of nitrogen oxides. The main injection (marked "2") primarily determines the engine load. By modulating the rate of injection, the heat release rate can also be influenced, similar to the effect of pre-injection [18]. Injections "3" and "4" indicate early and late post-injection, which are mainly used for exhaust gas temperature control. The associated supply of unburned hydrocarbons into the exhaust system causes an exothermic reaction in the diesel oxidation catalyst (DOC) due to the conversion of these additional HC emissions. For this function, the control of the post-injected fuel quantity is based on the DOC outlet temperature. This may also cause the ignition temperature of soot stored in the DPF to be exceeded, so that in the presence of oxygen, a burn-off of soot occurs. Active DPF regeneration is also started by introducing unburned HC into the exhaust gas. Figure 6 shows the temperature development before and after DOC and after DPF as a function of time for active DPF regeneration. [6]

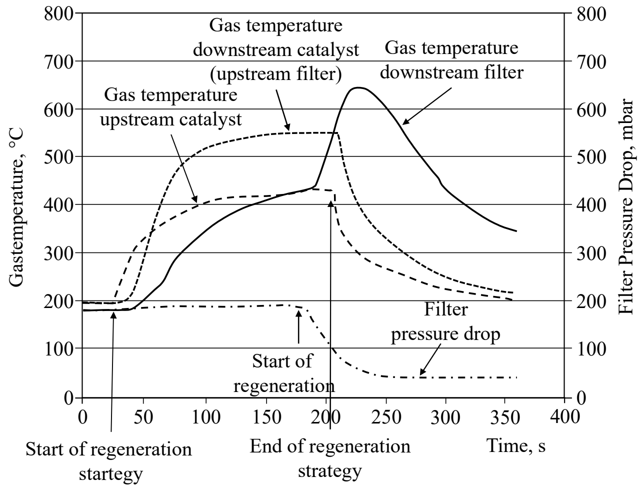

Figure 6: Time sequence of temperature and pressure drop across the DOC and DPF during active DPF regeneration [6]

At the initial state, the temperatures before DOC, after DOC (= before DPF) and after DPF are constant. With the start of the regeneration strategy, the DOC inlet temperature starts to rise. The DOC outlet temperature follows with a slight delay. Approximately 20 seconds after the start of the regeneration process, the DOC outlet temperature exceeds the inlet temperature. At about 175 seconds, the DOC outlet temperature has reached about 550 °C. Thus, it exceeds the ignition temperature of soot, and soot combustion starts. The heat release can be observed from the temperature increase after DPF. At this point, the maximum temperatures are reached. The development of the reaction can also be detected from the pressure difference across the particulate filter, which decreases in correspondence to the temperature increase. [6]

The temperature increase due to soot burn-off also represents the limiting factor for the soot loading of the particulate filter. If the particulate filter load exceeds a critical value and the soot is burned off actively, critical temperatures may occur, leading to damage of catalytic coatings or the destruction of the substrate. The particulate filter loading refers to the mass of soot bound in the filter normalized to the volume of the filter. Common limits are in the range of approx. 5-10 g/l. [7]

2.2 Continuous DPF Regeneration

Besides the active DPF regeneration, there is also the possibility of a "continuous" or "passive" DPF regeneration. In active regeneration, oxidation rates are so high that the released heat is sufficient to keep the process running without additional measures. Continuous regeneration, on the other hand, represents a state of equilibrium between the amount of soot on the filter, the amount of soot emitted by the engine and the oxidation rates as a function of temperature and residence time distribution. The reaction equations (1), (2), (3) and (4) show the continuous regeneration of soot in a reaction with nitrogen oxides. [7][19]

$$2\,NO_2 + C \rightarrow 2\,NO + CO_2 \tag{1}$$

$$NO_2 + C \rightarrow NO + CO \tag{2}$$

$$CO + NO_2 \rightarrow CO_2 + NO \tag{3}$$

$$CO + \frac{1}{2}O_2 \rightarrow CO_2 \tag{4}$$

These reactions already take place in a temperature range < 300 °C. However, relevant oxidation rates for continuous regeneration can only be expected above 300 °C. In addition to the oxidation in a reaction with nitrogen oxides, soot is also continuously oxidized via a reaction path with oxygen. Equations (5) and (6) show the reactions between soot and oxygen. [19]

$$C + O_2 \rightarrow CO_2 \tag{5}$$

$$C + \frac{1}{2}O_2 \rightarrow CO \tag{6}$$

The dominant reaction path depends on the temperature. At temperatures below 400 °C, the reactions with nitrogen oxides dominate. From about 450 °C oxidation is mostly taking place by reaction with oxygen. The relationship between temperature and the resulting oxidation rate is shown in Figure 7. [19]

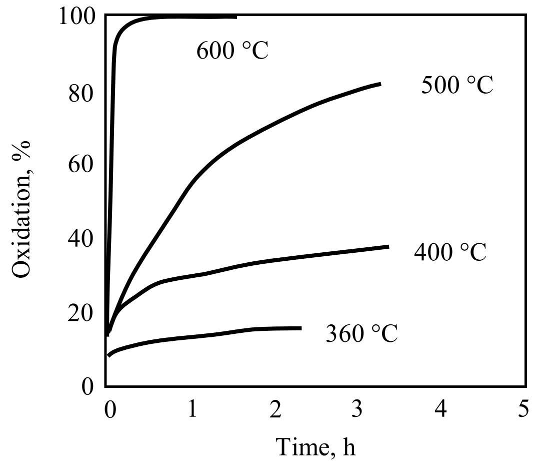

Figure 7: Soot oxidation rates as a function of temperature [6]

From Figure 7, it is obvious that at 600 °C, the oxidation is completed after a short period of time. The ignition temperature of soot is exceeded, so the heat released is sufficient to maintain the combustion of the soot. If the temperature is reduced to 500 °C, already several times the duration is needed for complete oxidation. The degressive behavior of the oxidation curves shows that the curves are approaching a limit value for oxidation. This is the so-called "balance point" between soot oxidation and the amount of soot emitted by the engine. This "balance point" is different for each operating point of the engine. [20]

2.3 Effect of RME and Rapeseed Oil on Active DPF Regeneration

2.3.1 Problem of Conventional Active DPF Regeneration with RME or Rapeseed Oil

During active DPF regeneration, fuel is injected into the combustion chamber after the main injection. This fuel quantity no longer participates in combustion and is emitted with the exhaust gas as unburned hydrocarbons. This is subject to the condition that the post-injected fuel is vaporized and thus leaves the combustion chamber with the exhaust gas. When using RME or rapeseed oil, the post-injection of fuel leads to extensive cylinder wall wetting. Even at gas temperatures of 600 °C, a fuel film of about 1 µm thickness remains on the colder cylinder wall. During the upward movement of the piston, this oil film is mixed with the engine oil and wiped off during the downward movement. This results in a dilution of the engine oil with RME or rapeseed oil. This behavior does not occur to any relevant extent with diesel fuel, on the contrary. The reason for this can be found in the different boiling curves of diesel, RME and rapeseed oil (Figure 8). [7][21][22][23]

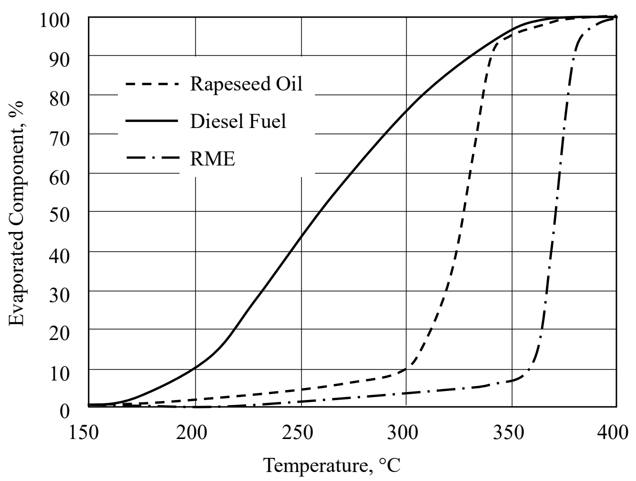

Figure 8: Evaporation curves of diesel, rapeseed oil and RME (data taken from [22][23][24])

In Figure 8, it is obvious that the initial boiling point of rapeseed oil is around 300 °C. At this temperature, 75 % of the diesel fuel is already evaporated. The boiling range for relevant fractions of RME starts at about 350 °C. Diesel fuel and rapeseed oil are both evaporated to 95 % at this temperature. Only at approx. 400 °C, RME as well as diesel and rapeseed oil are completely vaporized. The comparatively small boiling range of rapeseed oil and RME with a late start of evaporation is thus responsible for the higher fuel input into the engine oil. The higher vaporization enthalpy of rapeseed oil (209 kJ/kg @ boiling temp.) compared to diesel fuel (137 kJ/kg @ boiling temp.) additionally reinforces this effect. [25][26][27] In addition to the higher amount of fuel introduced into the engine oil, the typical temperatures in the oil pan of up to 120 °C are not sufficient to evaporate rapeseed oil or RME, while volatile fractions of diesel fuel may already evaporate at this temperature. [7]

The introduction of rapeseed oil or RME into the engine oil first of all leads to a reduction of the viscosity of the engine oil. Due to the double bonds of the fatty acid chains, an auto-oxidation of the fuel takes place. During this process, polymers are formed which lead to the formation of oil sludge. This increases the viscosity of the engine oil and can lead to clogging of the oil filter and the oil channels inside the engine, which typically results in fatal engine damage. Therefore, the DPF regeneration strategy imperatively has to be adapted to the use of rapeseed oil or RME. The oil change intervals also need to be adapted. [7][23]

In previous investigations on a commercial vehicle diesel engine, the rapeseed oil input into the engine oil was investigated when performing post-injection for active DPF regeneration. Figure 9 shows the engine oil dilution with rapeseed oil. [5]

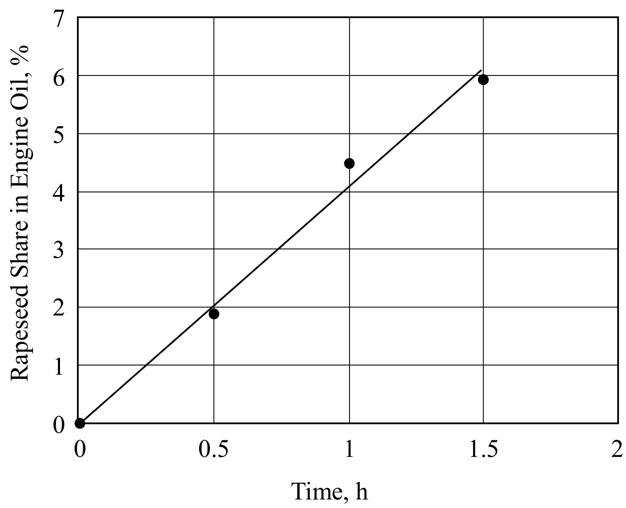

Figure 9: Engine oil dilution as result of rapeseed oil absorption [5]

In part load operation, a rapeseed oil input of 2 % of the engine oil was detected during a regeneration period of 0.5 h. This corresponds to an engine oil dilution of 4 %/h. To investigate the system behavior, injection timing, injection quantity and injection pressure were varied. None of the investigated measures led to a significant improvement of the dilution rate. [5]

2.3.2 Additional Systems for DPF Regeneration

In order to increase the exhaust gas temperature of rapeseed oil driven vehicles, usually additional systems are installed, such as catalytic burners, vaporizers or additional injectors in the exhaust system. These additional components are cost-intensive and require additional installation space. In the presented study, only internal engine measures for exhaust gas temperature increase are investigated. Therefore, such additional systems will not be explained in detail here. For further Information see [5].

3 Experimental Setup

A John Deere JD4045 engine serves as the test object. This is a non-road diesel engine for use in agricultural vehicles. Table 1 shows the technical data of the test engine. In addition to the engine, the corresponding exhaust aftertreatment system of a John Deere 6135R tractor is used. The system thus meets the EU Stage V emission standard (see Table 2), which compares to US TIER 4 emission standard. The schematic layout of the experimental set-up is shown in Figure 10.

Table 1: Technical details of MuSt5-Trak engine

	value	unit
maximum effective power	130	kW
maximum torque (@1600 rpm)	703	Nm
nominal speed	2100	rpm
displacement	4.5	dm^3
compression ratio	17.3: 1	-
bore diameter	106.5	mm
stroke	127.0	mm
firing order	1-3-4-2	(cyl. no.)
valves per cylinder	4	-
emission standard	EU stage IV	-

Table 2: EU Stage V emission standard [28]

	56 kW ≤ P < 130 kW	130 kW ≤ P < 560 kW
CO (g/kWh)	5.0	3.5
HC (g/kWh)	0.19	0.19
NO (g/kWh)	0.4	0.4
PM (g/kWh)	0.015	0.015
PN (#/kWh)	1×10^{12}	1×10^{12}

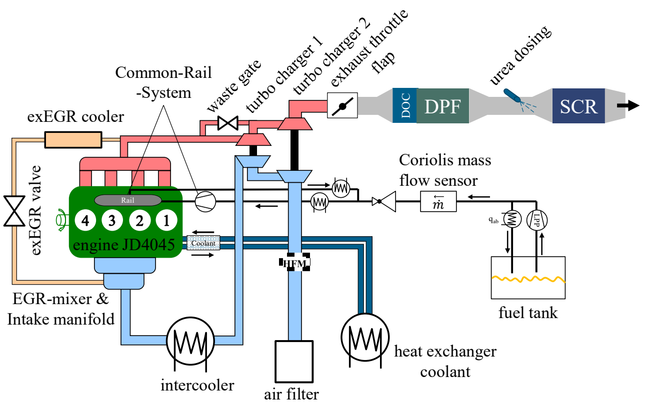

Figure 10: Schematic test bench setup

The engine has a common rail injection system with solenoid injectors and a dual piston high pressure pump. Two exhaust gas turbochargers connected in series are used to supercharge the engine. The high-pressure turbocharger has a wastegate for passive boost pressure control. An external exhaust gas recirculation (EGR) system with an EGR cooler and an electrically actuated EGR valve is installed for residual gas control. The external EGR is added to the fresh air by an EGR mixer. The air flow supplied to the engine is adjusted via an exhaust throttle valve located downstream of the turbochargers. The exhaust system consists of a DOC and a DPF combined in one canning and a downstream SCR catalyst including urea dosing. For fuel consumption measurement, the fuel system has been modified so that the return flow is no longer directed into the tank, allowing fuel consumption to be measured by a single Coriolis sensor.

The test engine has cylinder pressure indication for each cylinder. Also, the exhaust manifold on each cylinder is equipped with a lambda sensor. In the exhaust ports, the gas temperature is measured close to the valves in order to immediately counteract to a further increase in exhaust gas temperature when the temperature limit is exceeded. In the exhaust system, the temperature and pressure before and after each component are also determined. The pressure and temperature differences across the turbocharger, the exhaust flap, the DOC and the DPF are of particular interest here.

The exhaust gas components THC, CO, CO_2, O_2, NO_x, NO and the external EGR ratio are determined by an AVL AMA i60. An AVL MicroSoot is used to measure the particulate mass in a spectrum larger than 2 µm. A TSI EEPS analyzes the particle size distribution in the range from 5.6 nm to 560 nm. The sampling of the exhaust gas is located between the exhaust throttle flap and the DOC.

The tests were performed using the original ECU. The most important variables which were adapted are the position of the exhaust throttle valve, the position of the EGR valve and the fuel injection. With the current configuration, it is possible to carry out up to five injections per cycle. For all five injections, the timing can be modified, and for four of the injections, the injected quantity can be adapted. The access to the injection quantity of the fifth injection must remain available to the ECU for load control. In addition to the time and quantity, the injection pressure can be adjusted. The maximum injection pressure of the system is 220 MPa.

4 Increase of the Exhaust Gas Temperature

4.1 Procedure of the Experiment

Five different fuel blends of rapeseed oil fuel and diesel fuel are used for the tests. The properties of rapeseed oil fuel are defined according to DIN 51605. The diesel fuel used is a commercial diesel fuel according to DIN EN 590, which contains 93 % (m/m) fossil diesel and 7 % (m/m) biodiesel. The mixture ratio between the two fuels were altered in steps of 25 % (m/m) for the different blends. Table 3 gives an overview of the nomenclature and the definitions of the fuel blends used in this study.

Table 3: Fuel mixtures of the experiment

Name of fuel mixture	Description
D100	100 % (m/m) diesel fuel – corresponding to DIN EN590 standard: 93 % (m/m) fossil diesel, 7 % (m/m) biodiesel
R25 D75	25 % (m/m) rapeseed oil fuel (DIN 51605) and 75 % (m/m) diesel fuel (DIN EN590)
R50 D50	50 % (m/m) rapeseed oil fuel (DIN 51605) and 50 % (m/m) diesel fuel (DIN EN590)
R75 D25	75 % (m/m) rapeseed oil fuel (DIN 51605) and 25 % (m/m) diesel fuel (DIN EN590)
R100	100 % (m/m) rapeseed oil fuel (DIN 51605)

For the fundamental investigations of exhaust gas temperature management, a "high-idle" operating point was chosen, as such an operating point can be maintained permanently in real vehicle operation. For high-idle operation, engine speed is elevated without increasing the effective load. This results in an increase of gross indicated work by increasing pumping work and friction loss. The increase of gross indicated work also results in an increase of exhaust gas temperature. A speed of 1800 rpm was chosen as high-idle speed.

In order to determine the feasible temperature increase for this operating point, a systematic variation of different measures was carried out. For these measurements, the basic application, a deactivation of the EGR valve, a reduction of the air-fuel ratio and a retardation of various partial injections were examined to determine the influence on exhaust temperature. In order to eliminate the influence of varying EGR rate, the EGR valve was closed for all measurements except for the basic application. The nomenclature of the measurements performed is explained in Table 4.

Table 4: Nomenclature of the measurements

Measurement	Description
basic application	reference measurement of series engine application
exEGR deac	deactivation of external EGR; each further measurement is performed with deactivated external EGR
mt	minimum exhaust throttle flap position; determined by rise of soot concentration
ls	late single injection
pre+lm	single pre-injection with late main injection
dbl pre+lm	double pre-injection with late main injection
mt+ls	minimum exhaust throttle flap position with late single injection
mt+pre+lm	minimum exhaust throttle flap position with single pre-injection and late main injection
mt+dbl pre+lm	minimum exhaust throttle flap position with double pre-injection and late main injection

4.2 Results of Exhaust Gas Temperature Increase

The comparison of the measurements of the five fuel mixtures with standard engine setting shows that different exhaust gas temperatures result for the operating point. This is illustrated in Figure 11.

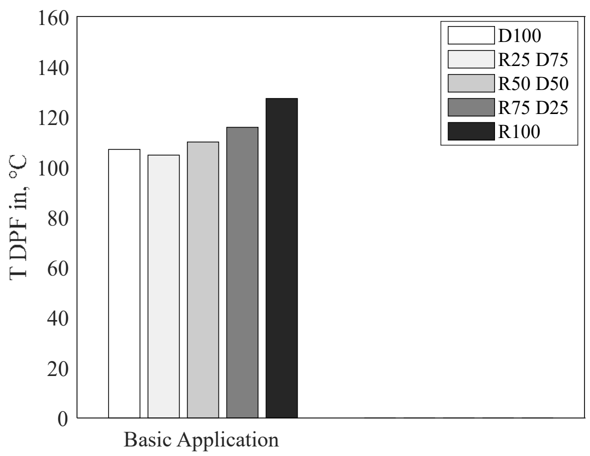

Figure 11: Comparison of DPF inlet temperatures for the different fuel mixtures with basic application

The reason for the different temperature levels if using the default settings can be found in the different volumetric energy densities of the fuel mixtures. The modeled power of the engine is based on the actuation duration of the injectors and the amount of fuel injected. If the energy density of the fuel differs from the energy density of diesel, this leads to a change in the actuation duration. As the fuel mixtures with rapeseed oil have a lower energy density and the viscosity increases compared to diesel, the actuation duration also increases. The engine thus operates at a higher modeled load point. This in turn causes an increase of the external EGR rate. The comparison of fuel mass flow and EGR rate illustrates this correlation (Figure 12).

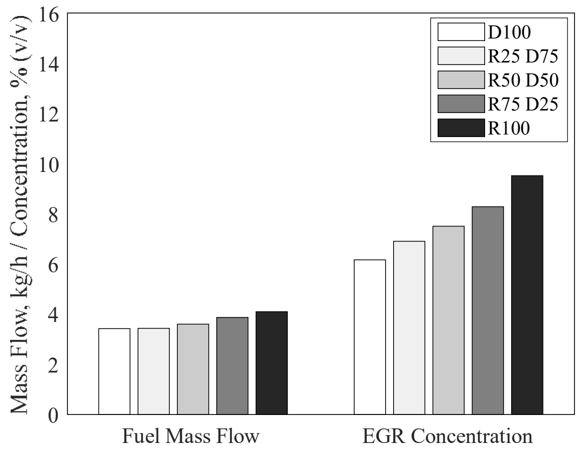

Figure 12: Fuel mass flow and EGR concentration of basic application

Figure 12 shows that with increasing rapeseed oil content, the fuel mass flow and the EGR rate increase. To minimize the influence of a shift in the modeled load point and the associated temperature difference, the EGR valve is closed for all further measurements. This results in an almost uniform temperature level for all fuels (see Figure 13).

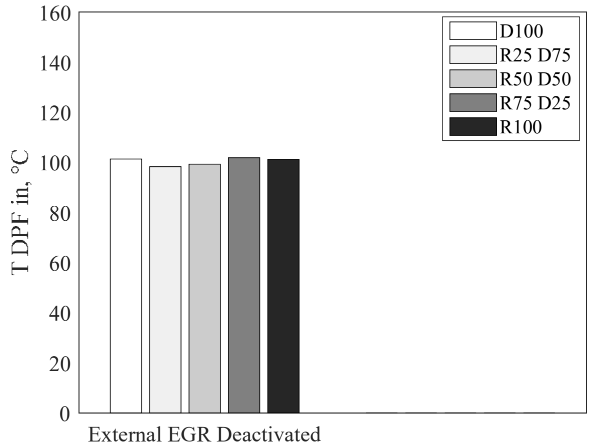

Figure 13: Comparison of DPF inlet temperatures for different fuel mixtures with deactivated external EGR

The temperature level shown in Figure 13 represents the initial state for further investigations on temperature increase. The DPF inlet temperatures of the individual fuel blends deviate by less than 2% from the average value.

With the measures listed, the exhaust gas temperature for the five fuels can be increased significantly. An overview of the temperatures achieved is given in Figure 14.

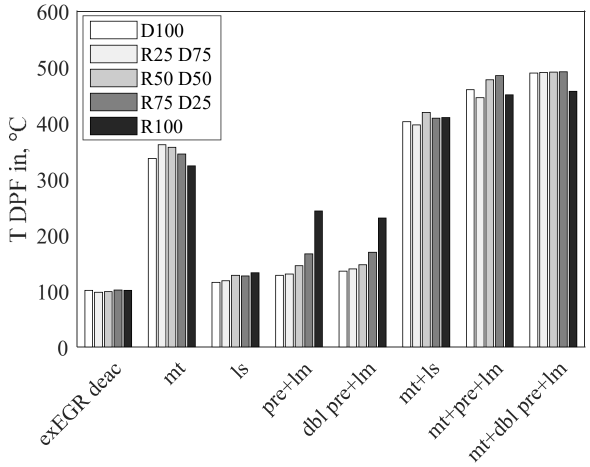

Figure 14: Overview of the resulting exhaust gas temperature of the measuring series

The measurement program is divided into two parts. The first part covers the separate evaluation of the individual measures. These measurements show the effect of closing the throttle valve (mt) and the late timing of injection (ls, pre+lm, dbl pre+lm). The second series of measurements comprises the combination of throttle valve closure and injection adjustment (mt+ls, mt+pre+lm, mt+dbl pre+lm). The influence of the individual measures on exhaust temperature can be determined from the separate measurements. However, a combination of different measures is necessary to achieve the temperature level required for a relevant oxidation rate.

4.2.1 Results of Individual Measures

An energy balance is carried out for the engine in order to evaluate the efficiency of the individual measures. The energy flows are shown schematically in Figure 15.

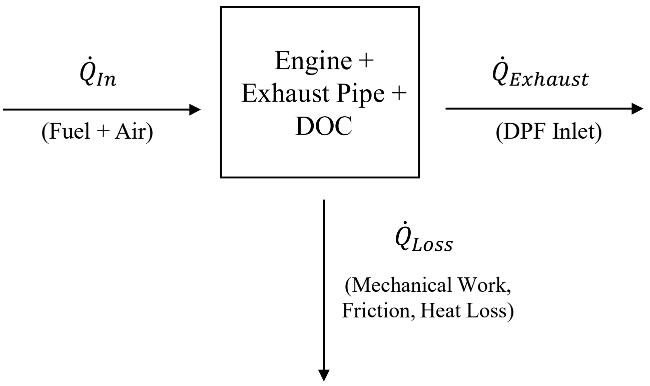

Figure 15: Schematic of energy balance

The heat balance determines how efficiently the fuel mass is used to raise the temperature of the exhaust gas. The ratio between the heat in the exhaust gas at the DPF inlet and the energy supplied to the combustion by fresh air and fuel represents the efficiency of the exhaust gas temperature increase. All other energy flows are considered as losses, as they are not relevant for increasing exhaust gas temperature. Figure 16 shows the efficiency of the exhaust gas temperature increase using D100 as an example.

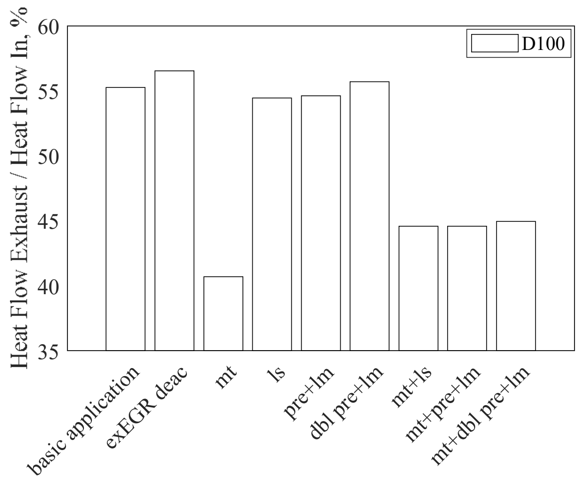

Figure 16: Efficiency of methods to increase exhaust temperature

Starting from the basic application, it can be seen that the deactivation of the external EGR increases the utilization of the fuel for exhaust gas temperature increase, although the absolute temperature decreases. Due to the original engine application, the throttle valve opens completely when the EGR valve is closed. This reduces the gas exchange work, so that the heat losses decrease. Increasing throttling results in a similar behavior, reducing the effect of the injected fuel on exhaust heat flow.

By throttling, the highest DPF inlet temperatures (330 °C to 370 °C) of all individual measures are achieved, since the exhaust gas mass flow and thus the thermal mass is reduced. However, this results in a significant increase of pumping loss. The effect on pumping work can be seen in Figure 17 from the comparison of "R100 exEGR deac" and "R100 mt".

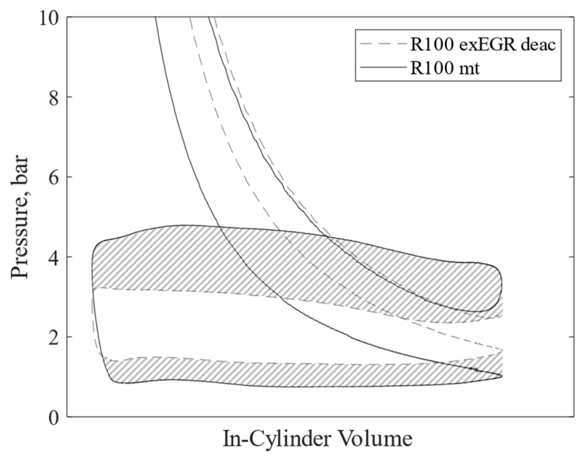

Figure 17: Comparison of gas exchange process between exEGR deactivation and minimum throttle with R100

The gas exchange process displayed in Figure 17 illustrates that the exhaust stroke is carried out at a higher pressure level with the exhaust throttle closed. At the same time, the exhaust gas enthalpy difference, available for the turbocharger, is reduced. This results in a reduction of the boost level of the intake air. In addition to the increased exhaust effort, the engine also aspirates the fresh gas at a lower pressure. The hatched areas represent the enlargement of the gas exchange cycle between “R100 exEGR deac” and “R100 mt”. This leads to a shift in the gross indicated work and to an increase in heat loss.

The increase of the exhaust gas temperature by closing the throttle is limited by the air-fuel ratio (λ). Diesel fuel starts to show a significant increase in the soot concentration in the exhaust gas already at $\lambda \leq 2.2$. For rapeseed oil, this limit is approximately $\lambda \leq 1.6$. In addition, an excess of oxygen is needed to oxidize the soot. Therefore, the increase of the exhaust gas temperature cannot be realized purely by throttling.

As shown in Figure 16, the late single injection (ls) and the pre-injection with late main injection (pre+lm) do not lead to a significant change in heat utilization. By using the double pre-injection with late main injection (dbl pre+lm) the heat utilization can be increased compared to the basic application. This confirms that it is advantageous to delay the start of combustion as far as possible to produce exhaust heat. The comparison between "ls" and "dbl pre+lm" (see Figure 18) shows the measures for late combustion phasing.

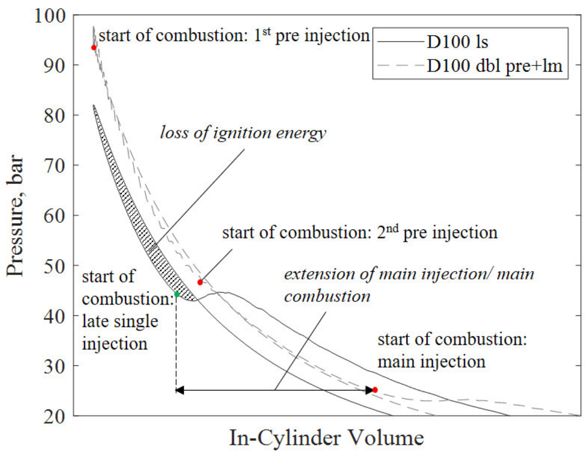

Figure 18: Comparison of high-pressure cycle of D100 ls and D100 dbl pre+lm

From the comparison of the high-pressure cycles of the measurements (D100 ls and D100 dbl pre+lm), suitable measures for a late start of combustion can be derived. The difference in peak compression pressure results from different starting conditions for the combustion. The solid line represents the high-pressure loop of D100 with late single injection. The hatched area shows the wall heat losses resulting from compression and expansion before the combustion of the late injected fuel starts. Due to this heat loss, a further delay of injection is not possible because the heat level is no longer sufficient for auto-igniting the fuel. In comparison, the dotted line represents the high-pressure cycle of the measurement with “D100 double pre-injection with late main injection”. The pre-injections compensate for the wall heat losses, so that the energy in the combustion chamber remains sufficient for auto-ignition of the main injection fuel for a longer time. However, the wall heat losses should not be overcompensated, as the additional heat released from the pre-injection is then converted to mechanical work. In this case, the late main injection needs to be reduced by a corresponding amount of fuel in order to maintain the engine speed. This results in less heat being transferred to the exhaust gas.

With increasing rapeseed oil content, a later combustion phasing is possible compared to a larger diesel fuel share. Despite similar injection timing, the injected fuel mass and the resulting exhaust gas temperature differ between the fuels. The corresponding values are compared in Table 5.

Table 5: Comparison of injection timing and quantity for late combustion with double pre-injection

	R100 dbl pre+lm	D100 dbl pre+lm
mass flow 1st injection (ECU model output)	2 mg/str 6 °BTDC	2 mg/str 6 °BTDC
mass flow 2nd injection (ECU model output)	3 mg/str -15 °BTDC	3 mg/str -15 °BTDC
mass flow main injection (ECU model output)	31 mg/str -27 °BTDC	19 mg/str -28 °BTDC
fuel mass flow/stroke (measured)	38.18 mg/str	25.35 mg/str
exhaust temperature @ DPF inlet	230 °C	135 °C

Table 5 shows that the start of the main injection of R100 and D100 is in a similar time frame. The injected fuel mass of R100 is about 150% of the fuel mass of D100. The resulting exhaust gas temperature is 95 K higher than for D100. The comparison of the cylinder pressure curves (see Figure 19) of the two measurements shows the difference in pressure increase due to combustion.

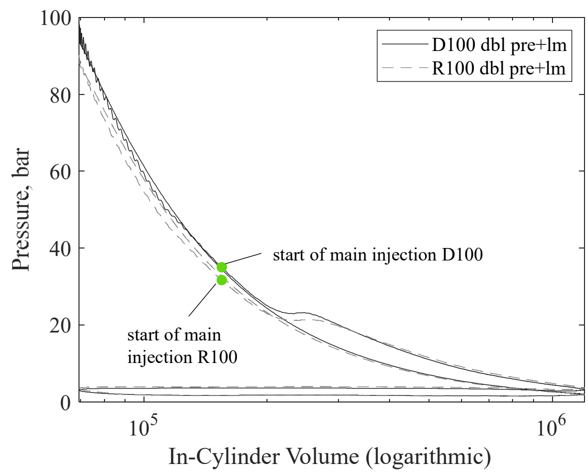

Figure 19: Comparison of p,V-diagram of D100 dbl pre+lm and R100 dbl pre+lm

In Figure 19, the start of the main injection is marked for both of the fuels. It can be seen that the pressure increase of diesel fuel combustion is larger than the pressure increase of rapeseed oil combustion. The center of combustion is located at about 42 °ATDC for both fuels. Due to the lower pressure increase with rapeseed oil, more fuel mass is needed to overcome the internal losses of the engine, resulting in higher exhaust gas temperatures. However, this effect could not yet be sufficiently clarified at the time of publication to continue the application for maximizing exhaust temperature.

4.2.2 Results of Combined Methods

Since the separate measures did not allow an exhaust gas temperature level at which sufficient oxidation rates are to be expected, a combination of the measures was investigated. For these investigations, exhaust throttling was combined with late combustion phasing. First application tests showed that first applying a late combustion phasing and then subsequent throttle valve closure is not productive, since combustion becomes unstable due to the retardation and therefore sensitive to changes in the air-fuel ratio, resulting in combustion misfires. Therefore, applying a certain level of exhaust throttling has to be subsequently followed by applying late combustion phasing for a successful combination of these two measures.

Similar to the findings when studying the individual measures, the highest exhaust gas temperatures are achieved by phasing combustion as late as possible. Reducing the air-fuel ratio increases the influence of late combustion phasing on the exhaust gas temperature increase. Due to the reduced thermal mass of the exhaust gas flow, the increased heat input causes a larger temperature increase than without throttling. The resulting temperatures are displayed in Figure 14 together with the results of the individual measures.

From the point of view of efficiency (see Figure 16), it can be noted that the improved heat utilization of late injection phasing can partially compensate for the efficiency losses due to throttling. By minimizing the throttle valve position and adjusting the injection with two pre-injections (mt+dbl pre+lm), not only the highest exhaust gas temperatures are reached, but also the throttling losses are compensated best. The achieved values for the DPF inlet temperature of 460 °C (when using R100), the air mass flow rate of 160 kg/h and the start of main injection of -15 °BTDC will be used to determine the potential for soot oxidation in the following section.

5 Use of Temperature Increase for Soot Oxidation

In order to investigate the effectiveness of soot oxidation at maximum exhaust gas temperature, the particulate filter was loaded with soot and the oxidation behavior was analyzed based on exhaust gas temperature and pressure difference across the DPF.

For determining the initial state of the particulate filter loading with soot, the reference point of 1800 rpm and 0 Nm was chosen. The pressure difference across the particulate filter in the unloaded state is 7 mbar when using diesel fuel. In order to ensure that the particle filter is free of soot, the test bench was operated at nominal power for one hour and the pressure difference across the particulate filter was observed. At this operating point the pressure difference is 24 mbar. The operation at nominal power for one hour, with DPF inlet temperatures of 440 °C, produced no change in the pressure difference over the course of the test. For a repeated check, the reference point 1800 rpm, 0 Nm was set and the pressure difference of 7 mbar was confirmed.

The particle filter was loaded with soot by continuous operation at an operating point of 2000 rpm and 70 Nm. This is the operating point with the highest soot raw emission at a DPF inlet temperature < 220 °C. In order to reduce the loading time, the injection pressure was lowered to further increase the soot emission. By shortening the loading phase at low exhaust temperatures, the influence of continuous regeneration can be neglected. The measurement of the pressure difference and the soot emission allows to estimate the filter loading to 4-5 g/l at the end of the loading phase. The resulting pressure difference across the particulate filter was 69 mbar for the reference point of 1800 rpm and 0 Nm.

In preparation for the soot oxidation test, the system was switched to rapeseed oil. The results of the regeneration are shown in Figure 20.

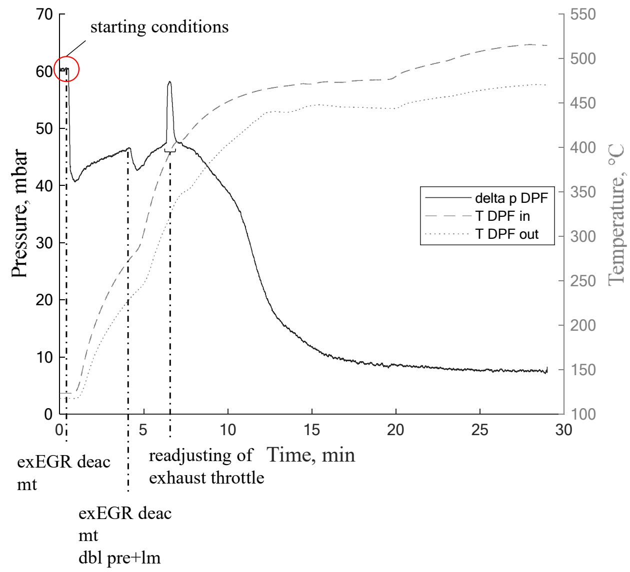

Figure 20: Result of soot oxidation with rapeseed oil exhaust temperature increase

To start the experiment, the reference point was set when using rapeseed oil fuel. The measured pressure difference of 61 mbar serves as a starting point for the subsequent test procedure. The oxidation test starts with the deactivation of the external EGR and a minimization of the throttle valve position. Approximately 4 minutes after the start of the measurement, the temperature increase at the DPF inlet saturates, and the late injection phasing with two pre-injections is applied. Between minute 6 and minute 7, the ECU readjusts the throttle position, so that manual correction is necessary. The exhaust gas temperature then stagnates briefly at about 420 °C at the inlet of the DPF. At the same time, the pressure difference across the particulate filter starts to drop. After correction of the throttle position, the exhaust gas temperature continues to rise further up to 480 °C. The drop of the pressure difference across the DPF takes about 12 more minutes. From minute 20 on, the pressure difference can be considered to be stationary. At minute 21, the ECU readjusts the injected fuel quantity so that a DPF inlet temperature of up to 515 °C is reached. This has only a minor effect on the pressure difference across the particulate filter. The temperature in the exhaust manifold increases proportionally to the DPF inlet temperature. The concentration of THC emissions shows an increase from 15 ppm to 40 ppm. Thus, the fuel is converted in the combustion and not in the DOC. The DPF outlet temperature never exceeds the DPF inlet temperature.

After completion of the oxidation test, the reference point of 1800 rpm and 0 Nm was set at default settings. The pressure difference across the particulate filter was determined to be 6 mbar. The filter loading can be reduced to the initial state by the measures of exhaust gas temperature increase within the regeneration period of a service regeneration. The temperature gradient across the DPF shows that the ignition temperature of the soot is not exceeded. A continuous supply of heat is therefore necessary to prevent an unwanted termination of the regeneration. Thus, it can be concluded that this is not an active but a continuous regeneration.

6 Determination of the Engine Oil Dilution

As described before in chapter 2.3.1, an active DPF regeneration with post-injection typically leads to a dilution of the engine oil by RME or rapeseed oil. In this work, an alternative procedure for DPF regeneration is described. The aim is to create a basis for a regeneration strategy with vegetable oil fuel and to document the effects on the dilution of the engine oil.

During the series of measurements, the quality of the engine oil was checked at defined times and the fuel content was determined. To ensure the reproducibility of the oil samples, the sampling procedure was standardized. Therefore, the oil is heated up to operating temperature during engine operation and the oil sample is taken 5 minutes after the test bench has been shut down. The sampling height is fixed at 50 mm below the oil level.

The results of the concentration of vegetable oil in the engine oil during the test series are shown in Figure 21.

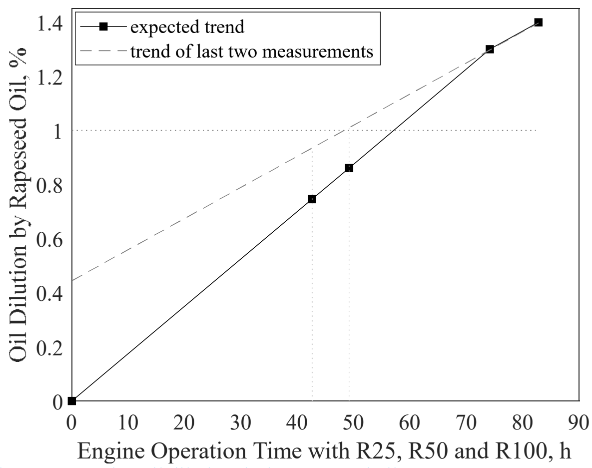

Figure 21: Engine oil dilution during rapeseed oil test

The initial state is defined by the measurement after the engine oil change, with a vegetable oil content of 0 %. After 42 hours of engine operation, the basic measurement of the engine maps with rapeseed oil fuel was completed. The vegetable oil content was determined to be < 1 %. After another 7 h (49 h total), the application tests with R100 were completed, resulting in a vegetable oil content in the engine oil of still < 1 %. The fourth measuring point is at 74 h and shows the vegetable oil content after the application with R75 D25. It amounts to 1.3 %. Afterwards, the tests were performed with R50 D50, R25 D75 and D100. The soot loading for the oxidation test was carried out correspondingly. Since the input of vegetable oil can be excluded when operating with D100, these operating times are not considered on the time axis. After the subsequent oxidation test with R100 (exact duration 1.5 h, regeneration 0.5 h), the fifth oil sample was taken after about 83 h. At this point, the vegetable oil content was 1.4 %.

As the vegetable oil content is only available with the declaration "< 1 %" in the measurements after 42 h and 49 h, the actual dilution rate during the application phase is unknown. However, the trend line over the oxidation test shows that its dilution rate must be lower than that of the high temperature application. One reason for this could be increased oil dilution during the late main injection without throttling, as this leads to a lower temperature level in the combustion chamber.

Assuming that the engine oil dilution exclusively originates from the phase when the exhaust gas temperature was elevated, the engine oil dilution would be at a maximum of 0.2 %/h. Compared to an assumed engine oil dilution rate of 4 %/h by a conventional active DPF regeneration with rapeseed oil [5], the described regeneration strategy is capable of reducing the engine oil dilution rate by 95 %.

7 Summary and Conclusions

In order to find a suitable DPF regeneration strategy for engines operated on rapeseed oil, the internal engine processes need to be investigated and their effect on the engine components must be determined. In the study described above, measures to increase exhaust gas temperature were investigated and evaluated with respect to their applicability for DPF regeneration.

When using rapeseed oil, diesel and mixtures of both of these fuels, it is possible to determine internal engine measures for increasing the exhaust gas temperature. In high-idle operation it is feasible to produce exhaust gas temperatures of about 500 °C at the inlet of the DPF. This temperature is sufficient to carry out the DPF regeneration within a reasonable time interval in real operation in tractors suitable for operation with rapeseed oil.

During the tests, the dilution of engine oil with rapeseed oil was investigated and documented. The measures taken to increase the exhaust gas temperature did not increase the engine oil dilution by rapeseed oil. No significant increase of oil dilution compared to the basic measurement with R100 could be noted.

The developed measures can be used to carry out further investigations for increasing exhaust gas temperature in part load operation. This is necessary to integrate the regeneration procedure into regular driving operation.

Since the engine oil dilution in rapeseed oil tractors is a decisive characteristic for the oil service life, it has to be examined whether the reduction of the fuel input of the developed regeneration strategy is sufficient to avoid a reduction of the oil change interval due to the regeneration. This could make previously required additional components for DPF regeneration in tractors adapted for rapeseed oil superfluous in the future, thus providing an advantage in terms of both manufacturing costs and reliability.

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Acknowledgments

The work presented here was carried out within the framework of the project "MuSt5-Trak", funded by the German Federal Ministry of Food and Agriculture (BMEL) with the Agency for Renewable Resources (FNR) as project management organization. The authors would like to express their gratitude for this funding.

Furthermore, the authors would like to thank the project partner John Deere GmbH & Co. KG, Mannheim (Germany) for providing the test engine and exhaust aftertreatment system, as well as for the comprehensive support throughout the project. The authors also acknowledge the Technology and Support Centre (TFZ) Straubing (Germany) for their excellent cooperation in the “MuSt5-Trak” consortium.

8 Definitions/Abbreviations

°CA	degree crank angle
ATDC	after top dead center
BSFC	brake specific fuel consumption
BTDC	before top dead center
CO_2	carbon dioxide
CO	carbon monoxide
DIN	German industrial standard
DOC	diesel oxidation filter
DPF	diesel particulate filter
EGR	exhaust gas recirculation
EN	European standard
FAME	fatty acid methyl ester
HC	hydrocarbons
HVO	hydrogenated vegetable oils
λ	air fuel ratio
NH_3	ammonia
NO_2	nitrogen dioxide
NO_x	nitrogen oxids
O_2	oxygen
$\dot{Q}$	energy/heat flow
RME	rapeseed oil methyl ester
SCR	selective catalytic reduction
TDC	top dead center
THC	total hydrocarbons