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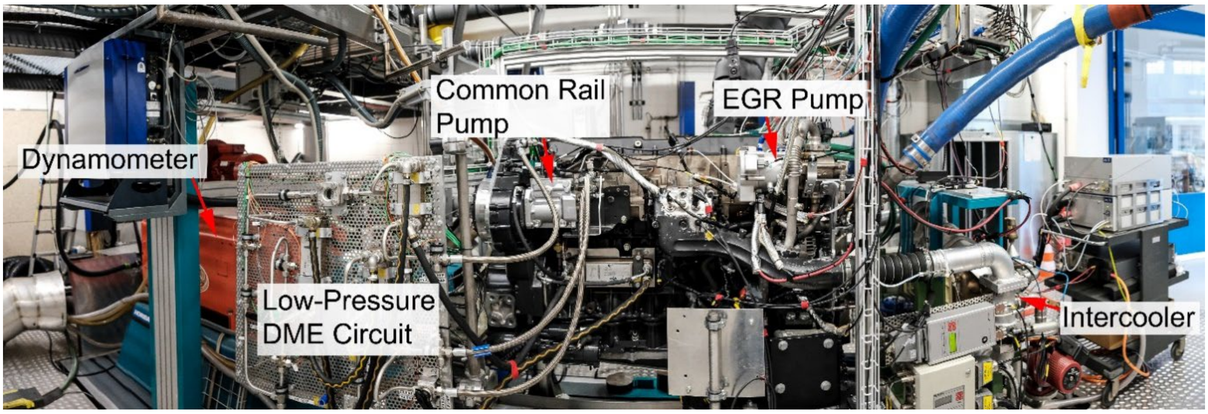
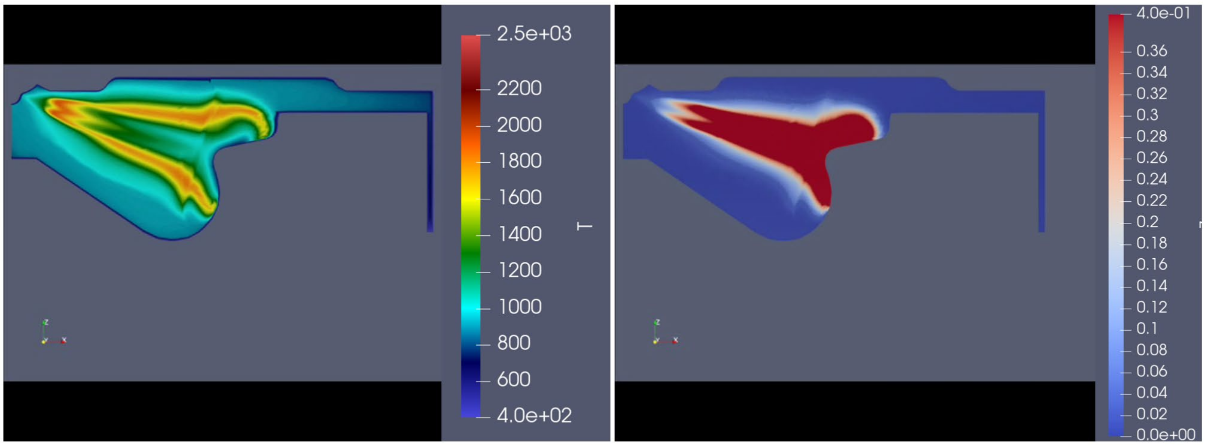
HDV-DME-Part Load

Efficient part-load operation of heavy duty DME engines





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Zusammenfassung

Nutzfahrzeuge und mobile Arbeitsmaschinen stehen unter Druck, sich von fossilen Primärenergieträgern zu befreien. Ein Teil wird elektrifiziert werden können, ein anderer Teil wird aber weiterhin auf chemische Energieträger angewiesen sein, dies speziell bei Anwendungen mit hohem täglichem Energiebedarf (Langstreckenanwendungen, Hochlastanwendungen auf und abseits der Strasse). Betrachtet man mögliche erneuerbare chemische Energieträger wird schnell klar, dass Bio-Treibstoffe zwar sinnvoll und wichtig sind, sie die grossen benötigten Energiemengen aber nicht abdecken können.

Motoren für Nutzfahrzeuge mit weniger als 1..3 gCO_2 per tonnen- oder personen-km werden, zusammen mit batterieelektrischen Fahrzeugen, in Entwürfen der Europäischen Gesetzgebung pauschal als "emissionsfreie Fahrzeuge" bezeichnet, ungeachtet der Herkunft der Primärenergie. Es ist darum sehr wahrscheinlich, dass neben Batteriefahrzeugen auch strombasierte, kohlenstoff-freie chemische Energieträger eine wichtige Rolle spielen werden. Gleichzeitig wird aktuell um eine adäquate Berücksichtigung kohlenstoffhaltiger solcher sogenannter "renewable fuels of non-biological origins (RFNBO)" gerungen, aber es ist aktuell unklar, ob Lebenszyklus-Betrachtungen in zukünftigen Gesetzen Einzug finden werden.

In der Wissenschaft ist das Thema erneuerbarer chemische Energieträger für Lastanwendungen ein wichtiges Thema, da eine Technologieoffenheit sehr wichtig ist zur Erreichung von ambitionierten Klimazielen. Betrachtet man mögliche Pfade der Erzeugung chemischer Energieträger aus den in der Atmosphäre vorhandenen Elementen H, C, O und N, stehen aktuell besonders Wasserstoff (H_2), Methan (CH_4), Methanol (CH_3-OH) sowie Ammoniak (NH_3) im Fokus. Wasserstoff und Ammoniak sind besonders attraktiv, da sie bei der energetischen Nutzung kein CO_2 freisetzen. Beide Energieträger haben allerdings recht hohe Herausforderungen zu meistern: bei Wasserstoff ist dies beispielsweise die tiefe volumetrische Energiedichte und sehr teure Betankungs- Transport und Speicherlösungen, bei Ammoniak die Giftigkeit. Es ist deshalb sehr wahrscheinlich, dass auch kohlenstoffhaltige erneuerbare Motor-Treibstoffe in Zukunft bei der Dekarbonisierung eine wichtige Rolle spielen werden müssen.

Methan und Methanol sind beides geeignete Motortreibstoffe, allerdings werden sie in aktuellen Brennverfahren vorgemischt umgesetzt, was im Vergleich zu Diesel tiefere Effizienz und höhere Betriebstemperaturen (kürzere Motorlebensdauer) mit sich bringt. Ein ebenfalls vergleichsweise einfacher chemischer Energieträger ist Dimethylether (DME, C_2H_6O), welcher sich direkt synthetisieren- oder praktisch verlustfrei aus Methanol gewinnen lässt. Seine physikalischen Eigenschaften sind ähnlich zu jenen von Flüssiggas, d.h. die Transport-, Betankungs- und Speicherlösungen sind sehr kostengünstig. Zudem hat DME eine sehr hohe Cetanzahl – es ist damit der einfachste chemische Energieträger, welcher sich in Diffusionsbrennverfahren (Dieselbrennverfahren) umsetzen lässt.

In einem Vorgängerprojekt wurde an einem 11 Liter Nutzfahrzeugmotor gezeigt, dass DME eine Effizienz auf Dieselmotorniveau ermöglicht und dies bei gleichzeitig sehr tiefem Schadstoffemissions-Niveau. Bei niedrigeren Lasten ergeben sich, wie beim Diesel auch, Schwierigkeiten aufgrund der niedrigen Temperatur-Dosiergrenze des Abgasnachbehandlungssystems. Um dieses Problem zu lösen, wird beim Diesel der Luftüberschuss reduziert und/oder der Wirkungsgrad durch eine nicht-optimale Verbrennung oder durch den Einsatz einer Abgasklappe verschlechtert, was die Abgastemperatur anhebt. Aufgrund der nicht russenden Verbrennung von DME wird im Rahmen dieses Projekts ein alternativer Ansatz untersucht: der sogenannte PCCI-Betrieb mit spätem Einspritzbeginn (SOI) und hohen Abgasrückführungsraten (AGR).

Die gewählte Strategie zeigt, dass hohe AGR-Raten in Kombination mit einer späten Einspritzung geringere Verbrauchseinbussen ermöglichen als der konventionelle Abgasklappenbetrieb. Darüber hinaus können hohe AGR-Raten die Kraftstoffeffizienz in bestimmten Niedriglast-/Leerlaufzuständen aufgrund geringerer Pumpverluste sogar verbessern. Geringere NO_x-Emissionen am Motorausgang bedeuten auch geringere Auspuffemissionen während der Warmlaufphase bis zur Aktivierung des SCR-Systems.

Schliesslich ermöglichen die höheren Auslasstemperaturen des Motors die Dosierung des Reduktionsmittels für die selektive katalytische Reduktion, damit zukünftige sehr strenge Emissionsgrenzwerte mit einem vergleichsweise einfachen Nachbehandlungssystem erreicht werden können.

Weitere Verbesserungen im Hinblick auf den Kompromiss zwischen NO_x und THC/CO bei instationären Bedingungen sind mit einer ausgefeilten AGR-Regelung möglich, während gleichzeitig Kompressorpumpen bei Lastabnahme minimiert werden kann.

Numerischen Ergebnisse unterstützen die experimentellen Erkenntnisse und zeigen, dass für die untersuchten Bedingungen eine gute bis sehr gute Übereinstimmung in Bezug auf Druckverläufe und Wärmefreisetzungsraten und sogar Emissionen zu erkennen ist. Angesichts der extrem geringen Reaktivität der untersuchten Bedingungen, insbesondere bei den sehr späten Einspritzzeitpunkten, kann die numerische Plattform dennoch als vielversprechend für die Untersuchung weiterer Betriebsstrategien angesehen werden.

Dieses Projekt hat also die Grundlagen geschaffen, einen DME-betriebenen effizienten Motor mir sehr tiefen Schadstoffemissionen im gesamten Betriebsbereich, inklusiv der sehr anspruchsvollen Kaltstartphase, technisch umsetzen zu können.

Résumé

Les véhicules utilitaires et les engins mobiles sont sous pression pour se libérer des sources d'énergie primaire fossiles. Une partie d'entre eux pourront être électrifiés, mais une autre partie continuera à dépendre de sources d'énergie chimiques, en particulier pour les applications nécessitant une grande quantité d'énergie quotidienne (applications à longue distance, applications à forte charge sur et hors route). Si l'on considère les sources d'énergie chimique renouvelables possibles, il devient rapidement évident que les biocarburants sont certes utiles et importants, mais qu'ils ne peuvent pas couvrir les grandes quantités d'énergie nécessaires.

Les véhicules utilitaires émettant moins de 1..3 gCO_2 / tkm ou pkm sont, avec les véhicules électriques à batterie, globalement qualifiés de "véhicules sans émissions" dans les projets de législation européenne, quelle que soit l'origine de l'énergie primaire. Il est donc très probable qu'en plus des véhicules à batterie, les sources d'énergie chimiques sans carbone et basées sur l'électricité joueront un rôle important. Parallèlement, on se bat actuellement pour une prise en compte adéquate des carburants carbonés dits "renewable fuels of non-biological origins (RFNBO)", mais on ne sait pas encore si les considérations relatives au cycle de vie seront prises en compte dans les futures lois.

Dans le domaine scientifique, le thème des agents énergétiques chimiques renouvelables pour les applications de charge est un sujet important, car l'ouverture technologique est essentielle pour atteindre des objectifs climatiques ambitieux. Si l'on considère les voies possibles de production de sources d'énergie chimiques à partir des éléments H, C, O et N présents dans l'atmosphère, l'hydrogène (H_2), le méthane (CH_4), le méthanol (CH_3-OH) et l'ammoniac (NH_3) sont actuellement au centre des préoccupations. L'hydrogène et l'ammoniac sont particulièrement attrayants, car ils ne libèrent pas de CO_2 lors de leur utilisation énergétique. Ces deux sources d'énergie doivent toutefois relever des défis de taille : pour l'hydrogène, il s'agit par exemple d'une faible densité énergétique volumétrique et de solutions de ravitaillement, de transport et de stockage très coûteuses ; pour l'ammoniac, il s'agit de sa toxicité. Il est donc fort probable qu'à l'avenir, les carburants renouvelables contenant du carbone devront également jouer un rôle important dans la décarbonisation.

Le méthane et le méthanol sont tous deux des carburants appropriés pour les moteurs, mais ils sont transformés dans les procédés de combustion actuels, ce qui entraîne une efficacité moindre et des températures de fonctionnement plus élevées (durée de vie plus courte du moteur) par rapport au diesel. Le diméthyléther (DME, C_2H_6O) est également un vecteur d'énergie chimique relativement simple, qui peut être synthétisé directement ou obtenu à partir du méthanol sans pratiquement aucune perte. Ses propriétés physiques sont similaires à celles du gaz liquide, ce qui signifie que les solutions de transport, de ravitaillement et de stockage sont très avantageuses. De plus, le DME a un indice de cétane très élevé, ce qui en fait le vecteur d'énergie chimique le plus simple à transformer dans des procédés de combustion par diffusion (procédé de combustion diesel).

Dans un projet précédent, il a été démontré sur un moteur de véhicule utilitaire de 11 litres que le DME permettait d'atteindre une efficacité équivalente à celle d'un moteur diesel, tout en maintenant un niveau d'émissions polluantes très bas. En cas de charges moteur plus faibles, des difficultés apparaissent, comme pour le diesel, en raison de la limite de dosage liée à basse température du système de traitement des gaz d'échappement. Pour résoudre ce problème, le diesel réduit l'excès d'air et/ou détériore le rendement par une combustion non optimale ou par l'utilisation d'un clapet d'échappement, ce qui augmente la température des gaz d'échappement. En raison de la combustion sans suie du DME, une approche alternative est étudiée dans le cadre de ce projet : le fonctionnement dit PCCI avec un début d'injection tardif (SOI) et des taux élevés de recirculation des gaz d'échappement (EGR).

La stratégie choisie montre qu'un taux élevé d'EGR, combiné à une injection tardive, permet des réductions de consommation plus faibles que le fonctionnement conventionnel du volet d'échappement. En outre, des taux élevés d'EGR peuvent même améliorer l'efficacité du carburant dans certaines conditions de faible charge/de ralenti en raison de pertes de pompage réduites. Des émissions de NO_x plus faibles à la sortie du moteur signifient également des émissions plus faibles à l'échappement pendant la phase de réchauffement jusqu'à l'activation du système SCR.

Enfin, les températures d'échappement plus élevées du moteur permettent de doser l'agent réducteur pour la réduction catalytique sélective, de sorte que les futures limites d'émission très strictes pourront être atteintes avec un système de post-traitement relativement simple. D'autres améliorations concernant le compromis entre NO_x et THC/CO dans des conditions instationnaires sont possibles avec une régulation EGR sophistiquée, tout en minimisant le pompage du compresseur en cas de baisse de charge.

Les résultats numériques soutiennent les conclusions expérimentales et montrent que, pour les conditions étudiées, il existe une bonne, voire une très bonne concordance en termes d'évolution de la pression et de taux de dégagement de chaleur, voire d'émissions. Compte tenu de la réactivité extrêmement faible des conditions étudiées, en particulier pour les moments d'injection très tardifs, la plateforme numérique peut néanmoins être considérée comme prometteuse pour l'étude d'autres stratégies de fonctionnement.

Ce projet a donc jeté les bases de la réalisation technique d'un moteur efficace fonctionnant au DME et présentant de très bonnes émissions de polluants dans toute la plage de fonctionnement, y compris dans la phase très exigeante du démarrage à froid.

Summary

Commercial vehicles and mobile machinery are under pressure to free themselves from fossil primary energy sources. It will be possible to electrify some of them, but others will continue to rely on chemical energy sources, especially for applications with high daily energy needs (long-distance applications, high-load applications, on and off the road). Looking at possible renewable chemical energy sources, it becomes clear that although biofuels are useful and important, they cannot cover the large amounts of energy required.

Engines for commercial vehicles with less than 1..3 gCO_2 per ton- or person-km, together with battery electric vehicles, are generally referred to as "zero-emission vehicles" in draft European legislation, regardless of the origin of the primary energy. It is therefore very likely that electricity-based, carbon-free chemical energy sources will play an important role alongside battery vehicles. At the same time, adequate consideration is currently being given to carbon-containing so-called "renewable fuels of non-biological origins (RFNBO)", but it is currently unclear whether life cycle considerations will be included in future legislation.

In the scientific community, the topic of renewable chemical energy sources for load applications is important, as technological openness is crucial for achieving ambitious climate goal. Looking at possible paths for generating chemical energy sources from the elements H, C, O and N present in the atmosphere, the focus is currently on hydrogen (H_2), methane (CH_4), methanol (CH_3-OH) and ammonia (NH_3). Hydrogen and ammonia are particularly attractive as they do not release any CO_2 when used in energy conversion devices. However, both fuels have quite significant challenges to overcome: for example, the low volumetric energy density of hydrogen and very expensive refueling, transport and

storage solutions, and the toxicity of ammonia. It is therefore very likely that carbon-based renewable motor fuels will also have to play an important role in decarbonization in the future.

Methane and methanol are both suitable engine fuels, but they are pre-mixed in current combustion processes, which results in lower efficiency and higher operating temperatures (shorter engine service life) compared to diesel. Another comparatively simple chemical energy source is dimethyl ether (DME, C_2H_6O), which can be synthesized directly or obtained from methanol with virtually no losses. Its physical properties are similar to those of liquid gas, i.e., the transportation, refueling and storage solutions are very cost-effective. In addition, DME has a very high cetane number, making it the simplest chemical energy carrier that can be converted in diffusion combustion (diesel combustion process).

In a previous project, it was shown on an 11-liter commercial vehicle engine that DME enables efficiency at diesel engine level and at the same time very low pollutant emission levels. At lower loads, as with diesel, difficulties arise due to the low temperature dosing limit of the exhaust gas aftertreatment system. To solve this problem, the excess air in diesel engines is reduced and/or the efficiency is worsened by non-optimal combustion or by the use of an exhaust gas flap, which raises the exhaust gas temperature. Due to the non-sooty combustion of DME, an alternative approach is being investigated in this project: the so-called PCCI operation with late start of injection and high exhaust gas recirculation rates (EGR).

The chosen strategy shows that high EGR in combination with late injection enables lower fuel consumption losses than conventional exhaust gas flap operation. In addition, high EGR rates can even improve fuel efficiency in certain low load/idle conditions due to lower pumping losses. Lower NO_x emissions at the engine outlet also mean lower tailpipe emissions during the warm-up phase until the SCR system is active.

Finally, the higher exhaust temperatures of the engine make it possible to dose the reducing agent for the selective catalytic reduction so that very strict emission limits can be achieved in the future with a comparatively simple aftertreatment system. Further improvements with regard to the compromise between NO_x and THC/CO under transient conditions are possible with a sophisticated EGR control system, while at the same time compressor surging can be minimized during load reduction.

Numerical results support the experimental findings and show that good to very good agreement can be seen for the investigated conditions in terms of pressure curves and heat release rates and even emissions. In view of the extremely low reactivity of the investigated conditions, especially at the very late injection times, the numerical platform can nevertheless be considered promising for the investigation of further operating strategies.

This project has thus laid the foundations for the technical implementation of a DME-powered efficient engine with very low pollutant emissions over the entire operating range, including the very demanding cold start phase.

Contents

Abbreviations

ATS	Aftertreatment System
BDC	Bottom Dead Center (position of the piston closest to the crankshaft)
CFD	Computational Fluid Dynamics
CI	Compression Ignition (standard diesel combustion mode)
CO	Carbon Monoxide
DOC	Diesel Oxidation Catalyst
DPF	Diesel Particulate Filter
ECU	Engine Control Unit
DME	Dimethyl Ether
EGR	Exhaust Gas Recirculation
EO	Engine Out
HRR	Heat Release Rate
MFB	Mass Fraction Burned
LLCA	(Nonroad) Low Load Application Cycle
NO_x	Nitrous oxides (sum of NO, NO2)
NRTC	Nonroad Transient Cycle
PCCI	(Partially) Premixed Charge Compression Ignition
SCR	Selective Catalytic Reduction (NOx reduction technology using ammonia as reduction agent)
SOI	Start of Injection (injector energizing moment)
TP	Tailpipe (downstream aftertreatment system)
TDC	Top Dead Center (position of the piston farthest away from the crankshaft)
THC	Total Hydrocarbon
TCO	Total Cost of Ownership
WHTC	World Harmonized Transient Cycle (for Heavy Duty Trucks and Buses)

1 Introduction

1.1 Background information and current situation

In 2017, Switzerland has signed to the Paris agreement [3,4]. In 2019, the Swiss Federal Council has set a net-zero greenhouse gas emissions goal for 2050 and the Swiss people have accepted a law for implantation in 2023. Possible strategies are described in [5]. It has to be kept in mind that these regulations and strategies account only for domestic greenhouse gas emissions. Around 60% of Switzerland's CO_2 emissions are emitted abroad and about 6-9 times the Swiss domestic greenhouse gas emissions are controlled by Swiss companies [6]. This gives the Swiss society not only the committed responsibility to decarbonize domestically but also to ethical responsibility to provide solutions for decarbonization in an international context, including production and supply chains.

Switzerland is highly integrated in the European market as well as in European legislation in terms of vehicles and machinery. As Switzerland is a transit country as well, energy-related solutions for Switzerland have to focus at least on the European continent. 98% of the European on-road vehicle fleet are light duty vehicles (passenger cars and delivery vans) which use 59% of the total on-road end energy share. Heavy-duty vehicles account for only 2% of the on-road fleet but they use 41% of the end energy share [7]. Not accounted in these numbers are non-road heavy applications e.g. for construction, agriculture, mining as well as maritime applications. This points out the importance to find viable decarbonizing solutions for such very demanding heavy applications with high power demand and with typically daily operation for many hours. Such applications are the focus of the project described here.

Lowest possible fuel consumption, while fulfilling pollutant emission limits, was always a main development goal in the heavy-duty vehicle and on/non-road machinery segment. Fuel costs are often a dominating factor and efficient engines were always a competitive advantage. Since fuel consumption and CO_2 emissions are directly coupled for a given fuel, further CO_2 reductions - while keeping the fuel type - are particularly difficult to achieve in the segment of high-power or long-haul applications. In these segments, battery electrification is often not a viable option. Many use cases would need battery sizes far above 1’000 kWh capacity and very high charging power, which is not feasible for larger fleets and off-grid / field refueling applications. The large-scale introduction of renewable chemical energy carriers in combination with efficient energy converters is therefore a key technology for the decarbonization in these segments. It helps to speed up sustainable powertrain solutions as well as it is an important aspect to increase resilience of our society.

Different renewable chemical energy carriers can be produced in different pathways and from different feedstocks. Biogenic or waste-based feedstocks are a meaningful option but the quantities are too limited for a large-scale replacement of fossil fuels. Most likely, technologies have to be developed which produce large quantities of chemical energy carriers in regions of the world with abundant potential for renewable electricity (photovoltaics and wind). Usually, hydrogen is in focus as a potentially very important chemical energy carrier to decarbonize transport, machinery and industrial processes. While hydrogen, used in fuel cells or combustion engines will play a role in future mobility/machinery applications, its suitability for use cases which operate on long-range, for long-operating-hours or at high-continuous-power is rather limited. High costs, high complexities, large tank volumes and high tank masses are the main reasons. Therefore, the authors believe that renewable liquid energy carriers will be needed to decarbonize such demanding applications.

Drop-in approaches, which can renewably be produced via renewable syngas and followed by Fischer-Tropsch synthesis, are very appealing to decarbonize existing fleets, However, this technology is most likely too expensive to play a major role for a broad application and limited to applications which really need drop-in solutions. Looking into liquid (or liquid ad low pressure levels) renewable chemical energy carriers which can efficiently be produced from power, the scientific community seems to currently favor methanol and ammonia. Ammonia is too dangerous for use in populated areas and it is therefore mainly discussed as a potential future fuel for ocean shipping. This leaves methanol as an energy carrier with a high probability to gain on importance as a chemical energy vector in the future, which can easily be stored and transported.

The authors believe that methanol powertrain technologies will be important in the future for certain applications. However, methanol is a so-called low reactivity fuel which makes it feasible for spark ignited (Otto cycle) engines. For durable and highly efficient engines, compression ignition (Diesel cycle) engines are favorable as component temperatures are lower and real-life efficiency is superior to SI concepts. Compression ignition engines ask for a high-reactivity fuel and luckily, methanol can be converted very cost- and energy-efficiently[1] to such a high-reactivity fuel: dimethyl ether (DME).

DME is not so much in focus of the actual discussion about future fuels. However, recent pollutant emission limit proposals (e.g. Euro 7 [8] or Tier 5 [9]) pose a very high technical and economical challenge for classical heavy-duty approaches, basing on diesel- or diesel-like liquid fuels. Using DME, technologically much simpler and simultaneously energy efficient solutions can be developed as the authors have shown in the project "HDV-DME Investigation of the suitability of DME as an alternative fuel in HDV" [10]. This is why the authors believe that DME as a fuel for heavy-duty application should gain more attention and that mature technologies should be developed to demonstrate the feasibility.

Internationally, DME has gained on importance in recent years. One driver is the LPG industry which sees renewable DME a way to decarbonize LPG applications. Large-scale investments are underway to ramp-up the production of renewable DME with the companies Oberon Fuels (San Diego, CA, USA) and Dimeta (Leiden, Netherlands) being major drivers. California has approved the use of DME as a motor fuel in 2015, Germany has recently defined a fuel standard for DME (DIN/TS 51698 [11]) and major economies are discussing updated legislation frameworks for renewable fuels (e.g. [12,13] for the EU). Recently, the US DOE has initiated a number of projects in the field of DME for commercial vehicle propulsion and the UK Department for Business, Energy & Industrial Strategy are funding DME engine technology developments under the so-called "Red Diesel Replacement competition".

Recently, DME has been proposed as a more efficient way than ammonia to transport hydrogen or methanol over long distances in a closed carbon cycle [14]. It is important to note that the density of liquid hydrogen at a temperature of 33K is 71 g/l and 1 liter of liquid DME contains 87 g of hydrogen. In other words: 1 liter of DME contains 22% more hydrogen than one liter of liquid hydrogen. If renewable DME will turn out to become an important molecule for chemical energy transportation over long distances, the direct use of renewable DME will naturally gain much more attention than today.

Summarizing the current situation: a lot of renewable fuels are under discussion at the moment and it is very hard to predict, which paths will be economically viable in the future. Hydrogen, methanol and ammonia receive actually a lot of attention but the authors believe that DME should play a role in the future renewable fuel mix as it offers a number of essential practical, technological and economic advantages.

1.2 Purpose of the project

In a predecessor project [2], the project team has established the current state-of-the-art for heavy-duty DME engine technology. To do so, a DME supply infrastructure was set up at a heavy-duty engine test bench at Empa in Dübendorf and a FPT Cursor 11 engine was modified accordingly (see *Figure 1*).

[1] DME is today produced in considerable quantities, mainly as a solvent or an eco-friendly propellant for spray cans. From own experience buying DME, the price of DME is 10% above the price of methanol (comparing identical chemical energy content) which means that these 10% have to cover energetic losses, investment costs and profit. The energetic efficiency for converting methanol to DME is estimated to be around 95-97%.

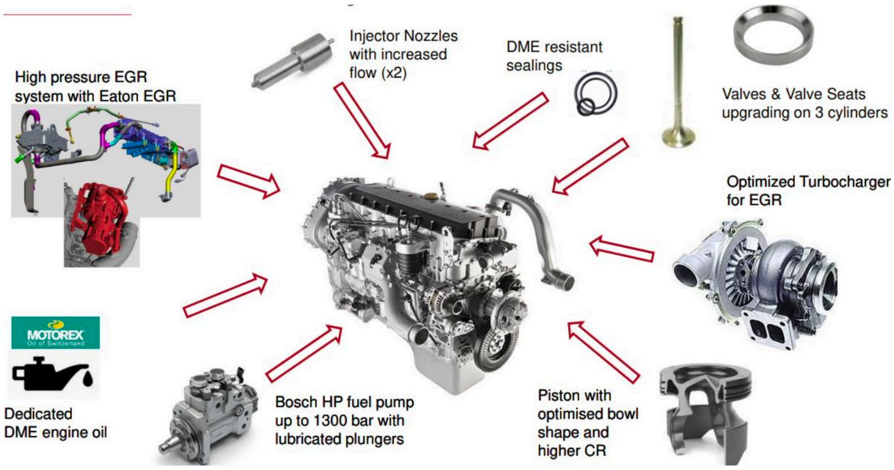

Figure 1: Modifications on the FPT Cursor 11 engine for R&D on DME combustion

The engine was equipped with a special system, which is commercially not yet available: an electrically driven volumetric pump for the control of exhaust gas recirculation EGR (see red parts in *Figure 2*). This setup enabled the control of high-pressure EGR independently from the pressure boundary conditions on the turbocharger's high-pressure side.

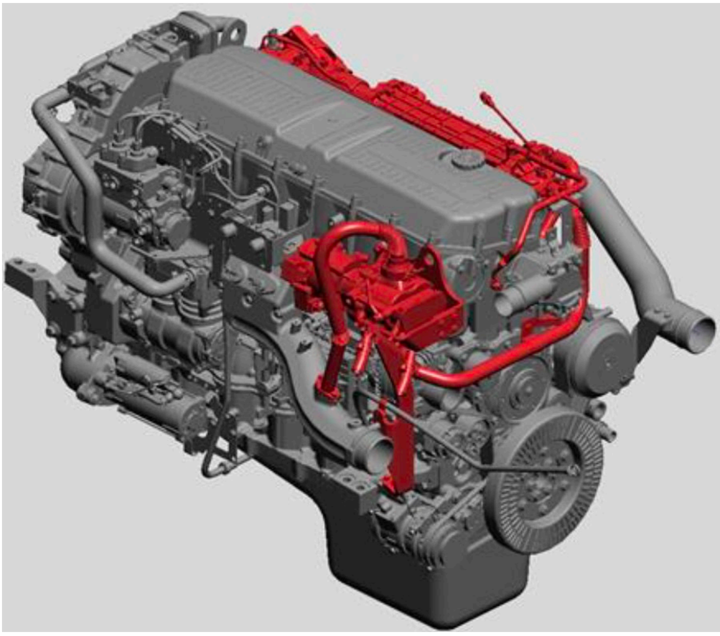

Figure 2: EGR pump setup

Due to a careful design of the combustion chamber geometry and the injection components, excellent results were achieved. Not only was it possible to increase engine efficiency versus Diesel operation, but also a completely soot-free combustion could be realized. Soot-free operation could be set independently from the desired NO_x level – meaning that the classical NO_x-soot trade-off known from Diesel operation basically does not exist anymore. The reason for soot-free diffusion combustion is the lack of C-C bonds in the DME molecule. *Figure 3* shows the resulting efficiency map, *Figure 4* shows the NO_x levels pre- and post-exhaust gas aftertreatment system and *Figure 5* shows according temperatures.

It can be seen that a very good efficiency level could be achieved with DME and that NO_x emissions at tailpipe position are very low. The exception is the region marked yellow where the $deNO_x$ function had to be reduced to prevent ammonia slip or the build-up of residuals from the $deNO_x$ fluid. In this yellow region, the temperature of the exhaust gas aftertreatment system is below about 200°C.

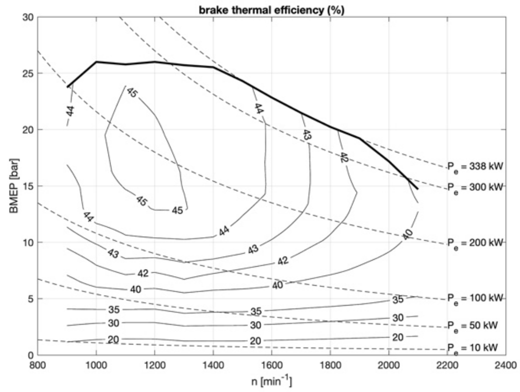

Figure 3: Efficiency map of the FPT Cursor 11 DME engine

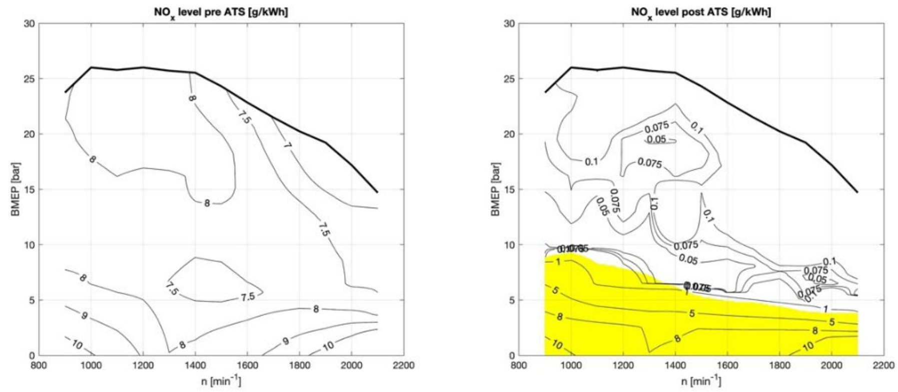

Figure 4: NOx levels pre- and post-exhaust gas aftertreatment system

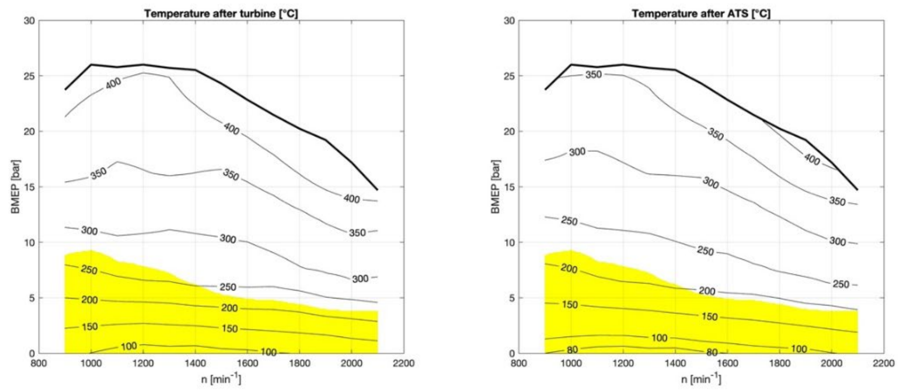

Figure 5: Temperature levels pre- and post-exhaust gas aftertreatment system

In order to enable $deNO_x$ functionality in the temperature region below approximately 200°C, the exhaust gas temperature would have to be increased, which means that the engine efficiency is deteriorated.

This is the usual approach for diesel as a fuel and can be achieved for example with throttling measures and post-injection strategies.

For DME, another approach exists: leave the diffusion-controlled combustion regime at part load conditions by introducing a considerable amount of “low temperature” premixed combustion, called late Partially Premixed-Charge Compression Ignition (PCCI). Partially homogeneous premixed conditions means that air-fuel mixture can approach the walls of the combustion chamber which then leads to a poor fuel conversion rate when classical Diesel fuel is used. A very detailed recent numerical and experimental study concluded [15]: “PCCI can therefore be considered particularly attractive for low boiling point fuels with high ignition propensity. In particular DME, which is injected at supercritical conditions, thereby promoting mixture formation and avoiding significant spray-wall interaction is an interesting candidate.”

The purpose of the project was therefore to quantify the potential of late PCCI on the FPT Cursor 11 engine setup, particularly to enable very low-NO_x combustion in the low engine load regions where deNO_x aftertreatment is critical. This was done experimentally in steady-state and transient conditions, as well as numerically.

1.3 Objectives

The main goal of the project described here is to find and understand partially premixed combustion schemes at part load conditions of a heavy-duty engine operated with DME. This breaks down to the following objectives:

- Understand low-temperature chemistry/combustion for DME, update DME combustion models for CFD use.
- Find optimal late PCCI combustion modes for lowest NO_x emissions and highest engine efficiency at relevant part-load conditions.
- Evaluate and understand the role of Exhaust Gas Recirculation (EGR) for reactivity and NO_x control in late PCCI combustion mode.
- Evaluate, if an adaption of the effective compression ratio is essential/beneficial.
- Compare late PCCI combustion with classical combustion modes and quantify the advantage for an on- and non-road vehicle mission.
- Assess the feasibility of late PCCI combustion of DME from a manufacturer's perspective.

2 Description of the experimental setup

The experimental setup of the test bench, the DME fuel supply as well as the engine setup were mostly taken over from a predecessor project [10]. The needs to address the low load temperature challenges where identified and different measures were analyzed in this project as mentioned in the introduction. It was therefore important that the experimental setup remained unchanged regarding measurement equipment and as far as possible engine components to allow consistency with the previous experiments.

2.1 Engine setup

The base engine as shown *Figure 6: FPT Cursor11 base engine with high pressure EG* is an FPT Cursor11 heavy-duty 6 cylinders in-line diesel engine with 338 kW peak power and a peak torque of 2300 Nm, equipped with a special cooled High-Pressure EGR system (see *Figure 2*). The peak cylinder pressure can reach 250 bar in order to accommodate a high compression ratio of 20.5:1.

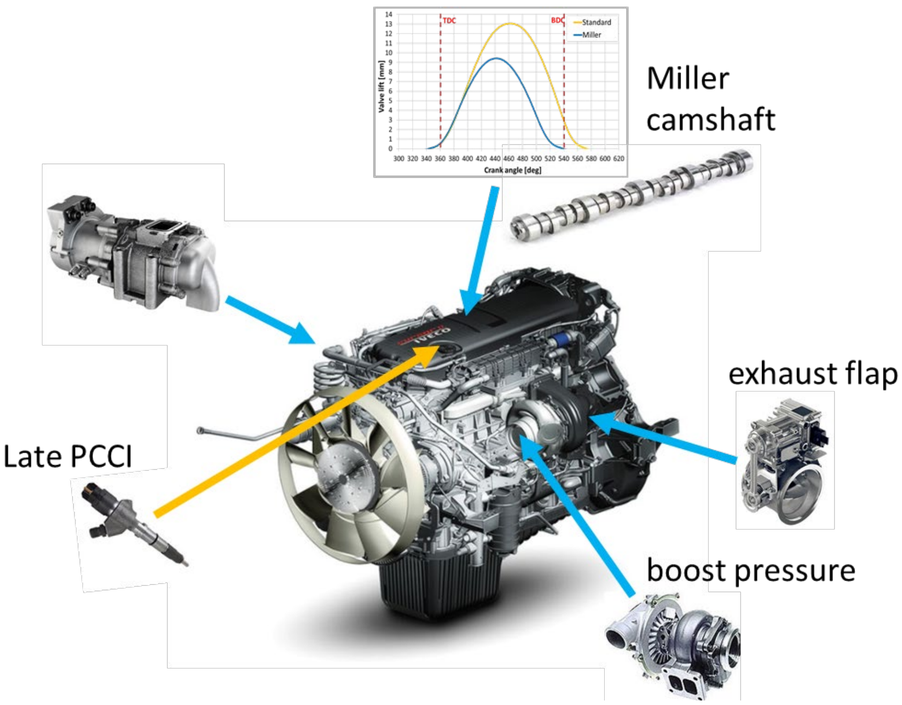

Figure 6: FPT Cursor11 base engine with high pressure EGR and components/parameters investigated for PCCI combustion and low load temperature management

Also shown in *Figure 6* are the several engine components/parameters that were investigated in this project to address the low load emissions and temperature challenges described in the introduction. These are described in detail in the following sections.

2.1.1 EGR-Pump

Apart from the DME specific changes the engine is equipped with an electrically driven 48V EGR-pump allowing full control over the recirculated exhaust gas quantity. The operation of this pump and the EGR rate are thus independent of the pressure ratios across the turbocharger as the pressures on the high-pressure side of the turbocharger is usually limiting the application of flexible EGR rates. Classically, backpressure levels have to be increased in certain load points to enable EGR, which deteriorated engine efficiency – this measure is not necessary with such a pump. The EGR rate is theoretically only limited by the combustion stability and the speed/power limits of the pump itself. This is especially true for a fuel like DME with a virtually soot free combustion.

This flexibility allowed the assessment of the influence of different EGR rates on the PCCI combustion mode investigated in this project. The ratio of exhaust gas recirculation is expected to influence the ignition delay and thus the premixed portion of the fuel. High EGR rates have the benefit of low NO_x emissions. In addition, through reduction of the fresh air flowing through the engine, the desired higher engine out temperatures can be reached at low-load conditions.

2.1.2 Miller camshaft

One way of influencing the conditions (temperature and pressure) inside the combustion chamber relevant for combustion modes like PCCI is to adapt the compression ratio of the engine. This can be done by changing the design of the piston or by changing the intake valve closing timing and thus adapting the effective start of compression. In this project it was chosen to change the camshaft of the engine to allow a so-called Miller valve timing, which is characterized by an early closing of the intake valve before piston bottom dead centre leading to a lower effective compression ratio. This allowed to retain the piston and the combustion chamber design. To investigate the influence of the adapted valve timing experiments were conducted with a standard and a Miller camshaft.

2.1.3 Exhaust flap

To address the low load temperature challenge for the exhaust aftertreatment system the base engine is equipped with an exhaust flap located at the turbine outlet of the turbocharger. It is used in production diesel engines to throttle the engine leading to lower efficiency and airflow of the engine and therefore to increase exhaust temperatures. As such it served as a reference measure for emission and temperature management at low load operation and also was combined with late PCCI combustion and EGR to investigate the potential of multiple strategies to reduce fuel consumption penalties usually encountered when applying throttling measures on an engine.

2.1.4 Boost pressure

One measure to influence the conditions (temperature and pressure) inside the combustion chamber for PCCI and also reduce air to fuel ratio leading to higher exhaust temperatures is to adapt the boost pressure. In production diesel engines in low load operation the turbocharger is set to produce more boost than needed to be able to deliver good transient response in load steps and low soot by providing enough airflow to the engine. Due to the properties of a soot-free combustion in a DME engine this need to hold available a certain airflow is less pronounced. Therefore, a boost reduction in low load and idle operation is expected to increase efficiency by lower pumping losses as well as better conditions for increased premixed fuel shares in PCCI combustion mode.

2.1.5 Engine instrumentation relevant for combustion analysis and CFD / 1D engine simulation

Figure 7 shows the relevant instrumentation used to provide the combustion and injection parameters needed for numerical simulations. They consist of in-cylinder, intake and exhaust pressure measurement for the reference cylinder 1 combined with a crank angle encoder to reference the swept volume to the pressure traces. Additionally, a needle lift sensor integrated in the Injector on cylinder 1 was also used to help define the hydraulic start of injection as the also applied current clamp gives only the energizing timing which can differ noticeably from the actual injection start. This is especially important for numerical simulation and engine models to describe the injection discharge rate, timing and therefore heat release and combustion reactions as precisely as possible to ensure good agreement with real engine operation.

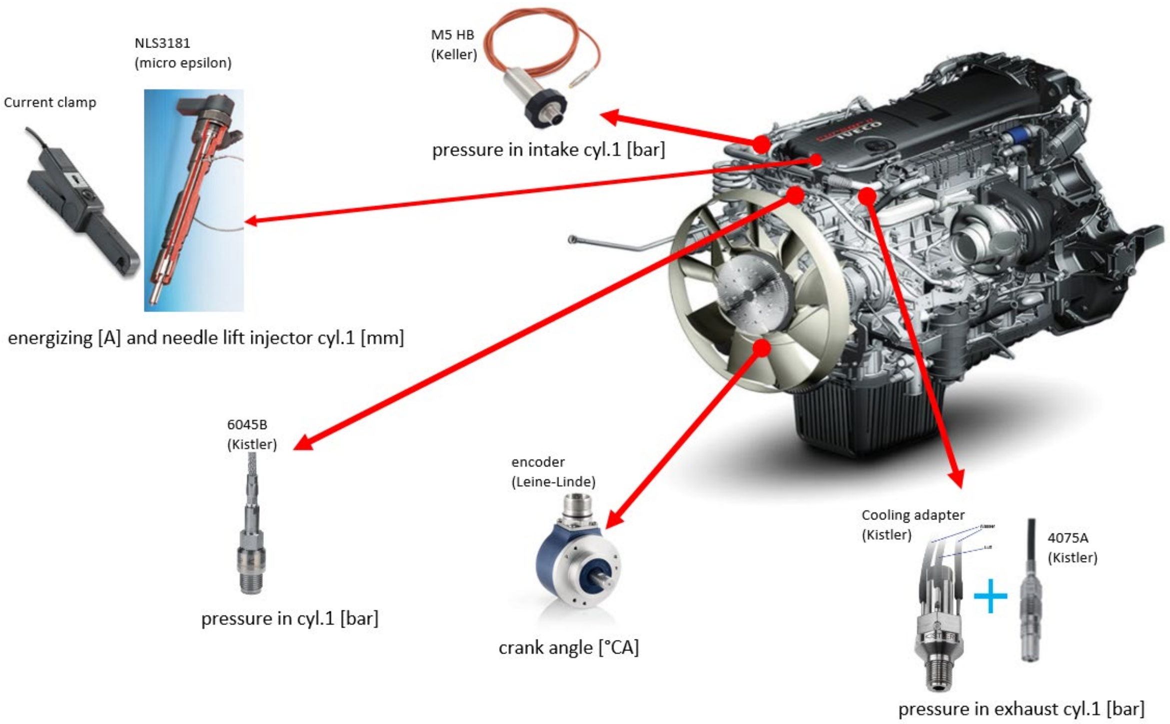

Figure 7: Relevant instrumentation for combustion analysis and 1D/CFD engine simulation

2.2 Engine test facility

Figure 8 shows a picture of the engine mounted on the test bench. The test bench and instrumentation specifications are listed in *Table 1* and *Figure 9* shows a schematic of the experimental setup.

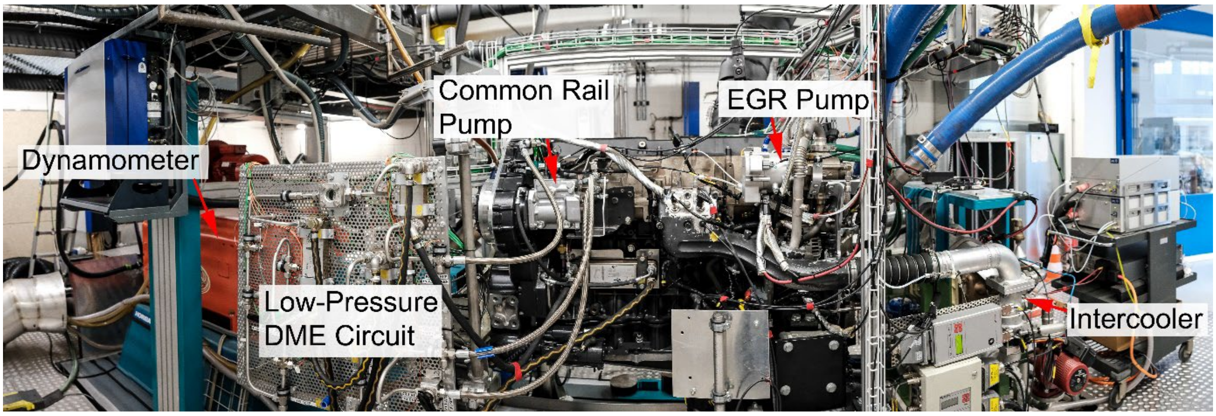

Figure 8: General view of the engine setup on the test bench

Table 1: Main parameters of the engine test bench and engine instrumentation

AC dynamometer	Horiba Dynas3 HD600 (600 kW / 3957 Nm peak absorbing power / torque)
Automation system	Horiba STARS Engine
Low-level test bench control	Horiba SPARC
Gaseous emission measurement system 1	Horiba Mexa 7500 DEGR (two emission measurement lines, one EGR measurement line)
Gaseous emission measurement system 2	Gasmet FTIR (calibrated for DME)
Gaseous emission measurement system 3	Siemens LDS (NH_3 measurement)
Particle measurement system 1	AVL Micro Soot Sensor (photoacoustic)
Particle measurement system 2	PMP compliant particle number measurement system with a volatile particle remover, counting all particles above 23 nm
Combustion air	Conditioned (T=22°C, rH=58%)
Air mass flow measurement	ABB Sensyflow P
Fuel flow measurement	Two Coriolis mass flow sensors (Endress+Hauser Cubemass 300)
DME supply	Feed at 26 bar_{abs}, backflow at tank pressure level
Intake channel pressure indication	Keller M5HB sensor
Cylinder pressure indication	Kistler 6045B sensor
Exhaust channel pressure indication	Kistler 4075A sensor (in cooling adapter)
Fuel	DME (purity 99.99 mass%), no additives

In order to guarantee reproducible conditions the combustion air is conditioned regarding temperature as well as humidity. Fuel mass flow is measured with a double-Coriolis-system which outputs the flow to the common rail pump as well as the fuel return to the DME cylinder.

EGR is quantified by measuring the CO_2 concentration in the intake manifold. Regulated emissions are measured with a certification-grade emission bench which sampled both the engine's raw emissions as well as the emissions after the exhaust gas after treatment system. In addition, a Fourier-Transform-Infrared-Spectrometer is used to measure gaseous emissions. Typically, automotive FTIR spectrometers are calibrated for Alcohols, Acids and Aldehydes. However, a DME-dedicated spectral emission evaluation method is implemented by the device manufacturer for this project. Not shown in *Figure 9* is the ammonia measurement system which is a non-sampling laser diode spectrometer (i.e., the laser diode and the respective receiver are directly placed in the exhaust gas stream after the exhaust gas after treatment system).

Particle emissions are quantified prior any exhaust gas treatment using a photoacoustic sensor (soot mass).

Thermocouples (type K) and pressure sensors (Keller 33X series) are placed at many points but the full instrumentation is not described here.

The test bench facilitates steady-state as well as dynamic operation of the engine. All time-based data is recorded with a sample rate of 10 Hz using adequate low-pass filtering to meet the Nyquist criterion. For steady-state measurements, the 10 Hz data is averaged over 75 s in post processing.

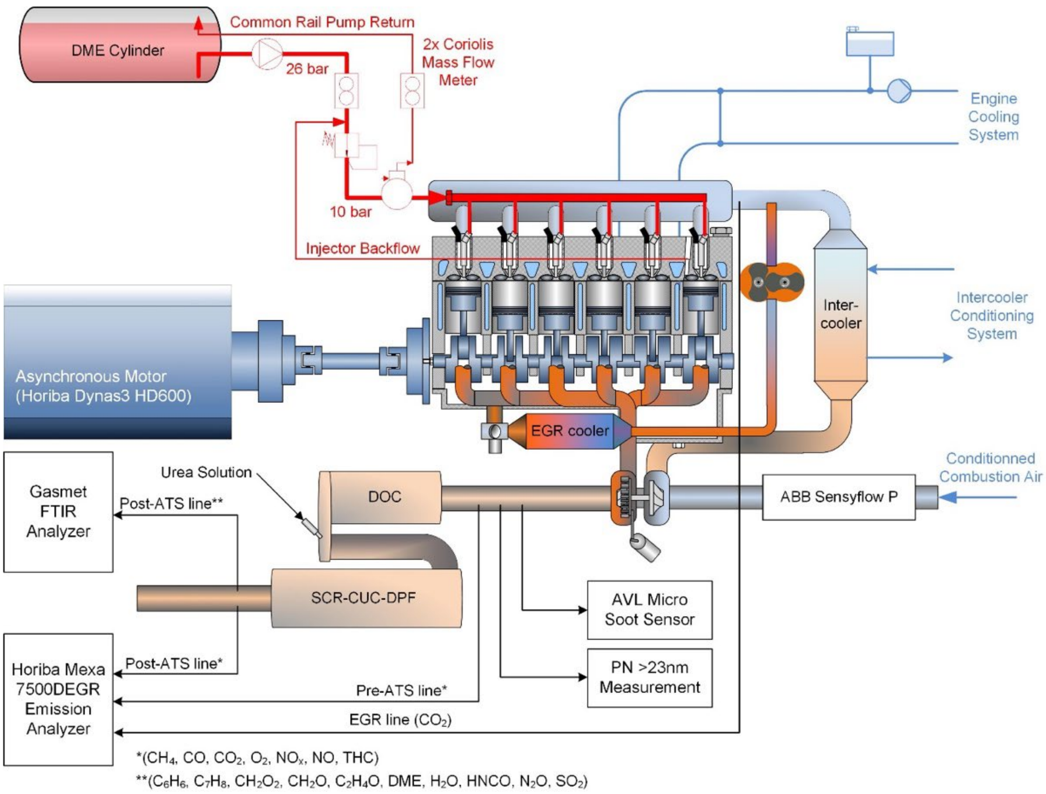

Figure 9: Schematic of the experimental setup (not shown: ammonia measurement post-ATS)

3 Experimental results and discussion

During the experimental campaign, different steps were taken to investigate late Partially Premixed-Charge Compression Ignition (PCCI) and its application on the DME engine in low load operation. The majority of the experiments were conducted in steady state conditions but also some transient engine test cycles experiments were examined.

In a first step, start of injection (SOI) variations were done for several low load operating points (including idling) in the range where exhaust gas temperatures are usually too low for efficient NO_x reduction by the SCR system as mentioned in the introduction. In those operating ranges, low engine out emissions are crucial. This is also especially true for cold start and warm-up conditions. Those variations were done for all the components/measures influencing reactivity and/or temperature and pressure conditions in the combustion chamber introduced in chapter 2.1 to investigate their isolated influence on PCCI combustion.

In a second step the different engine internal NO_x reduction and temperature increasing measures were combined to assess the potential of those combinations on PCCI combustion and low load emission and temperature behavior. Also a comparison was done to the classic approach of only operating an exhaust flap as a throttle to increase exhaust temperature at low load in current Diesel engines.

Finally, the late PCCI combustion strategy was integrated into a transient low NO_x control strategy incorporating dynamic EGR rate control using the 48V EGR pump as well as late injection (PCCI) in low load and idling operation. This also showed the high potential of the full EGR flexibility provided by the volumetric EGR pump compared to a very limited transient EGR control with a standard EGR flap to ensure lowest engine out emission levels for addressing future emission legislation in low load, cold start and warm-up conditions.

3.1 Late Partially Premixed-Charge Compression Ignition (PCCI) with DME in low load operation

3.1.1 PCCI combustion, emission and performances comparison for low load and idle point with EGR

To initiate PCCI combustion the start of injection (SOI) was varied towards later phasing to find the injection phasing where an important part of the injected mass would be premixed before combustion would start. With later injection phasing the ignition delay increases due to lower temperature and pressure and thus reactivity caused by the expanding volume of the combustion chamber after top dead centre (TDC).

Figure 10 shows the results for selected engine performance parameters fuel consumption, NO_x engine out emissions, temperature before SCR catalyst and burn duration for the variation of SOI at a low load point. Here negative values of SOI mean late injection after TDC and the reference injection phasing is on the right and set at 4.6° b. TDC for DME operation. The variation of SOI is shown for different EGR rates ranging from 0-50%. It has to be stated that 0% EGR refers to the target set point as the real EGR can be higher due to the EGR-flap and pump having some leakage. As the recirculation of exhaust gas is a proven method to lower engine out NO_x emissions mainly by reducing combustion temperatures and is also reducing the reactivity of the in cylinder charge it was chosen as a first influencing parameter for PCCI combustion to be looked at.

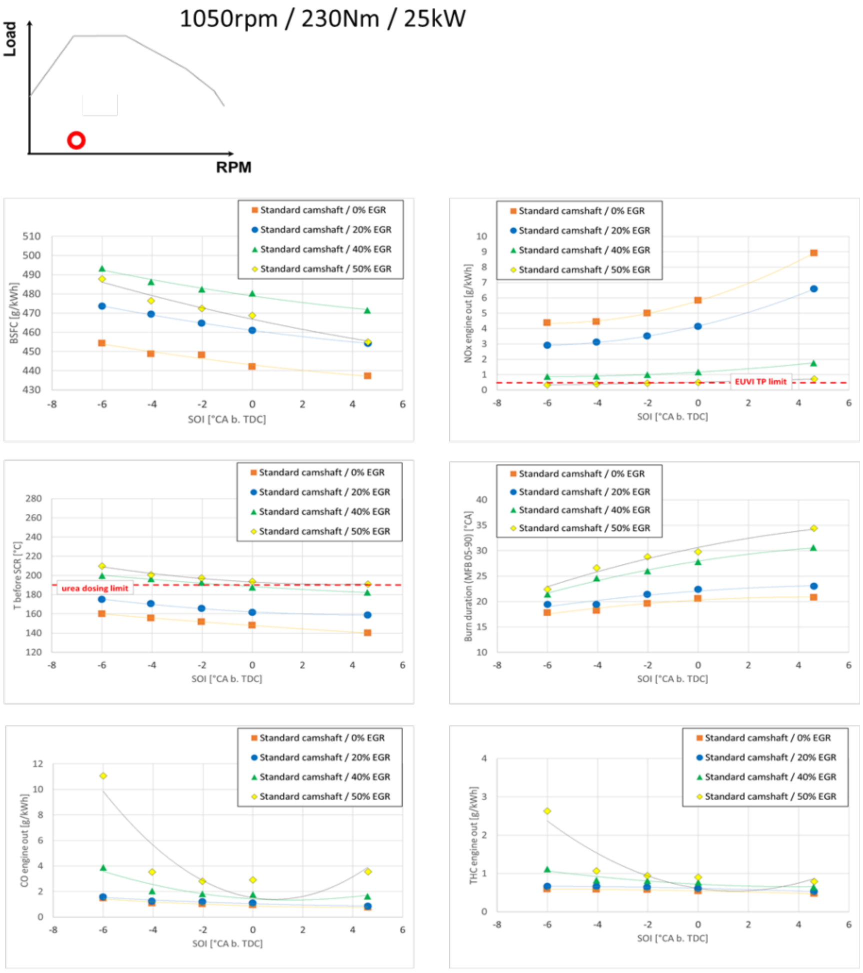

Figure 10: Influence of EGR on relevant parameters at different injection phasing for low load point

As *Figure 10* shows, the NO_x engine out emissions can be roughly reduced by half with late injection phasing in addition to the reduction produced by EGR. Also, later injections than 4° after TDC do not lead to significant NO_x reductions anymore and it was seen during the experiments that after this point CO and THC (Total Hydrocarbon Emissions, here mainly DME (70-90%) and CH_4 for high EGR rates) increase rapidly. It is worth noting that late injection strategy and high EGR rates allow engine out NO_x values in the range of Euro 6 tailpipe emission limits for this low load point. Furthermore, it is also possible to reach or exceed the dosing temperature limit for SCR systems resulting in very low tailpipe emissions. Fuel consumption is increased like expected for EGR and late phasing. Interestingly, at 50% EGR the fuel consumption improves again as the boost pressure is reduced even further by recirculating a considerable part of the exhaust gas and the burn duration is shortened thanks to a high premixed fuel mass share. The burn duration is reduced more rapidly with higher EGR rates as the PCCI combustion is intensified due to the lower reactivity of the in-cylinder charge promoting premixed combustion. The influence of EGR on PCCI combustion is already clearly visible in those results. *Figure 11* and *Figure 12* show the corresponding combustion analysis of cylinder pressure traces and heat release rates for the above-mentioned SOI variations for 0% EGR and 50% EGR. For the "no EGR" case a significant effect of partially premixed combustion is observed for SOI starting at 4°CA after TDC characterized by increased rates of heat release due to the higher premixed combustion portion of the injected fuel. Also visible are the clearly lower pressure (and therefore temperature) conditions for the late injection phasing responsible for the higher PCCI share. For the 50% case it can be observed that the heat release rates are noticeably higher and already present for earlier injection phasing starting at 0°CA (TDC). The SOI point of 6°CA after TDC shows already a decrease in heat release rate due to the very late combustion. This confirms also that the sensible limit for late injection is around 4°CA after TDC to guarantee stable combustion and limited emissions of CO and THC.

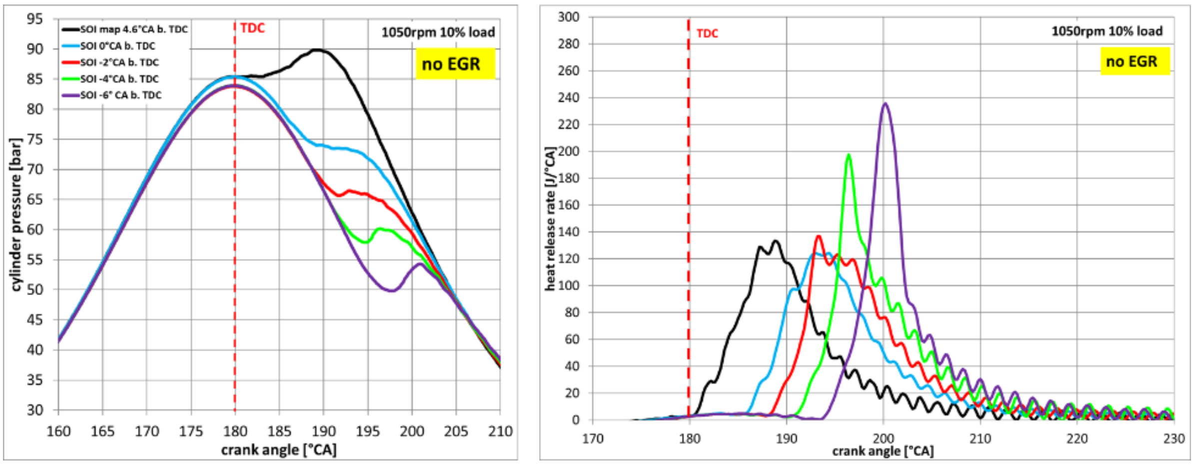

Figure 11: Cylinder pressure and HRR at different start of injection timings without EGR the 10% load point

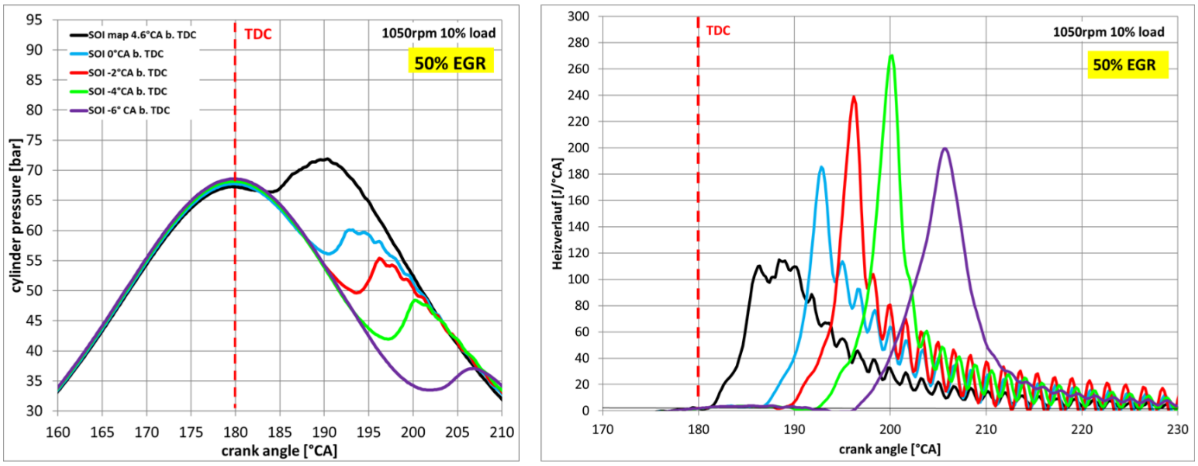

Figure 12: Cylinder pressure and HRR at different start of injection timings with 50% EGR / 10% load point

Figure 13 shows the heat release rates for the earliest and latest start of injection for both EGR cases. Additionally, the energizing and needle lift of the injectors are shown to further investigate the PCCI effect. The surfaces sketched into the needle lift signal represent the progress of the injection process (and thus also approximately the injected mass) until 5% of mass fraction burned (MFB) which represents the start of combustion. It can be seen that for both cases the premixed fraction must be higher with late injection. Especially for the 50% EGR case the injection process is nearly completed at start of combustion and therefor almost completely premixed combustion takes place. This confirms the important influence of EGR on PCCI combustion leading to a fast and low NO_x combustion with the additional NO_x reducing effect of EGR.

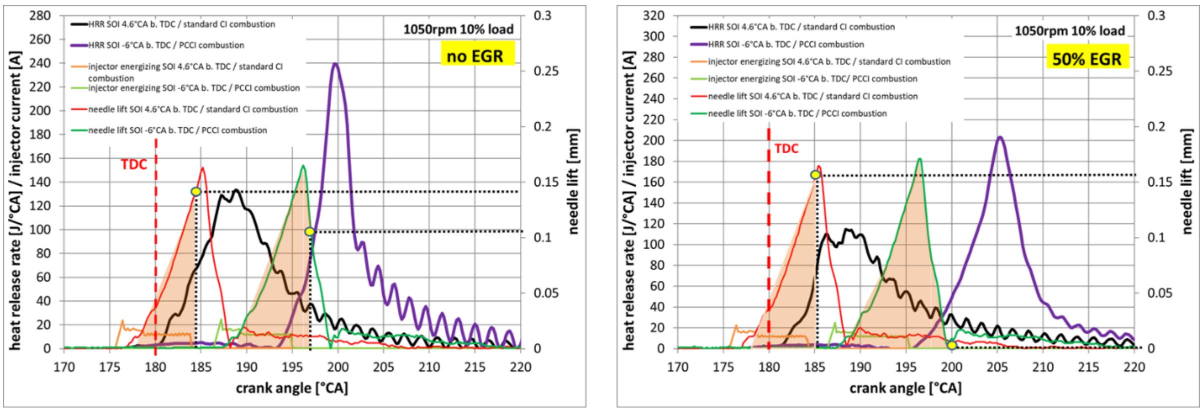

Figure 13: Injection progress at MFB 5% and heat release rate for no EGR and 50% EGR case in low load point

Figure 14 shows the results for selected engine performance parameters fuel consumption, NO_x engine out emissions, temperature before SCR catalyst and burn duration for the variation of SOI at low idling of the engine. To represent an idling point with auxiliary drive/power take-off a small torque of 70Nm was applied. Here negative values of SOI mean late injection after TDC and the reference injection phasing is on the right and set at 3.6° b. TDC for DME operation. The variation of SOI is shown for different EGR setpoints ranging from 0-70%.

As can be seen in *Figure 14* the fuel consumption improves with increasing EGR rates in idling conditions. This is due mainly to the reduced pumping losses as the excess air is reduced and therefore much less air has to be pumped through the engine when relevant parts of the exhaust gases are recirculated. Also in idle for dynamic operation reasons the turbochargers control vanes are set to a fully closed position to be able to ramp up boost pressure and fuelling without excess soot emissions in case of rapid load increase as e.g. in driveaway situation from a standstill. Again this can be optimized in the case of oxygenated fuels like DME. Interestingly the fuel consumption lines flatten out towards higher EGR rates. The NO_x reduction with late injection is therefore possible with less fuel penalty. The NO_x emissions are reduced by more than 75% for all the EGR settings. For 60% and 70% EGR the NO_x emissions are close to or well below Euro 6 tailpipe emissions. Looking at the temperatures at the SCR inlet it can be seen that maximal engine out NO_x emission reduction is key at idling since the temperatures are well below the dosing limit for urea. Interestingly CO and THC emissions decrease for EGR rates 60% and 70% and there seems to be a minimum for the 60% case. Especially for the 70% EGR point the CO and to a lesser extent THC emissions start increase clearly for SOI later than 4°CA after TDC. This SOI can therefore be regarded as a limit for stable combustion phasing. Looking at the combustion duration it can be seen that the duration is not increasing significantly with EGR except for 70% EGR which represents a quite extreme recirculation strategy and were the combustion seems to start slowing down even with higher premixed share.

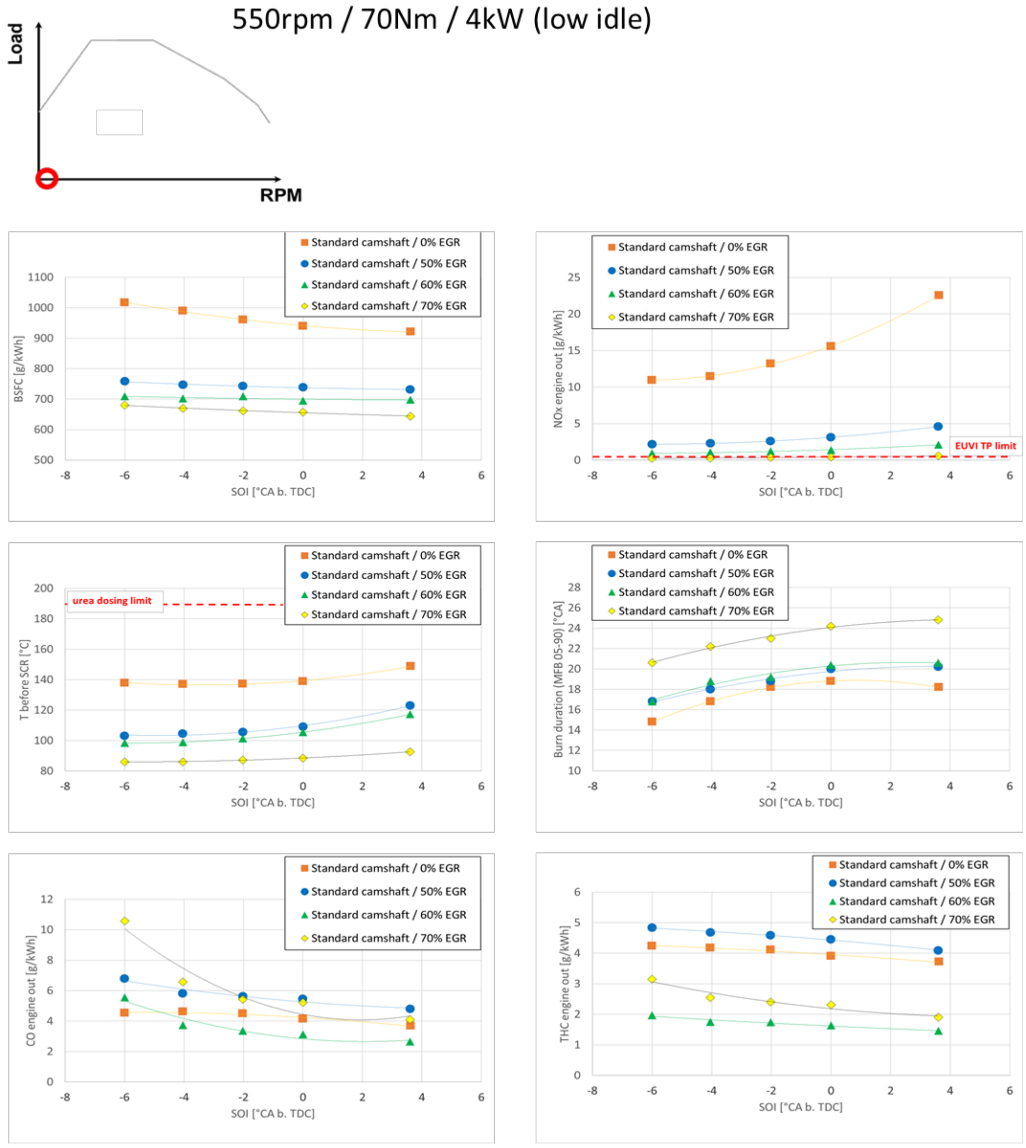

Figure 14: Influence of EGR on relevant parameters at different injection phasing for idle point

Figure 15 and *Figure 16* show corresponding combustion analysis of the SOI variations for the no EGR and the 70% EGR case. Again, it can be seen that EGR has a distinct effect on premixed combustion. With later injection phasing, the heat release rates show more characteristic of fast premixed combustion. For the "no EGR" case this starts for SOI around 2°CA after TDC. In the case of 70% EGR already the earlier injection phasing of 3.6° before TDC shows highly premixed combustion. Looking at the pressure traces it can be seen that the in-cylinder pressures (and therefore also temperatures) are much lower for the 70% EGR case leading to lower reactivity promoting low NO_x PCCI combustion.

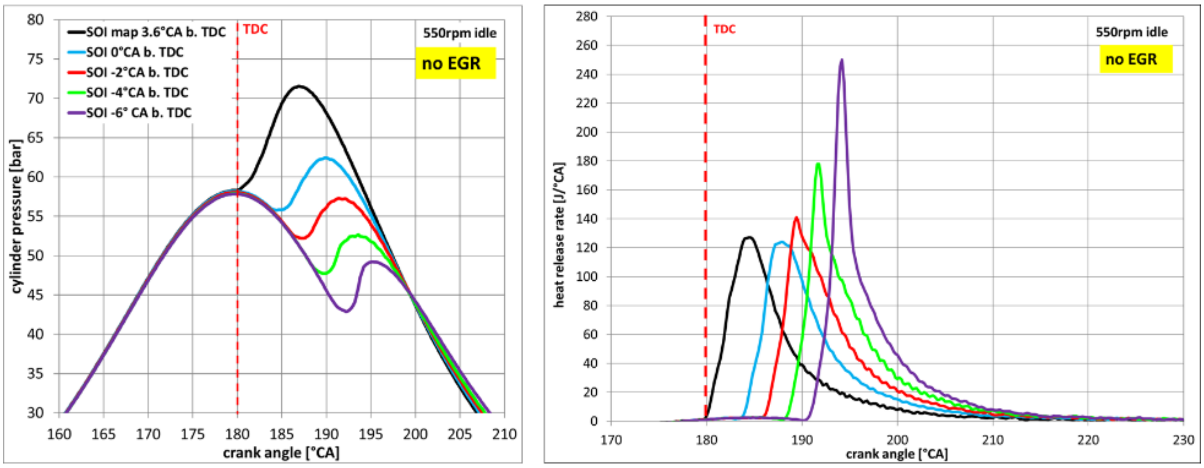

Figure 15: Cylinder pressure and HRR at different start of injection timings without EGR at idling

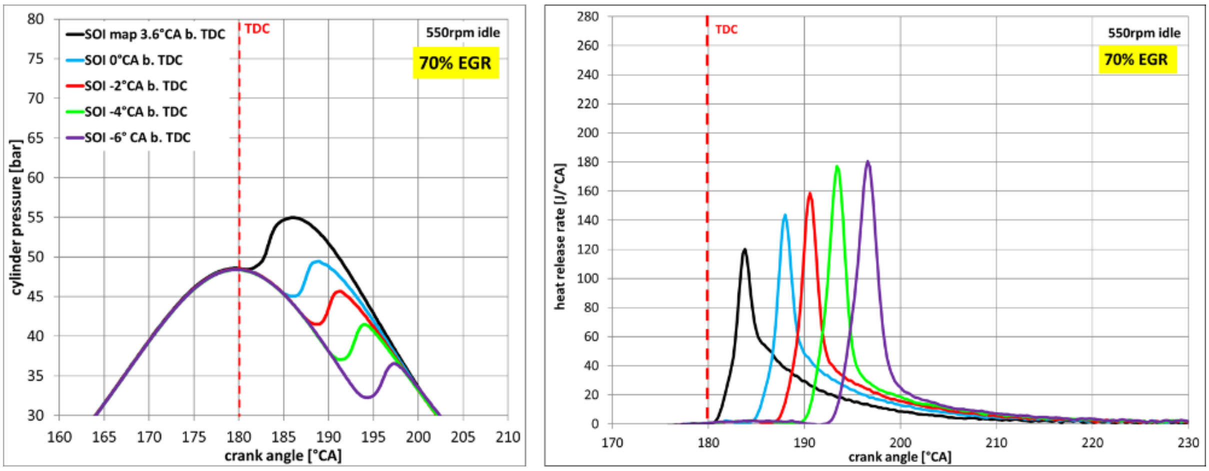

Figure 16: Cylinder pressure and HRR at different start of injection timings with 70% EGR at idling

This is confirmed in *Figure 17* where the detailed analysis of the injection process (injector energizing and needle lift) and combustion progress are shown. The coloured surfaces again represent the injection progress represented with the needle lift signal of the injector while the dots mark the start of combustion (5% MFB) point in the HRR. For the "no EGR" case the injection process is still ongoing at start of combustion although the heat release for the late injection phasing indicates higher premixed share. On the other side the 70% EGR case shows that the injection process is almost complete even for the standard earlier SOI at combustion onset while for the late injection phasing the injector needle is closed before even any heat release can be detected. This means that the combustion is completely premixed without a diffusion fraction anymore.

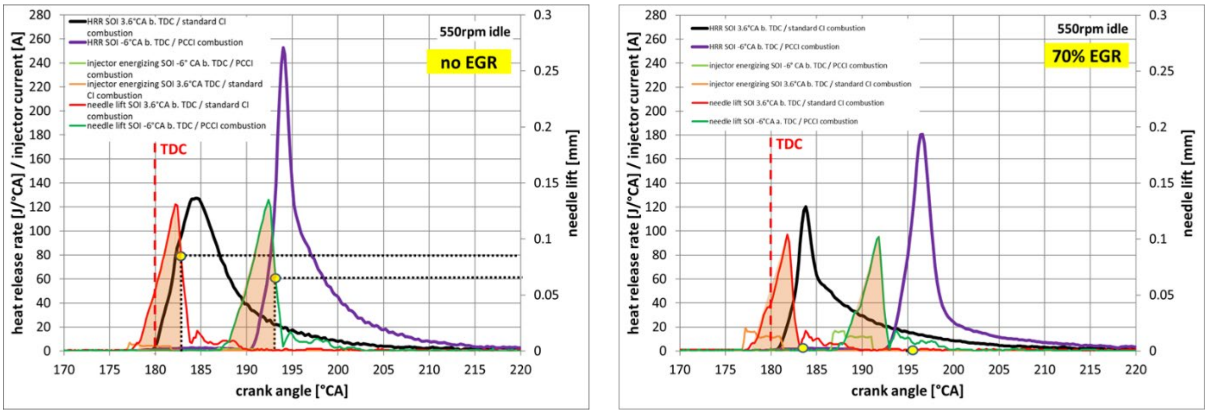

Figure 17: Injection progress at MFB 5% and heat release rate for no EGR and 70% EGR case at idle

3.1.2 PCCI combustion, emission and performances comparison for low load and idle point with Miller camshaft

The previous results were all conducted with the standard production engine camshaft. In this section the influence of a reduced effective compression ratio with a so-called "Miller camshaft" with an early intake valve closing on PCCI combustion, emissions and performance is presented. The early intake closing leads to an under expansion of the fresh air charge and thus to a shorter compression stroke with lower end pressure and temperature. This is expected to also influence the reactivity and efficiency of the engine. It is of course not feasible to change the valve timing depending on the operating point with a classical camshaft setup but systems with some sort of variability in the valve actuation system could provide variable compression ratios across the engine map (as for example studied in the FlexComb project [16]).

Figure 18 shows the valve profiles for the standard camshaft as well as for the Miller variant. The Miller camshaft has additionally also lower valve lift to keep the actuation slope of the cam in the same range as for the standard camshaft for the earlier closing timing. This prevents higher mechanical stress on the components related to a steeper actuation ramp if the lift of the standard cam would have been kept.

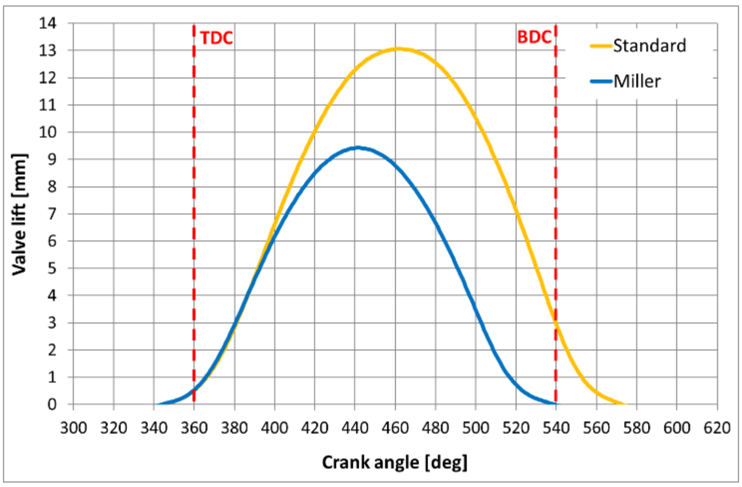

Figure 18: Inlet valve profiles for standard and Miller camshaft

Figure 19 shows the influence of this Miller camshaft on emissions and performance parameters while the boost pressure was held constant for these experiments. Generally, it can be stated that the Miller valve timing has a positive influence on all parameters shown here. Fuel consumption is slightly lower while also lowering NO_x emissions for earlier injection phasing. Later SOI show less influence of the valve timing for both of these values. Temperature before SCR is also positively influenced due to the reduced air to fuel ratio with less trapped air after intake valve closing. Burn durations show lower values for Miller valve timing and late injection indicating some influence on PCCI combustion although with small differences compared to EGR. CO and THC emissions are very similar except for the latest injection phasing which shows clear increase after 4°CA after TDC again indicating the limits of stable combustion when low reactivity measures are applied.

To summarize the influence of the selected Miller valve timing proved to be noticeable but rather limited. On the other hand also small differences of 10-20°C before SCR can lead to noticeable differences in Tailpipe NO_x emissions in this low load operation points and a fuel consumption improvement in the order of 1% is also not negligible. Furthermore the Miller valve timing could be combined with other measures to improve emissions and reduce fuel consumption in low load/idle operation. It has to be stated that with a switchable/flexible valve timing device the Miller effect could be further enhanced at low load operation without compromising performance at high load. The used profiles were adapted to suit the whole engine operating range and therefore the early intake valve closing potential on a system without flexibilities in the valvetrain is rather limited.

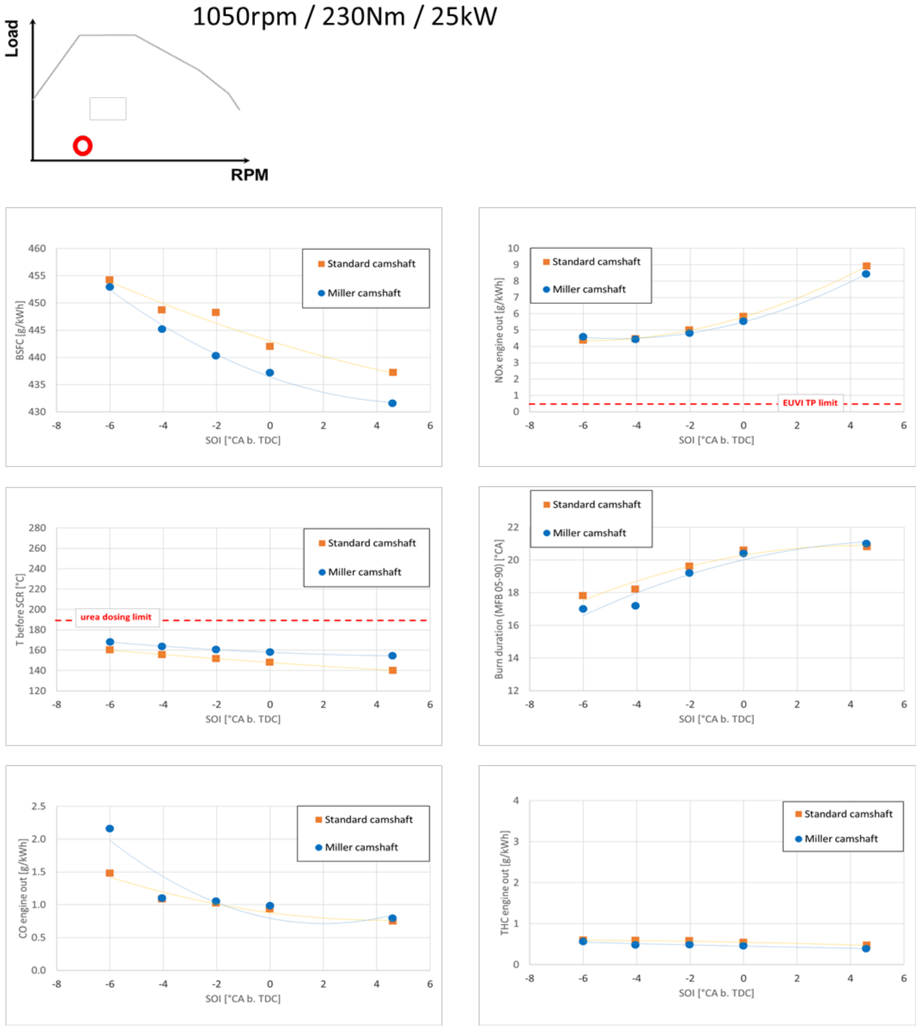

Figure 19: Influence of Miller valve timing on relevant parameters at different injection phasing for the 10% load point

Figure 20 and *Figure 21* show the corresponding combustion analysis for the Miller camshaft. These can be compared with *Figure 11* with standard camshaft. The heat release show an earlier onset of premixed combustion for SOI variations with the Miller camshaft due to the reduced pressures in the combustion chamber. However, both for the reference and late injection case in *Figure 21* the differences in the injection process at start of combustion and therefore PCCI combustion are small. To sum up, there is a noticeable influence of the tested Miller camshaft on emissions and combustion but it remains limited compared to the larger potential of using flexible EGR.

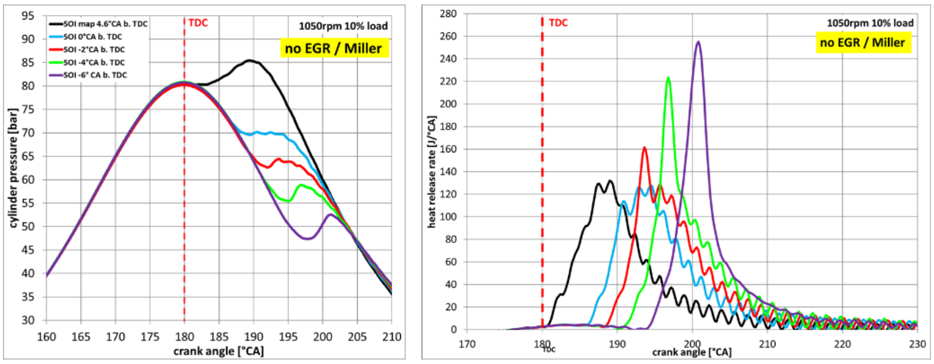

Figure 20: Cylinder pressure and HRR at different start of injection timings with Miller camshaft for low load point

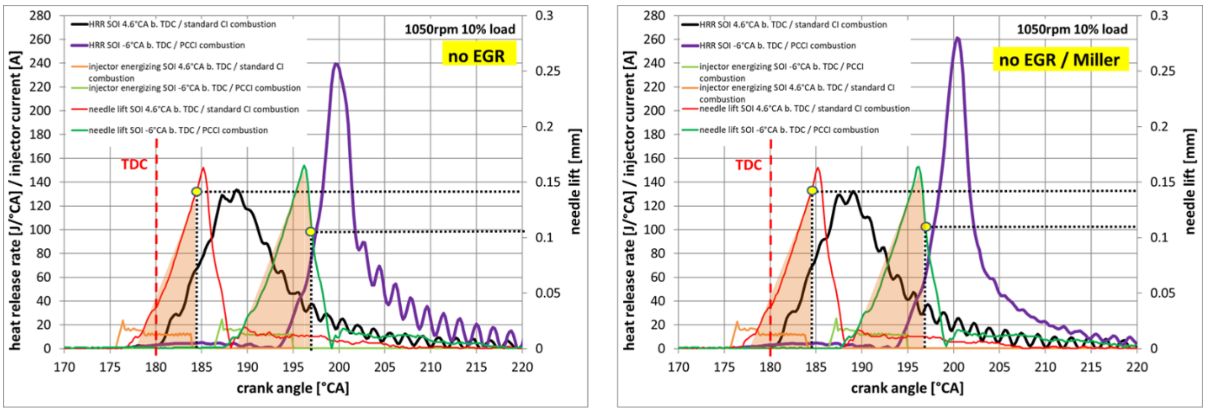

Figure 21: Injection progress at MFB 5% and heat release rate for standard and Miller valve timing at low load point

Figure 22 shows the influence of the Miller valve timing on idle operation. It can be seen that the specific fuel consumption is around 3% lower than with the standard camshaft. NO_x and temperature before SCR remain almost unchanged, the burn duration is slightly shorter compared with the standard cam profile. Additionally, the CO and THC emissions measurements show 20-30% lower values. The Miller valve timing has therefore a more pronounced positive effect on combustion efficiency at idling than the low load point at 1050 rpm.

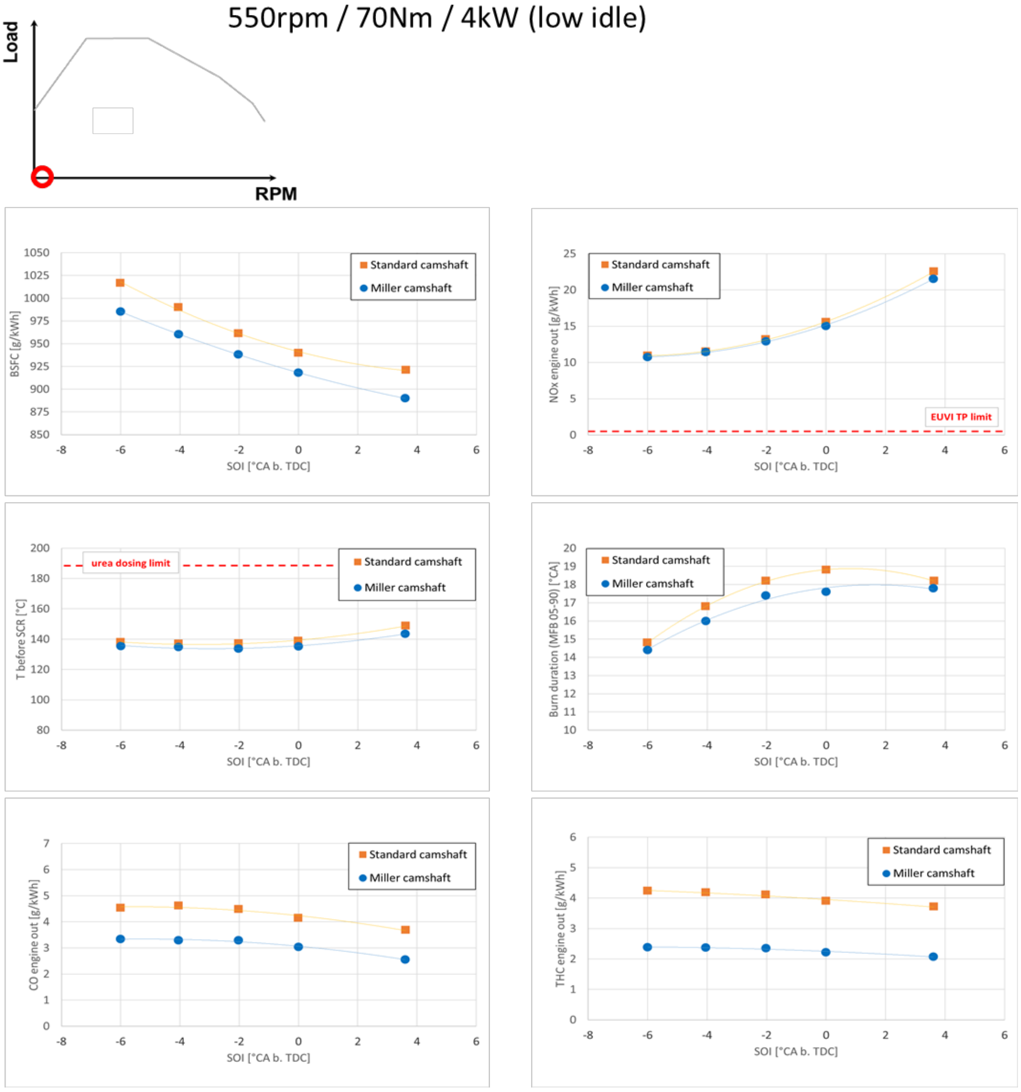

Figure 22: Influence of Miller valve timing on relevant parameters at different injection phasing for idle point

Figure 23 and *Figure 24* show also that the Miller valve timing has a more pronounced effect at idle on PCCI combustion than the low load point at 1050rpm. The premixed share is higher with Miller valve timing and the heat release rates show already premixed characteristics for earlier SOI than with the standard camshaft shown in *Figure 15*. The pressure at TDC is also lower. Compared to the EGR case the change in reactivity is less pronounced and also lacks the effect of substantial NO_x reduction from EGR itself. However, there is a positive effect on combustion efficiency with Miller valve timing.

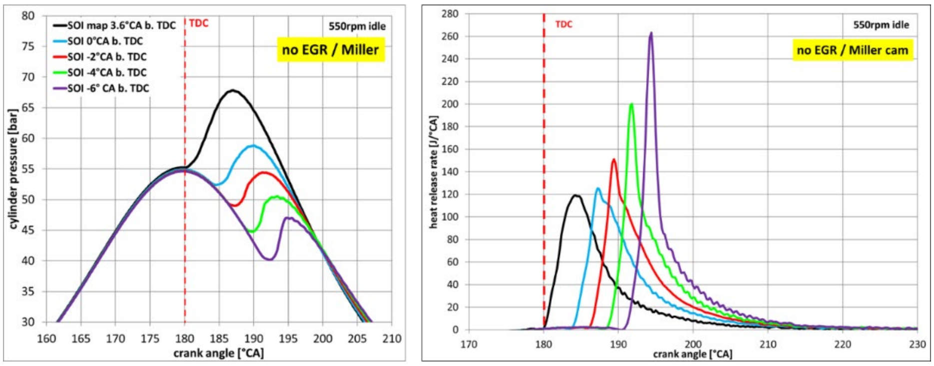

Figure 23: Cylinder pressure and HRR at different start of injection timings with Miller camshaft for idle point

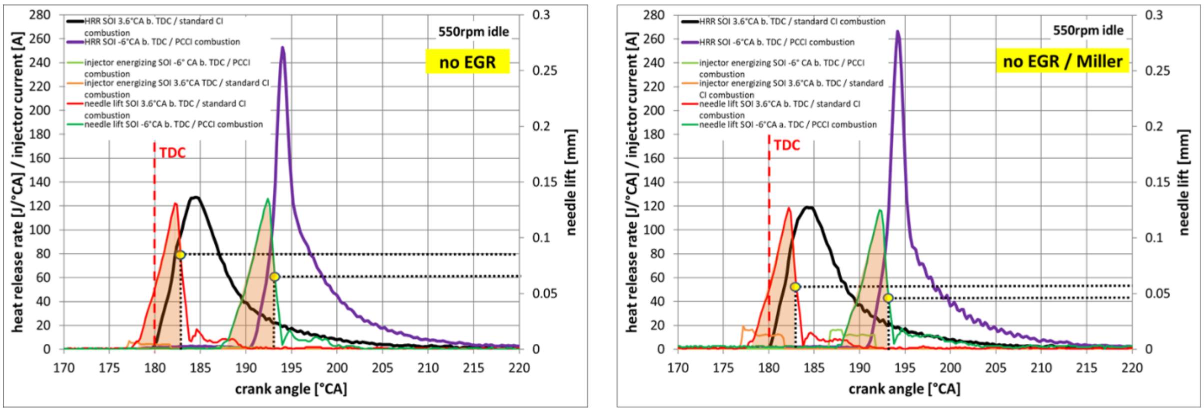

Figure 24: Injection progress at MFB 5% and heat release rate for standard and Miller valve timing at low load point

3.1.3 PCCI combustion, emission and performances comparison for low load point with boost reduction and exhaust flap

In this section the influence of the reduction in boost pressure as well as the application of an exhaust flap for the low load point at 1050rpm and 10% load (25 kW) are described. Such an exhaust flap is usually used in production engines for the thermal management of the exhaust aftertreatment system but it leads to drop in efficiency. Here the influence of this measure on PCCI combustion as well as on general engine parameters like in the previous chapters are shown.

For idling the boost pressures usually are already low and for transient operation reasons the VGT has to be kept almost close to ensure fast boost pressure build-up. Also, the exhaust flap has no significant influence on engine out temperatures and efficiency as the exhaust mass flow is very low and therefore the throttle effect is limited.

Figure 25 shows the relevant parameters of interest for the low load point at 1050rpm comparing standard boost setting and a case with reduced boost (-200mbar) without EGR.

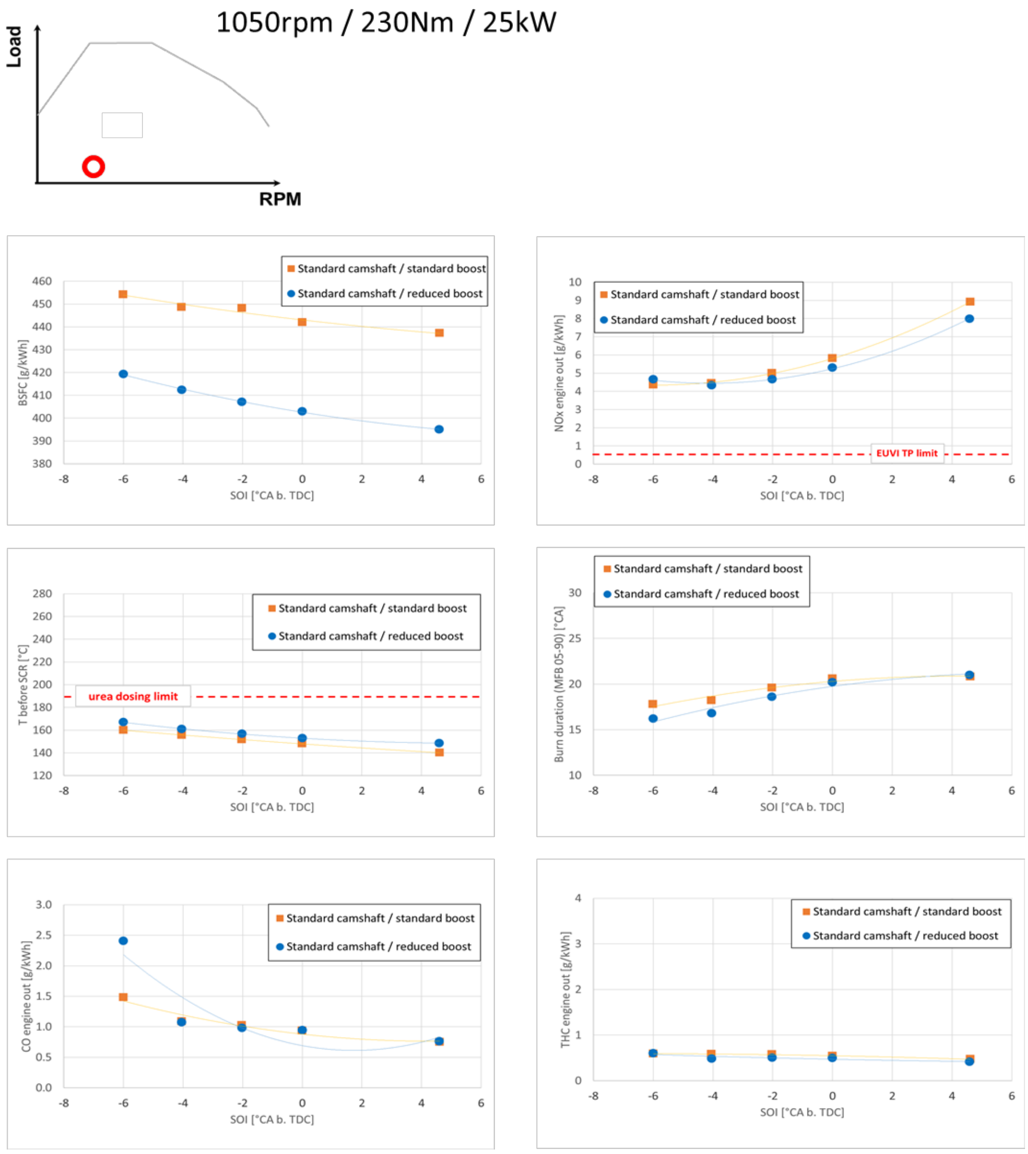

Figure 25: Influence of boost reduction on relevant parameters at different injection phasing for low load point

There is a clear reduction in fuel consumption of around 7-9% by reducing the boost pressure by 200 mbar for this operating point. As mentioned previously the boost is kept higher than necessary at low load operation to favour transient response. With DME this can potentially be reduced as due to the soot free combustion, less excess air has to be held available compared to Diesel engines. Also, the NO_x emissions are reduced slightly for earlier SOI phasing. The lower boosting leads to lower pressures at compression TDC and also lower excess air ratio thus influencing NO_x formation and reactivity. For later injection phasing the pressures are lower due to combustion onset in the expansion stroke and this effect is less pronounced. NO_x emissions reach a minimum at SOI of 4°CA after TDC and retarding SOI further leads to a trend reversal. Such effects have also been observed when lowering the reactivity by reducing the intake air temperature [2]. As discussed therein, the increasing ignition delays lead to higher levels of partial premixing and considerably reduced mixing rates at combustion onset. Hence, increasing portions of NO_x are formed under (partially) premixed conditions and the associated rapid compression heating of a large portion of the mixture.

CO which is a good indicator for combustion completeness is rising sharply after SOI of 4°CA after TDC again showing the limits in reactivity reduction with lower boost. Also the burn duration is shortened with reduced boost in the SOI range where PCCI occurs (starting from SOI 2°CA after TDC) indicating a stronger premixed combustion effect than with standard boost pressure setting.

Figure 26 and *Figure 27* show the corresponding combustion analysis and injection progress (needle lift). As seen in the combustion duration the PCCI combustion starts earlier with reduced boost pressure. This can be seen comparing with *Figure 11* for the standard setting. Looking at the injection progress in *Figure 27* the higher premixed share with reduced boost can be seen from the heat release rates and also the more advanced injection process at combustion start (MFB 5%).

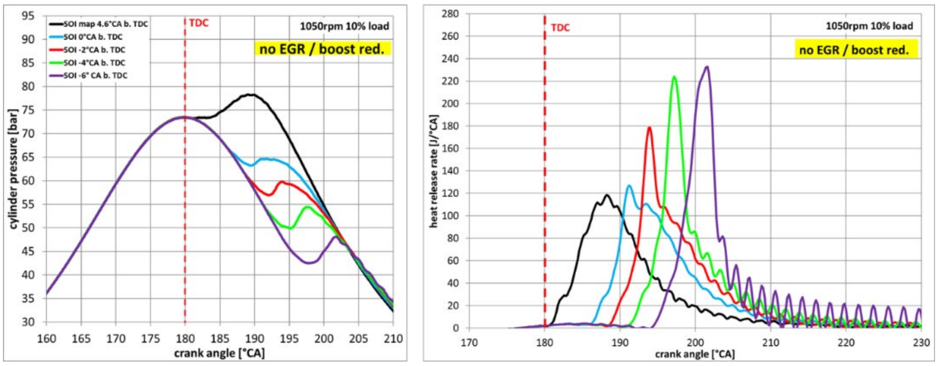

Figure 26: Cylinder pressure and HRR at different start of injection timings with reduced boost pressure for low load point

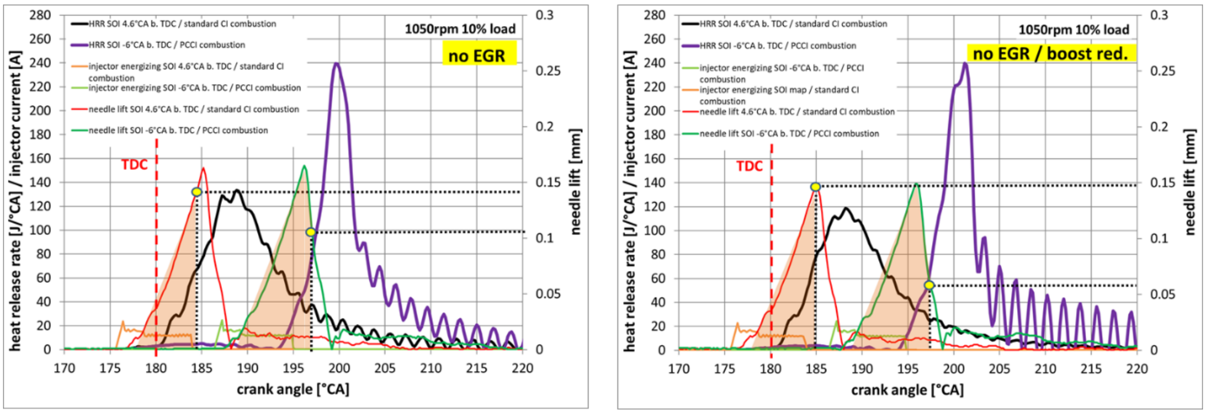

Figure 27: Injection progress at MFB 5% and heat release rate for standard and reduced boost pressure at low load point

Figure 28 shows the influence of the exhaust flap operation for the same low load point at 1050 rpm. This shows one of the classical approaches for thermal management of the exhaust after treatment system by throttling the exhaust side of the engine. This reduces the fresh air charge and increases the pumping losses of the engine leading to higher engine out temperatures. The temperature before SCR is raised by almost 100°C. However, as can be seen in the graph on the top left this measure leads to substantially higher fuel consumption in the order of 15-20% at this operating point. Additionally, the

NO_x emissions engine out are not influenced noticeably, therefore this measure only relies on heating up the system to the urea dosing limit of the SCR to be able to cut tailpipe NO_x emissions. CO and THC emissions are generally higher with activated exhaust flap, the difference is smaller for late injection phasing. When comparing the EGR cases in *Figure 10* it can be seen that with high EGR rates of around 40% and late injection timing substantially lower engine out emissions can be reached while also reaching the dosing limit for urea. Although the temperature cannot be raised as high as with the exhaust flap the substantially lower engine out NO_x emissions reduce the tailpipe emissions during warmup from cold start as well as allowing the SCR system to perform efficiently with lower dosing rates due to the reduced engine out NO_x emissions.

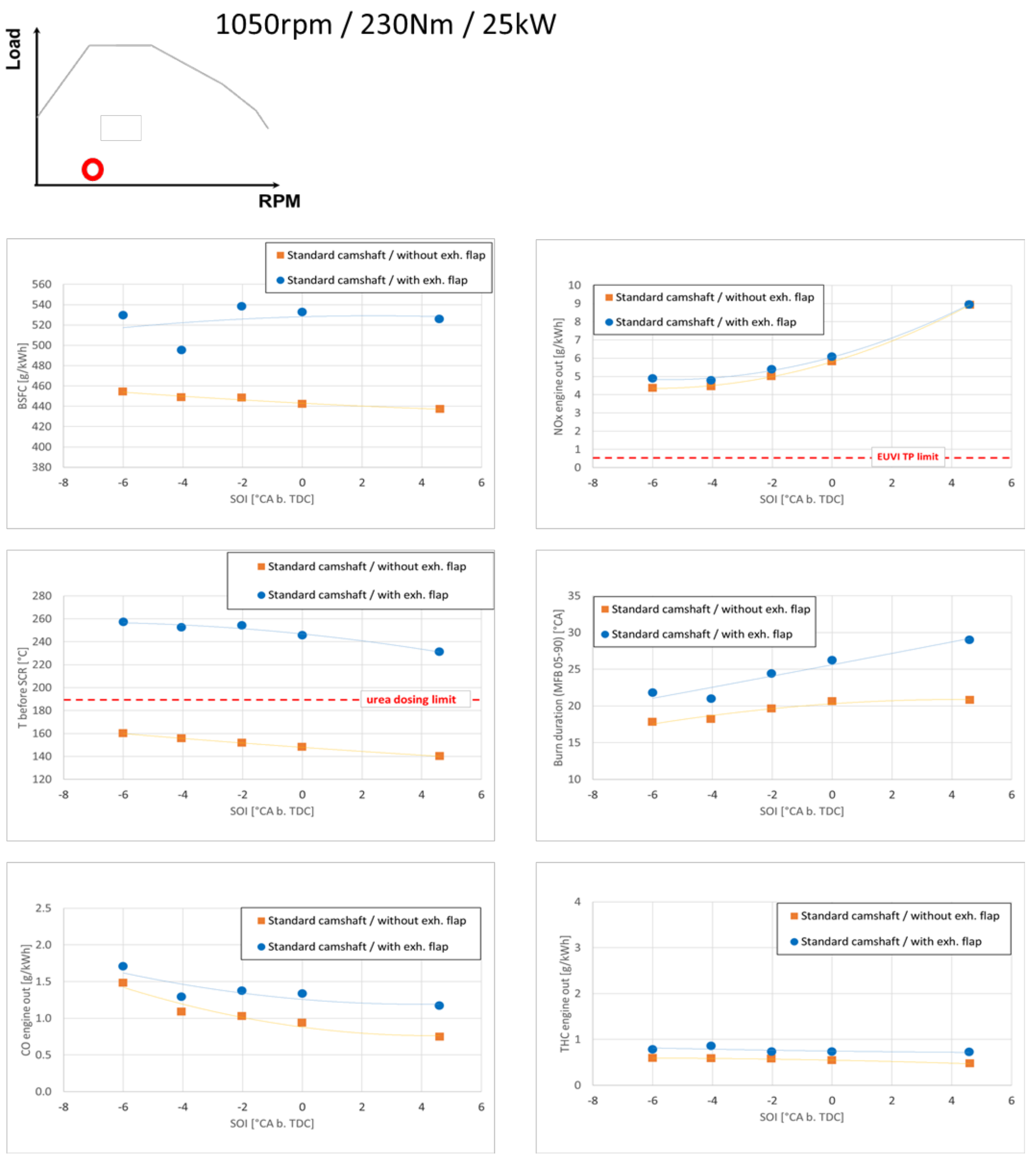

Figure 28: Influence of active exhaust flap on relevant parameters at different injection phasing for low load point

Figure 29 shows the respective heat release rates for the different SOI settings. If compared with *Figure 11* the effect on premixed combustion is relatively low, both cases with and without activated flap show similar heat release rate profiles starting with higher premixed peaks at SOI around 4°CA after TDC. This is also confirmed by *Figure 30* showing the related injection progress for the standard and the latest SOI. The supposed premixed share shown by the injection progress at start of combustion is similar. The higher heat release rate for the case with exhaust flap is mainly due to the clearly higher injected mass but the relative premixed share can be regarded as comparable between the cases.

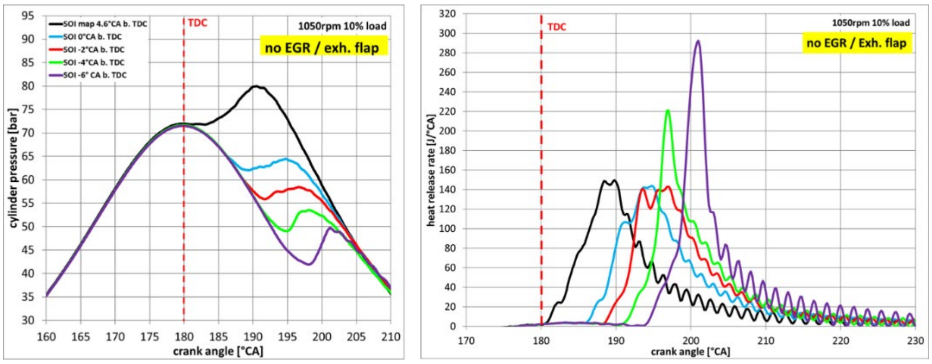

Figure 29: Cylinder pressure and HRR at different start of injection timings with active exhaust flap for low load point

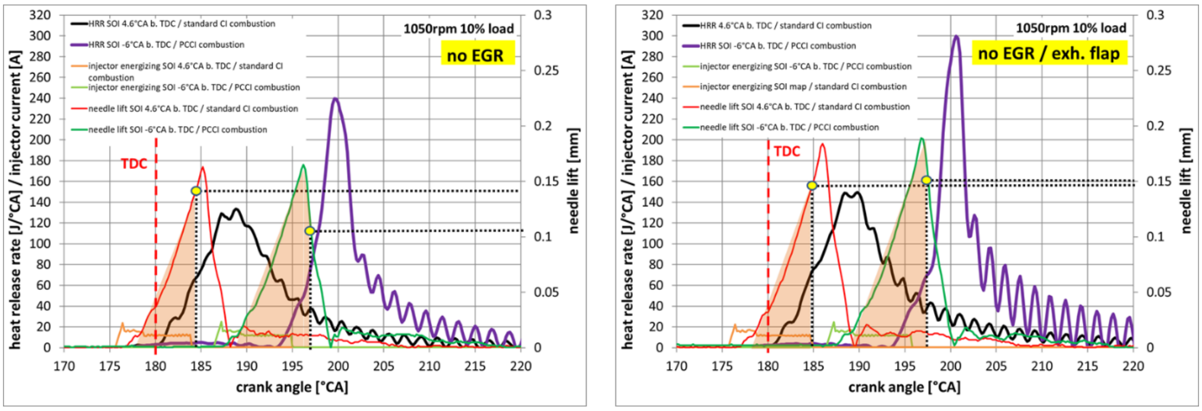

Figure 30: Injection progress at MFB 5% and heat release rate with and without active exhaust flap at low load point

3.2 Ideal engine out temperature and NO_x emission management measures using standard CI and PCCI combustion with DME in low load operation

After investigating the influence of different measures on PCCI combustion, a strategy can be developed how to address the NO_x emissions in low load modes in an optimal manner. This includes raising the temperature of the exhaust gas to reach the urea dosing limit of the SCR system and ideally reduce engine out NO_x emissions as well and at the same time limit the fuel consumption penalty. Here late PCCI combustion shows to be a very useful measure in reducing engine out NO_x emission with limited fuel penalty. In this section different measures are combined to investigate where late injection phasing can help reach the set targets when combined with other measures described in the previous chapters.

Figure 31 shows a summary of all the investigated combinations and their performance in terms of fuel consumption, tailpipe NO_x emissions as well as temperature before SCR system for the low load point 1050rpm/230Nm. The engine out NO_x target was set as 5 g/kWh which is about half of the calibrated value of 10 g/kWh for in service Euro 6 diesel engines with SCR only technology and around 60% of the base calibration value with DME of about 8 g/kWh. The comparison is shown for the application with standard camshaft as well as with the Miller timing camshaft. The graph shows measures combined with EGR and without EGR. Represented are the cases with the best fuel consumption with respect to the engine out NO_x target of 5 g/kWh.

The results show that without EGR the best cases use late injection to reach the target engine out NO_x. From those cases the one with reduced boost shows the lowest fuel consumption but without being able to reach substantial exhaust temperature increase and therefore NO_x tailpipe reduction compared to the reference case without EGR. The only measure which can reduce tailpipe NO_x to below regulatory levels without EGR uses the exhaust flap together with late injection. There is however a higher fuel consumption penalty linked to that. The Miller camshaft timing can have a small benefit in terms of fuel consumption and temperature in general. However, since the engine out NO_x levels are reduced with Miller cam it can lead to less SOI retardation need to reach the target NO_x emissions which in turn can have an added benefit on fuel consumption. The apparent tailpipe NO_x reduction seen with the Miller camshaft is due to different Ammonia loading states of the after-treatment system at the time of measurement. Therefore, the tailpipe NO_x emissions are only showed here to indicate the cases where dosing occurs and the tailpipe emission fall to or below regulatory levels.

On the right side of *Figure 31* EGR was added as a measure to reduce engine out NO_x emissions and increase exhaust temperature. For the selected engine out NO_x level there is no clear advantage using EGR for the reference and the reduced boost case. However, the combination of exhaust flap with EGR leads to a reduction in fuel consumption to the level of the reference cases with/without EGR while meeting the SCR temperature and NO_x tailpipe targets. This is due to the lower backpressure after engine caused by the exhaust flap when part of the exhaust gas is recirculated and also by the earlier (and more efficient) injection phasing possible with EGR to reach the engine out NO_x target. The beneficial combination of EGR and exhaust flap was also confirmed for lower engine out NO_x target setpoints. This represents thus an ideal measure to reduce NO_x emissions until the exhaust aftertreatment system is warmed up and also keep the SCR operational in low load conditions. Combined with late injection phasing PCCI very low NO_x engine out and tailpipe emissions are therefore possible while keeping fuel penalty limited.

Figure 31: Summary of temperature and NOx management measures for engine out NO_x target of 5 g/kWh at low load operation point 1050rpm/230Nm

Figure 32 shows combustion modes for the discussed case of using the exhaust flap without EGR from *Figure 31* (green arrow). Without late injection and PCCI combustion the engine out NOx setpoint of 5 g/kWh is not reachable. Here PCCI allows to reach around 40% NO_x reduction engine out while keeping engine efficiency losses very limited.

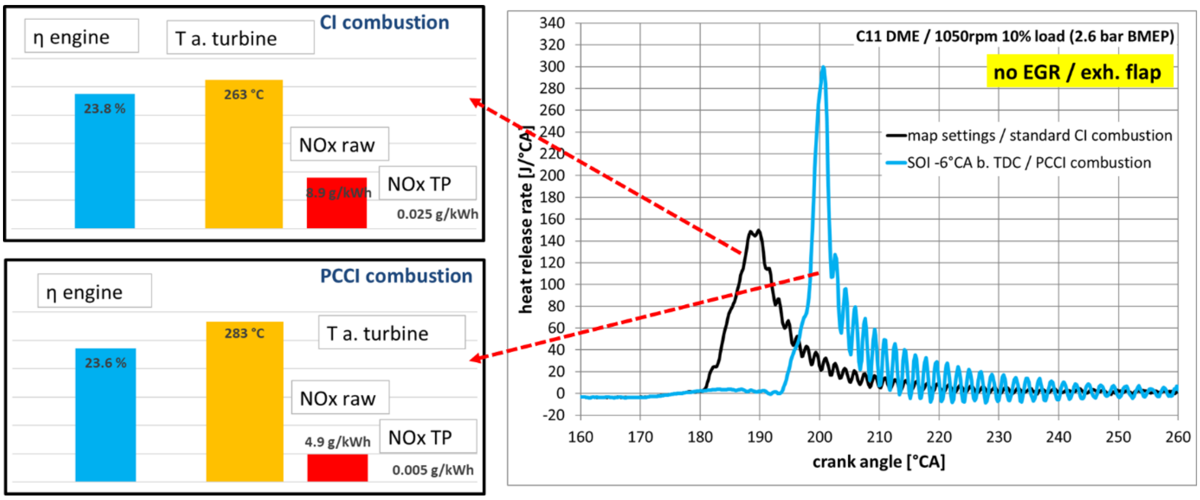

Figure 32: PCCI combustion mode for reduced engine out NOx target emissions at low load operation point 1050rpm/230Nm with active exhaust flap

Figure 33 shows the case with a combination of EGR and exhaust flap from *Figure 31* (blue arrow). It can be seen that with EGR the NO_x engine out target of 5 g/kWh is reached with a clearly better efficiency already with the standard CI combustion. Therefore, the use of late PCCI is not mandatory. However, with addition of late PCCI combustion mode engine out emissions can be reduced further by 40%. Compared with the PCCI mode in *Figure 32* without the use of EGR this reduction can be done even with better efficiency for an engine out NO_x value of around 3 g/kWh instead of 5 g/kWh.

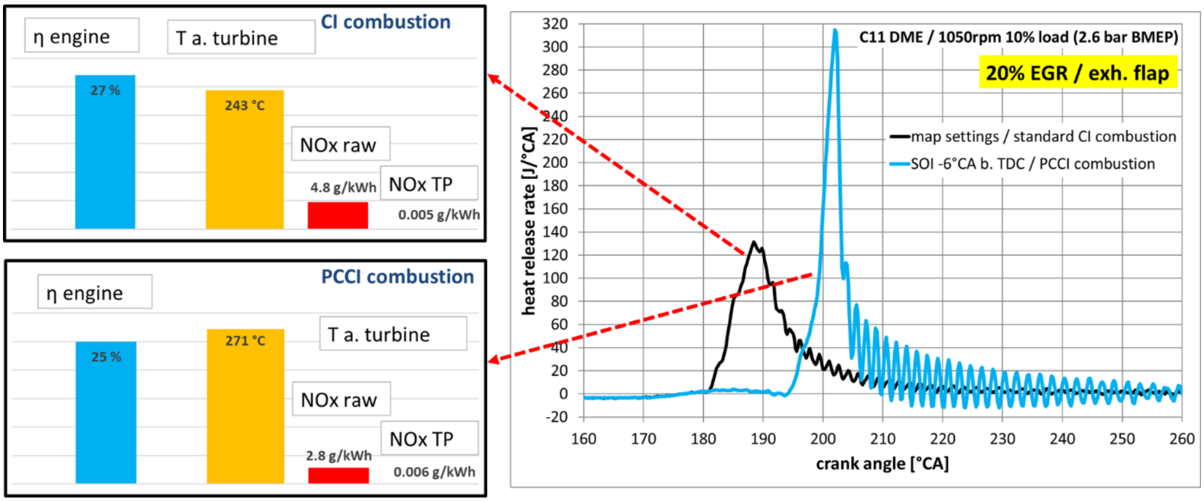

Figure 33: PCCI combustion mode for reduced engine out NOx target emissions at low load operation point 1050rpm/230Nm with EGR and active exhaust flap

Finally the data gathered from all the experiments with the different measures on the engine were also integrated into a software tool for data-based system modelling and optimization called ETAS ASCMO which was available at FPT. ASCMO stands for Advanced Simulation for Calibration, Modelling, and Optimization. It enables users to accurately model, analyse, and optimize the behaviour of complex systems with few measurements and advanced algorithms. In addition, the tool allows users to efficiently optimize parameters in physical models. ETAS ASCMO is used for model-based calibration in numerous applications, such as the optimization of fuel consumption and engine emissions. In our case the goal was to investigate fuel consumption optimal combinations of EGR, exhaust flap and SOI (injection phasing) for a given NO_x engine out target to derive a calibration strategy for the engine control unit.

Figure 34 shows an example of such a result from the data-based model created with this software tool. The x-axis shows the engine out target NO_x emissions. On the different y-axis there are the optimization parameters which are SOI (start of injection), EGR, and exhaust flap. The boost pressure is a model output and results from the different settings of the other parameters. The specific fuel consumption was chosen as the optimisation parameter to be minimized for a given NO_x engine out target together with not to exceed CO and HC engine out emissions. The graph shows two main results for minimal fuel consumption: for NO_x engine out target values between 8-5 g/kWh the optimal strategy is late injection phasing without the need of EGR or exhaust flap application. For NOx engine out targets lower than 5g/kWh a combination of exhaust flap and EGR is preferred without the need for late injection phasing. The results show also a case with deactivated exhaust flap. For this case the strategy is similar. However, the fuel consumption is clearly improved if the exhaust flap is allowed to operate together with EGR. It can be seen as well the Miller timing gives a slight advantage in fuel consumption with deactivated exhaust flap especially for very low NO_x targets due to the earlier applicable SOI and improved efficiency of engine cycle. Also, the Miller camshaft shows a fuel consumption improvement for NO_x target emissions >5 g/kWh where EGR and exhaust flap are not used.

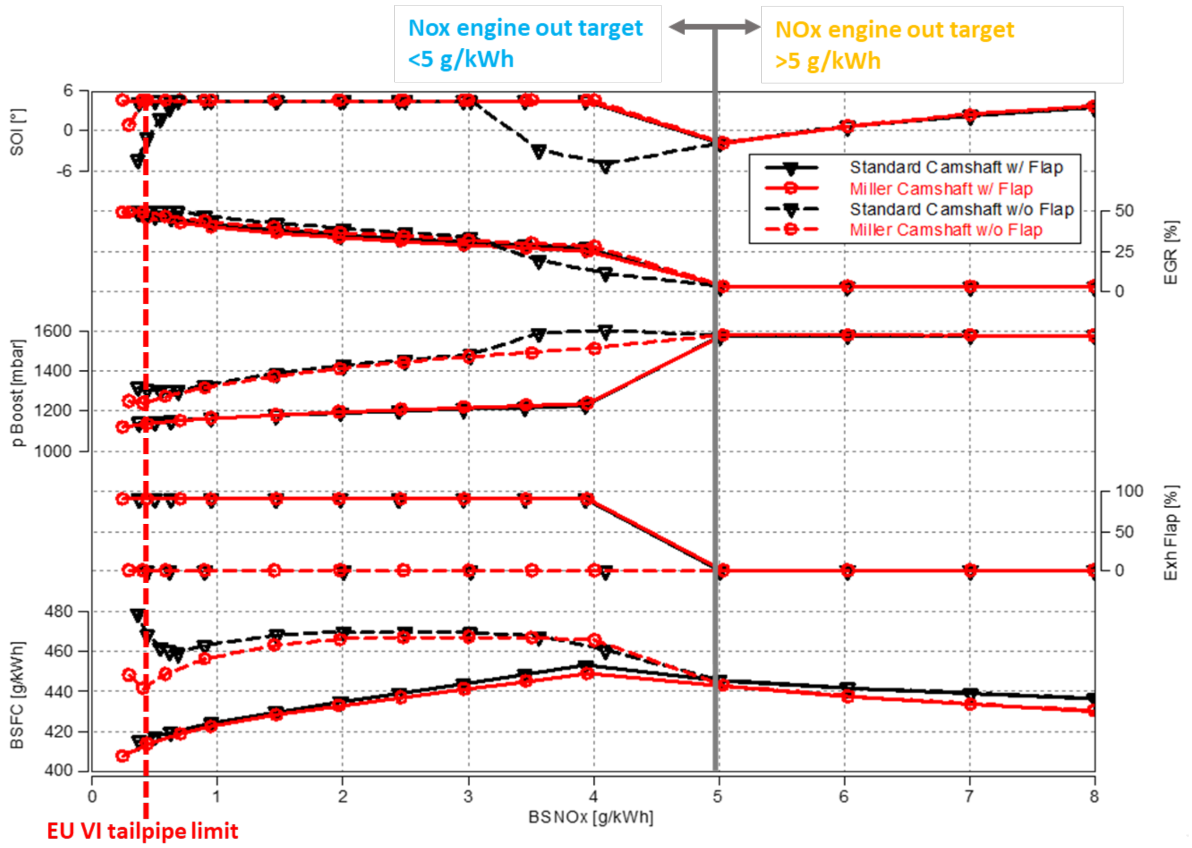

Figure 34: Example of the output from a data-based modeling and optimization tool for investigating fuel consumption optimal engine out NOx targets

3.3 Example of implementation of late PCCI combustion into low NOx transient engine operation control strategy

3.3.1 World Harmonized Transient Cycle (WHTC)

To understand the full benefit of late PCCI in combination with high EGR rates provided by the EGR pump, different ECU calibrations were tested and compared in a WHTC cycle. Based on the stationary measurements, three main modes are showed: A calibration without EGR, which is the baseline, since the original Cursor 11 with Diesel is configured without EGR. Seconds, a mild EGR calibration, which applies roughly 10% EGR over the whole engine map with a slight increase in the low load area and third, a strong EGR mode, which combines EGR rates up to 50% with late start of injection, representing actual PCCI combustion, for additional heat gain and NO_x reduction. It has to be pointed out that the software of the engine control system was not designed to take full advantage of the possibilities found in steady-state optimization. A proper transient calibration and combustion-strategy specific control functions, particularly also for the cold start phase, would have to be implemented to demonstrate the whole potential. However, such a task is very complex, time-consuming and expensive and it was not part of this project.

Luckily, the known and simple characteristics of the EGR pump enabled a straightforward control of the EGR rate. The desired EGR mass flow was defined in percentage based on the engine load and speed, with a transient reduction (adapted from transient soot limit functions of the Diesel-based-software) calculated on the derivative of the injection signal, to avoid CO peaks (load increase) or compressor surge (load decrease). In case of a high ratio over the pump, some throttling by the EGR flap was needed, to respect the allowed working conditions of the EGR pump.

In the measurement traces shown in *Figure 35*, the two main positive effects of EGR use are clearly visible: the engine out NO_x are reduced from around 7.5 g/kWh to 4.3 g/kWh in the mild NO_x calibration and further lowered to 2.1g/kWh with the strong EGR approach. In the same way as the NO_x are decreasing, the temperature after engine is raised, thanks to lower fresh air mass flow with applied EGR.

Soot mass or particulate number after engine was not critical at all, even compliant with proposed Euro 7 limits thanks to the oxygen-containing DME molecule without C-C bounds. The actual limit on the amount of EGR was coming from carbon monoxide, unburned hydrocarbons or even methane emissions, but with the transient EGR reduction and a simple DOC installed after the engine, the values are manageable.

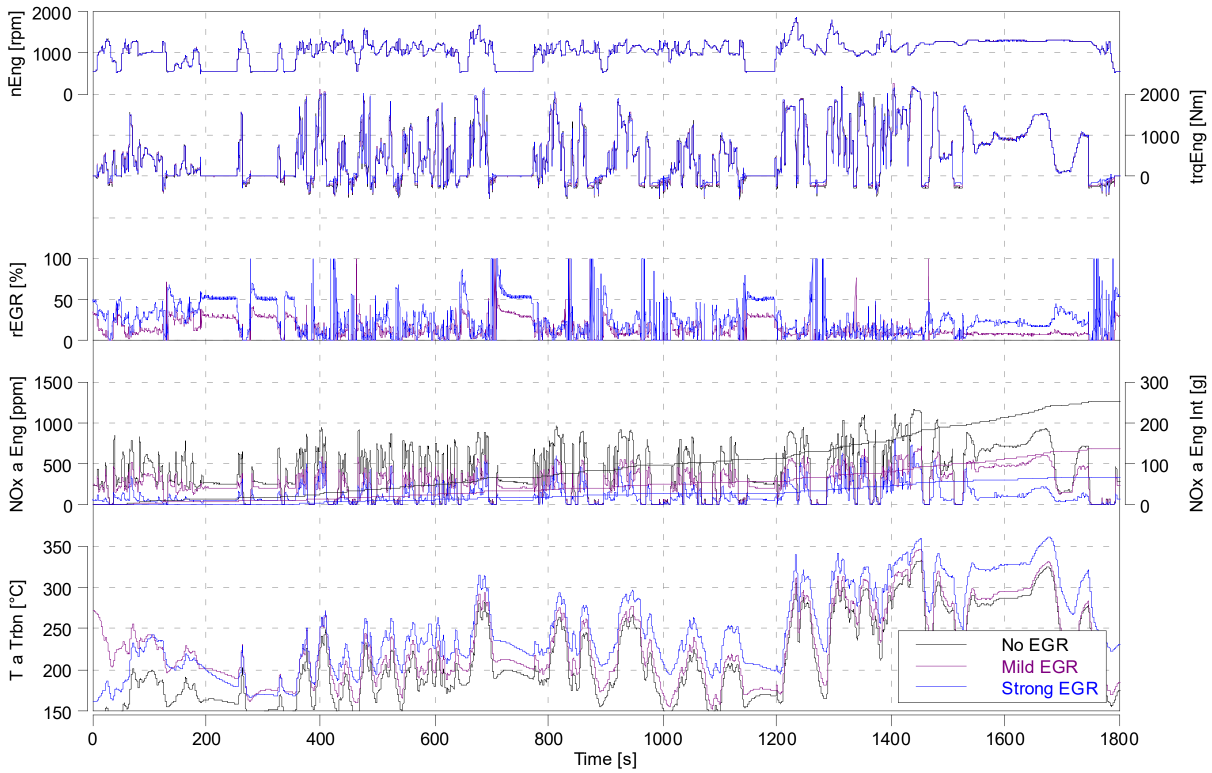

Figure 35: Engine behaviour during "WHTC hot"

The higher engine out temperature as seen in *Figure 35*, is very important for the working of the SCR system as shown in *Figure 36*. The few degrees higher temperature allow keeping the urea dosing active during the whole cycle. Without EGR and thermal management by the exhaust flap, the temperature in front of the SCR is falling below a critical threshold, where no further urea can be injected due to urea deposit formation. In addition, due to the higher NO_x to be reduced by the SCR, the stored ammonia inside the SCR is also faster consumed. In this specific cycle, the strong EGR mode does not bring meaningful further benefits, compared with the mild approach. Highest EGR rates in combination with a late SOI are only meaningful for low load cycles or in cold start.

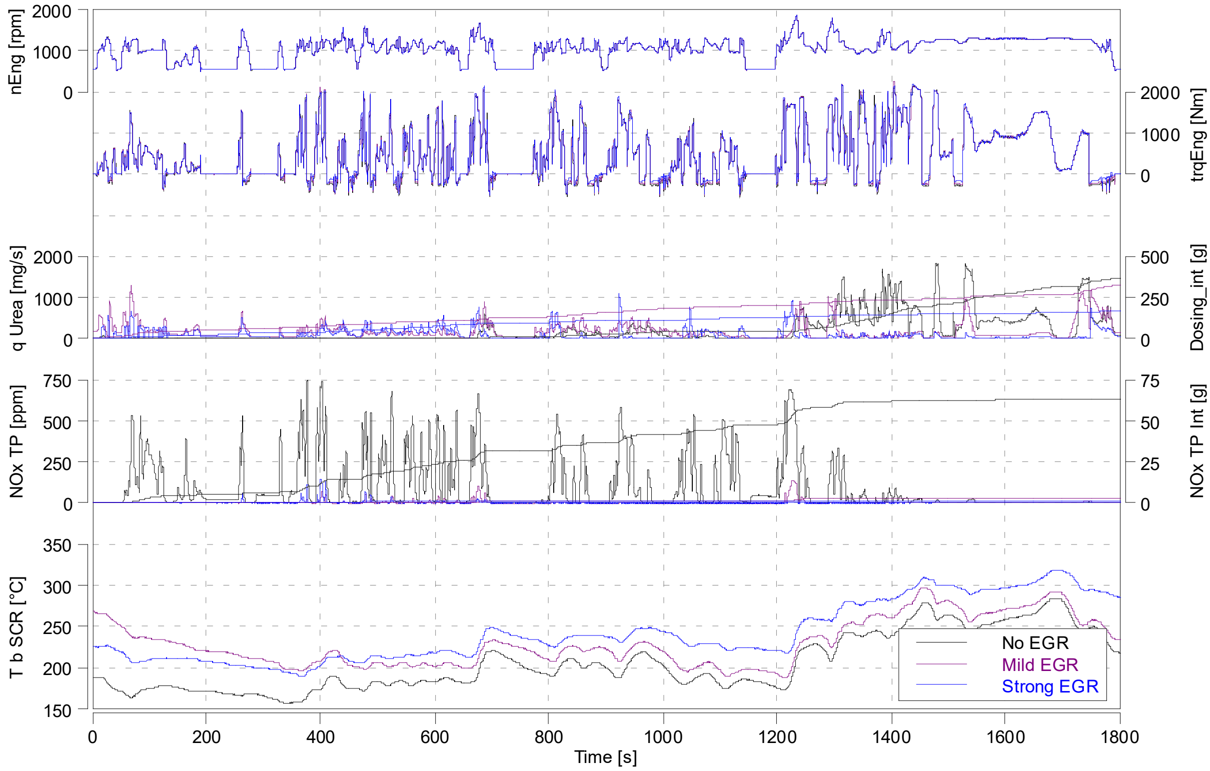

Figure 36: SCR behaviour during "WHTC hot"

Calibration	BSNOx EO	BSNOx TP	BSCO2 TP	BSCO TP
"No EGR" (baseline)	7.89 g/kWh	1.98 g/kWh	596 g/kWh	0.047 g/kWh
"No EGR with ExhFlap"	*7.70 g/kWh*	*0.15 g/kWh*	*613 g/kWh*	*0.009 g/kWh*
"Mild EGR"	4.29 g/kWh	0.09 g/kWh	603 g/kWh	0.657 g/kWh
"Strong EGR" (PCCI)	2.12 g/kWh	0.04 g/kWh	621 g/kWh	0.268 g/kWh

Table 2: A selection of brake specific emissions in "WHTC hot"

Table 2 lists break specific values from the hot WHTC. Beside the three modes showed in *Figure 35* and *Figure 36*, also the results from WHTC hot with conventional ("Diesel-like") thermal management with exhaust flap are showed. CO emissions for the "strong EGR" case are lower than for the "mild EGR" calibration due to an optimisation of the EGR controller in transient phases. The tailpipe NO_x with activated exhaust flap is fully in line with the Euro 6 legislation, but on the cost of an increase of 2.9% in CO_2 emissions / fuel consumption. In contrary, the mild EGR calibration with lower tailpipe NO_x only leads to a CO_2 / fuel consumption penalty of 1.2%. In case of a switch to the "No EGR" calibration after 1200 seconds, once enough heat is provided to the SCR, an Euro 6 compliant WHTC hot would be feasible with even less fuel penalty. But this would lead, due to the higher urea consumption, probably not to a further improvement in TCO. Assuming costs of 1.00 CHF/l for Urea and DME costs of 1.00 to 2.00 CHF/kg, the sweet spot for TCO is in the range of 5 to 6 g/kWh NO_x after engine.

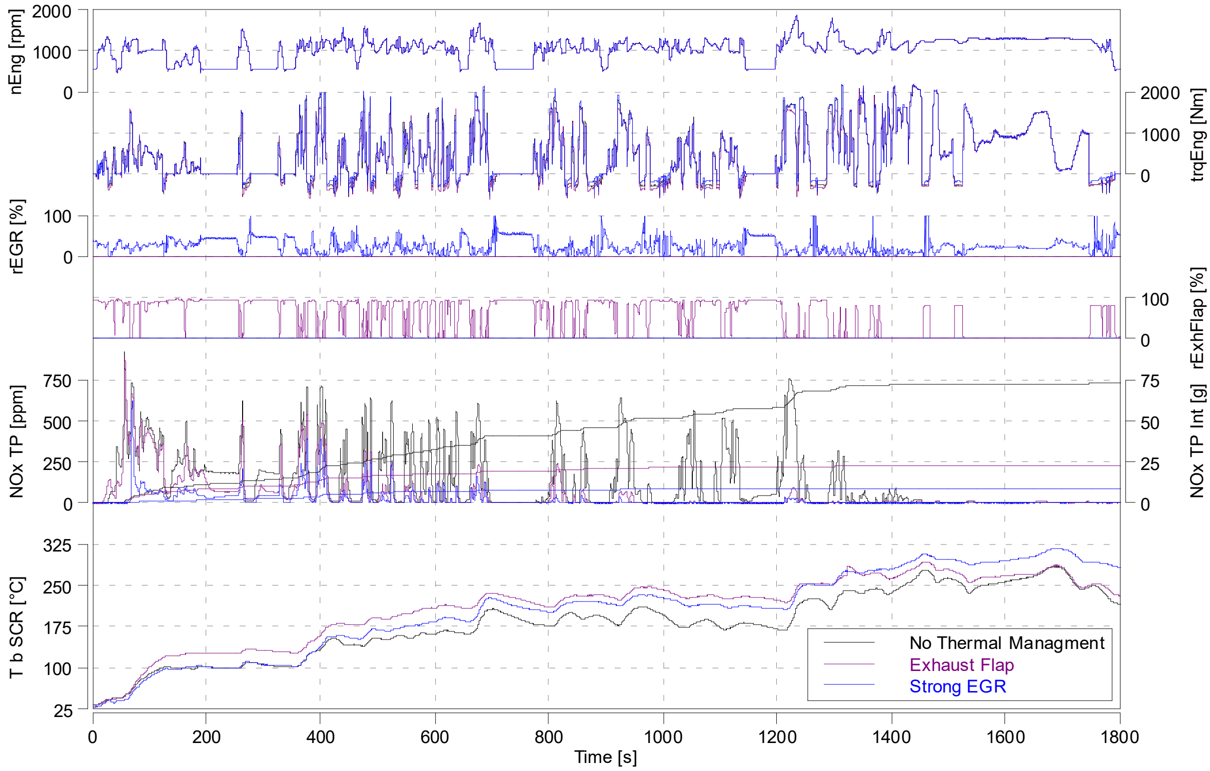

Figure 37: WHTC cold start comparison

In the cold started WHTC (*Figure 37*), the "strong EGR" approach is of higher interest. The lower engine out NO_x in the range of 2 g/kWh helps significantly to reduce the tailpipe NO_x in the first 700 seconds, when the SCR is too cold for operation. Thanks to the high cetane number of DME, the high EGR rates during cold start are not a particular problem. An increase in CO and THC emissions was noted, but especially the more critical amount of THC was reduced to an acceptable level of 0.162 g/kWh after engine by SOI advancing below 60 °C coolant temperature. The SOI shift led to an increase of the tailpipe NO_x from 0.28g/kWh to 0.33g/kWh over the whole cycle, what is absolutely no issue for a Euro 6 application, comparing with the conventional approach with 0.75 g/kWh NO_x.

Calibration	BSNOx EO	BSNOx TP	BSCO2 TP	BSHC EO
"No EGR" (baseline)	7.47 g/kWh	2.29 g/kWh	610 g/kWh	0.094 g/kWh
"No EGR with ExhFlap"	7.22 g/kWh	0.75 g/kWh	619 g/kWh	0.093 g/kWh
"Strong EGR" (PCCI)	2.04 g/kWh	0.28 g/kWh	631 g/kWh	0.426 g/kWh
"Strong EGR, SOI Cor" (PCCI)	2.11 g/kWh	0.33 g/kWh	630 g/kWh	0.162 g/kWh

Table 3: A selection of brake specific emissions in "WHTC cold"

CO_2 emissions and fuel consumption were higher over the complete cold WHTC with the "strong EGR" approach compared with exhaust flap application. But considering an operation mode switch to "mild EGR" after 700seconds, the EGR/PCCI strategies would show a clear advantage.

3.3.2 Nonroad Low Load Application Cycle (LLAC)

A further interesting case is the low load cycle, currently in discussion for the Tier 5 non-road regulation, which will take place approximately in 2029. This cycle was proposed by the Southwest Research Institute and contains multiple missions of typical non-road applications. Tier 4 or Stage V compliant vehicles would produce too high tailpipe NO_x emissions, due to the low average power of 15% (in comparison, the current US Non-Road Transient Cycle cycle has an average load of 35%).

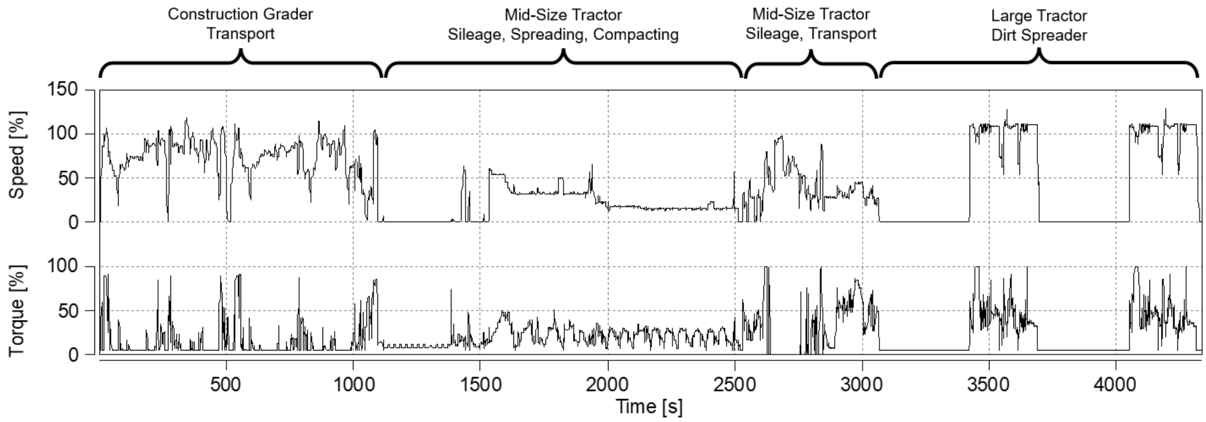

Figure 38 Low Load Application Cycle (LLAC) with normalized engine speed and torque

This cycle was reproduced with the C11 DME engine. Due to the lack of further DME reserves, only two calibrations could be tested in the first half of the cycle. “Strong EGR” is considered as most interesting calibration and “Mild EGR” as a baseline for an EGR application as it could be done on a Diesel engine.

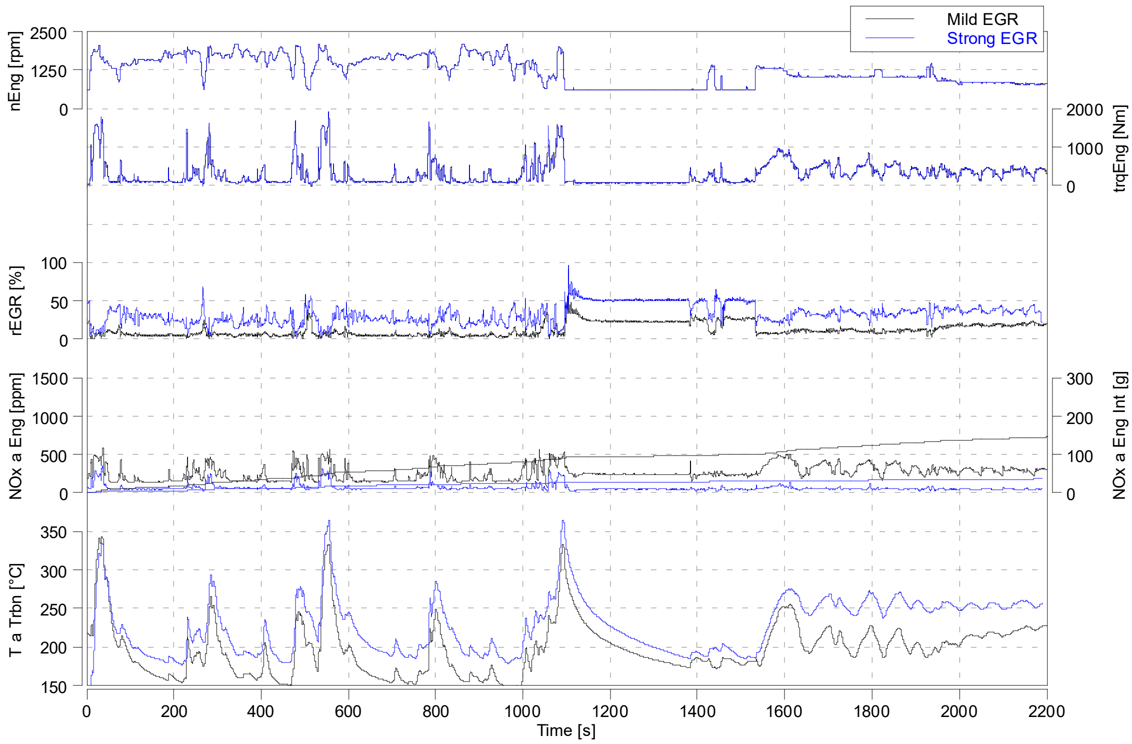

Figure 39: Engine behaviour during "LLAC"

The baseline measurements with a low EGR rate from 5% to 20% shows the main challenge of the LLAC: frequent low temperatures after the turbine in the range of 150 °C. Conventional thermal management with an exhaust flap could help in some parts of the cycles, but as example in the low idle phases, a temperature drop cannot be avoided with an exhaust flap. Either cylinder deactivation or strong valve timing adjustments would be needed.

Since the limit for the LLAC is proposed to be 2.5 times higher than the NRTC/RMC limit, NO_x in tailpipe needs to be below 100 mg/kWh. In the mild EGR approach with 6 g/kWh NO_x after engine, an SCR efficiency of 98.5% is needed, whereat with the strong EGR calibration with 1.5g/kWh NO_x, 93% is good enough.

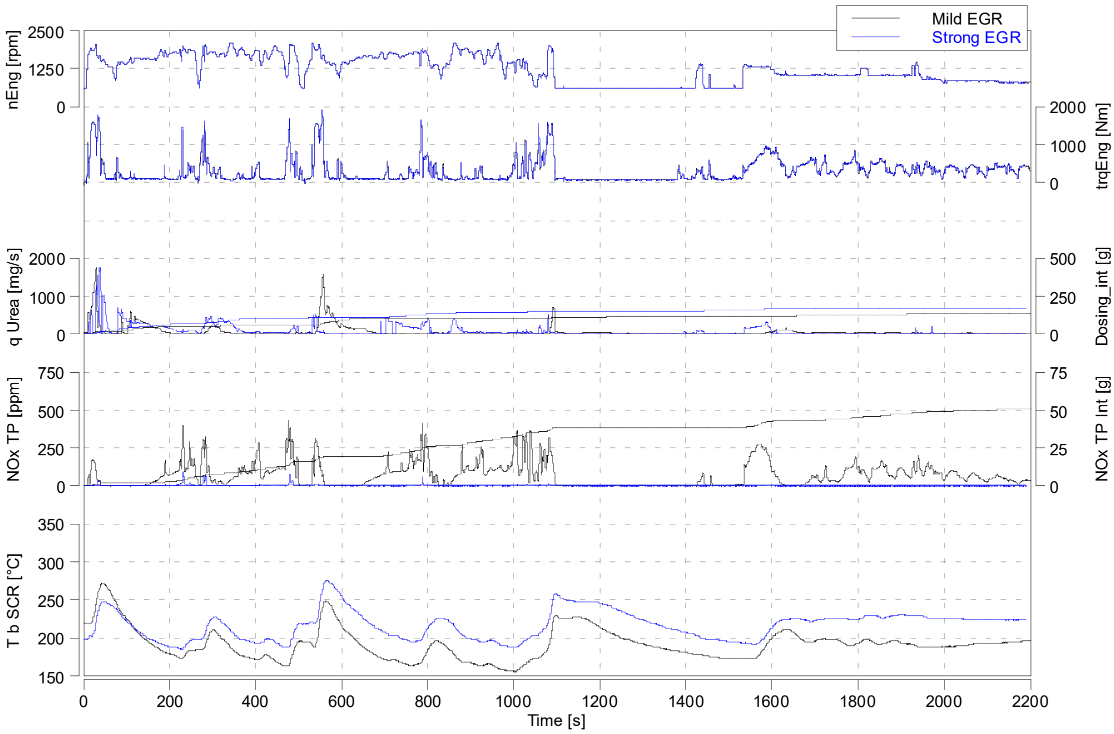

Figure 40: SCR behaviour during "LLAC"

The higher temperature after engine as seen in *Figure 39* with strong EGR are translating in an SCR temperature staying most time above 200 °C, which allows a continuous working of the catalyst. The urea dosing quantity needs to be limited in case of low temperature and due to the higher engine out NO_x with the mild EGR approach, it is a doubled disadvantage. Therefore, a significant reduction from 2.10g/kWh NO_x to a compliant value of 0.06g/kWh NO_x in tailpipe is possible with the strong EGR approach.

One critical point is the proposed Tier 5 limit for Methane of 0.017g/kWh. It was fulfilled with the mild EGR approach but exceeded by factor 5 on the strong EGR calibration. An optimization of the EGR control, or the use of a differently coated oxidation catalyst, could likely improve this situation.

These results are very promising, since they are giving a hint, that it is possible to have Tier 5 compliant engines, with Tier 4 like catalyst sizes (packaging advantage on non-road machinery) and without advanced technologies on the engine valvetrain.

3.4 Positive impact of high EGR rates on the SCR operation

From Diesel combustion, a correlation between NO_2/NO_x ratio after engine and EGR rates are known. With increased amount of EGR, NO is reduced after engine, whereat NO_2 stays relatively constant, leading therefore to a higher NO_2/NO_x ratio. Higher ratios have significant advantages for the aftertreatment system in low temperature area: the SCR can convert more NO_x by the fast SCR reaction instead of the standard one, needing less stored NH_3 for the same efficiency. Therefore, DME with the high EGR tolerance is perfectly suited to explore this opportunity.

On two different load points at 1050 rpm respectively 1200 rpm, the behaviour was tested with DME. For the 2.6 bar BMEP load point, EGR rate and SOI was modified to reach the target engine out NO_x of 4, 2, 1 and 0.5 g/kWh. At 6.5 bar BMEP, the SOI was constant and only EGR was varied to reach the target of 8, 4, 2 and 1 g/kWh NO_x after engine.

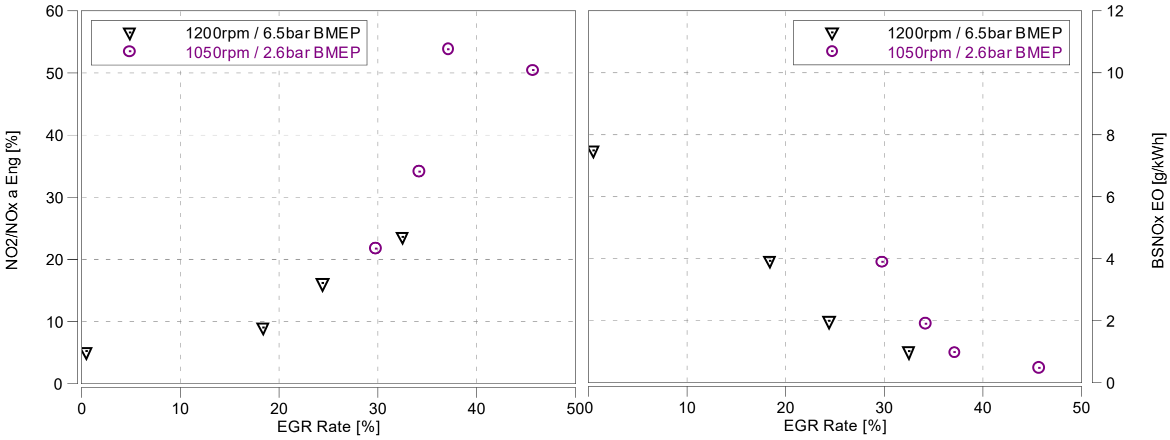

Figure 41: NO_2/NO_x ratio and engine out total NO_x after engine with different EGR rates in two load conditions

As it can be seen above, there is a clear trend between the NO_2/NO_x ratio after engine and the according EGR rate. There is one outliner with 53.8% NO_2, which is explained with measurement inaccuracies. The confirmation of the effect and the considerable amount of NO_2 makes it valuable to perform a brief simulation, how a system with a closed coupled SCR without a DOC in front, can benefit from it.

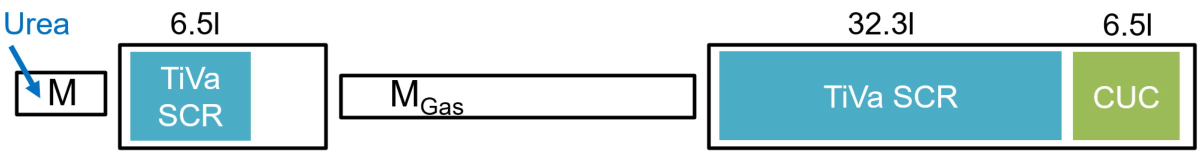

Figure 42: Layout of the simulated after treatment system (TiVA SCR = Titanium/Vanadium based SCR catalyst, CUC = Clean-Up Catalyst)

Figure 42 shows the chosen after treatment layout for simulation. Only one urea injector is needed, thanks to the possibility of a gas mixer for high NH_3 uniformity also in the downstream SCR. Therefore, such a layout can most likely fulfil Euro 7 or Tier 5 requirements with very limited complexity. The functionality of the DOC to reduce CO and HC is overtaken by the clean-up catalyst, even if probably with slightly less efficiency than a dedicated diesel oxidation catalyst (DOC) would have. Titan vanadium SCR do not need a periodic de-sulphation regeneration, allowing therefore the removal of the DOC in front of it and have a low N_2O selectivity below 450 °C (in line with the 338 kW rating of this engine), which is important for future legislations. The simulations are started with a cold aftertreatment system and without any prestored NH_3 in the SCR. The engine is kept constant on the according load point.

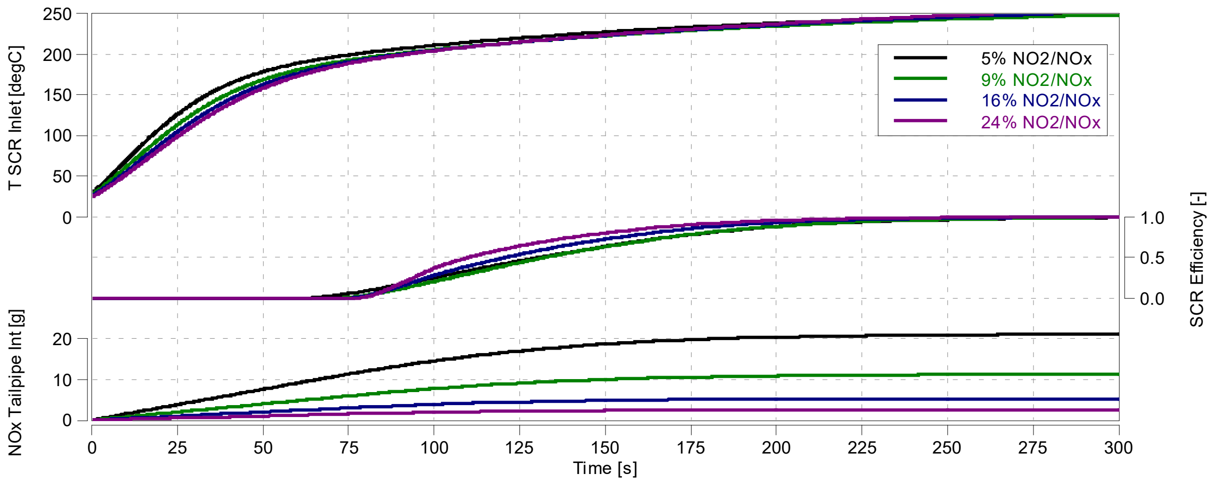

Figure 43: ATS simulation at 1200rpm and 6.5bar BMEP

On the higher load point in *Figure 43*, the effect of increase NO_2/NO_x ratio brings a small benefit. From second 100 to 200, an advantage of up to 20 percentage points in SCR efficiency is visible. But the main reason for the lower tailpipe NO_x is due to the reduced engine out NO_x. Also, it can be seen how the heat up of the system is slightly slower due to the reduced exhaust mass flow with higher EGR rates, delaying the urea injection which starts at 190degC.

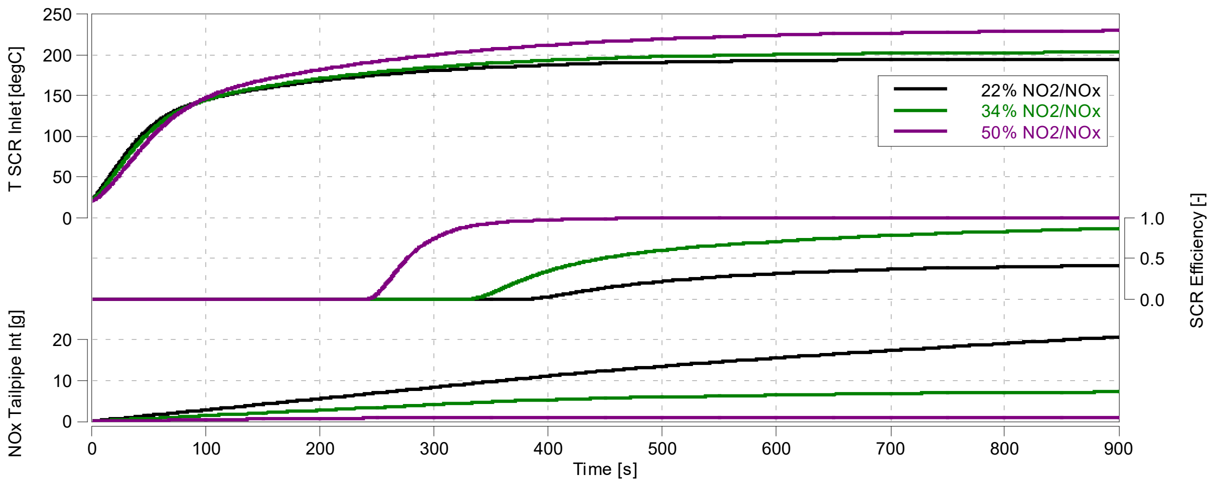

Figure 44: ATS simulation at 1050rpm and 2.6bar BMEP

The advantage of a high NO_2/NO_x by using EGR ratio gets much more visible on the lower load point in *Figure 44*. With 50% NO_2 after engine, we get remarkable SCR efficiency as soon as the dosing starts at 190 °C. The slight disadvantage of slower heat up can be neglected and is compensated very soon thanks to higher temperature level with higher EGR rates and the low thermal mass in front of the SCR. With 34% or even 22% NO_2, it takes much longer to get a certain SCR efficiency. Beside the effect of the different SCR reactions, also the limited dosing capabilities at this lower temperature plays a role. The lower the engine out NO_x, the less urea is needed for stoichiometric dosing.

It can be concluded, the PCCI combustion with high EGR rates leads not only to lower engine out NO_x and therefore lower need of SCR efficiency, but it also increases the SCR efficiency thanks to a beneficial NO_2/NO_x ratio as soon as dosing temperature is reached.

4 Conclusions of the experimental campaign

- PCCI combustion with DME

  The physical and chemical properties of DME favor the application of alternative combustion modes like late PCCI due to better mixture preparation and almost soot free combustion. EGR showed to be a strong enabler for increasing the premixed share of injected fuel compared to the application of Miller valve timing, exhaust flap and boost reduction. Moreover, flexible EGR control with an EGR pump allows for strong engine out NO_x reduction in low load operation and cold start/warmup conditions. Furthermore, high EGR rates can even improve fuel efficiency in certain low load/idle conditions due to lower pumping losses.

  The late partially premixed combustion allows for exhaust gas temperature increase and engine out NO_x reduction due to the late combustion phasing while limiting fuel consumption penalty due to the shorter combustion duration. For the presented setup and the investigated operating points a late SOI of 4°CA after TDC proved to be the sensible limit. Later injection timings lead to increase in CO and THC emissions without noticeable further NO_x reduction.

- Combined measures for exhaust gas aftertreatment thermal management

  To reach the lower dosing temperature limit of the urea/water solution for the SCR system with limited fuel consumption increase it is necessary to combine different measures. The standard approach of throttling the engine with an exhaust flap after the turbocharger turbine provides a strong increase in exhaust temperature but has the highest fuel penalty. However, if combined with EGR this measure shows a noticeable fuel consumption improvement while still reaching the dosing limit. In addition, unlike the sole application of the exhaust flap EGR reduces strongly the engine out NO_x which means less emission burden for the SCR system and thus reduced urea dosing quantity necessary and lower risk of urea deposits. Lower engine out NO_x means also reduced tailpipe emissions during warmup until activation of the SCR system. Late PCCI combustion can be an additional measure to further reduce engine out NO_x emissions for specific low NO_x strategies to comply with future emission legislations, especially targeting cold start and low load operation.

- Transient low NO_x control strategy incorporating PCCI combustion

  The results from the transient cycles are very promising. Late PCCI strategies with high EGR rates combining the two advantages of increase exhaust gas temperature and decreased engine out NO_x. This allows a thermal management functionality with only minor fuel consumption increase compared with an exhaust flap. In addition, there are first indications, that the future legislation limits of Euro 7 and Tier 5 can be reached with a Euro 6 like aftertreatment system, probably even without DPF.

  Further improvements regarding the tradeoff between NO_x and THC/CO in transient manoeuvres are possible with a more sophisticated EGR control, also compressor surge in load decrease could be minimized in the same step.

5 Numerical simulation tools

In order to model and to better understand the experimental result, computational fluid dynamics (CFD) was used. An open-source CFD solver (OpenFOAM, version 8) was chosen as a numerical platform. For mesh motion, the LibICE v8.x plug- was employed, which was developed over several decades by Prof. Tommaso Lucchini and his group at Politecnico di Milano and shared during the course of the project.

Prior to use, both the CFD solver as well as the LibICE plug-in required compilation on the EULER High Performance Cluster of ETH, running CentOS LINUX. This required considerable effort due to large number of combinations of openmpi libraries, gcc compiler versions and all required additional modules. Using the following combination, compilation could be successfully carried out:

- gcc 8.2.0
- openmpi 4.1.4
- cmake 3.25.0
- flex 2.6.4
- zlib 1.2.11
- openblas 0.3.15
- libxkbcommon 0.8.0
- python 3.10.4
- mesa 18.3.6
- paraview 5.9.1
- hdf 1.10.9

```
Currently Loaded Modules:
  1) StdEnv      3) openmpi/4.1.4   5) flex/2.6.4    7) hdf5/1.10.9      9) python/3.10.4       11) mesa/18.3.6
  2) gcc/8.2.0   4) cmake/3.25.0    6) zlib/1.2.11   8) openblas/0.3.15  10) libxkbcommon/0.8.0  12) paraview/5.9.1
```

For visualizing the three-dimensional flow fields, ParaView was adopted (version 5.9.1). ParaView has a built-in reader for “native” OpenFOAM data, avoiding lengthy conversions to other visualization formats as well as data redundancy. Another particularly useful feature of ParaView is that it can be executed in server mode (in parallel) on the cluster in combination with a client running locally on a laptop/PC as a front-end (also in mixed mode, i.e. LINUX server, windows client). This avoids lengthy data transfers to/from the cluster and potential problems due to memory requirements when post-processing on the client.

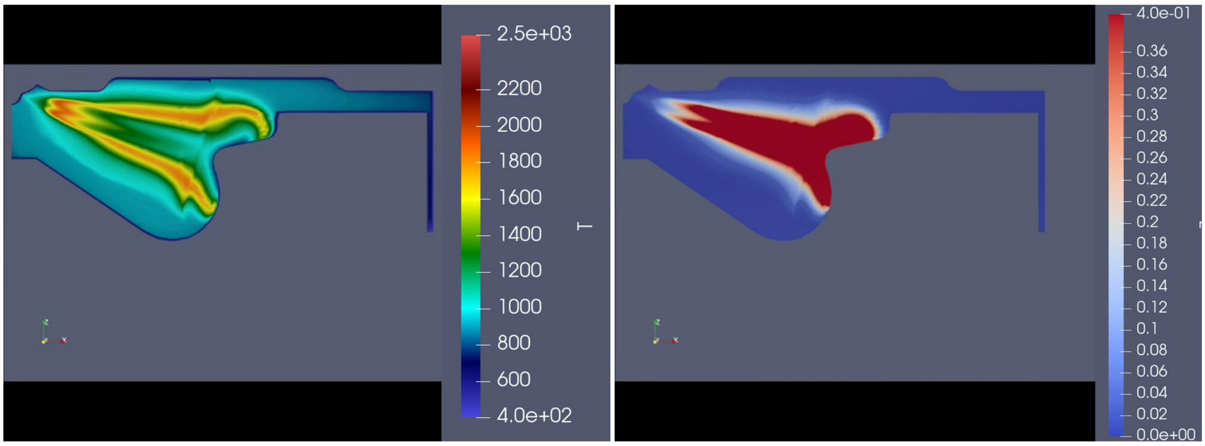

Figure 45: example of temperature (left) and mixture fraction (right) distributions shortly after the spray plume hits the stepped lip; visualized using ParaView 5.9.1 in client-server mode.

For post-processing of the pressure trace to obtain the apparent Heat Release Rates (aHRR); a python script was also provided. This reads in the measured as well as the simulated pressure evolutions and derives in a consistent way the heat release using the same wall heat losses (Woschni). It also simplifies processing of an entire set of operating conditions and provides a variety of plotting options.

To facilitate the learning process, very helpful tutorials were shared and the great support provided by Prof. Lucchini and Andrea Schirru is thankfully acknowledged. Andrea Schirru further visited Empa during March 2023, greatly speeding up the learning process.

Since PoliMI had previously simulated various C11 piston configurations [17] and had examined some higher load operating conditions [18], sector meshes for the DME engine were available and templates for two different combustion models provided.

All simulations were carried out with a sector mesh, i.e. only one eights of the closed volume is calculated exploiting the symmetric layout of the piston design and centrally located fuel injector. During compression and expansion, layers are removed and added to maintain good aspect ratios. The number of cells spans from roughly 463k cells at the start of the simulation (at Intake Valve Closure, i.e. -143° CA), to roughly 110k cells at TDC. This corresponds to an average cell size of approximately 1 mm during fuel injection and combustion. As can be seen in Figure 46, the mesh is spray-aligned for improved accuracy and boasts a considerably higher resolution in the spray region. These mesh resolutions and geometric configurations are based on previous sensitivity studies carried out by PoliMI and have been employed unchanged.

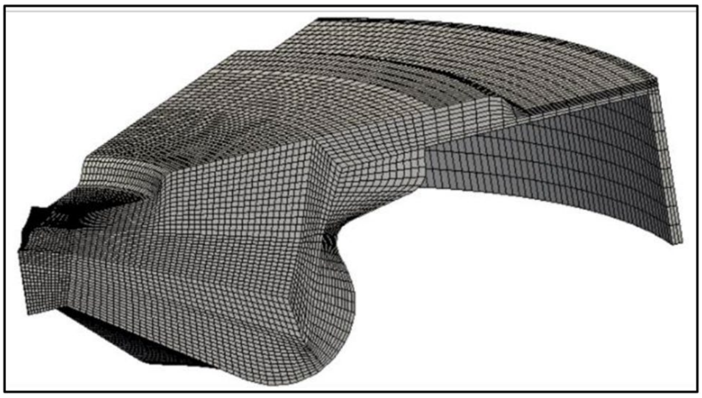

Figure 46: sector mesh close to top dead centre.

The “standard” k-epsilon two-equation turbulence model is used with a modified C1 constant (so-called round jet correction), in conjunction with a near-wall treatment.

Concerning fuel injection, a Lagrangian-Eulerian approach was adopted. It employs a “blob” injection model with initial droplet sizes equal to the nozzle diameter, while droplet secondary break-up is treated with the Kelvin-Helmholz-Rayleigh-Taylor (KHRT) as in previous investigations [17] [19].

Two combustion models have been tested, namely Representative Interactive Flamelets (RIF), which perform online integration of a counter-flow diffusion flamelet with imposed scalar dissipation rate and detailled chemical kinetics. This allows for capturing turbulence-chemistry interaction by employing a presumed PDF approach. Secondly, the so-called Tabulated Well Mixed TWM approach has been tested. As the name suggests, in this approach the species reaction rates are precomputed and stored in tables and retrieved at runtime for a subset of species. The model further assumes a well-mixed state and thereby neglects any potential effects of TCI. Since no on-line chemistry solution and PDF integration is required as for RIF, the TWM is considerably more cost-efficient. The main disadvantage of TWM is that for every initial composition, a separate table has to be built.

The updated, CRECK DME chemical kinetics from 2022 [20] [21] has been used for both RIF and to construct the TWM tables.

Concerning initial and boundary conditions, support from Alessandra Vaghini is thankfully acknowledged, who carried out the GT-Power simulations providing surface temperatures, species mass fractions as well as estimates for the hydraulic delay. In addition, the injection mass flow rates were provided, which are derived from the needle lift using an in-house methodology.

6 CFD simulation results and discussion

Of the vast experimental database, only a small selection of operating conditions could be investigated numerically. In close collaboration with the experiments, 5 operating conditions were chosen. These include the reference condition at 10% load (230 Nm torque) at an engine speed of 1050 rpm without EGR. The same condition is also assessed with 50% nominal EGR. In addition, three different EGR levels, 0%, 50% and 70%, have been studied at idle conditions 550 rpm and 70Nm. For all of them sweeps in SOI were carried out.

6.1 1050 rpm, 230 Nm, 0% EGR – reference condition

The ‘nominal’ injection timing has been used to study sensitivities to initial species composition, initial pressure and temperature levels, prescribed wall boundary temperatures as well hydraulic SOI offset. The RIF model was used as it can cater for changes in species composition, whereas the TWM model necessitates a new table whenever the composition changes.

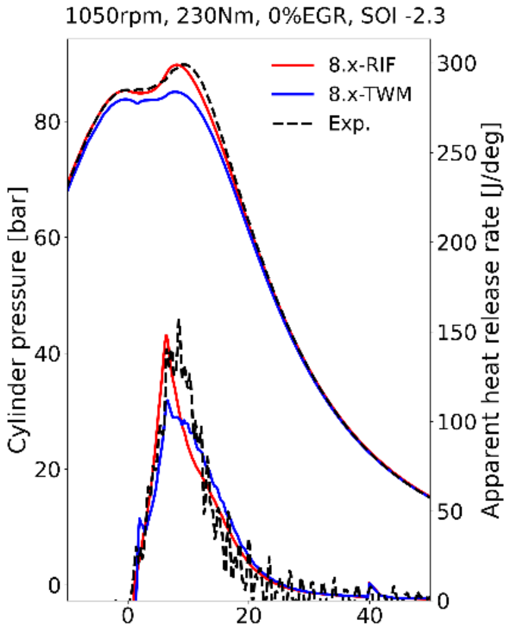

Figure 47: pressure and apparent heat release rate evolutions for the reference condition.

A small offset of one degree crank angle proved necessary to match the experimentally observed ignition delay, all other values, i.e. temperature, pressure levels and species composition (at IVC) could be carried over directly from the GT-Power calculations. As shown in Figure 47, good agreement concerning the pressure trace and heat release rates is evident for the RIF model, while the TWM model shows less good agreement.

The same settings have also been applied for the later injection timings, i.e. no case-by-case adjustments have been made.

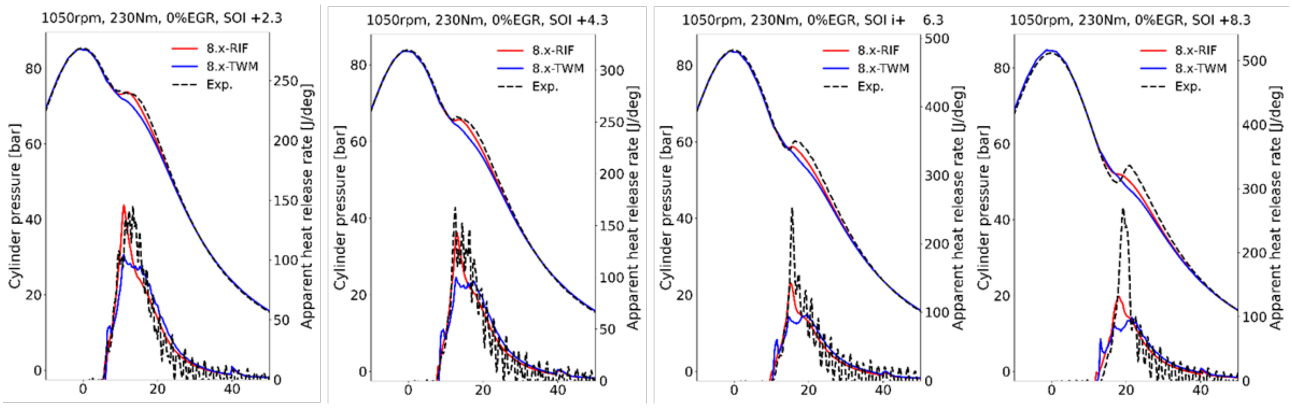

Figure 48: pressure and apparent heat release rate evolutions for a sweep in SOI at 1050 rpm, 230 Nm and 0% EGR.

As is evident from Figure 48, ignition delays are well matched for both RIF as well as the TWM throughout the full range of SOI. The lower reactivity for the latest two injection timings however leads to deteriorating predictions, and more so for TWM. It is important to note that these shortcomings could stem also from flow field effects, viz. turbulence levels and the two-phase flow and may not solely be due to the combustion model. Since there is no (optical) validation data available in this full metal engine, a step-by-step validation as performed e.g. it is unfortunately precluded. Furthermore, with increasing ignition delays, the potential errors of all models involved are likely to add up, resulting in the observed, worsening predictive capabilities at the latest two injection timings.

6.2 1050 rpm, 230 Nm, 50% EGR

At 50% EGR, ignition delays are slightly under predicted compared to 0% EGR with RIF and increasingly so for later injection timings. At the decreased reactivity, only the earliest injection timing is reproduced in terms of pressure evolution and only for RIF. From the second injection timing onwards, the increasingly sharp pressure rise is not reproduced for any model. On the contrary, the later timings have increasingly flatter pressure evolutions. TWM shows better ignition delay times for the latest two injections, however over predicts the remaining ones.

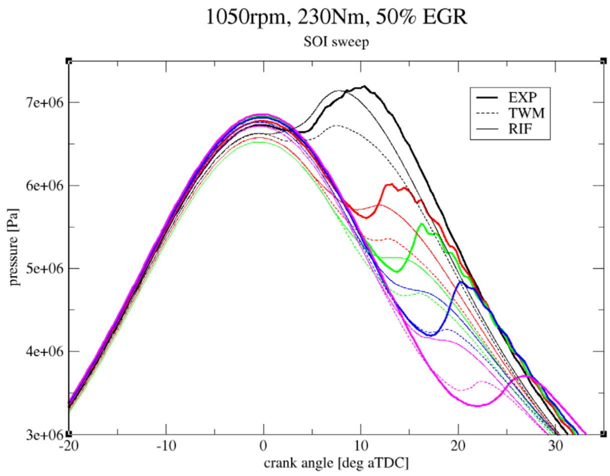

Figure 49: pressure and apparent heat release rate evolutions for a sweep in SOI at 1050 rpm, 230 Nm and 50% EGR (the different colours represent the five different SOI settings).

6.3 Idle conditions

Idle conditions constitute an important share of engine operation and were therefore considered important to study. In addition to late SOI, a means to increase the engine out temperature (hence allowing for ammonia dosing), using (very) high levels of EGR to lower engine-out NOx in the first place are pursued. The latter strategies could be important for lowering NOx during cold start, i.e. prior to reaching the operating temperature of the after treatment system.

At nominally zero percent EGR, similar findings as for the reference condition can be observed. Namely good to fair agreement of ignition delays, with the latest injection timings being slightly under predicted. At these considerably lower pressure levels, reactivity is reduced and somewhat worse performance can be expected.

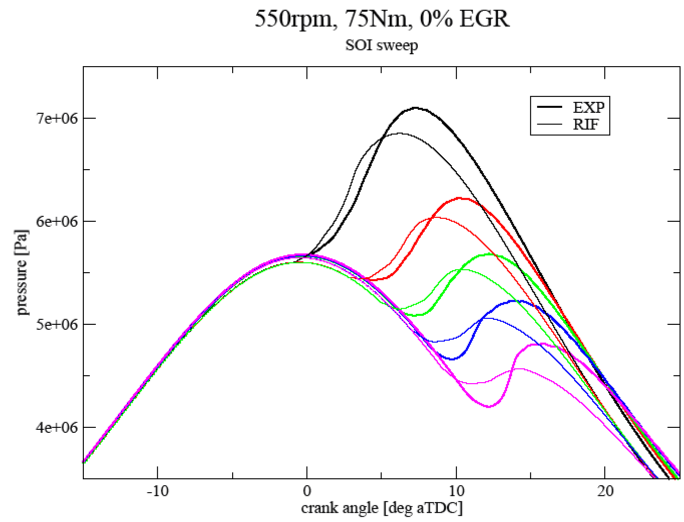

Figure 50: pressure traces for a sweep in SOI at 550 rpm, 75 Nm with 0% EGR.

When moving to 50% EGR, the under prediction of the ignition delays increases. The flattening of the numerical pressure evolutions (as opposed to steepening in the experiment) for later SOI is less pronounced than for the 50% EGR case at 1050 rpm.

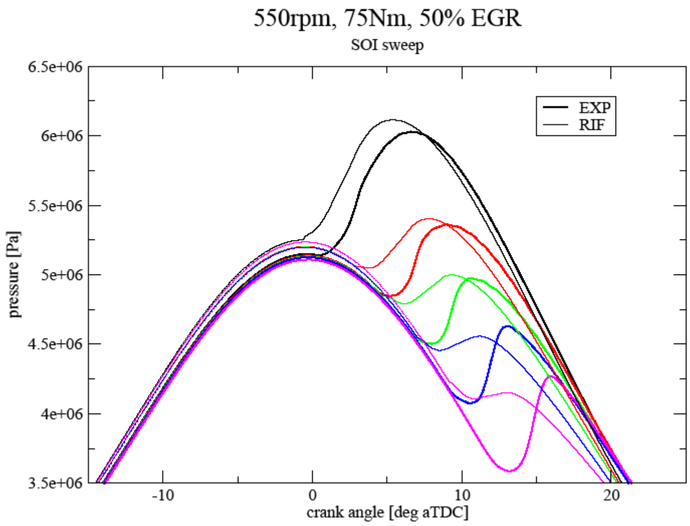

Figure 51: pressure traces for a sweep in SOI at 550 rpm, 75 Nm with 50% EGR.

It is important to note however, that the one degree crank angle offset for the hydraulic delay used above for the 1050 rpm cases has not been applied and better predictions can be expected for a new batch of calculation with this included.

At 70% EGR, good agreement in terms of ignition delays is seen for all SOI with RIF. The ensuing pressure increase however only shows fair agreement for the earliest injection timing and only when using RIF. With TWM, ignition delays are mixed: while the earliest SOI is over predicted, while the remaining ones are considerably under predicted. As a consequence, none of the pressure evolutions are well matched.

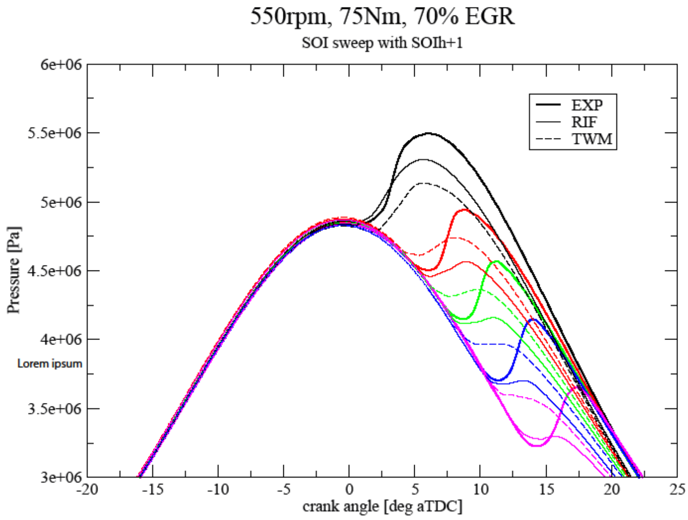

Figure 52: pressure traces for a sweep in SOI at 550 rpm, 75 Nm with 70% EGR.

It is important to note, that chemical kinetics are usually not developed specifically for such high EGR levels and hence some of the discrepancies may also be attributed to this.

6.4 Emission trends

6.4.1 Reference condition (1050 rpm, 230 Nm, 0% EGR)

For the reference condition, a selection of emission relevant species has been output at EVO, namely NO, NO_2, N_2O, DME, CH_4, CO and HCNO. In addition, the volume averaged temperature of the charge at EVO was studied as a proxy for the temperature increase at SCR entry. The experimentally observed temperature increase at the SCR entry from the earliest to the latest timing spanned roughly 20°C (cf. Figure 10, increasing from 140°C to 160°C). Numerically, a temperature increase of 65°C can be seen at EVO. Unlike the experiment at SCR inlet, this does not include heat losses to the exhaust valve and runner, temperature drop over the turbine and heat transfer to the piping all the way to the SCR entry. The general trend of reaching higher temperatures with late timings is however well reproduced.

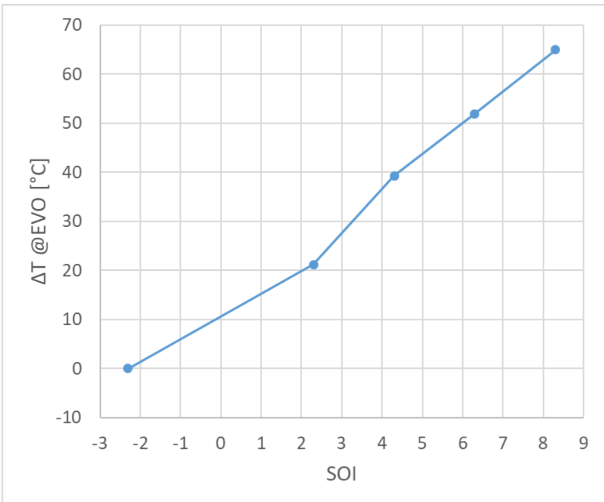

Figure 53: Increase of the volume averaged temperature at EVO for a sweep in SOI.

NO, NO_2 and N_2O emissions have been normalized with their respective values of the earliest injection timing. As was seen in the experiment (cf. Figure 10), the NO level decreases as the injection is postponed. Experimentally, the values decrease from 9 g/kWh to around 4 g/kWh or around 56%, which is in good agreement with the predicted decrease of roughly 60%. The NO_2 and N_2O emissions remain relatively constant over the sweep. This suggests, that later timings improve the NO_2/NO ratio which favours the fast SCR reaction. Constant N_2O levels are encouraging, since N_2O is an extremely potent greenhouse gas.

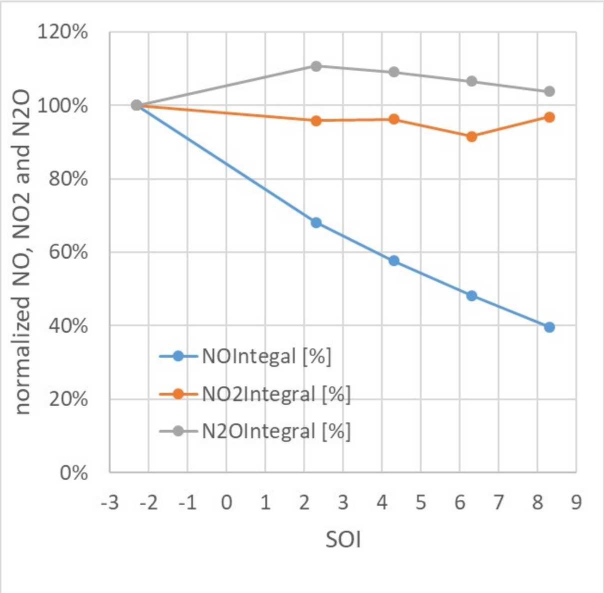

Figure 54: Normalized NO, NO_2 and N_2O levels at EVO for the sweep in SOI.

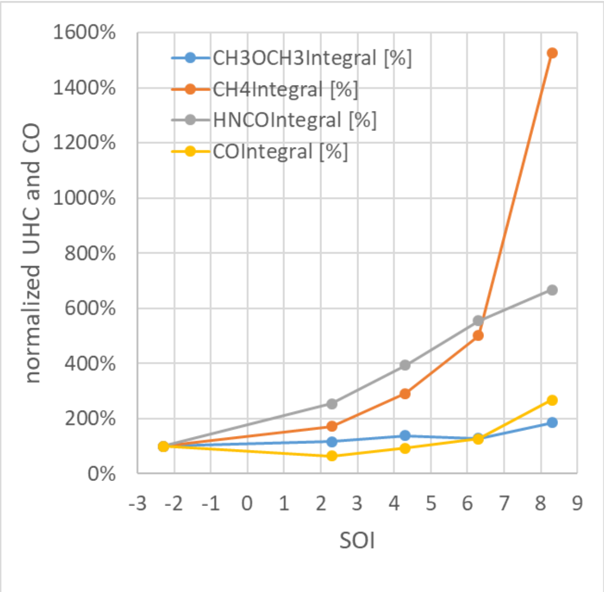

Figure 55: Normalized CO, HNCO, DME and CH_4 levels at EVO for the sweep in SOI.

Predicted CO levels increase only moderately over the SOI sweep, as is confirmed by the experiment (cf. Figure 10). DME (unburnt fuel) and CH4 have been chosen as potential main candidates to be compared to the UHC measured experimentally. While CH_4 increases dramatically, its absolute value for the earliest injection timing is 20 times lower than DME and only for the latest injection timing approaches the same order of magnitude. HCNO emissions are two orders of magnitude lower than those of DME for the earliest injection timings and therefore not considered critical.

7 Conclusions of the numerical campaign

The adopted CFD platform proved to capture correctly the impact of very high levels of EGR on the ignition timing at two different engine speeds and loads. For sweeps in SOI reasonable agreement was seen in terms of pressure traces and heat release rates at the lower EGR rates. For the two latest injection timings, especially in combination with high EGR, the significantly reduced reactivity leads to increasingly deteriorating predictive capability. For the reference condition, engine out emissions were examined and it was seen, that NO_x as well as UHC/CO trends were correctly reproduced. A comparison of two combustion models for two conditions suggests that RIF performs better than TWM at the conditions studied.

Considering this is the first application of the numerical platform to DME combustion at such high EGR levels, the simulation framework shows some promise for future design exploration. Nonetheless, further improvements of the platform are evidently necessary to improve its predictive capabilities at low reactivity before it can be used e.g. for product optimization purposes.

8 Assessment of late PCCI of DME from a manufacturer's perspective

For FPT Motorenforschung, this project studying PCCI in combination with the high EGR tolerance of DME showed very interesting results, considering upcoming legislation limits and the effort needed on engine and aftertreatment side to fulfil them, especially in low load cycles. Thanks to significant lowering of the engine out NO_x levels in combination with a rise on exhaust gas temperature, it seems that Euro 7 and Tier 5 standards can be fulfilled with a simplified Euro 6 or Tier 4 after treatment system (in contrast to diesel solutions, which will need complex, bulky and costly setups). This is interesting especially for the non-road customers with strong package challenges on their machines. Implementing a high-pressure-EGR loop including an EGR pump is a beneficial approach compared with alternatives like cylinder deactivation or heating elements to keep the SCR in the working area. Also, considering the beneficial fuel consumption compared with thermal management by an exhaust flap.

Beside all the technical benefits from DME, the challenge to introduce DME engines in series production are the actual poor availability of renewable DME and for the on-road applications on the “Tank to Wheel” approach in Europe. Even if trucks from Iveco Group are fuelled by renewable DME (or as already today with Bio-Methane), the CO_2 emissions from tailpipe counts fully to the truck manufactures CO_2 fleet budget.

For those reasons, the most likely use of DME engines could be in the national or international non-road market, especially for applications which are refilling always on the same fuel station as e.g. in mining business, construction machinery or agriculture. Also on the truck side, there could be interesting opportunities, but most likely outside Europe.

9 Outlook and next steps

As an immediate next step, the engine will be inspected in detail at FPT Motorenforschung in Arbon and the results will be discussed with the project partners as well as with suppliers. After approximately 450 hours of running on DME, some first conclusions about wearing are possible.

One major bottleneck for the industrialization of DME engine technologies is the lack of suitable high-pressure (common rail) pumps for DME. Within the project described in this report, a pump was used which is too limited regarding the intake pressure. The manufacturer of this pump has no plans to modify the pump as, for this particular supplier, the DME engine market is too small. Therefore, an alternative supplier was found who is mainly active in the lower-volume market or larger engines and this (Swiss) company is interested to develop such a pump for DME (and other low-lubricity fuels like Ammonia or Methanol). An innosuisse-funded follow-up project was set-up to develop such a pump and proof its functioning on exactly the engine used in this project. Once this critical key component will be available, it will be possible to build a P&D vehicle and to gain real-life experience on the road.

10 National and international cooperation

This project was a cooperation between the Swiss partners Automotive Powertrain Technologies Laboratory of Empa in Dübendorf and FPT Motorenforschung AG in Arbon. The Politecnico die Milano (Internal Combustion Engines Computational Laboratory, group of associate prof. Tommaso Lucchini) was closely involved in the development of tools for the reactive CFD simulation of in-cylinder processes with DME as a fuel. Within the framework of this project, a PhD student of Politecnico di Milano (Andrea Schirru) performed a research stay at Empa.

11 Publications

Within the framework of this project, the following publications have been made or are in the planning phase:

Schirru, A., Hardy, G., Wright, Y., Lucchini, T., D'Errico, G., Soltic, P., Hilfiker, T., "CFD Modeling of a DME CI Engine in Late-PCCI Operating Conditions," SAE Technical Paper 2023-01-0203, 2023, https://doi.org/10.4271/2023-01-0203.

Schirru, A., Hardy, G., Lucchini, T., D'Errico, G., Mehl, M., Hilfiker, T., Soltic, P., "Numerical Investigation on the use of Dimethyl Ether (DME) as an Alternative Fuel for Internal Combustion Engines", Fuel, 354, 2023, https://doi.org/10.1016/j.fuel.2023.129434

Soltic, P., Hilfiker, T., Wright, Y., Hardy, G., Fröhlich, B., Klein, D., "The potential of dimethyl ether (DME) to meet current and future emissions standards in heavy-duty compression-ignition engines", Fuel, 355, 2024, https://doi.org/10.1016/j.fuel.2023.129357

Soltic, P., "The Potential of Dimethyl Ether (DME) to Meet Current and Future Emission Standards in Heavy-Duty Compression-Ignition Engines", Tagung "Fahrzeugantriebe mit erneuerbaren Energieträgern, 28.-29.11.2023, Stuttgart

12 References

[1] Soltic P, Hilfiker T, Wright Y, Hardy G, Fröhlich B, Klein D. The potential of dimethyl ether (DME) to meet current and future emissions standards in heavy-duty compression-ignition engines. Fuel 2024;355. https://doi.org/10.1016/j.fuel.2023.129357.

[2] HDV-DME – Investigation of the suitability of DME as an alternative fuel in HDV. BFE SI/501794 2021. https://www.aramis.admin.ch/Grunddaten/?ProjectID=41773.

[3] N/N. Paris Agreement. United Nations 2015.

[4] Delbeke J, Runge-Metzger A, Slingenberg Y, Werksman J. The paris agreement. Towar. a Clim. Eur. Curbing Trend, 2019, p. 24–45. https://doi.org/10.4324/9789276082569-2.

[5] Bundesamt für Energie BFE. Energieperspektiven 2050+ 2020:36.

[6] Wenger F, Ziegler M, Wulkop A, Keberle A. Klimastandort Schweiz - Schweizer Unternehmen als globale Treiber für Netto-Null. 2022.

[7] Ulrich Kramer, David Bothe, Christoph Gatzen, André Pfannenschmidt, Carolin Baum, Fabian Schrogl, et al. Future Fuels: FVV Fuel Study IVb - Transformation of Mobility to the GHG-neutral Post-fossil Age. Frankfurt 7 Germany: 2022.

[8] N/N. Proposal for Regulation of the European Parliament and of the Council on Type-Approval of Motor Vehicles and Engine of such Systems, Components and Separate Technical Units Intended for Such Vehicles, with Respect to thir Emissione and Battery Durability . vol. 2022. 2022.

[9] Lowry J, Matt J. Tier 5 Rulemaking Workshop Potential Amendments to the Off - Road Diesel New Engine Regulations. CARB 2021.

[10] Klein D, Hardy G, Soltic P, Hilfiker T. HDV-DME Investigation of the suitability of DME as an alternative fuel in HDV. 2021.

[11] N/N. Nationale Plattform Zukunft der Mobilität: Schwere Nutzfahrzeuge – Standards und Normen für alternative Antriebe. 2021.

[12] European Parliament, Council of the European Union. Regulation (EU) 2018/1999 on the Governance of the Energy Union and Climate Action. Oj L 328 2018;2018:1–77.

[13] Skov IR, Schneider N. Incentive structures for power-to-X and e-fuel pathways for transport in EU and member states. Energy Policy 2022;168. https://doi.org/10.1016/j.enpol.2022.113121.

[14] Schühle P, Stöber R, Semmel M, Schaadt A, Szolak R, Thill S, et al. Dimethyl ether/CO2 – a hitherto underestimated H2 storage cycle. Energy Environ Sci 2023. https://doi.org/10.1039/D3EE00228D.

[15] Doll U, Altantzis C, Garica MM, Barro C, Wright YM, Boulouchos K. Lowest-emission compression ignition combustion concepts using partial stratification and multiple injections (PCCI). BFE Final Rep 2022;SI/501744-.

[16] N/N. FlexComb – Flexible engine platform with new combustion concepts for renewable fuels, SFOE, SI/502333 2021.

[17] Schirru A, D'Errico G, Lucchini T, Zhou Q, Hardy G, Soltic P, et al. Combustion Modeling in a Heavy-Duty Engine Operating with DME Using Detailed Kinetics and Turbulence Chemistry Interaction. SAE Tech Pap 2022. https://doi.org/10.4271/2022-01-0393.

[18] Schirru A, Hardy G, Wright Y, Lucchini T, D'Errico G, Soltic P, et al. CFD Modeling of a DME CI Engine in Late-PCCI Operating Conditions. SAE Tech Pap 2023-01-0203 2023:1–11. https://doi.org/10.4271/2023-01-0203.

[19] Arcoumanis C, Bae C, Crookes R, Kinoshita E. The potential of di-methyl ether (DME) as an alternative fuel for compression-ignition engines: A review. Fuel 2008;87:1014–30. https://doi.org/10.1016/j.fuel.2007.06.007.

[20] Brückner C, Pandurangi SS, Kyrtatos P, Bolla M, Wright YM, Boulouchos K. NOx emissions in direct injection diesel engines – part 1: Development of a phenomenological NOx model using experiments and three-dimensional computational fluid dynamics. Int J Engine Res 2018;19:308–28. https://doi.org/10.1177/1468087417704312.

[21] Pelucchi M, Schmitt S, Gaiser N, Cuoci A, Frassoldati A, Zhang H, et al. On the influence of NO addition to dimethyl ether oxidation in a flow reactor. Combust Flame 2022:112464. https://doi.org/10.1016/j.combustflame.2022.112464.