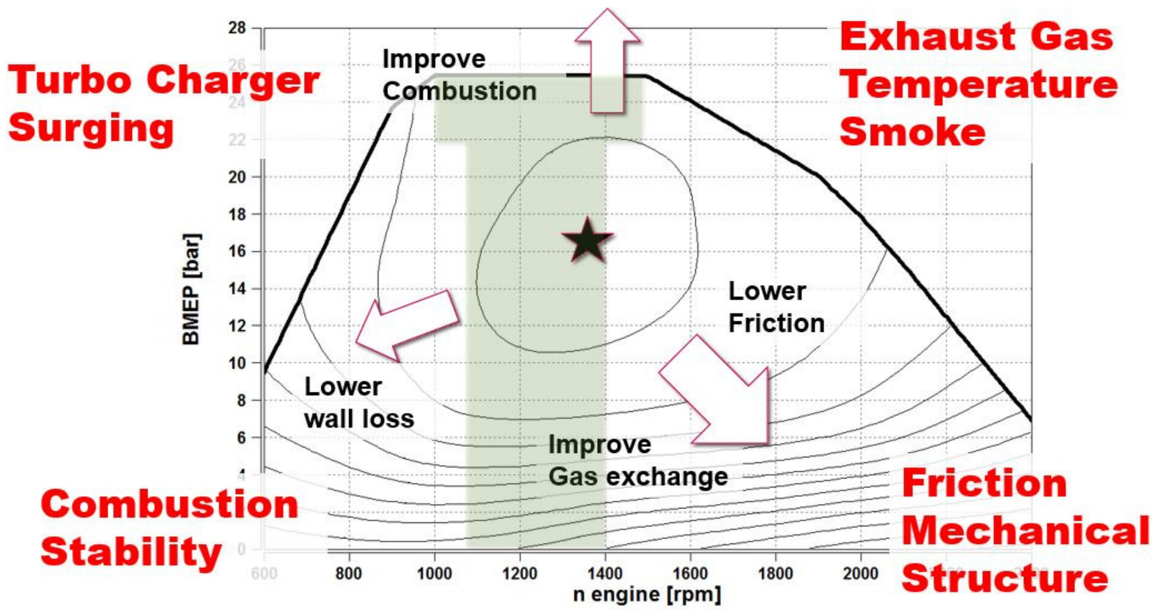
Final report dated 18 October 2019

Increasing Brake Thermal Efficiency of Heavy-Duty Diesel Engines for Long-Haul On-Road Vehicles up to 50%



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Zusammenfassung

Dieselmotoren für Nutzfahrzeuge, Baumaschinen und landwirtschaftliche Maschinen wurden im letzten Jahrzehnt aufgrund der Senkung der Abgasgrenzwerte intensiv weiterentwickelt.

Wegen der hohen Entwicklungsaufwände und der intensiven Forschung an Nachbehandlungssystemen wurden sauberere Motoren entwickelt, ohne dass der thermische Gesamtwirkungsgrad beeinträchtigt wird. Heute konzentrieren sich die Weiterentwicklungen auf eine höhere thermische Effizienz damit es zu einer Reduktion von CO2-Emissionen führt, um die von der EU festgelegten Ziele für 2030 zu erreichen.

Mit Rücksicht auf den momentanen Stand der Technik, sind die mechanische und Verbrennungseffizienz die dominierenden Punkte zur Verbesserung des thermischen Wirkungsgrades von Dieselmotoren. Ohne die Verwendung von Abwärmerückgewinnungs-Technologien, wie z.B. eTurbo oder Rankine-Zyklus, ist ein thermischer Wirkungsgrad von 47% bis 48% mit einem spezifischen NOx-Wert nach Motor von 10 bis 12 g/kWh erreichbar, um den AdBlue-Verbrauch auf dem aktuellen Niveau der Euro-VI-Motoren zu halten. Dies konnte im Projekt mit Verbrennung bei höheren Spitzendrücken, einer verbesserten Brennraumform, neuen Injektorendesigns sowie mit der Verwendung flexibler AGR-Technologien experimentell nachgewiesen werden.

Die Verwendung von Abwärmerückgewinnungs-Technologien für eine hocheffiziente Motorbasislinie wie eTurbo hat nur begrenzte Auswirkungen auf den thermischen Wirkungsgrad, da die Abgastemperaturen niedrig sind. Um dies zu entschärfen, kann der Motor verkleinert werden, was zu höheren Abgastemperaturen führt.

Résumé

Poussés par la réduction des limites d'émissions à l'échappement, les moteurs diesel pour véhicules utilitaires, engins de chantier et machines agricoles ont connu un développement intensif au cours de la dernière décennie.

En raison des efforts de développement importants et des recherches intensives menées sur les systèmes de post-traitements, des moteurs plus propres ont été mis au point sans nuire à l'efficacité thermique globale. Aujourd'hui, de nouveaux développements sont axés sur une plus grande efficacité thermique afin de réduire les émissions de CO2 et ainsi respecter les objectifs fixés par l'UE à l'horizon 2030.

Compte tenu de l'état actuel de la technologie, les rendements mécanique et de combustion sont les deux principaux contributeurs à l’augmentation du rendement thermique global d’un moteur diesel de poids-lourds. Sans l'utilisation de technologies de récupération de chaleur perdues telles que e-Turbo ou cycle Rankine, un rendement thermique de 47 à 48% est réalisable tout en maintenant le NOx spécifique à 10-12 g/kWh en sortie moteur afin de maintenir la consommation d'AdBlue au niveau actuel des moteurs Euro VI.

L'utilisation de technologies de récupération de chaleur perdue telles que eTurbo sur une base de moteur très efficace a un effet limité sur l'efficacité de rendement thermique final car les températures d'échappement sont relativement basses. Pour contourner ce fait, la capacité volumétrique du moteur pourrait être réduite pour permettre des températures d'échappement plus élevées.

Sommario

I motori diesel per veicoli commerciali, le macchine movimento terra e i macchinari agricoli hanno subito un intenso sviluppo nell'ultimo decennio grazie alla spinta esercitata dalla continua riduzione dei limiti di emissione dei gas di scarico.

Grazie ad elevati sforzi della ricerca condotta sui sistemi di post-trattamento e al loro sviluppo, sono stati sviluppati motori sempre più puliti senza compromettere l'efficienza termica. Oggi, ulteriori sviluppi si concentrano su una maggiore efficienza termica per ridurre le emissioni di CO2 al fine di essere in linea con gli obiettivi fissati dall'UE per il 2030.

Considerando l'attuale stato dell'arte, la combustione e l'efficienza meccanica sono i due principali fattori che contribuiscono all'ulteriore aumento dell'efficienza effettiva di un tipico motore diesel per veicoli pesanti. Senza l'uso delle tecnologie di recupero del calore residuo come l'e-Turbo o il ciclo Rankine, è possibile raggiungere un'efficienza termica del 47-48% mantenendo il livello di NOx specifico al di sotto di 10-12 g/kWh al fine di mantenere il consumo di AdBlue all'attuale limite prescritto dalla normativa Euro VI dei motori.

L'uso di tecnologie di recupero del calore residuo come l'eTurbo come soluzione per motori ad alta efficienza ha un effetto limitato sull'efficienza effettiva causa temperature di scarico molto basse. Una soluzione sarebbe rappresentata dal ridimensionamento del motore per avere delle temperature di scarico più elevate.

Summary

Driven by the reduction of the exhaust emission limits, diesel engines for commercial vehicles, construction equipment's and agricultural machineries have undergone intensive development over the last decade.

Due to high development efforts and intensive research carried out on after treatment systems, cleaner engines were developed without impairing their overall thermal efficiencies. Today, further developments are focused on higher thermal efficiency to reduce CO2 emissions in order to be in line with targets set up by the EU for 2030.

Considering the current state-of-the art, the combustion and mechanical efficiencies are the two main contributors to the further increase of the brake thermal efficiency of a typical heavy-duty diesel engine. Without the use of Waste Heat Recovery technologies such as e-Turbo or Rankine cycle, a thermal efficiency of 47-48% is achievable while keeping specific NOx engine out at 10-12 g/kWh in order to keep AdBlue consumption at current Euro VI engines level.

The use of Waste Heat Recovery technologies such as eTurbo on a highly efficient engine baseline has limited effect on the Brake Thermal Efficiency because the exhaust temperatures are on the lower side. To circumvent this fact, the engine could be downsized for higher exhaust temperatures.

Main findings

- Increasing the engine geometric compression ratio to reach a peak cylinder pressure of 250 bars in combination with an open piston bowl has a beneficial effect on the combustion efficiency as well as on the specific fuel consumption / NOx trade-off.
- An EGR Volumetric Pump allows Exhaust Gas Recirculation on-demand to improve the coupling between the engine and the after treatment system. A precise EGR flow can be controlled, in most of the engine operating points, independently of the Turbocharger condition while the power requirement of the pump is below 2kW when installed within a high pressure EGR loop.
- The use of Waste Heat Recovery technologies such as eTurbo brings only benefits if a high voltage installation (> 400V) is already installed and this in conjunction with a downsized engine able to recover higher exhaust gas temperatures.
- To bring the thermal efficiencies to further limits, thermal insulation of the in-cylinder is needed but this would require more fundamental R&D to find out functional and durable coatings, in particular for highly loaded diesel engines.
- The reduction of the fuel consumption by 8% for a fleet of 20000 long haul heavy-duty (40T) commercial vehicles traveling 140000 km per year would lead to about 184 kT of CO_2 saving per year in comparison with 2019.

Contents

Abbreviations

AHRR	Apparent Heat Release Rate (also called net HR) [J/CA] or [J/s]
ATS	After-Treatment System
BMEP	Brake Mean Effective Pressure [bar]
BSFC	Brake Specific Fuel Consumption [g/kWh]
BS NOx	Brake Specific NOx [g/kWh] (in general raw, after engine in this study)
BTE	Brake Thermal Efficiency [%]
CA	Crank Angle [°aTDC]
CR	Compression Ratio [-]
CUC	Clean-Up Catalyst
DOC	Diesel Oxidation Catalyst
DPF	Diesel Particulate Filter
DT	Dwell-Time [us]
ECU	Engine Control Unit
EGR	Exhaust Gas Recirculation [%]
eTurbo	electrical Turbocharger
eTCD	electric Turbo compound (= power turbine)
EVO	Exhaust Valve Opening [°CA aTDC]
FEP	Fuel Efficiency Package (FPT notation of the development stage of an engine)
FMEP	Friction Mean Effective Pressure [bar]
FPT	Fiat Powertrain Technologies
HCCI	Homogeneous Charge Compression Ignition
HR	Heat Release [J]
HRR	Heat Release Rate [J/CA] or [J/s]
IMEP	Indicated Mean Effective Pressure [bar]
IMEPg	gross Indicated Mean Effective Pressure [bar]
IMEPn	net Indicated Mean Effective Pressure [bar]
ISg NOx	gross Indicated Specific NOx [g/kWh]
ITEg	gross Indicated Thermal Efficiency [%]
IVC	Intake Valve Closing [°CA aTDC]
MCE	Multi-Cylinders Engine
mfb50	CA at mass fuel burnt reaching 50% [°CA aTDC]
NOx	Nitrogen Oxide (NO + NO2)
ORC	Organic Rankine Cycle
P Boost	Boost Pressure [bar] (Intake Manifold)
P Exh	Exhaust Pressure just after the engine [bar] (Turbine Upstream)

PCP	Peak Cylinder Pressure [bar]
PPC	Partially Premixed Combustion
PMEP	Pumping Mean Effective Pressure [bar]
pRail	Injection Rail Pressure [bar]
q_inj	fuel volume quantity [mm3]
Qch	Chemical Heat Release [J]
Qnet	net Heat Release [J]
RPM	Rotation per Minute [1/min]
SCR	Selective Catalytic Reduction
SOIe	electric Start Of Injection [CA]
t	time [s]
T	Temperature [K]
T/C	Turbo charger
TCB	Thermal Coating Barrier
TCD	Turbo Compound
TtW	Tank-to-Wheel
VGT	Variable Geometry Turbine
WHR	Waste Heat Recovery
WtW	Well-to-Wheel

1 Introduction

1.1 Background information and current situation

Brake Thermal Efficiency

The Brake Thermal Efficiency (BTE) tells about the conversion efficiency from the heat coming from a chemical source (Fuel) to a mechanical output work.

In order to facilitate the understanding of the various phenomena involved in the thermodynamic process, various definitions have been proposed.

At FPT Motorenforschung, the following efficiency split-up definition is used for respectively: Mechanical, Gas Exchange, Combustion and Wall Heat loss, see Equation 1. These four efficiencies are not independent from each other. For example, there is a strong link between combustion and wall heat loss efficiencies as higher in-cylinder gas temperature would lead to improved combustion but on the other hand increase wall heat loss (lower efficiency).

Nevertheless, it is a quite practical way to identify areas for further improvements and develop technologies accordingly.

$$BTE = \eta_{Eng} = \frac{P_e}{\dot{Q}_{fuel}} = \underbrace{\frac{P_e}{P_i}}_{\eta_{Mech}} \cdot \underbrace{\left(1 + \frac{P_{iLP}}{P_{iHP}}\right)}_{\eta_{GasEx}} \cdot \underbrace{\frac{P_{iHP}}{\dot{Q}_{fuel} - \dot{Q}_{wall}}}_{\eta_{Combustion}} \cdot \underbrace{\frac{\dot{Q}_{fuel} - \dot{Q}_{wall}}{\dot{Q}_{fuel}}}_{\eta_{Wall}}$$

η_{Mech} … Mechanical Efficiency (if >100%: ok)
η_{Comb} … Combustion Efficiency
η_{Wall} … Wall Heat Loss Efficiency
η_{GasEx} … Gas-Exchange Work Efficiency (if >100%: ok)

P_e ……. Brake Power (at flywheel)
Q_{Fuel} …. Fuel Energy
P_{iHP} …… Ind.Power during high pr. Cycle
P_i ……. Indicated Power (Cylinder pressure)
Q_{Wall} …. Total Wall Heat Flux
P_{iLP} ….. Ind.Power during low pr. cycle

Equation 1: Brake Thermal Efficiency Split-up definition

Gas Exchange Efficiency can be above 100% when the Boost pressure is higher than the Exhaust pressure which is the case for a large area of the engine map of a typical heavy-duty diesel engine.

Mechanical efficiency can also be higher than 100% when, for example, a turbine from a Waste Heat Recovery coupled to the engine provides additional work.

The figure 1 gives an idea of the efficiencies split up (η mech, η gas exch. η comb, η wall HT) for a typical heavy-duty diesel in the engine map (BMEP as function of RPM). The white star shows the location of the maximum respective efficiency.

If the gas exchange and mechanical efficiencies have a similar pattern in the engine map, wall heat loss and combustion efficiencies have a completely different pattern resulting in lots of compromises to reach a high BTE (η engine) in the engine map combined with a large area surrounding the maximum BTE (sweet spot).

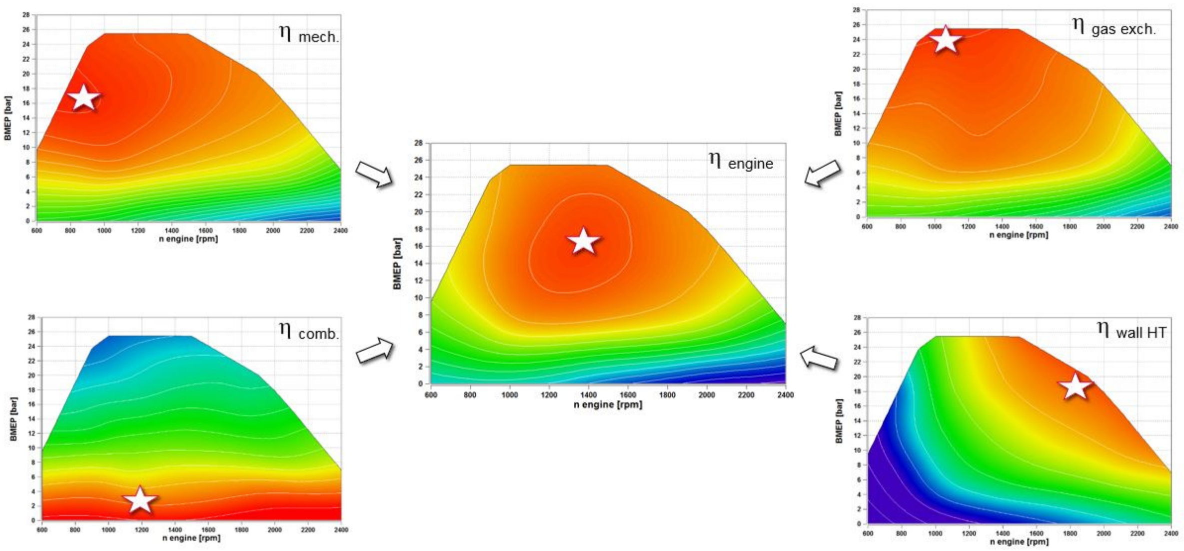


Figure 1: Brake Thermal Efficiency Split-up in the Engine Map

The figure 2 shows the restrictions in the engine map due to mechanical and thermal limitations such as the maximum exhaust gas temperature limited by the turbine or the engine friction increasing at high RPM. The T-shape represents the area where an on-road heavy-duty diesel engine spend most of its missions. It is centred around 1200 rpm because the gearbox and differential ratio are chosen to accommodate a cruising speed of 90 km/h. The white arrows show the actions / countermeasures required to enlarge the area around the sweep spot which would have a beneficial effect on the fuel consumption measured over one cycle.

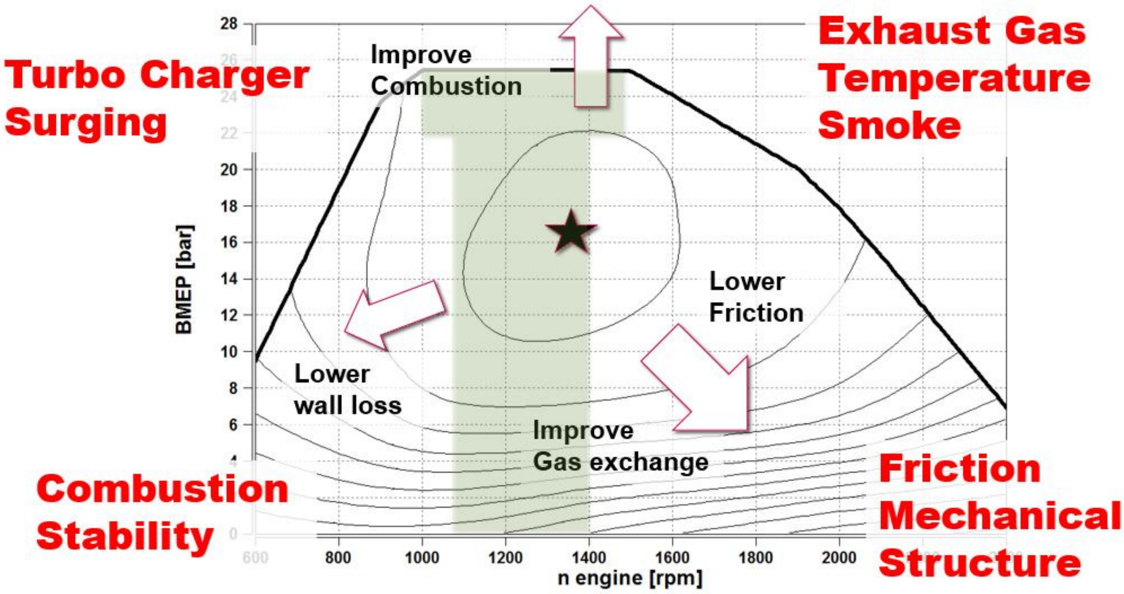


Figure 2: Limitations in the Engine Map leading to counter measures

Today's, current FPT Cursor 11 Euro VI at BS NOx engine out = 10g/kWh, reaches about 43.6% of BTE for the best point in the engine map, at 1200rpm x 75% load.

Approach for 1200x75% load single stage T/C Eng. Out: BS NOx = 10g/kWh	Mechanical Efficiency [%]	Gas Exchange Efficiency [%]	Combustion Efficiency [%]	Wall Heat Loss Efficiency [%]	Brake Thermal Efficiency [%]
EURO VI no EGR	93.1 %	99.0 %	55.7 %	84.9 %	**43.6 %**

The combustion efficiency tells how much of the heat available (Fuel energy - wall Heat loss) is transformed into gross work (indicated high pressure of cylinder loop in p-V diagram). A wall heat loss efficiency equalling 100%, would represent an adiabatic engine. The specific fuel consumption of an engine without any loss (BTE = 100%) would be 84.1 g/kWh

SuperTruck Program in U.S.

The U.S. Department of Energy SuperTruck program is a public-private partnership (284 million US$) that promotes R&D to improve the efficiency of long haul Class8 heavy-duty trucks. The goal of this project is to develop and demonstrate state-of-the-art feasible and efficient technologies for long haul tractor-trailer

From "Committee to Review the 21st Century Truck Partnership, Phase 3", in 2015, the following BTE claimed by OEM were the following:

- Cummins 15L ISX: 51.1% engine with ORC BTE demonstrated (ORC = +3.6%pt)
- Daimler 11L DDC11: 48.2% BTE engine-only demonstrated
- Volvo 11L D11: 48.0% BTE engine-only demonstrated
- Navistar 13L MaxxForce (DAF): 47.4% BTE engine-only with 2-Stage T/C demonstrated

No indication of BS NOx engine out were mentioned. The maximum Brake Thermal Efficiency (BTE) is an indication of the best efficiency of the engine in the best point only and some technologies like turbochargers can be tuned to perform well on the best point, but they could introduce disadvantages in terms of drivability, power and smoke during transients.

1.2 Purpose of the project

Driven by the reduced emissions limits, diesel engines for commercial vehicles, construction equipment and agricultural vehicles have undergone strong development over the last decade.

Due to high development effort and intensive research, clean engines were developed without impairing the efficiency and the consumption. Further improvement is now focused on thermal efficiency.

The purpose of the project is to demonstrate through measurements on testbed and 1-D cycle simulations, the potential of various enhanced technologies improving the combustion, gas exchange and friction.

1.3 Objectives

In the present project, the stated goal is to achieve a brake total efficiency of 50% by increasing simultaneously the Combustion and Wall Heat Loss Efficiencies alongside the Gas exchange efficiency. Testbed measurements have been carried out to demonstrate combustion improvements, whereas 1-D cycle simulations are used to show the technological path to achieve the target BTE with Waste Heat Recovery technologies and Thermal Boundary Coating.

At the onset of the project, the emphasize was purely on CO2 reduction by improving BTE whereas now, due to the fact that in the meantime future even stricter NOx emissions have been announced, the emphasize is more on finding a suitable engine platform from gas handling and combustion layout to fulfil new emission regulations with minimum impact on fuel consumption.

2 Description of facility

2.1 Base Engine: Cursor11 FEP2 Euro VI

- The base engine, figure 3, used for the project is a 6 cylinders Cursor11 FEP2 Euro VI equipped with a high pressure EGR circuit. It is currently powering the Heavy-Duty Iveco Stralis vehicles. Rated power is 353 kW @ 1900 rpm and Torque is 2300 Nm @ 950 rpm to 1465 rpm
- The base engine was used for the development of an open combustion chamber layout (New Bowl, enhanced injectors, Lo-Swirl, Hi-CR…), high peak cylinder Pressure up to 270 bars, and an EGR Pump based on a Volumetric Pump. The upgraded base engine is called Cursor11 FEP2 Empa
- A low friction package Cursor11 FEP3 engine was developed at FPT-Arbon. The measurement results are used to extrapolate a low friction technology on the Cursor11 FEP2 Empa

Figure 3: Base engine and performance curves

2.2 Modified Engine: Cursor11 FEP2 Empa

The base engine was fitted with a smaller turbine T/C, more suitable for higher EGR at low RPM to avoid compressor surging. A volumetric Eaton compressor, V250A, used as a EGR pump is installed between the EGR cooler outlet and the intake manifold. It is driven by an electric motor whose speed can be accurately controlled from the test bed. The EGR cooler is connected to a conditioning unit to control the gas temperature above the dew point to avoid water droplets condensation which could harm the compressor high speed rotors. Non-Return Valves at the EGR cooler outlet were kept installed during the measurement performed for the Work Package 3. The EGR valve was kept in fully open position in order that only the EGR pump controls the flow rate. Figure 4 shows the installation of the main air handling components.

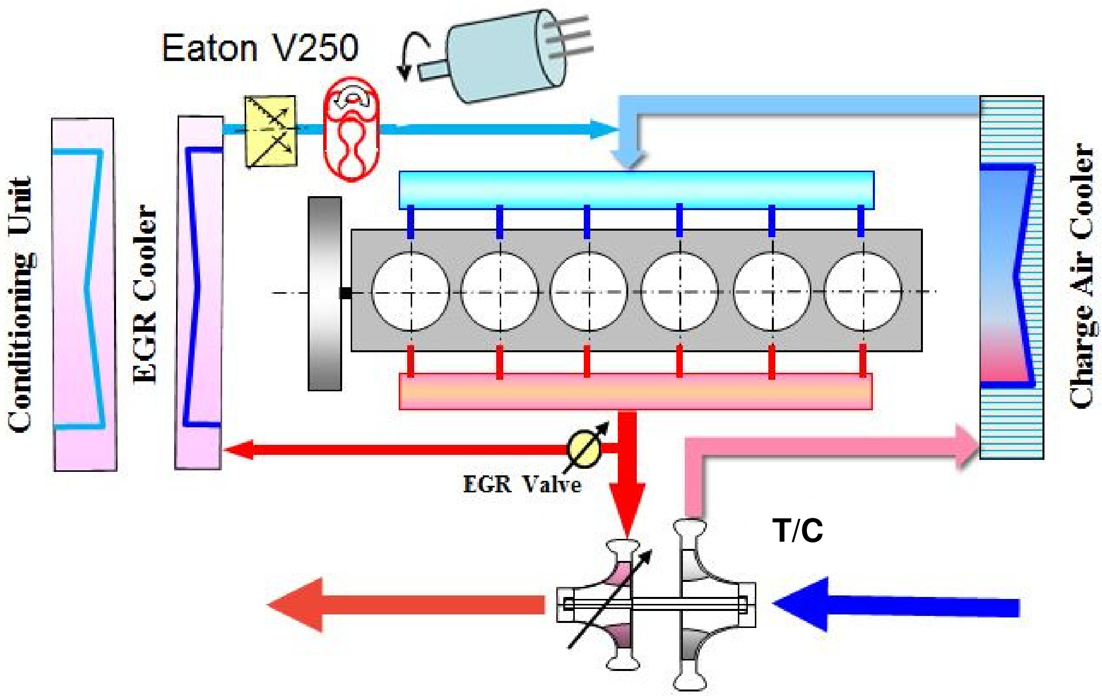


Figure 4: C11 FEP2 Empa Air handling circuit layout configuration

2.3 Testbed Measurement setup at Empa

The Cursor 11 engine output shaft is linked to a Horiba asynchronous dyno motor. A high coolant temperature conditioning unit controls the engine coolant temperature whereas a low temperature unit controls the charge air cooler, see figure 5. The ECU used is a ETK version accessible via INCA software allowing a complete remote control on the engine (Fuel Injection, Turbo charger)

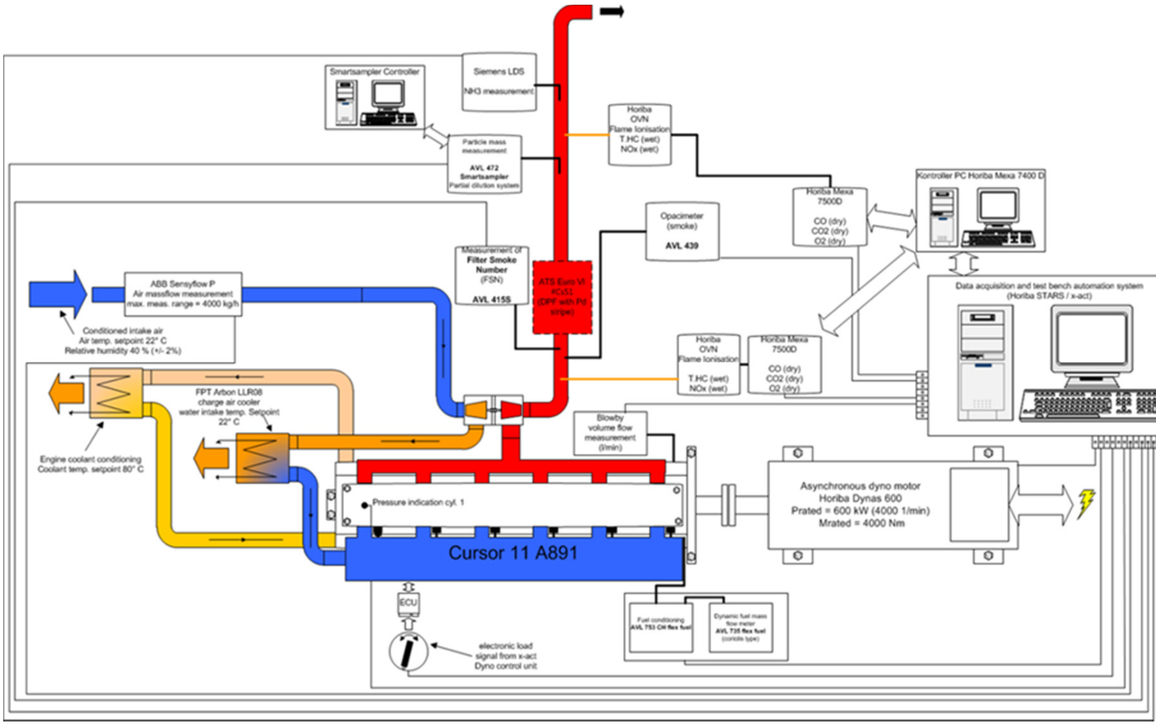


Figure 5: Schematic of measurements set up at Empa, engine side

At the engine outlet, see Figure 6, soot is measured through two different methods: Smoke Smoke FSN AVL415 and MicroSoot AVL. The other species such as NOx, HC, CO, CO2 and O2 are measured before and after the after-treatment system. CO2 is measured in the intake manifold to calculate the EGR rate. At the tail pipe, NH3 slip and urea mass flow are measured as well.

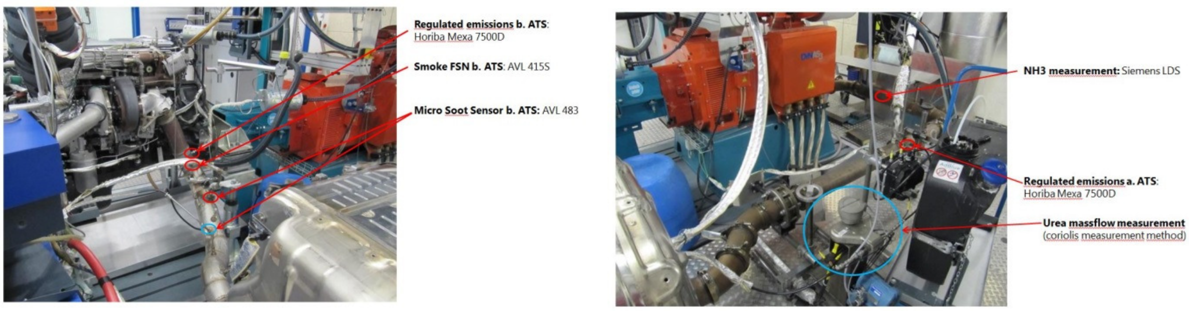


Figure 6: C11 FEP2 Empa Tail Pipe measurements

For each load point, about 170 measurement data (low frequency / averaged) are stored by the test bed together with the high frequency dynamic data for the cylinder pressure, intake + exhaust pressures and the solenoid current with a resolution of 0.2°CA.

3 Procedures and methodology

3.1 Work Package description

WP1: Fuel Injection characterization of new injectors suitable for use with Open chamber piston bowl shape. 3D CFD optimization of Open chamber combustion chamber for low Swirl and High PCP.

WP2: Ordering prototypes and instrumentation of High PCP, Low-Swirl cylinder head as well as high pressure injection system. Turbocharger and nozzles upgrade based on WP3 Investigation. Second calibration with new Turbocharger and Nozzle.

WP3: 1-D cycle simulation led design of a novel exhaust gas recirculation system using a volumetric pump (Eaton Roots Blower) and testbed demonstration at Empa for steady state performance. Potential demonstration of the Open Chamber piston bowl designed in WP1. On-demand EGR for Hi-PCP and Hi-pRail engine to find optimum between SOI, pRail, pBoost with three to four different BS NOx level at engine outlet for 24 load points in engine map. Investigation of Post injection to reduce soot.

WP4: GT-power simulation of WHR solution for integration with Cursor11 layout taking into consideration a reduced friction package.

3.2 1-D Cycle Simulation for Components Optimization

In order to optimise individually air handling components such as Turbocharger, EGR pump, EGR cooler, eTurbo, … a detailed model of the Cursor11 engine in GT-power was built, see figure 7. Such models are required to simulate the potential gain of friction reduction or use of waste heat recovery technologies.

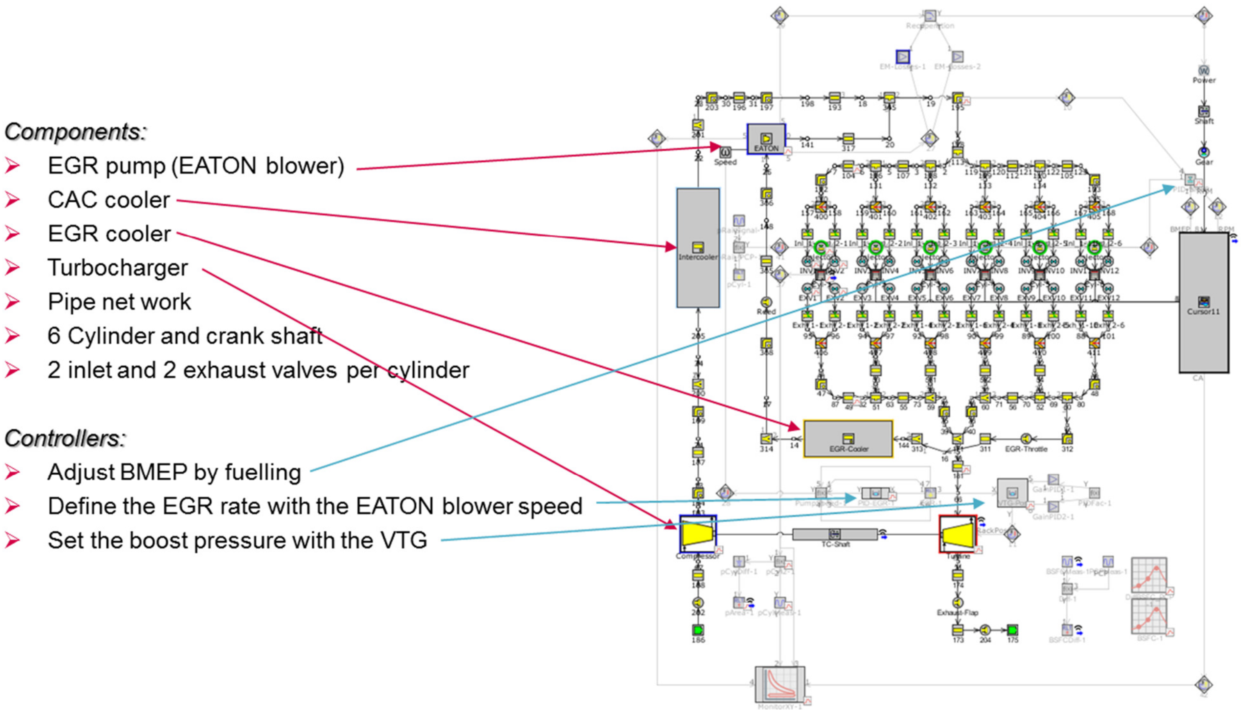


Figure 7: Accurate GT-power model of the Cursor11

The GT-power model went through an intensive validation activity against engine measurements in order to capture the main trends of the important engine characteristics such as BSFC, Peak Cylinder Pressure, …

In general, we see a good correlation of measured and predicted data for the validated engine however, BSNOX shows a deviation of ±20%, see figure 8.

The most relevant data for the turbo charger are predicted well: exhaust gas temperature and pressure before turbine (for a defined boost) show a good trend capture. This is important to evaluate potential gains from waste heat recovery by a turbine, since the extractable energy increases with the exhaust gas temperature.

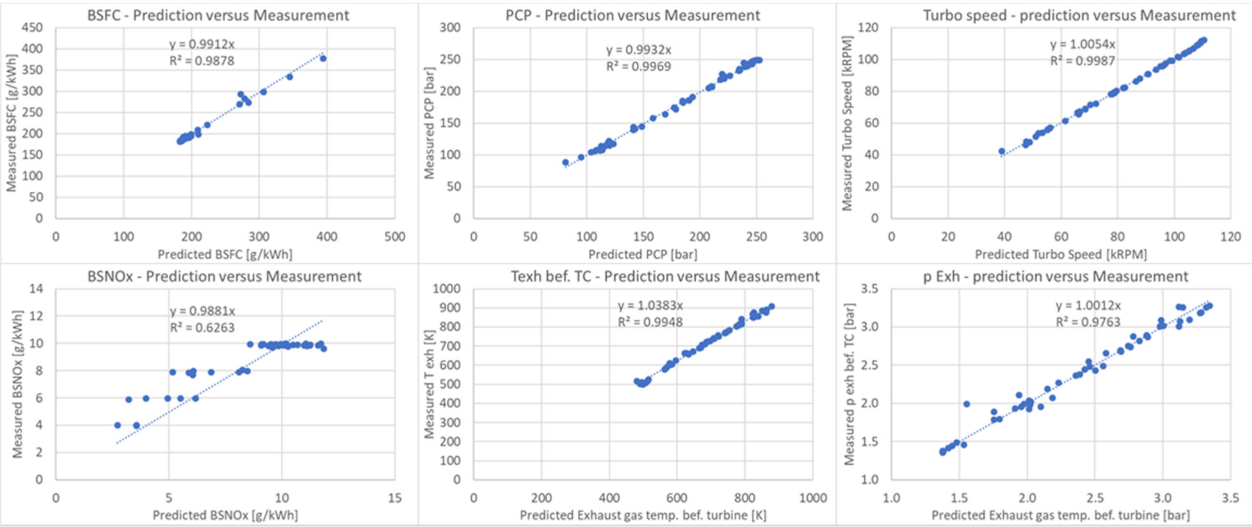


Figure 8: GT-power model correlation between predicted and measured data

3.3 Combustion Efficiency related Hardware

Engine Head

Hi-PCP: the engine head is made out of CGI (Compact Graphite iron) which is a material characterised by an elastic modulus of 140 Gpa that is about 1.4 times higher than currently used grey cast iron. This allows increasing the current head's design peak cylinder pressure (PCP) up to 250 bars in the field and 270 bars for research purpose. This enables the exploration of the benefits on thermal efficiencies of running with higher compression ratio in combination with higher cylinder pressure.

Low-Swirl: the selected swirl generated by the intake ports was halved in order to gain from higher flow coefficients to reduce the gas exchange loss contribution from the head and be able to increase the number of injector nozzle holes without impacting the spray-to-spray interaction.

Instrumentation: Thermocouples were located between the exhaust valve on the flame plate side in order to monitor the temperatures.

Fuel Injection System

Increased Common Rail Pressure: the fuel injection system was upgraded up to 2500 bar pressure with possibilities to reach 3000 bar for testing purpose. The combined higher rail pressure with higher EGR level has the potential to improve the BSFC-NOx trade-off.

Injector with fast opening/closing: the injectors were modified by the fuel injection laboratory in Arbon to maximise the opening / closing gradient of the injection rate to reduce dwell time between successive injections for pilot-Main and Main-post.

Post-Injection: injecting a quantity of fuel representing 10 % to 15 % of the total injected mass just after the main injection has the potential to reduce the engine's soot raw emissions.

Ks and Alt- nozzles: Two types of nozzles were tested in order to select the best NOx-Soot Trade-off. The Ks1.5 type nozzle which is currently installed on Euro VI engines has convergent holes and is subjected to hydro-erosive grinding during manufacturing process to suppress cavitation as much as possible. The liquid spray shall in theory be penetrating more into the combustion chamber. On the other hand, the Alt- nozzle have parallel holes and has a quite reduced sac hole while the guiding of the needle is improved. The liquid spray shall be less penetrating at low injection pressure as turbulence induced at the nozzle exit is higher, resulting in a wider spray angle. This shall reduce liquid and gas interaction with the surrounding walls of the combustion chamber at partial load when charge density is reduced. The injectors have a hydraulic flow equals to 1100cc / 30s for a ΔP injection = 100 bars.

Pistons

High Compression ratio: the bowl shape volume in the piston was defined to reach a compression ratio of about 20.5 which was found from a previous FPT internal study to be the best compromise between combustion gain, PCP and friction loss.

Open Chamber with recess (Figure 9): a 3D CFD optimisation study was performed in WP1 to optimise a bowl shape suitable for higher injection pressure and higher EGR. At increased BMEP (> 26 bars), the flame wall interaction with the piston walls (especially the recess) leads to higher heat loss. An open Chamber could be beneficial to reduce heat transfers as the impact length can be increased by 30% (from 30mm to 40mm), compared with the currently used H-bowl, to reduce local heat flux. Emphasises on soot reduction at part load was the main target to avoid potential EGR blower soot contamination and mitigate soot increase with higher EGR levels. The recess has a function to split the flame into two parts: one towards the head and one towards the piston bowl. This avoids soot contamination on the liner. The pistons used for WP3 and in most of the next planned activities of WP2 are standard steel piston with standard conrod length.

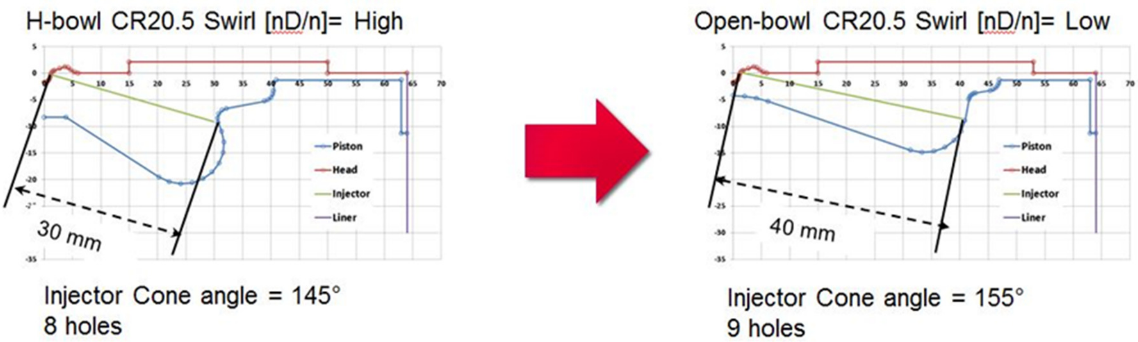


Figure 9: Geometry comparison of H-bowl (left) and Open chamber (right)

3.3 Gas Exchange Efficiency related Hardware

EGR is an effective way to reduce the engine's raw NOx emissions. An EGR rate of 10% has the potential to reduce the NOx after the engine by 50 % (Figure 10), therefore EGR on-demand as function of the SCR operating requirements could help to realize a further NOx reduction while maintaining or even lowering CO2 emissions.

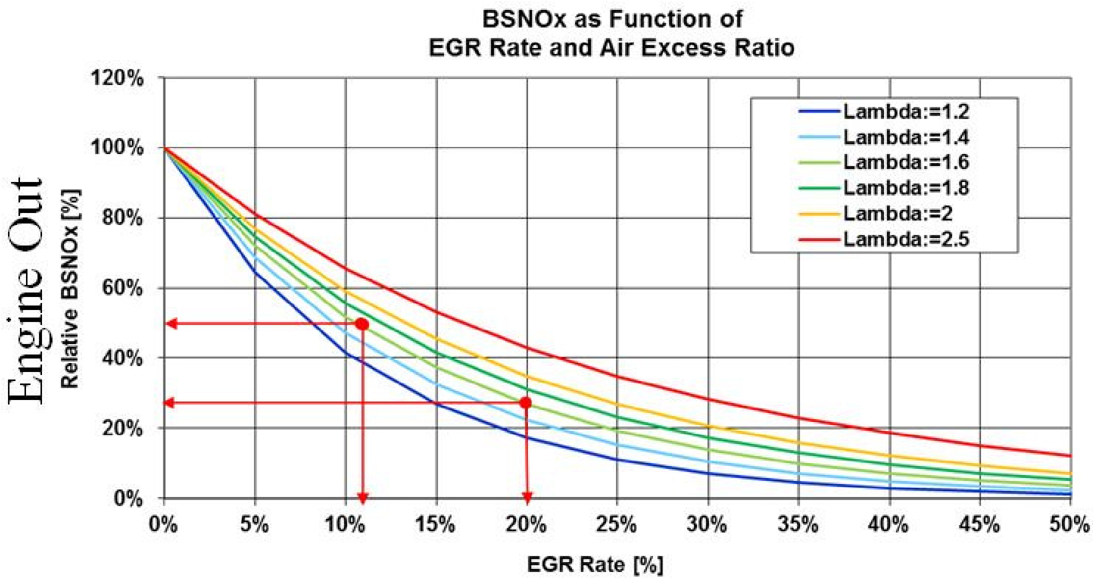


Figure 10: Engine NOx out in function of EGR

In general, heavy duty diesel engines are equipped with an EGR system of High Pressure loop configuration (Figure 11) leading to several limitations:

- To recirculate EGR, the Exhaust Pressure must be higher than the Boost Pressure, limiting the EGR level in the engine map. Non-Return valves located just after the EGR cooler permits to capture the pulsations peaks of the exhaust flow allowing EGR even if averaged boost pressure is slightly higher than the exhaust pressure.
- An exhaust flap located after the turbine is mandatory to increase the back pressure for certain operating conditions.
- A rather complex air path control is required in order to achieve the necessary air excess (Lambda) and EGR level.

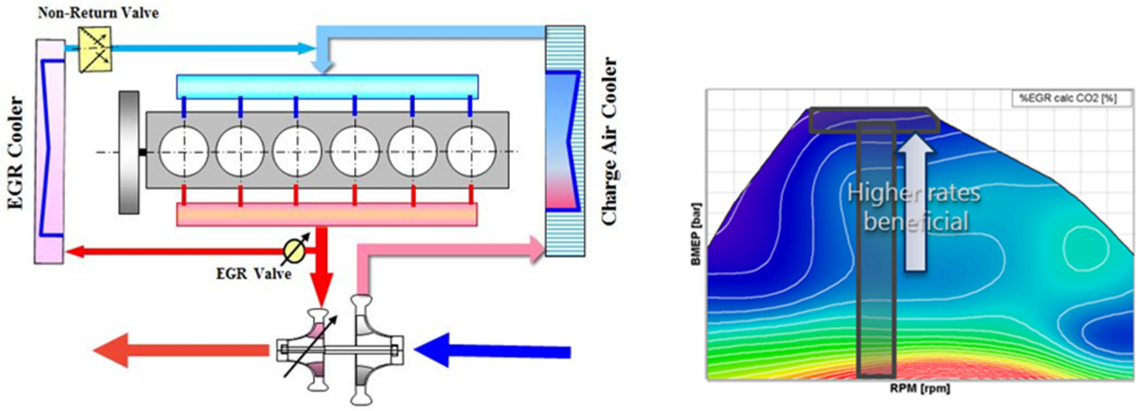


Figure 11: Typical HP-EGR Configuration used on HD Diesel engines and resulting EGR level

A volumetric pump applied to a High Pressure loop EGR layout would allow to control the EGR independently from the Engine Operating points.

- EGR on-demand to improve the coupling between the ATS system and the engine.
- The volumetric pump requires a limited amount of power as only a small pressure increase of the order of 0.3 to 0.4 bar is required.
- The EGR flow control could be simplified as it is mainly a function of the speed of the volumetric pump.
- In operating points where the T/C can deliver a positive pressure ration between exhaust and intake, a volumetric pump can still control EGR while it can recover power.

EGR Pump

Compared with reciprocating, piston type, compressor, the rotary compressors have: a smaller size, require no valves, a more uniform gas flow and are more suitable for lower pressure ratio.

There are two main types of compressors: the lobe or Roots type and the screw or Lysholm type.

The first EGR blower provided by Eaton is an available on-the-shelf helix roots type volumetric blower, type V250A, see Figure 12, modified with special sealing's to sustain higher absolute pressure, around 3.30 bars, at the inlet of the compressor which is required for the dedicated application on the Cursor 11. Such unit is normally used for large pressure ratio PR in the order of 2.4 to Boost performance cars whereas we need only about 300 to 400 mbar max. of pressure rise (Exhaust to Boost) for the Cursor 11. Installation on the testbed is shown Figure 13.

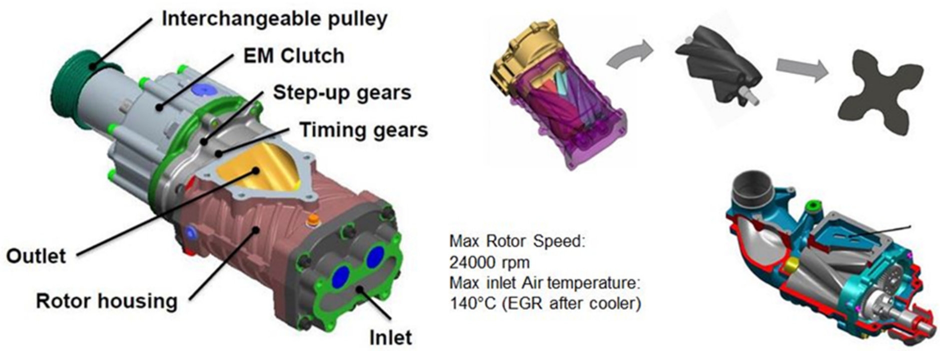


Figure 12: V250 Eaton Compressor (source: Eaton)

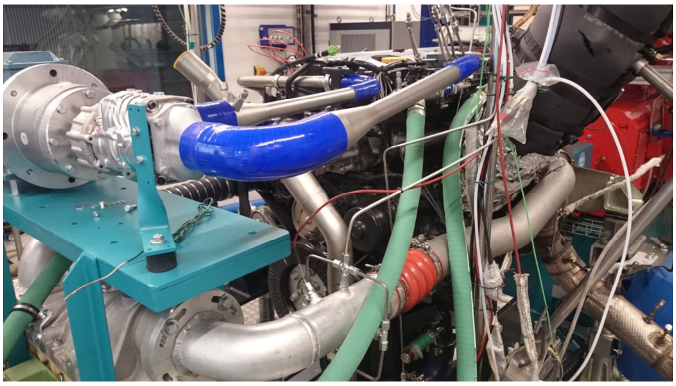


Figure 13: V250 Eaton Compressor installation on test bed

The volumetric flow of EGR is in theory a linear function of the EGR pump drive speed for low leakage, see figure 14. This is a major advantage for controlling the EGR rate independently of the operating load and T/C condition.

For most operating loads, the EGR blower power consumption is below 0.5% of the total engine power output. At part load, with an EGR rate above 20%, the EGR pump power consumption can reach 2%, see Figure 14. This shows the benefit of putting the pump in the High Pressure EGR loop. In certain operating points, the blower power is slightly negative which means, the pump is able to deliver power.

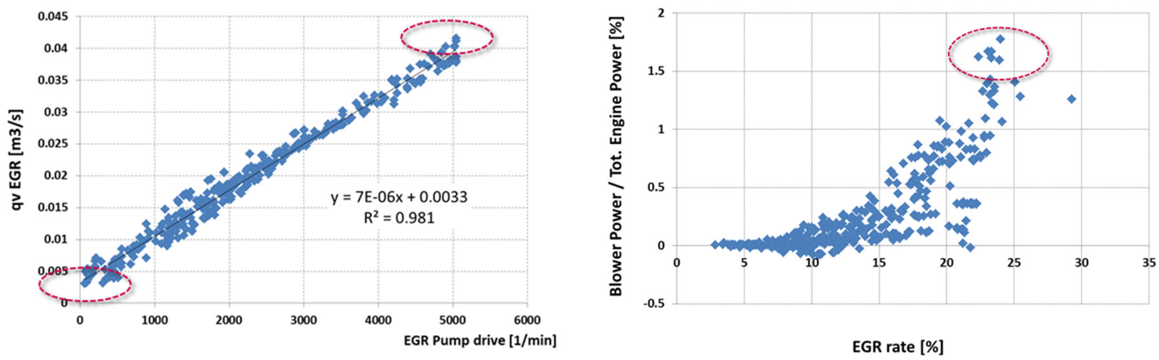


Figure 14: Eaton Compressor Volumetric flow and Power requirements

Following the encouraging results obtained with the original compressor, during the project, Eaton proposed to develop a dedicated EGR pump prototype with steel case and rotor to cope with the high temperature demands as well as reducing risk of erosion from water droplets occurring if temperature drops below the dew point. A 3kW 48V electric motor with direct drive is integrated with the pump to improve efficiency and easier package, see figure 15.

This dedicated EGR pump is able to recover energy (Turbine mode) in a more efficient way for Pressure Ratio below 1.0.

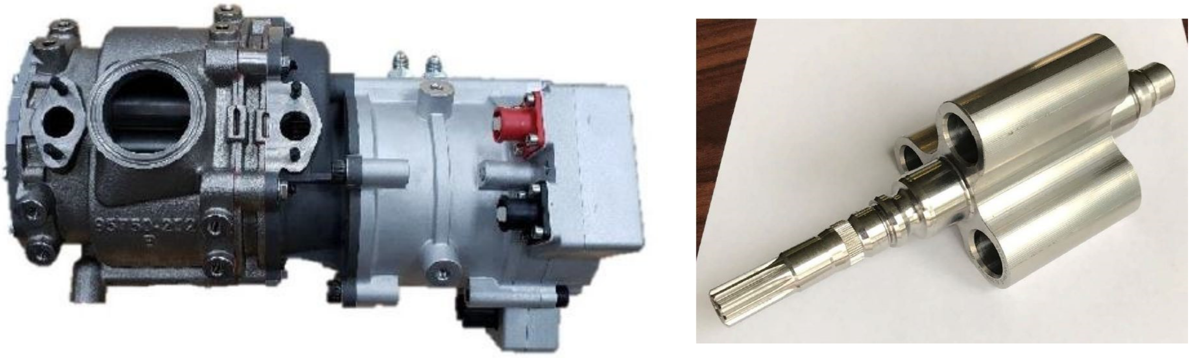


Figure 15: Dedicated EGR pump prototype with integrated 48V e-motor from Eaton

Main functionalities of the pump were tested as well as defining the operating map for a GT-power 1-D cycle study. Due to lack of time within this project, it was not possible to make a full investigation of the pump, but this will be carried out in a follow-up project using DME as a fuel. A dedicated control system is being developed, giving the choice of the pump speed control as function of estimated EGR, measured O2 in Intake Manifold or NOx

The Installation of the prototype EGR pump, figure 16, shows the minimum pipe length between the EGR pump and the engine in order to improve dynamic performances.

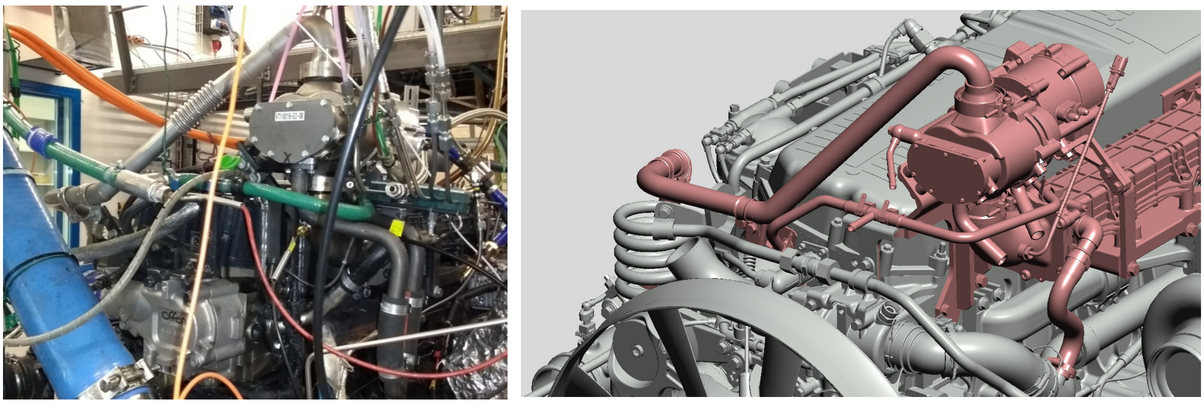


Figure 16: Installation of the new Eaton EGR pump

Turbo Charger

The original Turbocharger of the Cursor11 FEP2 Euro VI has been replaced by a Turbocharger with a smaller Turbine and equipped with ball bearing. This allows more EGR at low revolution / high boost pressure while avoiding the surging line of the compressor and to benefit from the higher compressor efficiency. Comparison of the compressor maps between the original and upgraded turbocharger is shown figure 17.

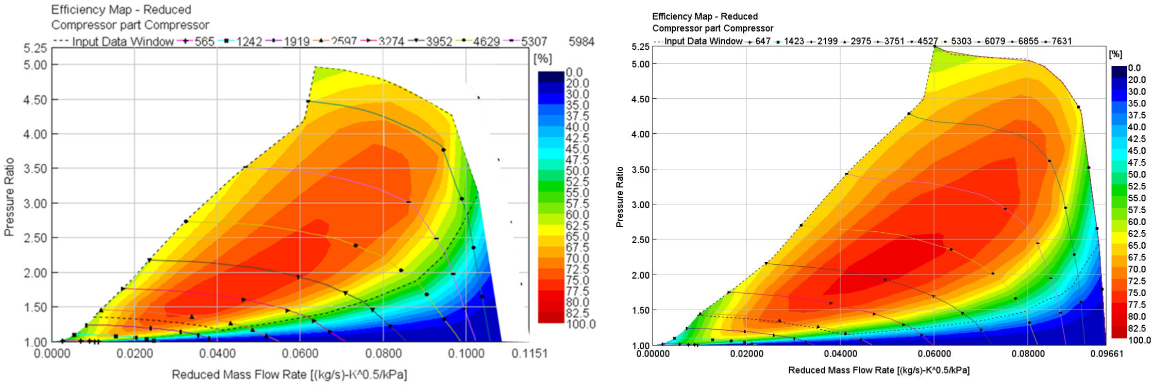


Figure 17: Compressor Maps, Baseline (left) vs Upgraded (right)

O2 sensor in Intake Manifold Sensor (IAO2)

In order to estimate accurately the EGR for the development of the control logic of the EGR pump, an O_2 sensor was position in the intake manifold, see figure 18. Such sensors are usually installed on the exhaust side, at the tail pipe to provide information about the air excess ratio (Lambda).

Two failures of the sensor after just 8 hours of operation were experienced. The possible cause was small water droplets forming in the intake pipe (low dew point temperature) interacting with the heated crystal leading to local thermal shock.

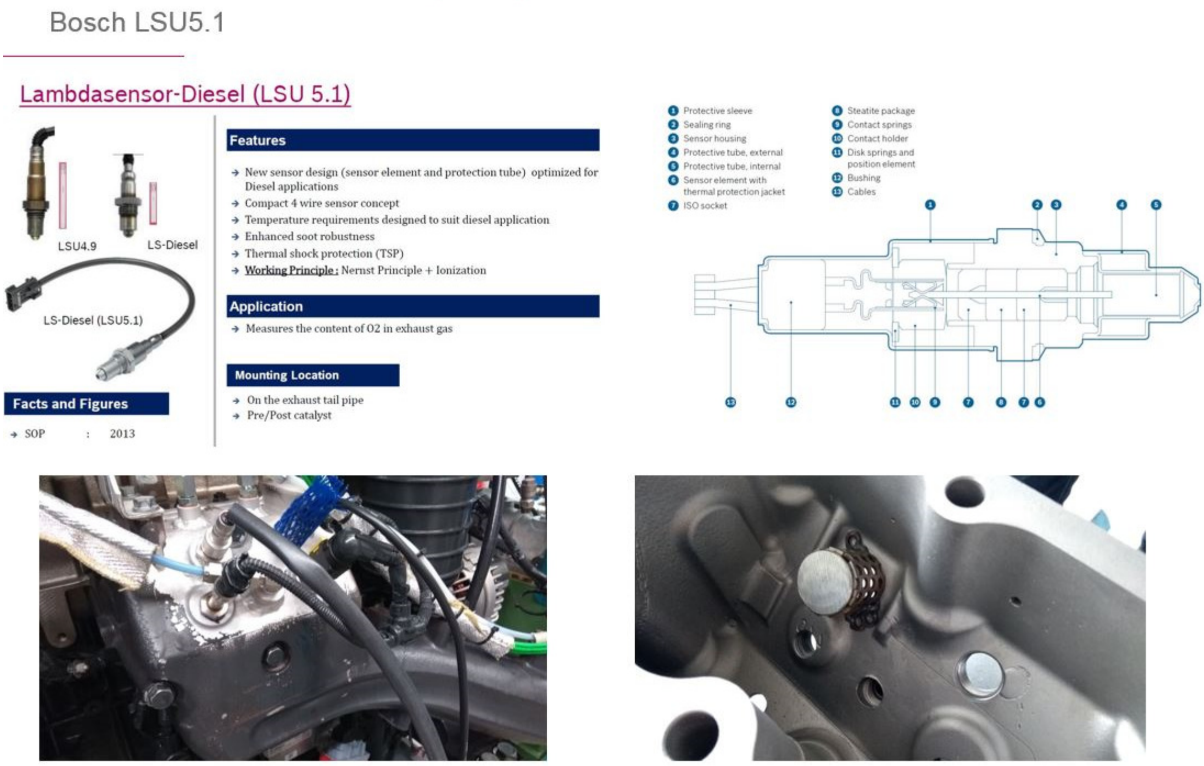


Figure 18: Lambda sensor LSU5.1 used in the Intake Manifold

CFD analyse of the Intake manifold and EGR pipe was carried out to find the optimum sensor position, see figure 19. In the CFD domain, small droplets were introduced near the EGR pipe inlet and tracked down to follow their path.

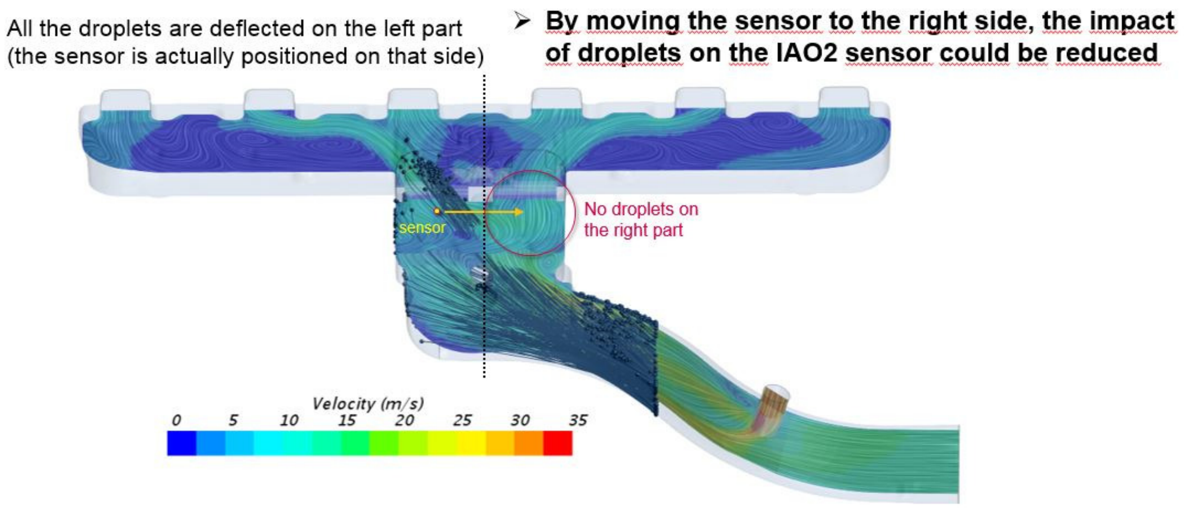


Figure 19: Lambda sensor LSU5.1 used in the Intake Manifold

After the repositioning of the sensor and adding a protection to avoid direct droplet impact, see figure 18 on bottom left, the O2 sensor has operated for about 100 hours without problems.

A testbed vs O2 sensor LSU5.1 measurement comparison of O2 molar concentration, see figure 20, shows an error in the order of 1.6% for about 500 measurement points. This validates the use of this O2 sensor for the EGR pump control logic in a follow up project with DME fuel.

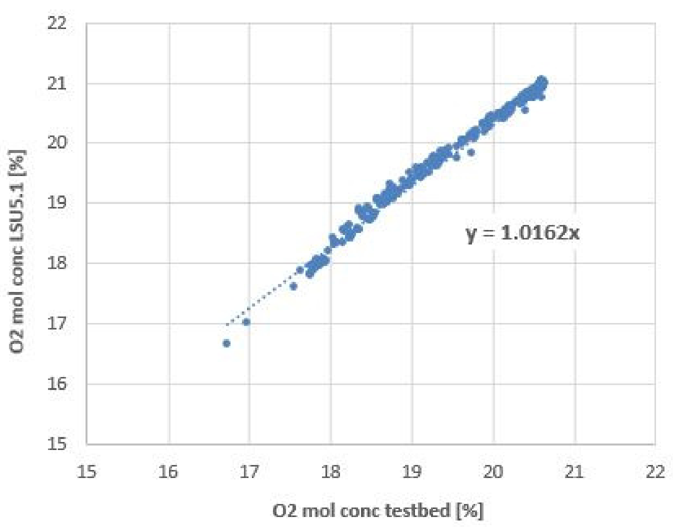


Figure 20: O2 molar concentration comparison between testbed and LSU5.1 measurement

3.4 Waste Heat Recovery related Hardware for Mechanical Efficiency improvement

A turbine is used to extract energy from the exhaust gas. This energy is transferred to the crankshaft through a direct mechanical connection or a motor generator unit. In order to enable this additional energy extraction from the hot exhaust gas, the back pressure (= exhaust pressure after the engine) needs to be increased.

In this study, two options were investigated: eTurbo and Power Turbine (or eTCD) as seen figure 21.

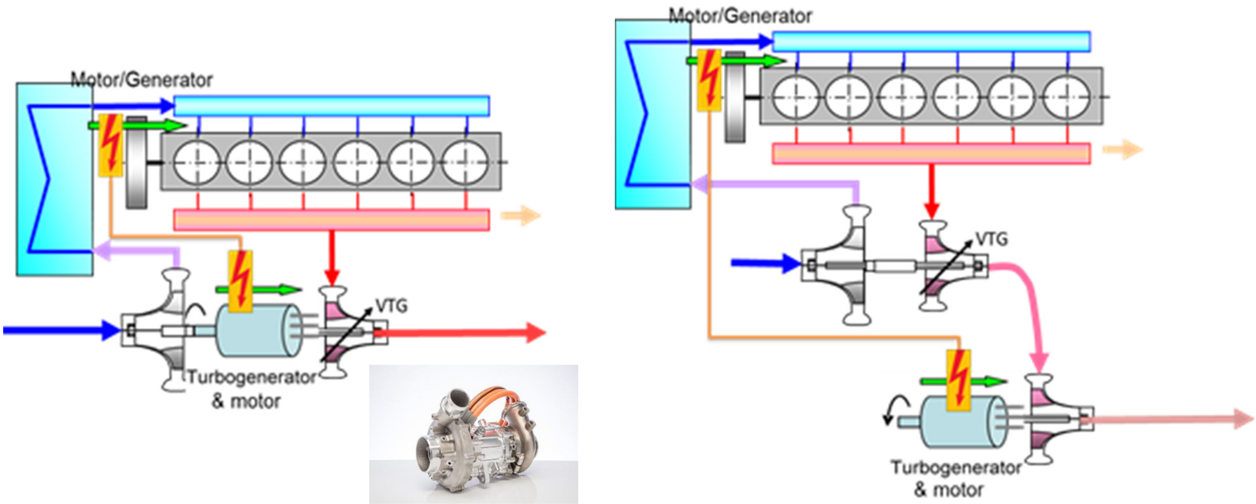


Figure 21: eTurbo (left) vs power turbine (right)

The eTurbo has an e-motor linking mechanically the compressor and the turbine with the advantages of being rather simple to install in terms of packaging and faster during transient operations. The eTurbo provides a very fast response to a Boost pressure demand as it can support the turbine with extra power which is quite beneficial for soot reduction.

The eTCD is a second turbine connected to an e-motor which is electrically linked to the crankshaft. It can be optimized to operate at higher efficiency since the speeds are different for both turbines. In theory, the eTCD could also be installed downstream the after-treatment system resulting in an increased ATS pressure and thus lower spatial velocity which would allow a more compact unit to be designed. Both solutions require a high voltage (> 400V) motor generator unit, coupled with an inverter and a battery to stabilize the load which significantly increase the cost of the engine unit.

The power developed by a turbine is largely a function of the turbine inlet temperature and the expansion ratio P_{out} / P_{in}. This ratio depends upon the flow capacity of the turbine (turbine width, inlet and outlet diameter)

$$\textbf{\textit{Turbine Power:}} \quad P_t = \eta_t \cdot \dot{m}_{exh} \cdot c_p \cdot T_{in} \cdot \left[1 - \left[\frac{P_{out}}{P_{in}}\right]^{\frac{\gamma-1}{\gamma}}\right]$$

Waste Heat Recovery by means of a turbine is a subtle balance between gaining energy by increasing back pressure and the loss due to a larger pumping power. This requires some optimisation to size the components.

The turbine flow capacity needs to be reduced to increase the extracted power, leading to a higher exhaust pressure after the engine. The higher exhaust pressure increases the exhaust gas temperature before the turbine but reduced it after. A larger request of waste energy recovery (or extracted power) leads to a worsening in pumping power due to the higher exhaust pressure. The benefit for the engine is the recovered exhaust gas energy at the crank shaft minus the additional pumping power as shown figure 22.

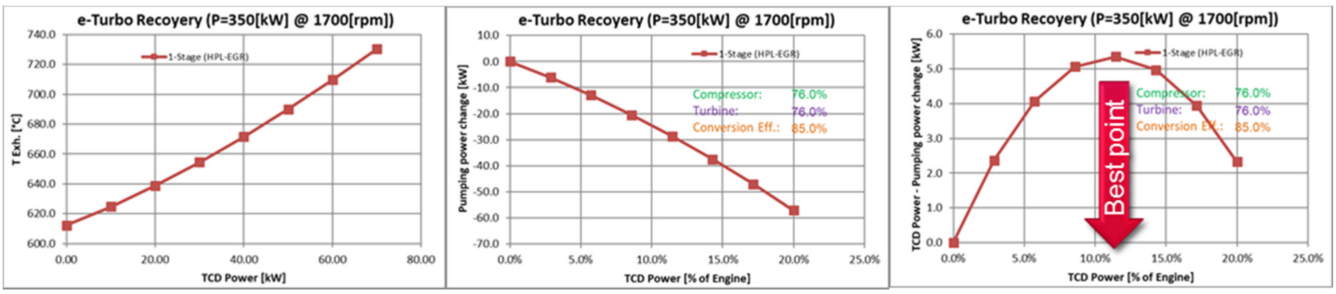


Figure 22: eTurbo power recovery and effect on pumping loss

The electrical conversion efficiency (Inverter, generator, motor) is about 85%. For a mechanically linked turbo compound, the loss would be similar because of the high gear ratio involved between the turbine and the crank shaft.

An optimisation of the turbine size (scaling factor TSFM) and the compressor wheel diameter (DWC, u/c ratio) has been carried out to get an indication of the general trend and gradients.

The optimisation diagrams, figure 23, show that the turbine flow capacity needs to be scaled in between 75% and 105% of the current size Turbocharger. In each diagram, the red dot is showing the minimum BSFC. In combination with the VGT position used for the optimisation it makes sense to apply a scaling factor of 80% (TSFM) to the base turbine for best overall performance.

The optimum compressor diameter is in between 84mm and 92mm. This is in combination with the 80mm turbine.

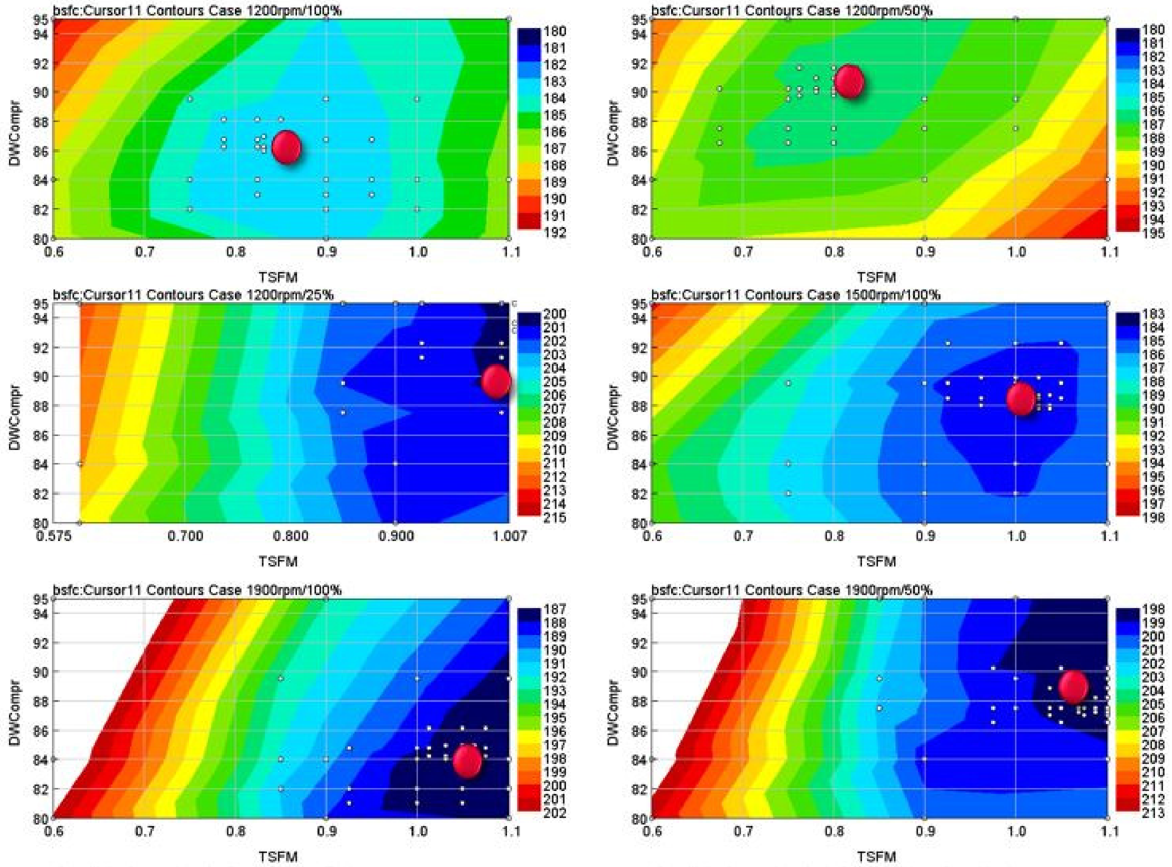


Figure 23: Turbocharger size optimisation diagrams

Waste Heat Recovery technologies based on Organic Rankine Cycle were not reviewed within this project.

4 Results and discussion

4.1 Improvement of the BTE / NOx Trade-Off with Higher CR

Theoretical Thermal Efficiency in function of the Compression Ratio based on a Seiliger cycle is shown figure 24. From a CR of 16.5 to 20.5, the thermal efficiency increases by 3.5%pt which translates into a theoretical 6% fuel consumption improvement.

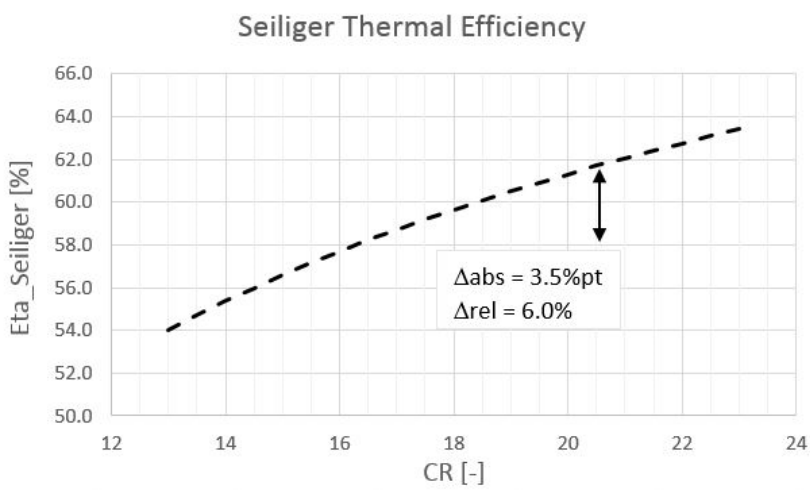


Figure 24: Thermal Efficiency as function of the Compression Ratio

To evaluate the actual potential in thermal efficiency, measurements were performed on a Single Cylinder Engine equipped with a Cursor11 combustion chamber (H-bowl) for two compression ratios: 16.5 and 20.5. A trade-off of SOI (mfb50) and of EGR at constant rail pressure were carried out.

Cylinder pressure traces, figure 25 shows the effect of increased CR for a given high load point at 1170rpm.

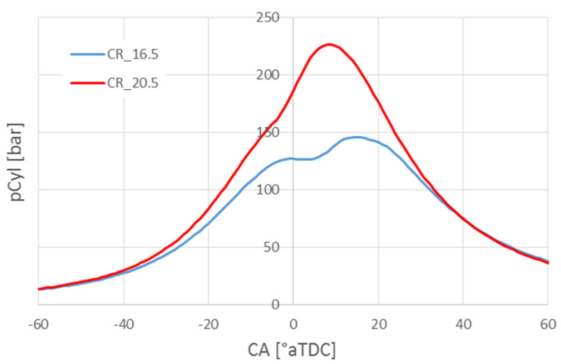


Figure 25: Cylinder Pressure for two compression ratio

Single Cylinder at 1170 rpm x 22.7 bar, representing a high load with no EGR has undergone a trade-off of mfb50 with constant rail pressure by varying SOI, see figure 26. Gross indicated thermal efficiency improves by 1%pt while the indicated specific NOx was reduced by about 2g/kWh for the same mfb50. Cylinder pressure increased by about 32-35 bars.

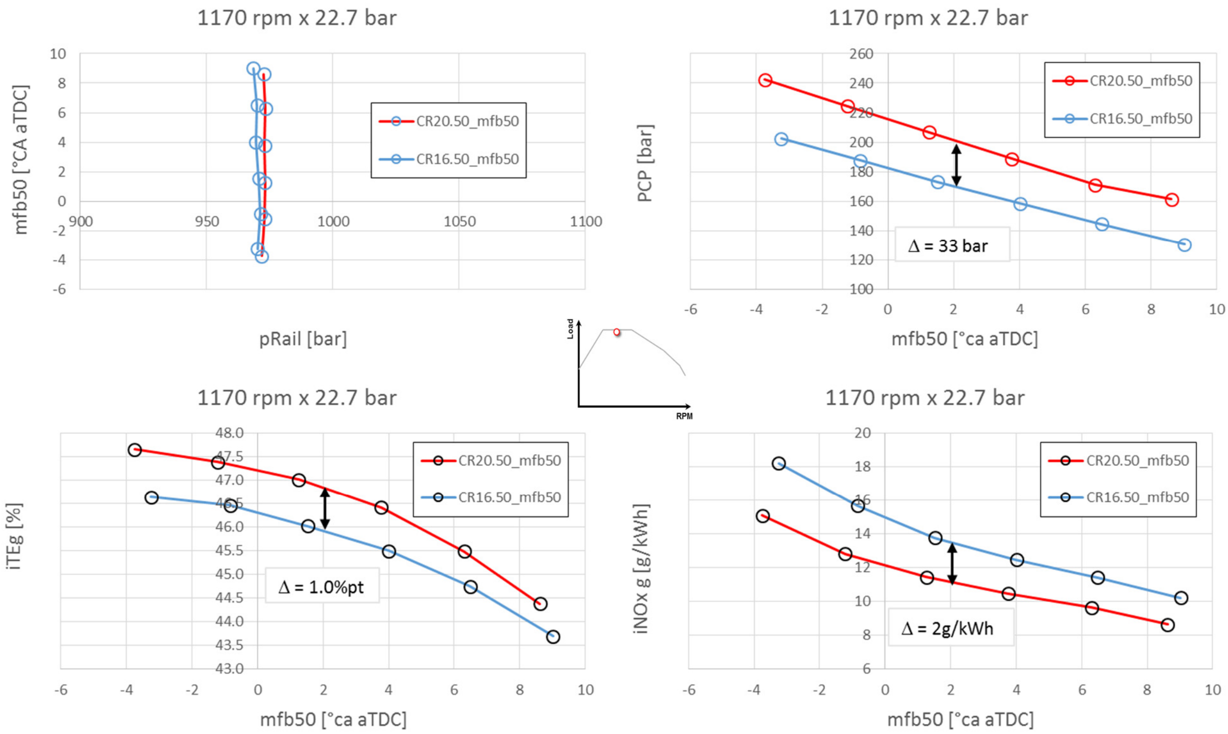


Figure 26: Effect on Thermal efficiency and NOx with a mfb50 Trade-off

For a Single Cylinder at 1170 rpm x 22.7 bar, a trade-off of EGR with constant rail pressure and mfb50 shows an improvement in thermal efficiency of 1.4%pt, figure 27. In contrast, for EGR above 10%, there are no difference in terms of NOx between the two compression ratios. This will be later investigated using 3D CFD simulations.

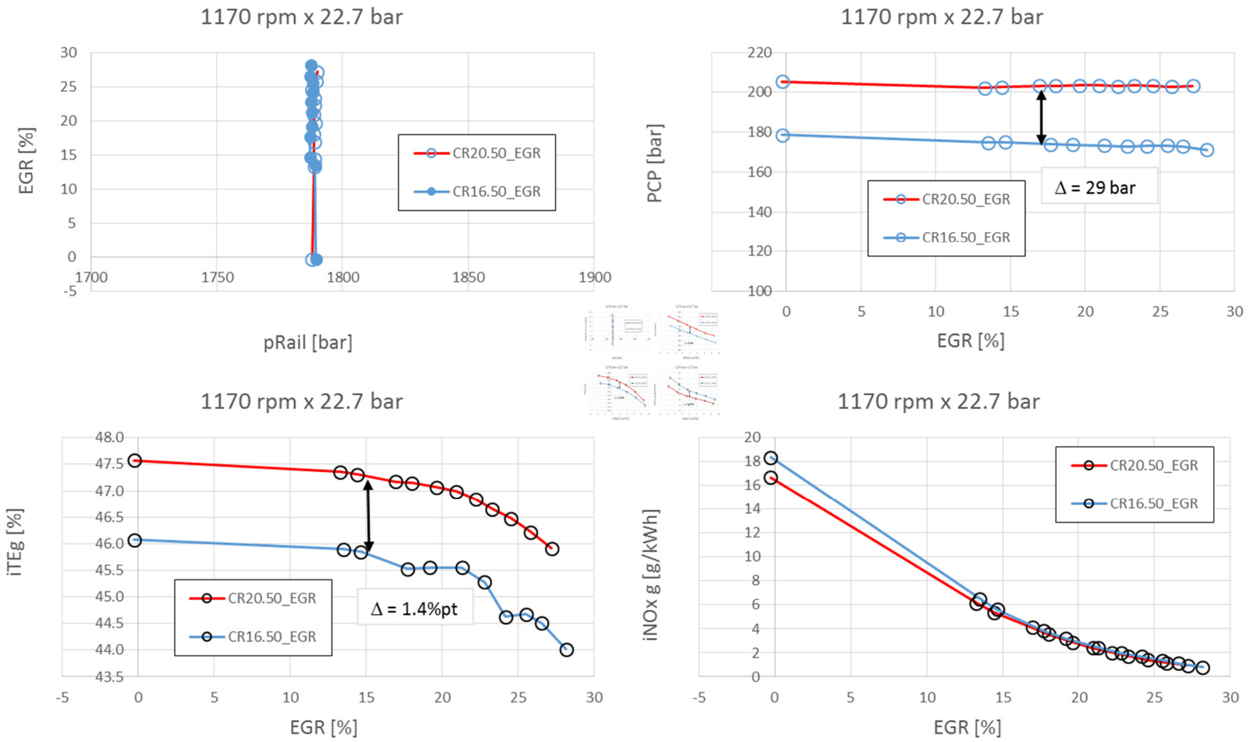


Figure 27: Effect on Thermal efficiency and NOx with a EGR Trade-off

Single Cylinder gross indicated thermal efficiency versus indicated specific NOx for four load points with mfb50 and EGR variation is shown figure 28. At constant NOx, the gain in thermal efficiency is comprised between 1.2%pt and 1.7%pt depending upon the load.

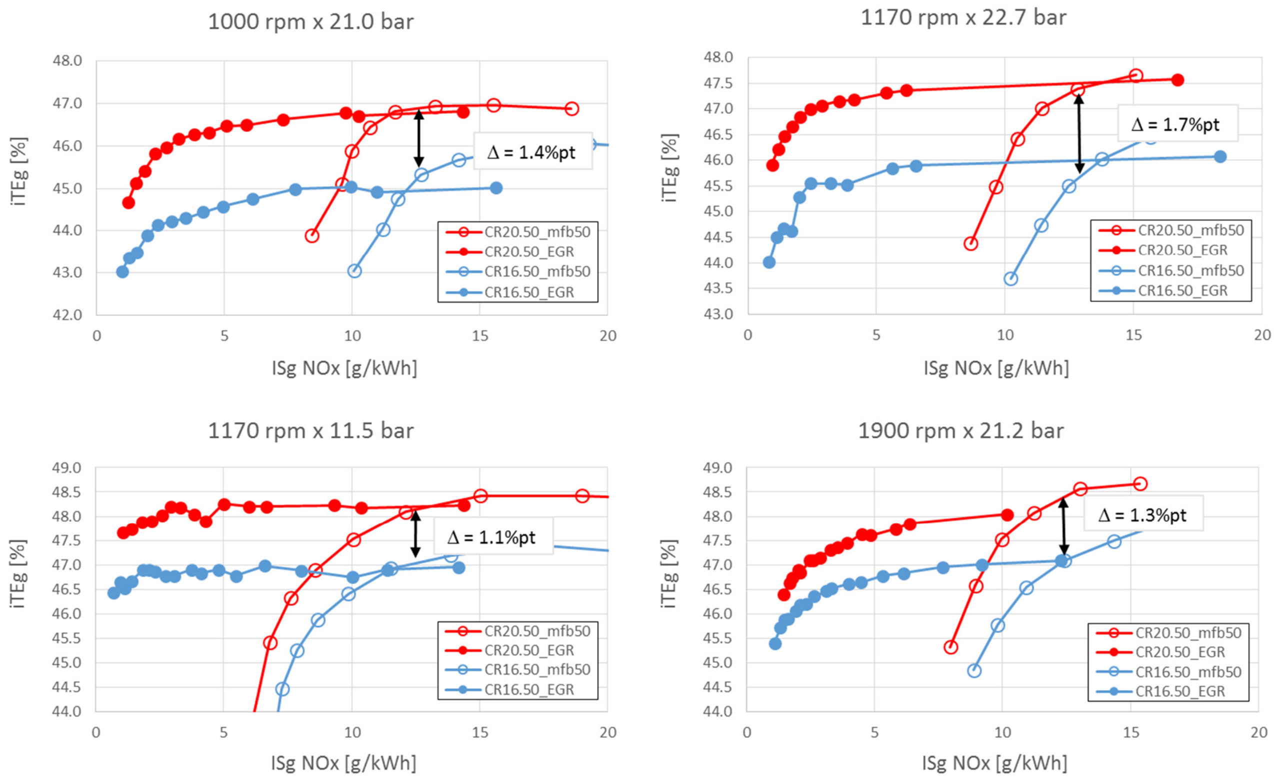


Figure 28: mfb50 and EGR influence on Thermal efficiency and NOx of four load points

The real gain measured on the thermal efficiency is about 1/3 of the theoretical gain which can be explained by increased wall heat loss with the higher compression ratio.

4.2 Open Combustion Chamber development

An alternative combustion chamber with lower swirl and increased free flame path was developed in order to reduce the wall heat loss which is challenging when combustion efficiency improves.

For the investigation of gain in efficiencies with the open chamber in comparison with the H-bowl, two distinct load points were chosen representing load collective and fuel consumption respectively as seen figure 29.

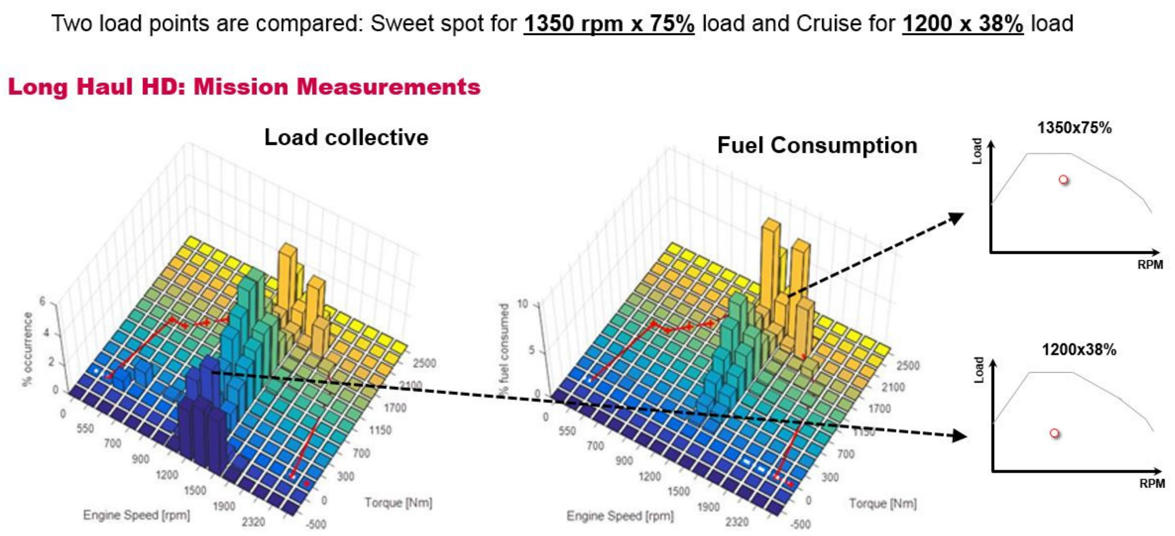


Figure 29: Long Haul Mission in Engine Map

At 1350x75%, representing an important fuel consumption load, the gain with the Open Chamber in terms of fuel consumption is about 1% (or 0.5%pt BTE improvement) at a constant BS NOx = 12 g/kWh. No penalties on Soot were observed, see figure 30.

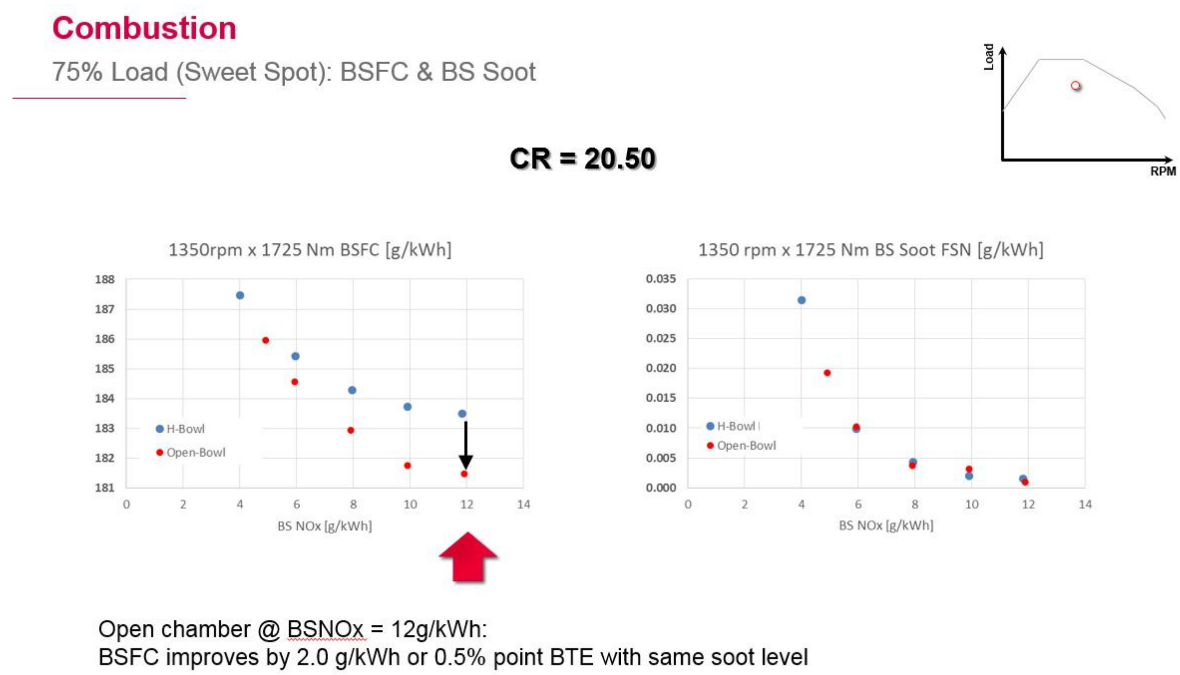
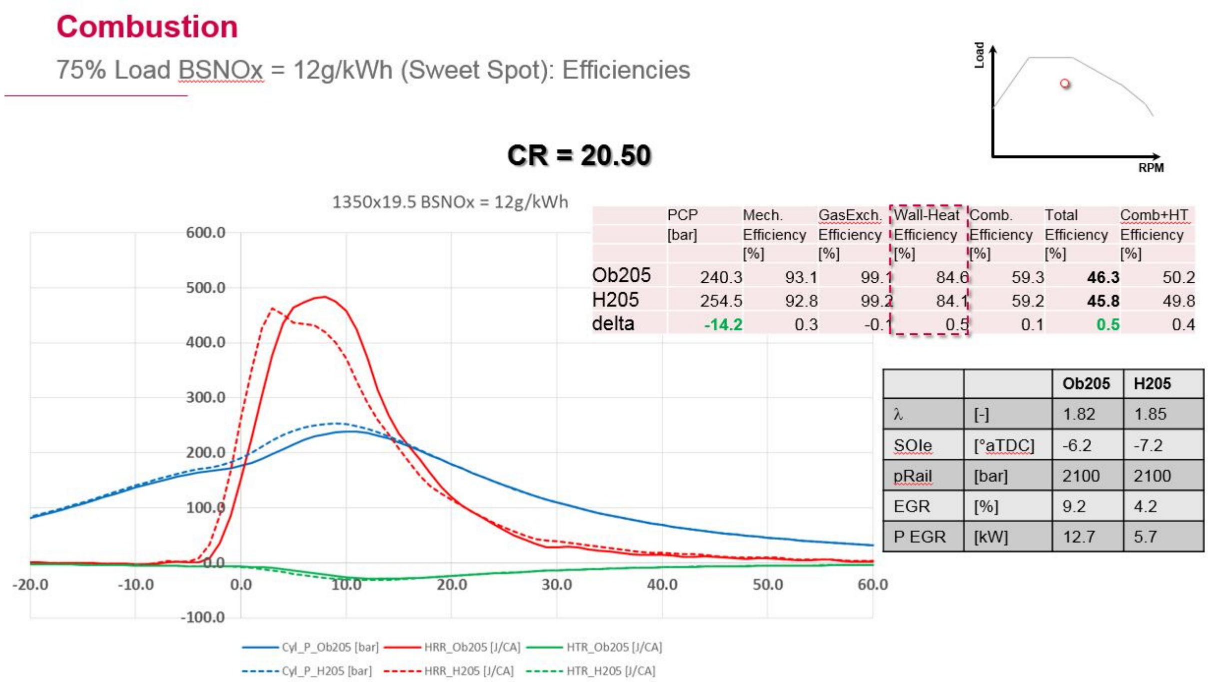


Figure 30: Bowl shape effect on BSFC and Soot for 1350rpmx1725Nm

The following Heat Release Rate and efficiencies investigation for the same load point shows that the gains with the Open chamber (Ob205) come from the wall heat loss which increases by 0.5%pt, see figure 31.



	PCP [bar]	Mech. Efficiency [%]	GasExch. Efficiency [%]	Wall-Heat Efficiency [%]	Comb. Efficiency [%]	Total Efficiency [%]	Comb+HT Efficiency [%]
Ob205	240.3	93.1	99.1	84.6	59.3	**46.3**	50.2
H205	254.5	92.8	99.2	84.1	59.2	**45.8**	49.8
delta	-14.2	0.3	-0.1	0.5	0.1	0.5	0.4

		Ob205	H205
λ	[-]	1.82	1.85
SOIe	[°aTDC]	-6.2	-7.2
pRail	[bar]	2100	2100
EGR	[%]	9.2	4.2
P EGR	[kW]	12.7	5.7

Figure 31: Heat Release Analyses of the two bowls at 1350rpmx1725Nm

The 1200x38% load represents an important collective mission point and the gain in fuel consumption with the Open Chamber is about 1.4% (or 0.7%pt BTE improvement) for a constant BS NOx = 3g/kWh but with a 40% increase in Soot, see figure 32.

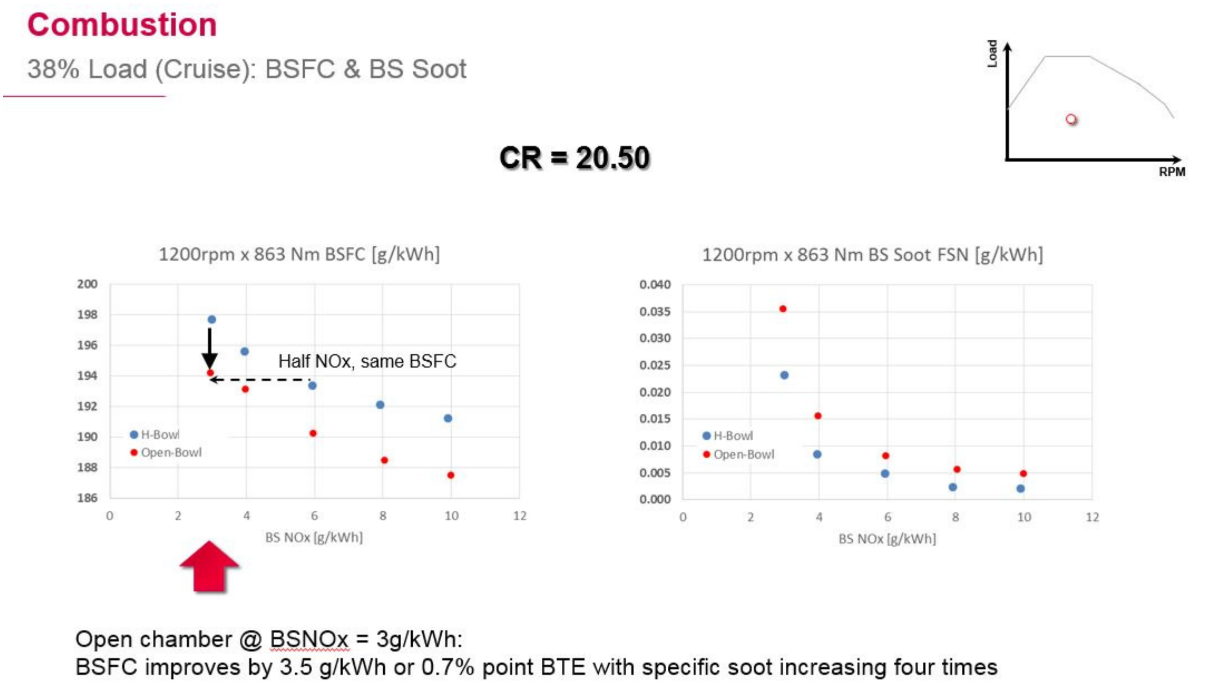
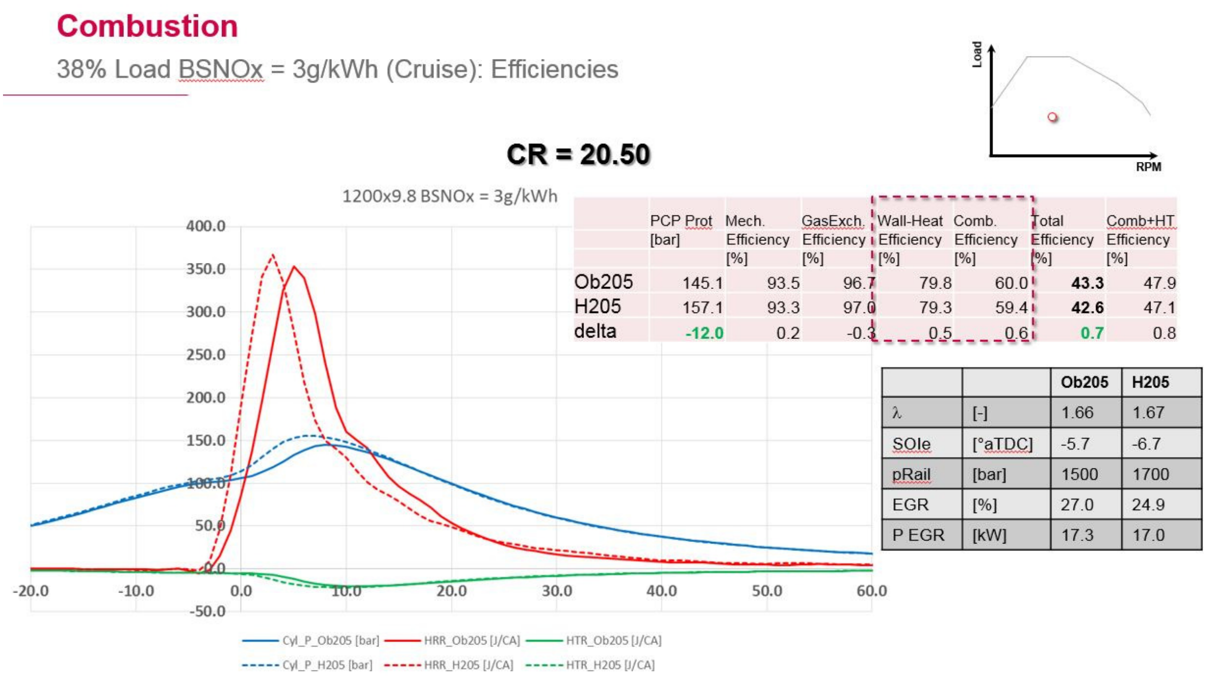


Figure 32: Bowl shape effect on BSFC and Soot for 1200rpmx863Nm

The following Heat Release Rate and efficiencies investigation shows that the gain comes mainly from the combustion and wall heat loss efficiencies which improves both by about 0.5%pt, see figure 33.



	PCP Prot [bar]	Mech. Efficiency [%]	GasExch. Efficiency [%]	Wall-Heat Efficiency [%]	Comb. Efficiency [%]	Total Efficiency [%]	Comb+HT Efficiency [%]
Ob205	145.1	93.5	96.7	79.8	60.0	**43.3**	47.9
H205	157.1	93.3	97.0	79.3	59.4	**42.6**	47.1
delta	-12.0	0.2	-0.3	0.5	0.6	0.7	0.8

		Ob205	H205
λ	[-]	1.66	1.67
SOIe	[°aTDC]	-5.7	-6.7
pRail	[bar]	1500	1700
EGR	[%]	27.0	24.9
P EGR	[kW]	17.3	17.0

Figure 33: Heat Release Analyses of the two bowls at 1200rpmx863Nm

The gain in terms of combustion efficiency with the open chamber, in particular at part load is attributed to an increased free flame length, reducing the interaction with the piston. Lower swirl and reduced heat flux with surrounding walls, improve the wall heat loss efficiency of the engine equipped with the open chamber.

4.3 Use of Post injection to improve Soot and Fuel Consumption

The use of a post injection just after the Main to improve the Soot emissions has been reported by R. Dreezen from TU Eindhoven. In figure 34, the soot at tail pipe as function of post injection proportion and dwell time is shown.

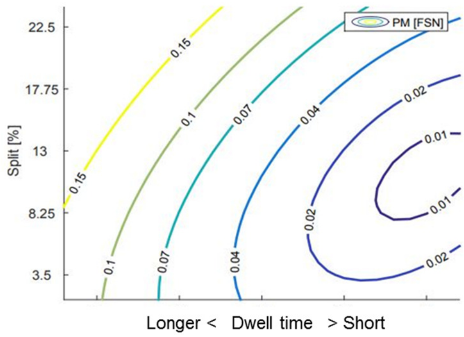


Figure 34: Effect of Post injection on the Soot

After upgrading the ECU injection control software, the potential benefits of post injection were investigated at full load with the Open Chamber for three different speed: 1900, 1500 and 1200 rpm, see figure 35. Soot was reduced by about 60% with the enabled post injection corresponding to a minimum dwell time and about 12% injected quantity in the post injection. Moreover, a fuel consumption reduction of 0.7% was observed for the rated power load point.

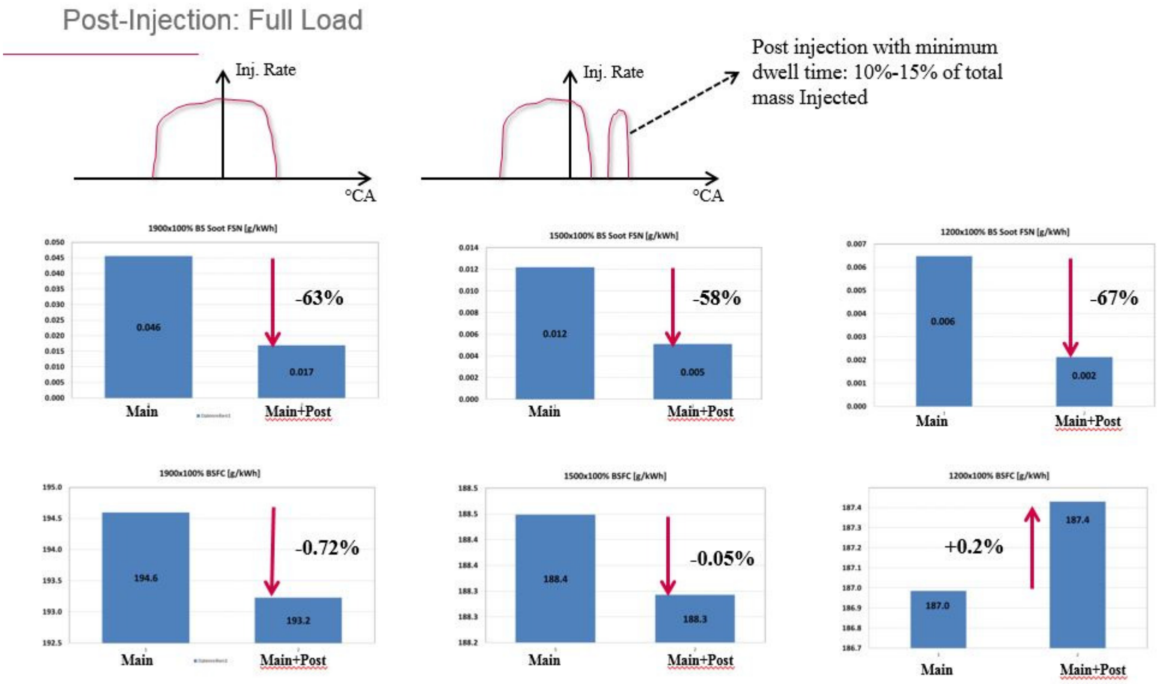


Figure 35: Effect of post injection at high load for 1900 rpm (left), 1500 rpm and 1200 rpm (right)

The post injection increases the in-cylinder turbulence for a short time which improves mixture with air and pollutants such as CO or Soot. As this additional injection occurs during the expansion stroke, the additional mixing effect is lasting longer.

4.4 BTE as function of BS NOx engine out

The use of an EGR pump allows studying the effect of EGR on the Brake Thermal Efficiency as function of the raw NOx (engine out). Below is an investigation where each load point is optimised in terms of boost pressure and injection setting (pRail and SOI). Four engine speeds and four loads ranging from 25% to 100% were measured as seen in figure 36. We can see that between BS NOx 4 g/kWh and 12 g/kWh, the BTE increased by 0.2-0.25%pt per unit of specific NOx [g/kWh].

An increase of BS NOx engine out by 4 to 5 g/kWh should lead to an improvement of BTE by about 1%pt.

In general BS NOx at engine out in the range of 10 to 12g/kWh are considered as an upper limit for the optimal operation of a SCR system.

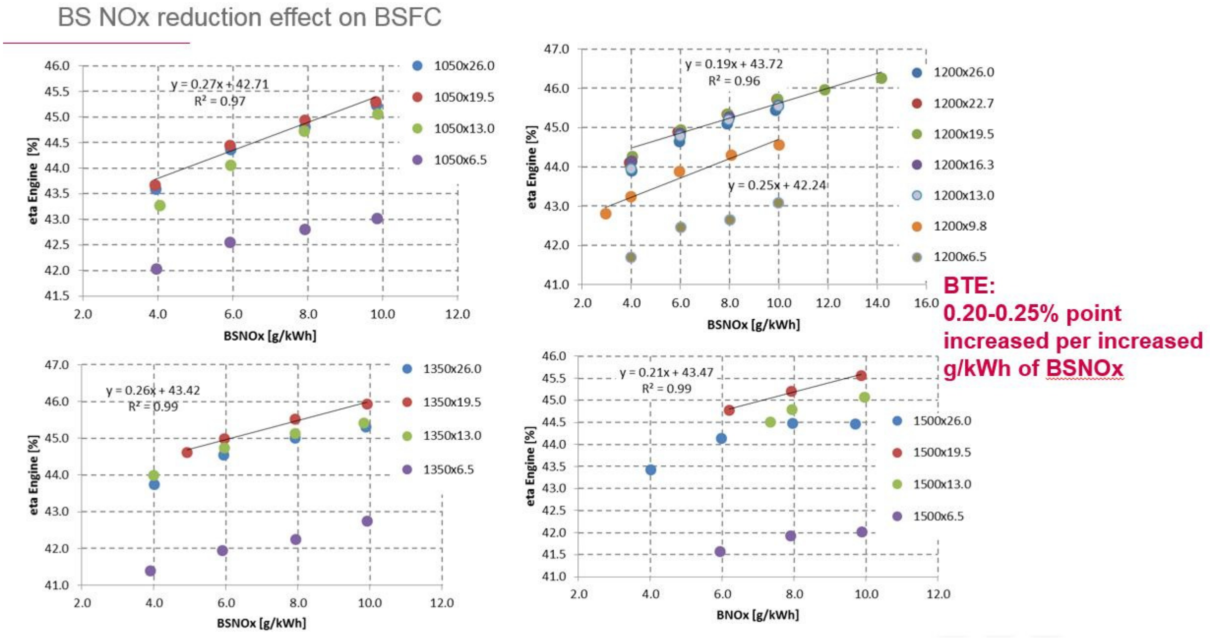


Figure 36: BTE as function of raw BS NOx for four different engine speeds

Looking at the total cost of ownership (diesel-fuel + AdBlue) if the cost of AdBlue is taken into account: we can see that as the cost ratio of diesel-fuel to Adblue is close to 1 (US Scenario), the engine NOx out shall be in the range of 4 g/kWh in order to offset the Adblue consumption required to reduce the NOx to meet emission limit, see Figure 37.

For the EU Scenario, with a cost ratio of diesel-fuel to Adblue close to 3, the engine shall be tuned for a BS NOx engine out = 6 to 8 g/kWh.

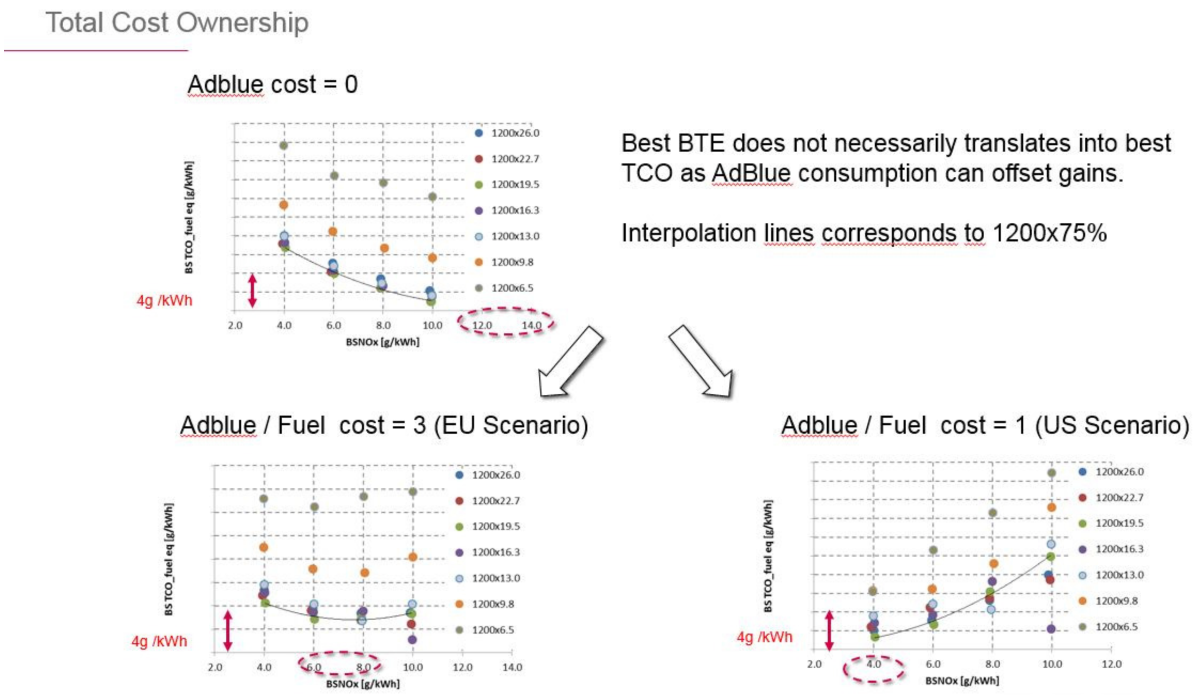


Figure 37: Effect of raw BS NOx on Total Cost Ownership

4.5 Improvement of the BTE on a MCE (6-cylinders) without WHR

The final result obtained for the Brake Thermal Efficiency is 47.2% by improving:

a) Combustion efficiency with higher compression ratio, open combustion chamber and enhanced injectors.

b) Mechanical efficiency with low friction package

c) Gas Exchange efficiency for higher EGR with upgraded T/C and EGR pump

Load point at 1200x75% representing one of the sweet spots in the engine map with a specific NOx limited to 10 g/kWh raw emissions after engine.

Approach for 1200x75% load single stage T/C Eng. Out: BS NOx = 10g/kWh	Mechanical Efficiency [%]	Gas Exchange Efficiency [%]	Combustion Efficiency [%]	Wall Heat Loss Efficiency [%]	Brake Thermal Efficiency [%]	Δ%
EURO VI no EGR	93.1 %	99.0 %	55.7 %	84.9 %	**43.6 %**	8.3%
Opt.L-EGR without WHR	95.0 %	99.0 %	59.3 %	84.5 %	**47.2 %**	
Delta (Opt.L-EGR – EURO VI)	**+1.9% point**	**+0.0% point**	**+3.6% point**	**-0.4% point**	**+3.6% point**	

Without the use of Waste Heat Recovery (e-Turbo, Rankine Cycle, …), the BTE increased from the engine baseline is 3.6%pt, translating into a 8.3% improvement in fuel consumption or equivalent CO2 reduction TtW. The main contribution to the improvement comes from the combustion efficiency and the mechanical efficiency. Despite the improved Turbocharger, gas exchange efficiency remains stable because more EGR is used to counterbalance increased injection pressure to optimise air-fuel mixture. Wall Heat loss Efficiency is expected to be reduced as higher compression ratio leads to higher in-cylinder gas temperature.

To optimise the mechanical efficiency: the re-dimensioning of Crankshaft and piston pins, increasing of conrod length, smart switchable water and oil pump, … requires a lot of development on the engine base itself which was carried out at FPT, Arbon. A Mechanical Efficiency of 95% is in line with Cummins 94.7% target within the SuperTruck project.

The contribution of the EGR pump alone to the BTE can be estimated to 1% fuel consumption improvement or 0.5%pt BTE.

4.6 Improvement of the BTE on a MCE with WHR

The waste heat recovery benefits in terms of fuel consumption with an eTurbo is similar to the option with High pressure VGT T/C plus a separate e-power turbine (eTCD), see comparison in figure 38. With the later arrangement, we have the possibility to position the eTCD downstream the After Treatment System which leads to a higher pressure in the ATS. This allows to reduce its size due to a reduced space velocity.

The gain in terms of fuel consumption is in the order of 1.0% to 1.2 % fuel consumption or 0.6%pt BTE for 1200 rpm and 75% load.

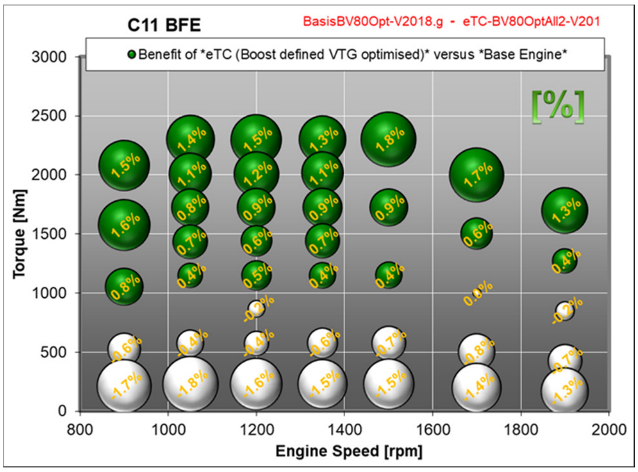


Figure 38: Fuel consumption benefits for eTurbo (left) and eTCD (right)

Despite a detailed investigation to have the right size for the eTurbo (eTC), the gain in terms of fuel consumption is not significant because the exhaust gas temperature is rather low due to the rather highly efficient engine itself. At 1200x75% load, the power delivered by the eTurbo's turbine to the crankshaft is about 8 kW but the added pumping loss is around 5 kW, meaning that the net gain is 3 kW or about 1.4% of the engine power (217 kW).

Figure 39 shows the BSFC in the engine map for the C11 with the low friction package FEP3 and the eTurbo. Overall gains in terms of BSFC with the eTurbo are on the full load curve which means that an engine equipped with it could be downsized by 10% in terms of volume displacement leading to higher exhaust gas temperatures. This could be considered for a future optimisation.

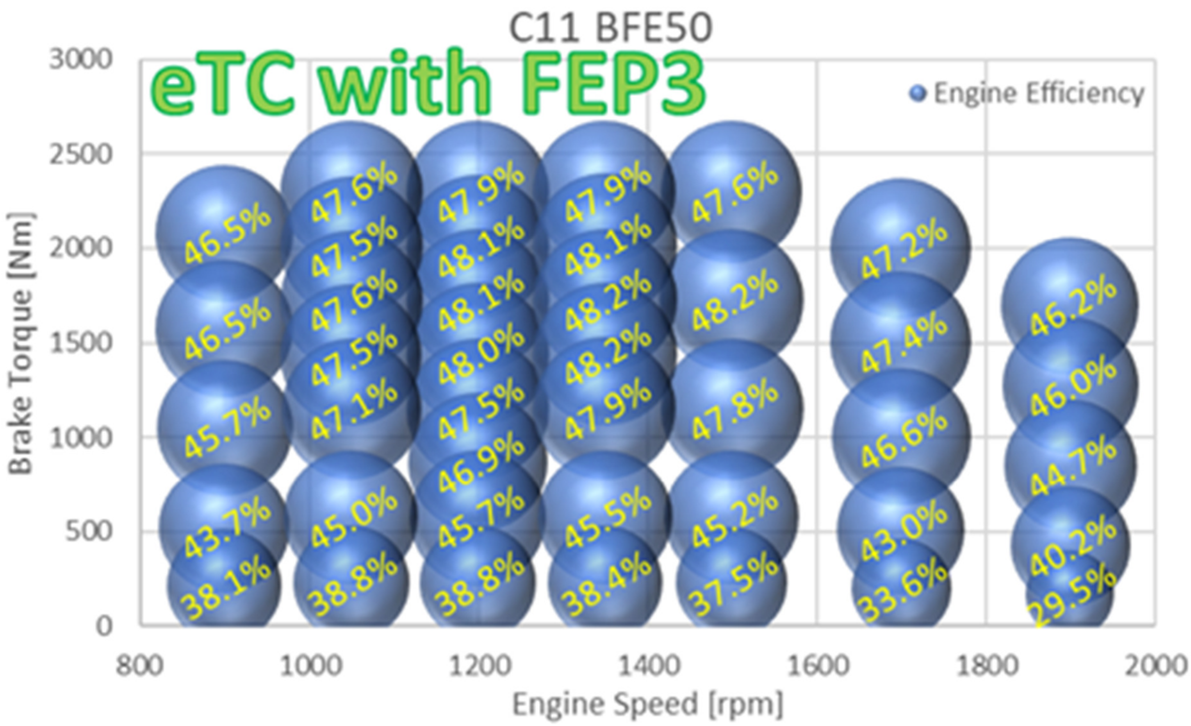


Figure 39: C11 Empa FEP3 estimated BSFC in the engine map with eTurbo

Interesting to note is the power delivered by the EGR pump to the crankshaft thanks to the eTurbo increasing the back pressure. Resulting ∆P engine (P Boost – P Exhaust) leads to the EGR pump working in turbine mode (pressure ratio < 1), see figure 40.

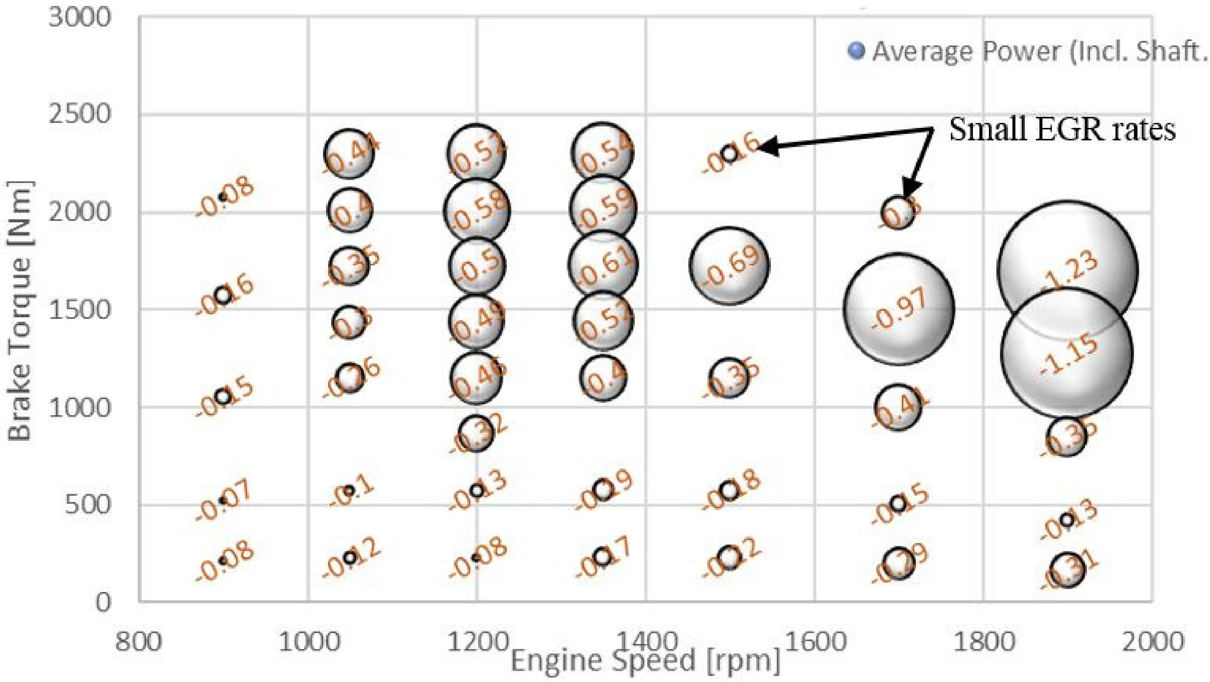


Figure 40: Eaton EGR pump power delivered to the crankshaft in the engine map for eTurbo

The final result obtained for the Brake Thermal Efficiency with the eTurbo is 47.8%. If the benefit in terms of BTE is limited, the eTurbo improves the transient response, especially for soot, which could potentially lead to some further optimisation of the injector nozzle diameter (increase hydraulic flow) and therefore combustion efficiency. eTurbo is only an option if the engine itself is ready for high voltage to allow some hybridisation but a proper volume displacement for higher BMEP could be considered to maximise the benefits of the Waste Heat Recovery by the eTurbo.

Approach for 1200x75% load single stage T/C Eng. Out: BS NOx = 10g/kWh	Mechanical Efficiency [%]	Gas Exchange Efficiency [%]	Combustion Efficiency [%]	Wall Heat Loss Efficiency [%]	Brake Thermal Efficiency [%]
EURO VI no EGR	93.1 %	99.0 %	55.7 %	84.9 %	**43.6 %**
Opt.L-EGR without WHR	95.0 %	99.0 %	59.3 %	84.5 %	**47.2 %**
Delta (Opt.L-EGR – EURO VI)	**+1.9% point**	**+0.0% point**	**+3.6% point**	**-0.4% point**	**+3.6% point**
Vision w/ eTurbo	98.3 %	97.0 %	59.3 %	84.5 %	**47.8 %**
Delta (eTurbo- Opt.L-EGR)	**+3.3% point**	**-2.0% point**	**+0.0% point**	**0.0% point**	**+0.6% point**

∆%

8.3%

9.6%

4.7 Improvement of the BTE on a MCE with WHR and improved wall Heat loss efficiency.

The last frontier for improvement lies into the reduction of the wall heat loss by using in-Cylinder thermal coating barrier.

For example, thermos-swing wall insulation technology reduces heat loss from the combustion chamber as it was introduced by Toyota for light duty diesel engines but with much lower load and durability requirements compared to the heavy duty applications discussed here.

By reducing the thermal conductivity (λ) of the insulation layer formed on the wall surface and the volumetric specific heat capacity (ρ.C), the surface temperature of the insulation coat can quickly vary during a short cycle time. This leads to a high wall surface temperature (hence lower wall heat losses) during combustion and a low wall surface temperature (hence good volumetric efficiency) during gas exchange.

Unfortunately, no breakthrough has yet been reported with such thermal swing material and, in general, measured fuel consumption improvement is limited to 0.5% or less than 0.25%pt BTE. Such result with six coating variants was recently confirmed by Volvo cars into a SAE paper: 2019-24-0062

Nevertheless, heat loss reduction from the combustion chamber leads to the following improvements:

- Increased piston work
- Higher exhaust gas temperature providing more enthalpy for the turbine, especially eTurbo
- Reduction of the cooling heat rejection leading to smaller radiators

In the following table, based on the C11 Empa with a low friction package (FEP3) and an eTurbo it was estimated how much the Wall Heat loss efficiency needs to be improved in order that BTE reaches 50%.

Simulations have been carried out with TBC at valves, top liner regime and the piston, see figure 41.

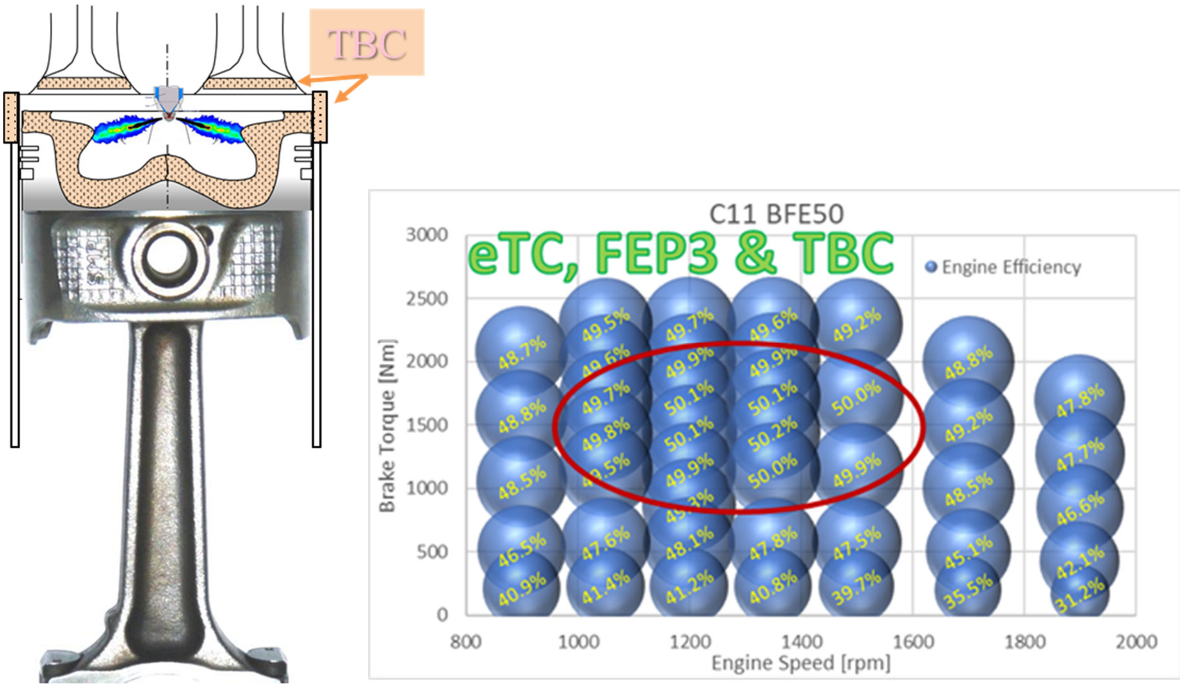


Figure 41: C11 Empa FEP3 with TBC predicted BSFC

The wall Heat loss efficiency needs to be improved by 2.3%pt from 84.5% to 86.8%, but this could be quite challenging and more research should be dedicated to this task.

Approach for 1200x75% load single stage T/C Eng. Out: BS NOx = 10g/kWh	Mechanical Efficiency [%]	Gas Exchange Efficiency [%]	Combustion Efficiency [%]	Wall Heat Loss Efficiency [%]	Brake Thermal Efficiency [%]
EURO VI no EGR	93.1 %	99.0 %	55.7 %	84.9 %	**43.6 %**
Opt.L-EGR without WHR	95.0 %	99.0 %	59.3 %	84.5 %	**47.2 %**
Delta (Opt.L-EGR - EURO VI)	**+1.9% point**	**+0.0% point**	**+3.6% point**	**-0.4% point**	**+3.6% point**
Vision w/ eTurbo	98.3 %	97.0 %	59.3 %	84.5 %	**47.8 %**
Delta (eTurbo - Opt.L-EGR)	**+3.3% point**	**-2.0% point**	**+0.0% point**	**0.0% point**	**+0.6% point**
Vision w/ eTurbo & TBC	99.0 %	97.0 %	60.0 %	86.8 %	**50.0 %**
Delta (eTurbo+TBC - eTurbo)	**+0.7% point**	**+0.0% point**	**+0.7% point**	**2.3% point**	**+2.2% point**

Δ% in fuel consumption

8.3%

9.6%

14.7%

5 Conclusions

The maximum BTE achieved without the use of any WHR technologies is 47.2% @ BS NOx engine out = 10 g/kWh, representing a + 8.3% improvement in fuel consumption from the Euro VI engine baseline.

With further optimisation of mechanical and gas exchange efficiencies and calibrating the BS NOx engine out to 12 g/kWh, a BTE = 48% shall be achievable.

An increased compression ratio, enhanced injection system, low friction package and improved turbocharger performances are required.

The use of an EGR Pump and the choice of piston bowl shape depends upon the strategy chosen to reach the future EU7 ultra low NOx requirements at tail pipe.

The EGR pump allows to directly couple the engine with the ATS system, in particular for the NOx control, and permits the use of a smaller turbine without affecting much the gas exchange efficiency at high load while giving more room for the compressor surging. On the other hand, the EGR pump requires a 48V electrical system to cope with a power demand of about 2kW.

Regarding the combustion chamber, the open bowl brings benefits in terms of BTE by reducing the wall heat loss due to the lower swirl and the longer free flame path. On the other hand, it requires higher EGR level for a same level of BS NOx engine out as the H-bowl. The increase in heat rejection would affect the fuel consumption of the whole vehicle (less aerodynamic with more radiator areas and fan switching increase). A compromise would have to be found based on a system engineering approach.

In Europe, in combination with a highly efficient SCR system reaching a high NOx conversion rate, an open combustion chamber with high compression ratio should be more suitable.

The BTE improvement with an eTurbo is limited to about 0.6%pt when apply on an engine already well optimised because the exhaust gas temperatures are rather low, limiting the turbine power.

Waste Heat Recovery technologies would require High Voltage electrification which substantially increases the cost of the powertrain. eTurbo is rather expensive therefore it is difficult to justify its adoption if initial investment payback from the gain in fuel economy is limited to two years.

To reach the target of a BTE 50%, wall Heat loss efficiency has to be improved by about 2% to 3% using Thermal Boundary Coating for the in-Cylinder walls. This is very challenging and further R&D is required on this topic.

On the other hand, the use of fuels made of entirely or partly of renewable sources, eFuel or Bio-mass base, would allow a smooth transition towards total decarbonisation without large infrastructure investment while keeping current powertrain which have proved their reliability these last 20 years. Moreover, current engines are made by up to 90% from recycle materials avoiding further mining in sensitive areas.

Following table summarize the technologies that have been reviewed during the project and indicates potential for BTE improvement along estimated cost and Payback time.

In Appendice, a simulation of CO2 saved (baseline = 2019) per year for a fleet of 20000 long haul 40T vehicles for three different hypothetical scenarios based on the fuel improvement calculated in this work is described.

Hardware	Feature	Potential	ΔBTE [%pt]	Payback [y]$_{140k\ km/y}$
Head	Peak Cylinder Pressure up to 250 bars. CGI material used in foundry.	Higher Combustion Efficiency. (Carnot)	1.2	0.15
	Lower Swirl	Reduced wall heat loss and allows more nozzle hole numbers (avoid spray interaction)	0.5	0.03
Piston	Open bowl with recess and CR = 20.5	Reduced wall heat loss by reducing flame wall interaction		
Fuel Injection System	Up to 2500 bars injection pressure (pRail)	Soot reduction and more compact heat release around TDC	0.25	0.1
	Enhanced Injectors with fast opening/closing	Reduction of dwell time between successive injection		
	Alternative nozzles with 9 holes	Soot reduction at part load		
	Post injection directly after main injection	Potential to reduce Soot up to 50% by enhancing oxidation turbulent mixing during expansion phase.		
EGR Blower	EGR on demand independently of T/C condition. Possibility to match ATS requirements.	Higher EGR and pRail improves BSFC / NOX trade off	0.5	1.7
Turbo Charger	Upgrade Hi-Efficiency T/C to allow more EGR at low RPM / high load	Gas Exchange Efficiency improvement from higher compressor efficiency and avoid surging area.	0.25	0.4
eTurbo	Waste Heat Recovery High Voltage	Improve transient response to Boost pressure target but recovery limited by exhaust gas temperatures	0.5	2.7

6 Outlook and next steps

3D CFD investigation:

Following simulation activities based on the measurements collected during the project will be carried out:

a. Further increase of the Compression ratio and deeper understanding of the NOx / BSFC trade-off at higher Compression ratio.

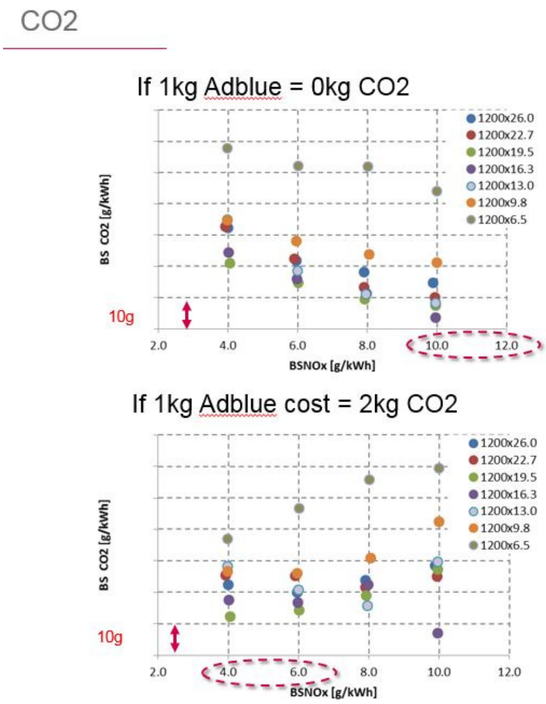
b. Effect of post injection to improve Soot and BSFC

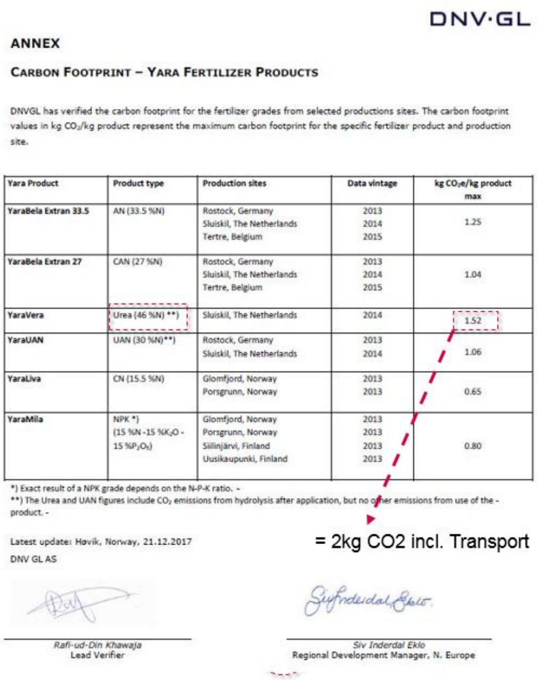
c. Effect on heat release by using increased hydraulic flow injector

d. Effect of Thermal Barrier Coating on wall Heat Loss

EGR pump:

If the EU regulations regarding CO2 emissions is moving towards a Well-To-Wheel, and if Adblue production and logistics is taken into account in the total CO2 balance of a vehicle, the EGR pump can be an interesting technology as it can bring the raw BS NOx emissions below 5g/kWh with ease of control and no throttling to increase the back pressure, see figure 42. The production of 1kg of Adblue required the release of about 2kg of CO2 according to carbon footprint survey if we include transport.





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ANNEX

CARBON FOOTPRINT – YARA FERTILIZER PRODUCTS

DNVGL has verified the carbon footprint for the fertilizer grades from selected productions sites. The carbon footprint values in kg CO_2/kg product represent the maximum carbon footprint for the specific fertilizer product and production site.

Yara Product	Product type	Production sites	Data vintage	kg CO_2e/kg product max
YaraBela Extran 33.5	AN (33.5 %N)	Rostock, Germany Sluiskil, The Netherlands Tertre, Belgium	2013 2014 2015	1.25
YaraBela Extran 27	CAN (27 %N)	Rostock, Germany Sluiskil, The Netherlands Tertre, Belgium	2013 2014 2015	1.04
YaraVera	Urea (46 %N) **)	Sluiskil, The Netherlands	2014	1.52
YaraUAN	UAN (30 %N)**)	Rostock, Germany Sluiskil, The Netherlands	2013 2014	1.06
YaraLiva	CN (15.5 %N)	Glomfjord, Norway Porsgrunn, Norway	2013 2013	0.65
YaraMila	NPK *) (15 %N -15 %K_2O - 15 %P_2O_5)	Glomfjord, Norway Porsgrunn, Norway Siilinjärvi, Finland Uusikaupunki, Finland	2013 2013 2013 2013	0.80

*) Exact result of a NPK grade depends on the N-P-K ratio. -
**) The Urea and UAN figures include CO_2 emissions from hydrolysis after application, but no other emissions from use of the - product. -

= 2kg CO2 incl. Transport

Latest update: Høvik, Norway, 21.12.2017
DNV GL AS

Rafi-ud-Din Khawaja
Lead Verifier

Siv Inderdal Eklo
Regional Development Manager, N. Europe

Figure 42: Specific NOx if Adblue is part of the CO2 bilan, bottom left.

Combustion Chamber:

Active pre-chamber with injection of air-fuel mixture running lean with EGR to control the fuel reactivity. Low emissions and high efficiency because of lean burn combustion

Air Handling Technologies:

Compressor with variable geometry could improve gas exchange efficiency.

Two-stage Turbo charging system improves BTE by 0.5%pt and is suitable for engine downsizing.

Variable Valve Actuation technology to improve gas dynamics and able to switch towards Miller cycle (Late IVC) with high Boost pressure

Alternative fuels:

1. DME with low boiling point allows more pre-mixed combustion to further reduce NOx. As the fuel is oxygenated and no C-C bound exists, soot is ultra-low. Modification on the fuel injection system is required. A follow-up project is supported by BFE.

2. Gasoline with low Octane number (RON60): partially premixed combustion (PPC) applied to conventional diesel engine allows diesel engine like BTE with substantial specific NOx reduction. RON60 is a possible "transition fuel" candidate from fossil fuel based to low viscosity based renewable fuels. The use of gasoline in compression ignition engine may help as well to alleviate the surplus of gasoline in Europe and thus shortage of Diesel fuel. The more favourable gasoline RON60 fuel H/C ratio of 2.15 compared with Diesel fuel of 1.85 translates into a TtW CO2 emission reduction by 5-6% if BTE remains unchanged.

3. Methanol fuel use in compression ignition engine with high CR has been reported and showed high potential, especially in combination with PPC combustion. Methanol, being the simplest fuel that can be produced from renewable energy and that is liquid at ambient conditions, offers huge potential for substantial CO2 reduction in the transport industry.

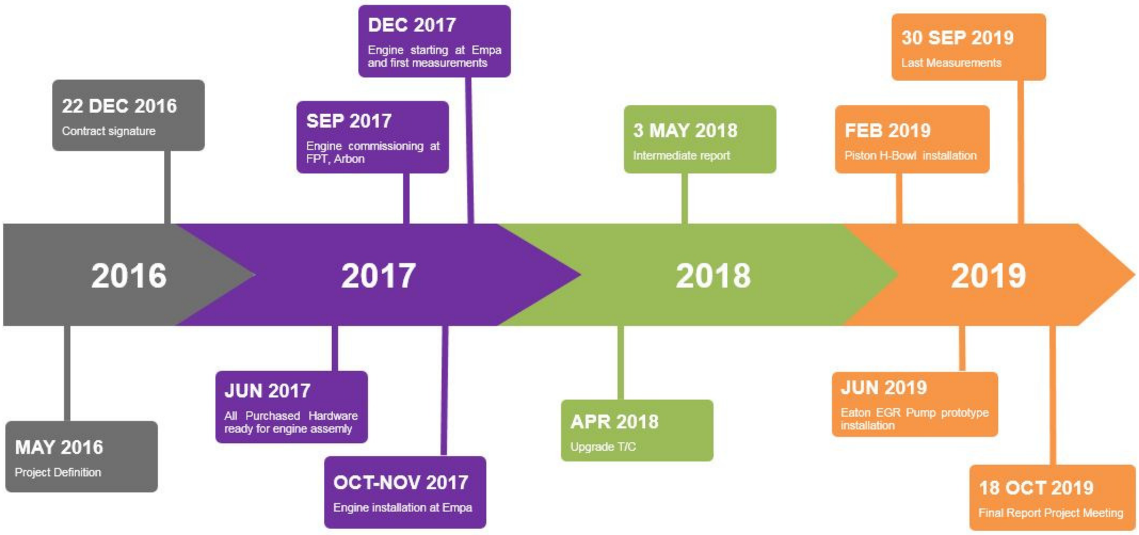
Tier I suppliers should provide high pressure injection pump compatible with low viscosity fuels by developing further lubricated plunger technology.

7 National and international cooperation

a) National cooperation with Empa, Swiss Federal Laboratories for Materials Testing. Empa takes over WP3, the investigation of the potential of exhaust gas recirculation.

b) Cooperation with FHNW, Windisch Kai Hermann and Silas Wüthrich to build a conditioning system to keep the EGR blower gas inlet temperature above dew point.

8 Project Timeline



9 Publications

1. Conference on Combustion Research in Switzerland, 7th of September 2017 at ETH Zurich
   Heavy duty diesel engine combustion with fully flexible EGR configuration: panel
2. Conference on Combustion Research in Switzerland, 24th of June 2019 at ETH Zurich
   BTE50% Project: presentation

10 References

1. THE U.S. SUPERTRUCK PROGRAM
   https://theicct.org/sites/default/files/publications/ICCT_SuperTruck-program_20140610.pdf
2. System Simulation of Partially Premixed Combustion in Heavy-Duty Engines : Gas Exchange, Fuels and In-cylinder Analysis, E. Svensson 2019
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8. Lastauto Omnibus: 4/2014

11 Appendices

In the table below, a fictive scenario of CO2 saving per year (baseline = 2019) is described for 20'000 vehicles based on the fuel consumption reduction potential due to the introduction of new technologies described p41 (bottom table).

These figures correspond to a fleet of long-haul heavy-duty (40T) commercial vehicles traveling about 140000 km / year. The fuel consumption for the EU6 baseline in 2019 is based on a survey from vehicle tests published by various European magazines (see 3-8 in 10. Referrences). The deducted averaged fuel consumption for a typical long-haul mission equals to 30.9 l/100 km, corresponding to a Brake Thermal Efficiency of 42.1% and a load factor of 28% (= averaged_Mission_Power / Max_rated_Power).

The Brake Thermal Efficiency target used within this project was corresponding to the maximum achievable within the engine map (about 1200 rpm / 75% load). In general, we can consider that improvement for the best point can be scaled to any point in the engine map as long as the load is above 25%.

Yearly Truck distance	[km]	140000			
Averaged Speed	[km/h]	80			
Time driving	[hours]	1750.0			
	[day]	218.8			
Power @ 80 km/h	[kW]	100.0			
Energy Required / year	[MWh]	175.0			
LHV diesel	[J/kg]	42800000			
Scenario fuel consumption reduction		**EU6 base**	**-8.3%**	**-9.6%**	**-14.7%**
Year introduction (fictive)		2018	2023	2025	2028
BSFC	[g/kWh]	200.0	183.4	180.8	170.6
BTE	[-]	0.421	0.459	0.465	0.493
Mass Fuel Amount / year / veh.	[kg]	35005	32100	31645	29859
Fuel density	[kg/l]	0.81	0.81	0.81	0.81
Vol. Fuel Amount / year / veh.	[l]	43216	39629	39067	36863
Mass CO2 / year /veh.	[T]	111	102	100	95
delta CO2 / year / veh.	[T]	-	9	11	16
fuel consumption	[l/100km]	30.87	28.31	27.91	26.33
Vehicle fleet / year	[units]	20000	20000	20000	20000
delta CO2 / year / fleet	**[kT]**	**-**	**184.2**	**213.1**	**326.2**

A 8.3% fuel consumption reduction leads to a saving of 184.2 kT of CO2 per year in comparison to 2019 and a 14.7% fuel consumption reduction leads to a saving of 326.2 kT of CO2 per year.

By 2030, a mandatory 30% CO2 reduction will lead to substantial vehicle modifications to reduce aerodynamic drag, rolling friction, vehicle weight, …in order to reduce the power requirements by about 20% to move a long-haul vehicle at 80 km/h.

In below simulation, it is considered that the Brake Thermal Efficiency of the engine can achieve 49.3%.

Year Period		< 2030	> 2030
Yearly Truck distance	[km]	140000	140000
Averaged Speed	[km/h]	80	80
Time driving	[hours]	1750.0	1750.0
	[day]	218.8	218.8
Power @ 80 km/h	[kW]	100.0	82.0
Energy Required / year	[MWh]	175.0	143.5
LHV diesel	[J/kg]	42800000	42800000
Scenario fuel consumption reduction		EU6 base	-30.0%
Year introduction (fictive)		2018	2030
BSFC	[g/kWh]	200.0	170.6
BTE	[-]	0.421	0.493
Mass Fuel Amount / year / veh.	[kg]	35005	24485
Fuel density	[kg/l]	0.81	0.81
Vol. Fuel Amount / year / veh.	[l]	43216	30228
Mass CO2 / year /veh.	[T]	111	78
delta CO2 / year / veh.	[T]		33
fuel consumption	[l/100km]	30.87	21.59
Vehicle fleet / year	[units]	20000	20000
delta CO2 / year / fleet	[kT]	-	667.0

A 30.0% fuel consumption reduction translates into a saving of 667.0 kT of CO2 per year in comparison to 2019.

12 Acknowledgements

At first I would like to thank my manager Harald Fessler who always gave me support for sharing new ideas and lengthy technical discussions. Harald contribution to this project is related with the Waste Heat Recovery technologies such as eTurbo, Turbocompound, … running countless GT-power simulations.

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I am very grateful to the whole Powertrain Systems Group at Empa led by Patrik Soltic for the great collaboration and support during this project. A special mention goes to Thomas Hilfiker and Roland Graf for their work ethic and dedication to the task.

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Finally, I am thankful to Stephan Renz and Carina Alles from BFE to provide the financial support for this project.