Use of EGR e-Pump for Dual-Mode Dual-Fuel Engines in Mild Hybrid Architectures

Antonio García, Javier Monsalve-Serrano*, Rafael Lago Sari, Álvaro Fogué-Robles

CMT - Motores Térmicos, Universitat Politècnica de València, Camino de Vera s/n, 46022 Valencia, Spain

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Corresponding author (*):
Dr. Javier Monsalve-Serrano (jamonse1@mot.upv.es)
Phone: +34 963876559

Abstract

Highway transport sector is still expected to be dominated by internal combustion engines in the future, especially due to limitations in the introduction and applicability of full electrification for heavy-duty propulsive systems. Nonetheless, this sector is still under the objectives imposed in the roadmap towards carbon neutrality. For this, engine efficiency and low-emission combustion modes must be improved, and the introduction of synthetic fuels, low electrification levels and devices for engine optimization and energy management have proven to be a great advance towards these objectives. This work studies the application of an EGR e-pump for energy recovery and combustion optimization on a powertrain running on Dual-Mode Dual-Fuel combustion mode as a substitute of a complex dual route EGR system. The results include the evaluation of the impact of using energy recovery devices on combustion performance and emissions levels, as well as a numerical evaluation of driving conditions considering a medium-duty application in the transport sector under different driving scenarios. The results point out that the inclusion of the EGR e-pump can contribute to mitigate the drawbacks of removing the complex dual route system in terms of equivalent fuel consumption without greater impact in terms of emissions. Additionally, its application in mild hybrid platforms can promote a significant improvement in CO_2 emissions, especially for urban areas (20% compared to a conventional powertrain based on HP EGR).

Keywords

Powertrain electrification; EGR e-pump, advanced air management systems; Dual-Mode Dual-Fuel combustion; Mild hybrid trucks.

1. Introduction

Carbon neutrality is the ultimate goal to be obtained in the next few years ahead to fulfil with the green deal agreed by the G7 nations [1]. This imposes several restrictions concerning the road transportation sector since it represents a share of ≈12% in the total CO_2 produced globally [2]. In Europe, this is even more relevant since the road transportation share increases up to 18% [3]. While battery electric vehicles are gaining attention as a possible solution for light-duty transport [4], the decarbonization of heavy-duty transportation has no clear solution. The usage profile and energetic demand of heavy-duty applications leads to several barriers regarding the full electrification of this sector [5]. The significant reduction in the truck payload (≈ 20%), long charging times and increased total cost of ownership are some of the problems that hinders the use of BEVs for heavy-duty applications. On the other hand, hydrogen-fuelled engines seem to be an alternative, since it provides zero CO_2 emissions in a tank -to-wheel basis, with reduced barriers for its introduction [6]. Nonetheless, this fuel still lags with respect to conventional fuels in its distribution infrastructure and requires dedicated powertrain design [7]. A mid-term solution could be the use of hydrogen-derived fuels with similar characteristics than those of conventional fuels, enabling the use of the current powertrain and distribution systems [8][9]. Clearly, the benefits would be reduced since higher energy demand is required to process hydrogen to fuels such as e-diesel and e-gasoline. Nonetheless, these fuels can be produced outside the country where they are consumed, benefiting from places where renewable energy is found in abundance such as Chile, Kingdom of Saudi Arabia, etc. This allows to reduce the Capital Expenditure (CAPEX) of the production system while having no operating expenses (OPEX), which enables to compensate the lower efficiency that these fuels could have compared to direct electrification [10][11].

Such scenario is favourable to further investigate alternatives to improve the efficiency of internal combustion engines, which could favour the prompt transport decarbonization by means of e-fuels [12]. Concerning the recent advances on high efficiency combustion, low-temperature combustion concepts have demonstrated in the past the capacity to provide higher efficiency than conventional diesel combustion while attaining ultra-low soot and nitrogen oxides (NO_x) [13][14]. Several of the early challenges regarding combustion controllability and load extension have been dealt during the last years, attaining solutions that reached technology readiness levels (TRL) up to five [15]. One of the most promising solutions is the Dual-Mode Dual-Fuel combustion concept [16]. This combustion mode relies on the former Reactivity Controlled Compression Ignition combustion (RCCI) in a great extent (from low to medium load) [17]. Nonetheless, additional stratification paths are added to the combustion chamber whenever the operation starts to be limited by pressure gradients. The most effective strategy is to increase the equivalence ratio stratification by means of shifting the high-reactivity fuel injection towards the top dead center, allowing to extend the engine operational range up to full load conditions [18]. This requires an increase in the oxygen concentration while maintaining the EGR levels, to avoid soot and NO_x formation. Nonetheless, this implies a wide range of mass flows through the turbine and compressor which are not possible to be achieved by conventional turbochargers, since they either operates near to the surge or the choke zones. Recently, these limitations started to be overcome by introducing mild hybrid electrification in the vehicles [19]. This has enabled the use of electrified air management systems such as e-Pumps for EGR and e-turbochargers [20][21]. Both devices allow to decouple the dependence of the air management system on the exhaust energy. Recent studies carried out by Smith [22] in a 4-cylinder heavy-duty engine representative of call 8 trucks have highlighted the potential of the EGR e-pump in providing the required EGR concentrations for NO_x control while enabling the simplification of the air management system from variable geometry turbines (VGT) to fixed geometry turbines (FGT). García et al. [21] also performed a numerical investigation aiming at identifying the benefits and challenges of using electrified air management systems such as EGR e-pump and e-turbocharger in a mild hybrid vehicle. The simulation results allowed to conclude that the combination of the system allows both fuel consumption savings and simplification of the geometry towards a fixed geometry turbine. Additionally, it was suggested that the use of e-pump allows to reduce the transient response of EGR while enabling energy recuperation in specific cases.

Despite the potential of this system, there are only few works available in the literature. In addition, they are fully based on numerical assessments and focused exclusively on consumption and emission investigations. Nonetheless, it is believed that experimental investigations need to be carried to investigate not only performance and emissions parameters but also the complex interplay that the EGR e-pump may introduce in the system regarding combustion, fuel injection and air management. Additionally, the extension of this investigation towards a full map calibration is mandatory to draw the complete scenario of this system as a potential solution for electrified powertrains. In this sense, this investigation aims at evaluating the application of an e-pump EGR system in combination with an advanced low temperature combustion concept as a pathway to deliver a clean and efficient powertrain solution to comply with the regulations that are to come. Experimental assessments are performed in a multi-cylinder engine platform, previously modified to run under DMDF combustion. A prototype e-pump EGR system was included to the engine fed by a 48 V power source system, allowing to monitor and quantify the energy conversion and management of the different power sources. This means that the possible energy recovery and its conversion to electrical energy can be assessed as well as the energy needed to deliver the required EGR amount can be also quantified. The investigations were performed first at representative conditions by applying parametric studies to assess the impact of the e-pump EGR system on the energy management and the combustion process. Next, considering the conclusions from the first investigation, a full map calibration was sought followed by a driving cycle evaluation. The last was performed considering a mild-hybrid 48 V vehicle platform to attain a real scenario in terms of the benefits that may be obtained combined mild-hybridization and the EGR e-pump.

2. Experimental materials and methodology

This section intends to detail the engine and experimental facility employed during this investigation as well as the EGR e-pump description, heat exchanger configuration and the testing methodology developed to carry out this work.

2.1. Engine and test cell facility description

Air management performance is highly dependent on the thermodynamic conditions that are found in the inlet and outlet of compressor and turbine. Low temperature combustion concepts have particularities such as low exhaust gas temperatures and high exhaust recirculation levels, which may decrease the energy availability for the turbine [23]. In this sense, the real representation of this boundary conditions is of utmost importance. One of the most representative way to account the limitations and the conditions that may be found in the system is by means of multi-cylinder engine testing. Therefore, this evaluation was performed using a multi-cylinder engine, representative of medium-duty applications. This engine was calibrated in the past for a Dual-Mode Dual-Fuel combustion having different targets such as EUVI NO_x [24][25]. In addition, it has a combustion system optimized for this concept [26]. The main characteristics of the engine are presented in Table 1.

Table 1. Main engine characteristics.

Engine Characteristics	
Engine Type	4 stroke, 4 valves, direct injection
Number of cylinders [-]	6
Displaced volume [cm^3]	7700
Stroke [mm]	135
Bore [mm]	110
Piston bowl geometry [-]	Bathtub
Compression ratio [-]	12.75:1
Rated power [kW]	235 @ 2100 rpm
Rated torque [Nm]	1200 @ 1050-1600 rpm

This engine has been modified to enable its operation in DMDF mode. First, the original compression engine was reduced from 17.5:1 to 12.75:1 by means of piston machining. Additionally, six port fuel injectors were added to deliver the low reactivity fuel. More detail about fuel injection systems that are used in the engine and previous calibrations can be found in previous works. A fully instrumented test cell facility was used to perform this investigation. An AVL active dynamometer with an embedded PUMA system was used to control the load and engine speed for each operating condition.

Specific instrumentation for instantaneous in-cylinder pressure measurement, gas analysis, etc. were also included. A detailed description on the test cell instrumentation and devices that were used can be obtained in previous works from the authors [27]. Table 2 summarizes the different sensors used and their associate accuracy. The addition of the EGR e-pump system required some modifications with respect to the base experimental setup. First, a heat exchanger was added prior to the e-pump EGR system to avoid excessive temperatures at the pump inlet. Additionally, specific control systems were added to the LabVIEW routine aiming at regulating the pump speed according to the operating condition required. Figure 1 illustrates the experimental facility used in this investigation.

Table 2. Accuracy of the instrumentation used in this work.

Variable measured	Device	Manufacturer / model	Accuracy
In-cylinder pressure	Piezoelectric transducer	Kistler / 6125C	±1.25 bar
Intake/exhaust pressure	Piezoresistive transducers	Kistler / 4045A	±25 mbar
Temperature in settling chambers and manifolds	Thermocouple	TC direct / type K	±2.5 °C
Crank angle, engine speed	Encoder	AVL / 364	±0.02 CAD
NO_x, CO, HC, O_2, CO_2	Gas analyser	HORIBA / MEXA 7100 DEGR	4%
FSN	Smoke meter	AVL / 415	±0.025 FSN
Gasoline/diesel fuel mass flow	Fuel balances	AVL / 733S	±0.2%
Air mass flow	Air flow meter	Elster / RVG G100	±0.1%

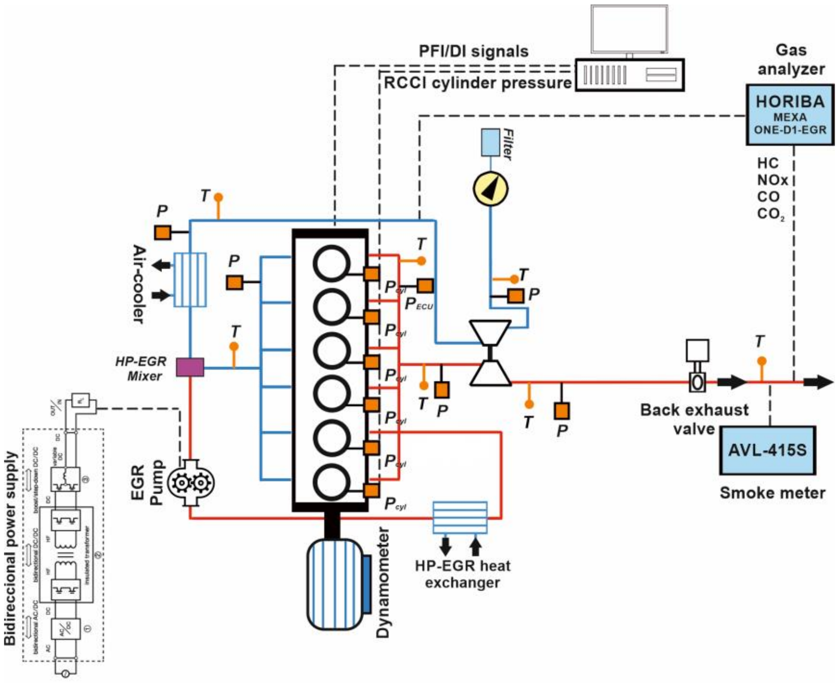


Figure 1. Schematic of the equipment used in the experimental facility.

2.2. EGR systems considerations

This section describes the different EGR system configurations that have been evaluated in the current engine with DMDF combustion up to arrive to the EGR e-pump solution.

2.2.1. Dual route EGR system

Dual route EGR systems are composed by a low pressure EGR and high pressure EGR route. The high pressure EGR is derived before the turbine inlet while the low-pressure system is taken after the turbine, passing through a set of heat exchanger and dryer to avoid water to enter in contact with the compressor blades. This system provides benefits regarding the maximum quantity of EGR that can be done, the temperatures achieved at the cylinder inlet and the pumping losses required to drive the EGR [28].

Due to these advantages, this system has been investigated for low temperature combustion concepts where high EGR concentrations are required. Figure 2 illustrates the EGR system layout that was used in the early investigations of the DMDF combustion concept, which allowed to obtain engine-out tailpipe EUVI NO_x emissions with ultra-low soot emissions and similar efficiency as the one from the original diesel calibration.

Despite the benefits, dual-route EGR systems present some practical challenges. One of the most important one to be considered is the packaging issue due to the space required to accommodate all the pipes, dryer, and heat exchanger for the system. Apart from that, dual-route EGR systems also lead to higher costs, being not the preferred option for wide-scale applications.

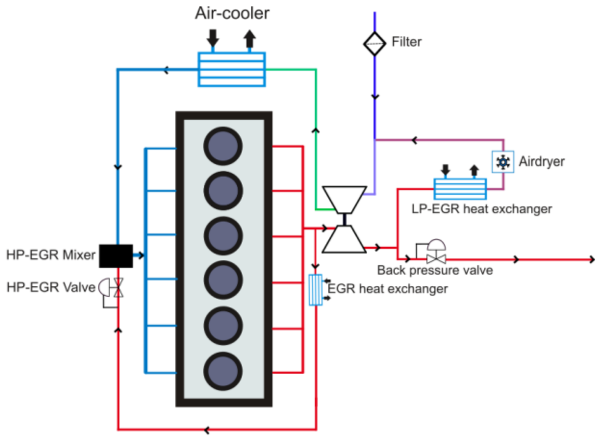


Figure 2. Diagram of the setup used with the dual-route EGR architecture.

2.2.2. High pressure EGR

High pressure EGR is the simplest way to increase the dilution levels in the combustion chamber. The exhaust gas is derived before the turbine and directed towards the intake manifold, where its flow rate is regulated by means of an electric valve. The maximum EGR output is highly dependent on the pressure difference between the exhaust and intake pressure [29]. Due to its simplicity, this EGR system equips most of the current engines, including the commercial version of the engine used in this investigation. Figure 3 depicts the EGR scheme that was used in this investigation.

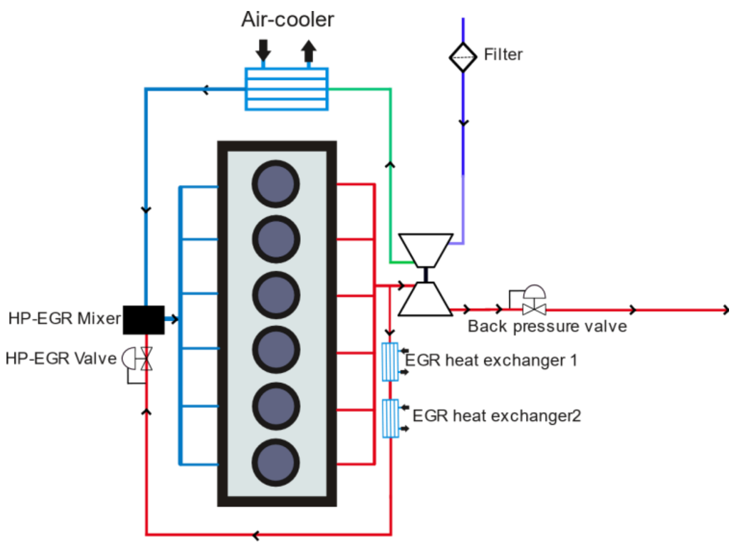


Figure 3. Diagram of the setup used with the HP EGR architecture.

2.2.3. e-Pump EGR system

Considering the points referred in the introduction section, it is clear that the e-pump based EGR system can deliver benefits in conventional applications such as the decrease of transient times and the possibility of recovering energy from the exhaust gases in specific operating conditions. In spite of the limited capacity of the system, numerical investigations have demonstrated up to 2% of improvement in the fuel consumption using this system [21]. For this investigation, a prototype e-pump EGR system was used [30], with the specifications presented in Table 3.

Table 3. Technical specifications of the EGR e-pump.

Specification	Value
Displacement volume [c.c./rev]	400
Nominal power [kW]	6
Power supply [VDC]	48
Maximum speed [rpm]	10000
Power Recovery	Yes

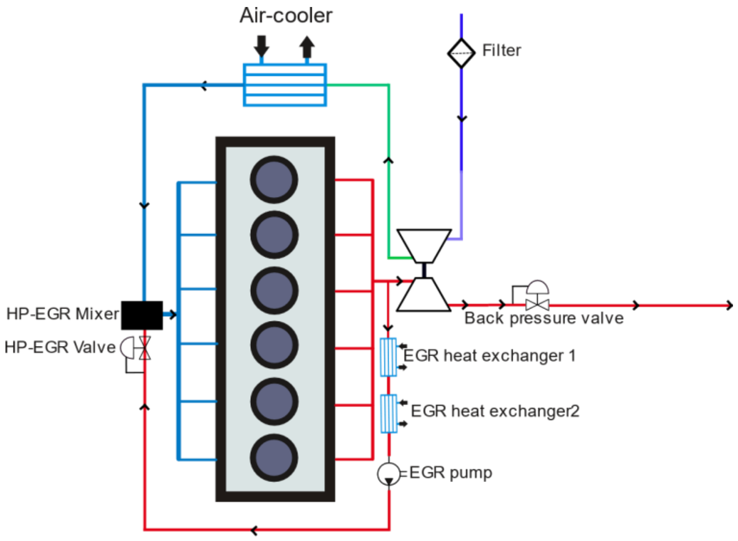


Figure 4. Diagram of the setup used with the HP EGR and e-pump architecture.

2.3. Assessment methodology

The use of the EGR e-pump was assessed by means of both experiments and numerical simulations to understand its impact on driving cycles. In this sense, this section intends to describe the experimental and numerical methodology employed in this investigation.

2.3.1. Experimental evaluation

The experimental investigations were performed in two steps. First, a parametric sweep of e-pump velocity was done in specific operating conditions. This analysis allows to understand the complex interplay among the boundary conditions in the e-pump inlet, the original air management system and the intake manifold conditions that will govern the combustion development. The points to be assessed where chosen considering the particularities of the DMDF combustion concept. As shown in Figure 5, this concept relies on different strategies. From low to medium load, high premixing degrees are employed with low to high grade of equivalence ratio stratification. These zones also require EGR levels up to 50%. By contrast, from high to full load, a dual-fuel diffusive combustion is used, being characterized by high reactivity fuel injections near to the top dead center and EGR levels around 20%. In this sense, different engine speeds at 50% of engine load (the most demanding engine load in terms of EGR requirement) were evaluated. For each operating condition, and EGR e-pump speed sweep was performed. The limits of the sweep were experimentally determined by the points where the combustion stabilities were too high, or the e-pump started to be limited by maximum speed.

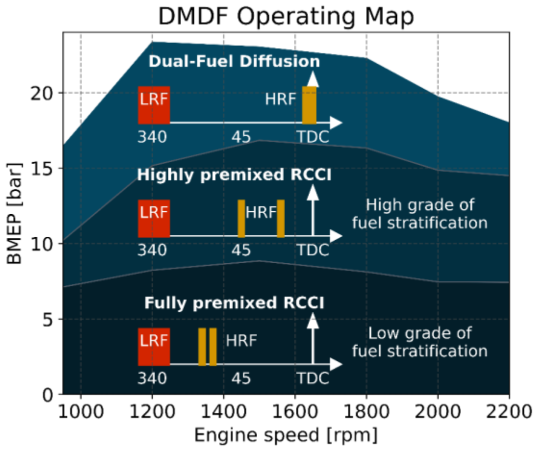


Figure 5. Conceptual injection strategy of the dual-mode dual-fuel combustion concept.

Next, a full map calibration is proposed, considering only the optimized operating conditions using the new system. The calibration methodology was based on that presented in previous papers such as García et al. [31] and Benajes et al. [32]. Nonetheless, slight modifications were done to account for the energy spent or generated by the e-pump. This was accomplished by means of including the power signal from the e-pump to the main LabView controller interface. This signal was used to calculate a final brake specific fuel consumption according to the equation Eq. 1, which allows to include all the energy paths of the system.

$$BSFC_{eq} = \frac{\dot{m}_{LRF} \cdot \frac{LHV_{LRF}}{LHV_{LRF,ref}} + \dot{m}_{HRF} \cdot \frac{LHV_{HRF}}{LHV_{HRF,ref}}}{\dot{P}_{engine} - \dot{P}_{pump}} \qquad \text{Eq. 1}$$

In this formula, it is considered that the power consumed or regenerated by the pump is greater than zero when EGR is being forced by the pump and it is consuming electric energy, and lower than zero for those conditions where the pump is regenerating energy and producing electric energy that is stored in the battery.

It is important to remark that the increase of the EGR levels that flows through the HP EGR system leads to higher demands on the heat exchanger of the original EGR system. In this sense, the cooling system needs to be redesigned to enable reasonable temperatures at the intake manifold. To do this, a specific study was performed to design a cooling system for the HP-only architecture by means of an experimental characterization of the original heat exchanger and numerical simulations of the system. This has allowed to define the number of heat exchangers by considering their ability on remove heat from the exhaust gases. The result of this investigation is presented in Figure 6. As it is can be evidenced, the addition of an extra heat exchanger with the same characteristics of the original one allows to remove a significant additional quantity of heat from the EGR. Nonetheless, further increase of heat exchangers does not provide any appreciable benefit on EGR cooling. In this sense, the new heat exchanger proposal has considered the use of two heat exchangers in series (the original plus an extra one). This system was used for both the HP EGR system and EGR e-pump system.

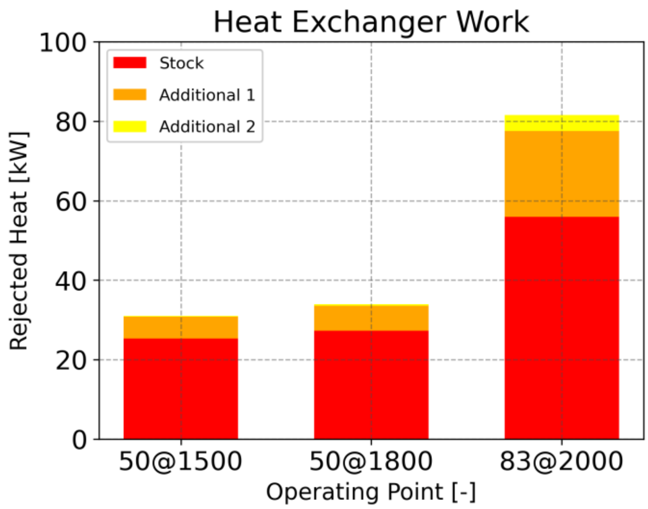


Figure 6. Performance of different number of heat exchangers in the EGR circuit.

2.3.2. Numerical analysis

A numerical analysis is also proposed to assess the influence of the e-pump EGR system on different driving scenarios. To do this, a mild hybrid platform was designed considering the methodology proposed by García et al.[33] A Volvo FE 350 truck was used, since it is originally equipped with the engine used in these investigations [34]. Battery size and electrical motors were defined by means of an optimization process. Details about the design process can be verified in García et al [33][35]. Figure 7 illustrates the final truck architecture containing the modifications for accommodating the DMDF combustion and the mild hybridization. Two fuel tanks are included in the system as well as a small battery pack and a belt assisted starter (BAS). The main characteristics of the truck platform are presented in Table 4.

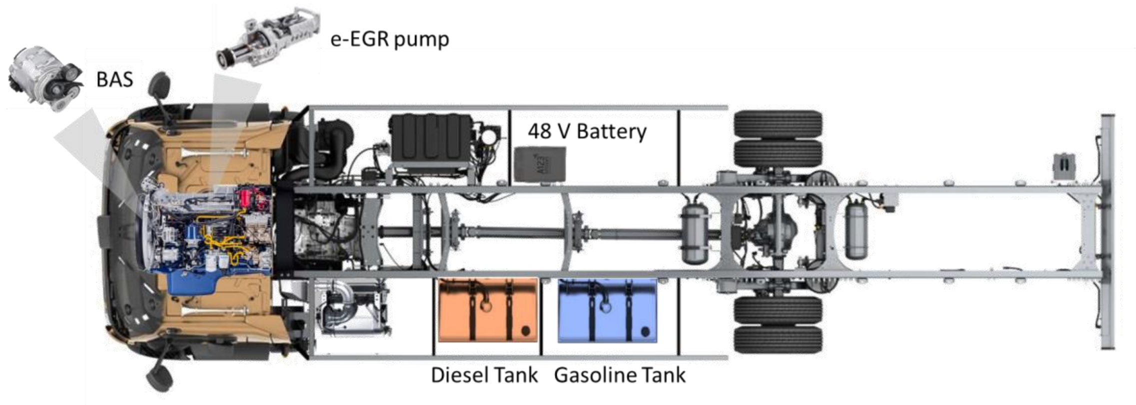
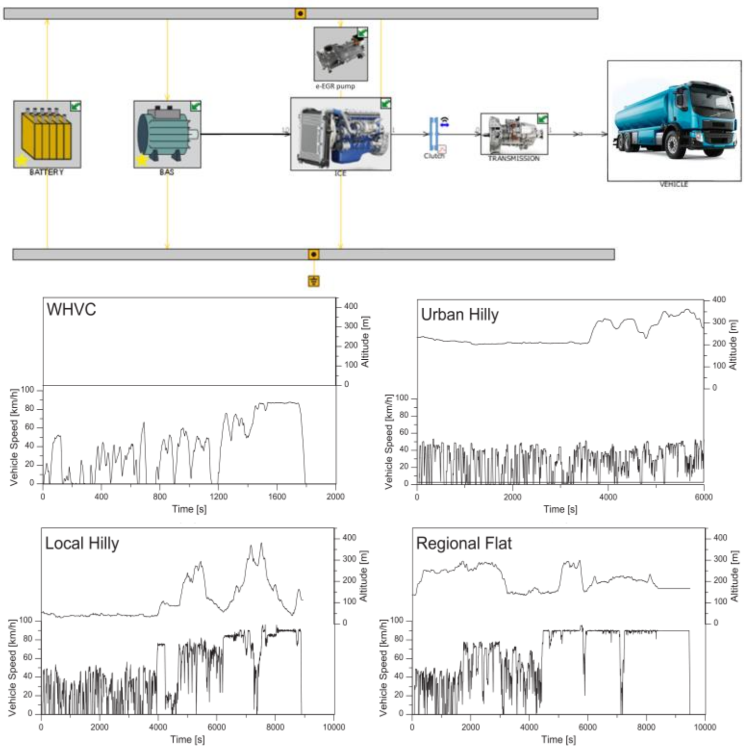


Figure 7. Schematic of the vehicle modifications assumed for the mild hybrid architecture.

Table 4. Main characteristics of the vehicle model and electric components.

Characteristic	Value
Base vehicle Mass [kg]	7035
Max Payload [kg]	8982.5
Frontal Area [m^2]	6.89
Tires Size [mm/%/inch]	295/80/22.5
Gear Box type [-]	12 gears manual
Differential ratio [-]	3.08
ICE rated power [hp]	350@2200rpm
ICE rated Torque [Nm]	1400
Battery size [kWh]	2
Electric motor size [kW]	22.4

A 0-D numerical model was built in GT-Drive to enable the simulation of different driving cycles and payloads with and without EGR e-pump and the comparison to the conventional truck architectures such as the conventional diesel combustion and the DMDF. The final model is represented in Figure 8. As it can be seen, it consists of different objects that represent the components of the truck. The ICE object allows the use of the experimental fuel consumption and emission maps as inputs to the simulation. This approach has demonstrated the capability of reproducing transient results with good accuracy [36][37]. In addition to the ICE object, other components such as the battery model, BAS, transmission, and vehicle geometry are defined in the remaining objects. An explicit solution scheme is used to integrate the differential equations with respect to time whereas different integrators are used to obtain the cumulative results regarding fuel consumption and emissions [38]. The simulation was performed considering four different driving cycles (flat, local, urban and WHVC) and three different truck payloads (0%, 50% and 100%). The selected driving cycles are representative of both in-service conformity and normative driving cycles, while the payload of 50% is the payload used for homologation purposes [39].



(a)

(b)

Figure 8. Schematic of the vehicle architecture implemented in GT-Drive (a) and the different driving scenarios considered for the study (b).

3. Results and discussion

The results section is divided into different parts. First, the simplification of the original DMDF EGR system by removing the low pressure EGR circuit and the assessment of its impact on performance and emissions is presented. Next, the effect of introducing the EGR e-pump system is discussed from a combustion, performance, and emissions basis. Finally, a full map calibration considering the EGR e-pump as the preferred alternative to drive EGR is proposed, followed by the assessment of the calibration maps in a mild hybrid electric vehicle (MHEV) and its comparison to the conventional DMDF and diesel combustion concepts in a conventional powertrain architecture.

3.1. Simplifying the EGR circuit: removing the low pressure EGR

Prior to any evaluation with the EGR e-pump system, preliminary investigations were done removing the low pressure EGR system to understand the impact of using this simplified approach and to set a benchmark to be used as comparison for the electrified architecture. As previously commented, this approach would benefit the application of the engine architecture in terms of packaging and production costs. In this sense, a full map calibration was performed using the HP EGR system. The results for performance and emissions were then compared with those from the LP + HP pressure EGR system reported in previous results.

Figure 9 presents the brake specific fuel consumption (BSFC) results obtained with the new EGR configuration compared to the original LP + HP EGR circuit. As it is shown, the removal of the low pressure EGR circuit leads to penalizations in fuel consumption in most of the cases, with absolute fuel consumption increases of up to 25 g/kWh. These results are a consequence of the higher pumping losses that are obtained with the new

EGR configuration, since the exhaust gas are derived prior to the turbine, requiring it to work in closed VGT positions and low efficiency points. This is mainly perceived from low to medium load conditions. As the load is increased, the differences are also lower since the exhaust gases energy availability allows to have better operating conditions inside of the turbine.

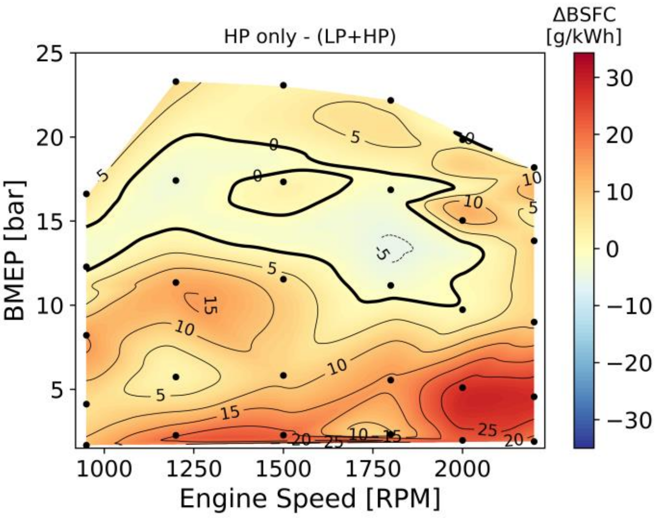
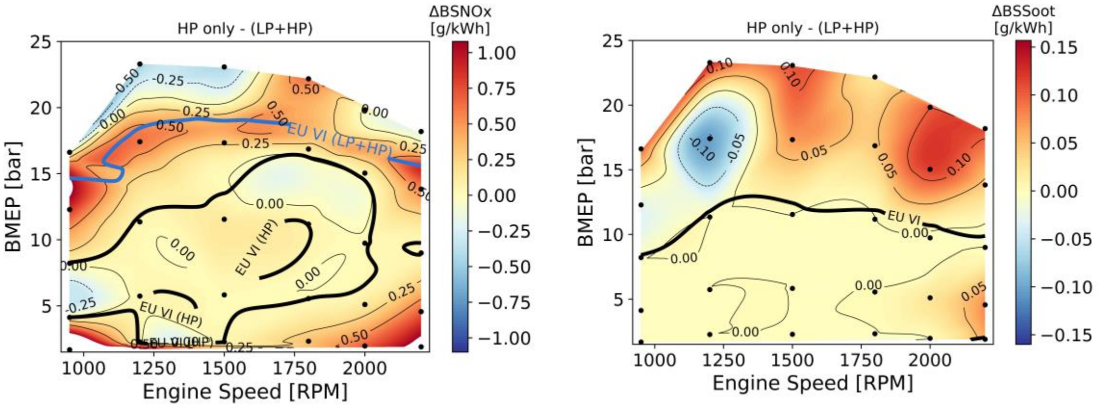


Figure 9. Differences in BSFC between the dual-route EGR architecture and the single-path EGR architecture.

It should be also remarked that the calibration strategies from both architectures is not the same. While the LP+HP EGR system aims at obtaining EUVI NO_x values with ultra-low soot, the new HP system cannot comply with this strategy. Therefore, the calibration with HP EGR was a compromise between fuel consumption optimization and low NO_x and soot emissions. However, as it is depicted in Figure 10 (a), the region where EUVI legislation is fulfilled is much lower than that from the LP+HP EGR calibration considering NO_x emissions. While the original calibration was able to fulfill the EUVI targets in terms of NO_x up to 80% of engine load, the HP EGR calibration has only a narrow zone located in the middle speed middle load conditions. This is a consequence of higher inlet temperatures that are obtained with the new EGR system since the heat flow through the EGR heat exchanger increases, even with the additional heat exchanger system. Moreover, the low pressure EGR system cooling effect is not present anymore. Both reasons contribute to higher inlet temperatures and consequent higher NO_x formation. By contrast, the ultra-low soot emission zone (soot<10 mg/kWh) seems to not be significantly affected by the EGR system modification (Figure 10 (b)).

From this analysis, it can be concluded that the removal of LP EGR system does not bring any benefit in terms of neither emissions nor fuel consumption. In this scenario, the use of strategies that may improve the EGR system operation such as the e-pump EGR system are of interest. To explore this, the next sections deal with the introduction of this system in the combustion concept and its evaluation using different methodologies.



(a) (b)

Figure 10. Differences in $BSNO_x$ (a) and BSSoot (b) between the dual-route EGR architecture and the single-path EGR architecture.

3.2. EGR pump as alternative to improve engine performance.

As previously demonstrated, removing the LP EGR route leads to several drawbacks concerning fuel consumption and NO_x and soot emissions. Therefore, and alternative must be realized to attain a mid-term solution regarding packaging, while still offering the required boundary conditions to obtain a proper DMDF combustion. This may be attained by including the EGR e-pump system, also enabling a second route of efficiency gain by means of the energy recovery. To investigate the opportunities of improving the concept with this EGR system, a sweep-based investigation was performed, evaluating different pump speeds at 50% of engine load and several engine speeds (from 950 rpm to 2200 rpm), some of the most demanding conditions in terms of EGR and air management settings. It is important to remark that the sweeps aimed at delivering the same engine power output and EGR levels, which is obtained by tuning the VGT position.

Figure 11 illustrates the effect of the EGR pump speed on the combustion development by means of the heat release rates for 50% of engine load and 1800 rpm of engine speed, condition that presents one of the highest EGR levels. As it is shown, the increase of the e-pump speed leads to a delayed combustion process. This result is a consequence of the complex interplay between the air management and injection systems and their impact on the combustion development. This can be evidenced in Figure 12, where the intake and exhaust manifold pressures as well as the EGR levels and total LRF injected mass are presented. While the EGR levels are maintained constant, both intake and exhaust pressures are modified. The variation of the pump speeds requires to modify the VGT position, leading to lower exhaust pressures and a consequent different pressure ratio in the turbine. Since the compressor operation is directly impacted by the turbine settings, differences in the compressor outputs are also expected.

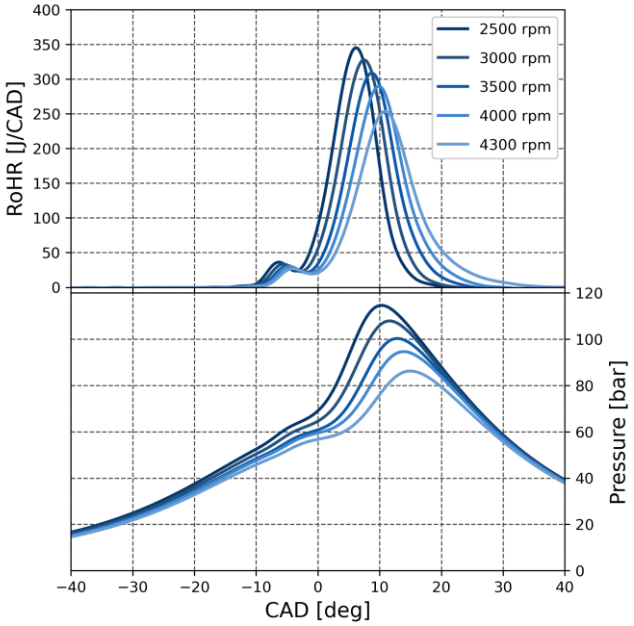


Figure 11. Effect of the EGR e-pump on in-cylinder thermodynamic evolution.

As it is shown also in Figure 12, despite having similar EGR and power output levels, the air management cannot deliver the same air mass flow values, which means a modification of the operating condition towards richer equivalence ratios. Despite this, the combustion seems to be negatively affected by the increase of the EGR e-pump speed. This can be explained by the decrease of both inlet pressure and temperature as the e-pump is accelerated, resulting in worse conditions for the fuel oxidation.

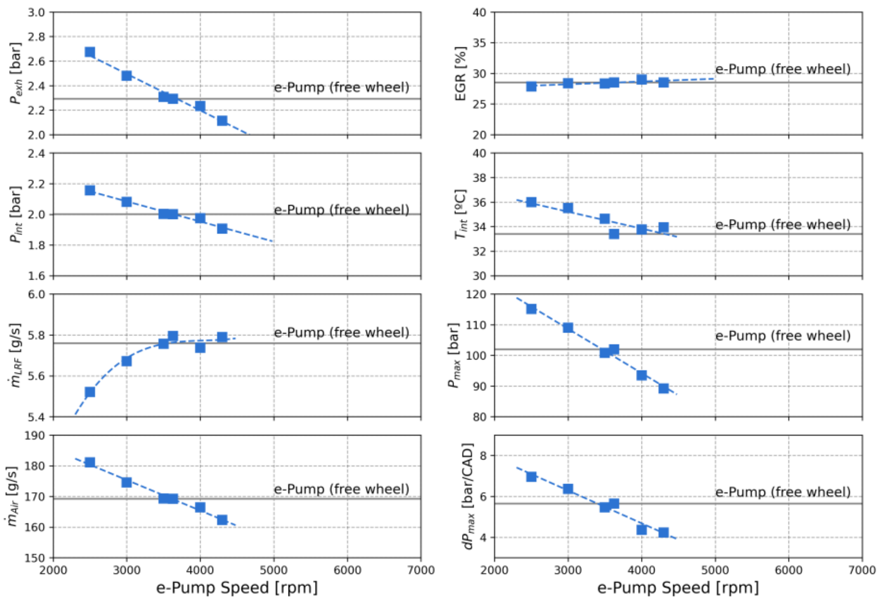


Figure 12. Effect of the EGR e-pump on different engine boundary conditions.

Other secondary phenomena also occur as the inlet pressure is modified. As it is shown, compression ratio reduction in the compressor modifies the total fuel mass injected by the low reactivity fuel injection system. This modification is a consequence of the higher-pressure difference between the injection pressure and the manifold pressure. This phenomenon is expected to be found in real applications and therefore was not corrected during the experiments since it did not impact the target variables (EGR levels and power output). Lastly, the use of the EGR e-pump adds another degree of freedom to mitigate maximum pressure and maximum pressure gradients within the cylinder, that are directly related to noise emissions and mechanical durability of the engine. As shown in Figure 12, it is possible to maintain power output and EGR levels while reducing noise emissions and lowering the mechanical demand over the engine structural components.

The use of the EGR e-pump also opens new paths to recover energy from the exhaust gas. This means that this energy path must be included in the total efficiency of calculation. Additionally, the variation of the air management configuration is expected to have a significant influence on the pumping losses of the engine. The combination of all the discussed parameters such as combustion phasing, pumping losses, fuel mass, etc., will culminate in the outcome of energy conversion that is the brake specific fuel consumption equivalent, presented in Eq. 1. This parameter allows to characterize the global effect of the EGR e-pump, which is depicted in Figure 13. This figure summarizes the effect of EGR e-pump speed for 50% of engine load and engine speeds from 950 rpm to 2200 rpm considering the values from HP EGR calibration as references. Each square represents the $BSFC_{eq}$ with the EGR e-pump, while the solid horizontal bars refer to the reference value from the LP+HP EGR original calibration.

As it can be observed, there is a trend inversion regarding the effect of the pump speed on the $BSFC_{eq}$ values as the engine speed is increased. At low engine speeds (950 rpm), the EGR e-pump operation results in a better overall energy conversion when it is operating at low speeds. This means that the pump is operating in a regenerative mode, extracting energy from the flow. In general, at these conditions, an improvement on $BSFC_{eq}$ of 2.5 g/kWh can be obtained. It is worth mentioning that as the methodology seeks to maintain the EGR and emissions level, when the pump is regenerating energy, it is necessary to further close the VGT and increase the volumetric pump inlet density by means of higher pressure. To have a wider range of regeneration capacity on the e-pump it would be necessary to assess the turbo-matching and sizing of the VGT to ensure that there is enough margin to maneuver with the rack position. As the engine speed is increased, the trend starts to invert towards higher EGR e-pump speeds, meaning that the most advantageous operation comes from providing energy to the EGR flow. This enables the decrease of the pumping losses of the engine by means of opening the VGT rack position. This type of operation can provide benefits of up to 8 g/kWh, much higher than those when the EGR e-pump works at regenerative mode. It is also worth to mention that, in most of the cases, the introduction of the EGR e-pump system allows to have better $BSFC_{eq}$ results than those from the HP EGR system. Additionally, at high engine speeds, the improvements can be higher enough to achieve almost the same $BSFC_{eq}$ results than those from the dual-route EGR system. In this sense, it can be concluded that the addition of the EGR e-pump system can enable a more efficient operation by different paths (regeneration or reduction of pumping losses). Nonetheless, a wider assessment must be done to identify if the effect is preserved also on other engine loads. To do this, a full map calibration is proposed in the next subsection.

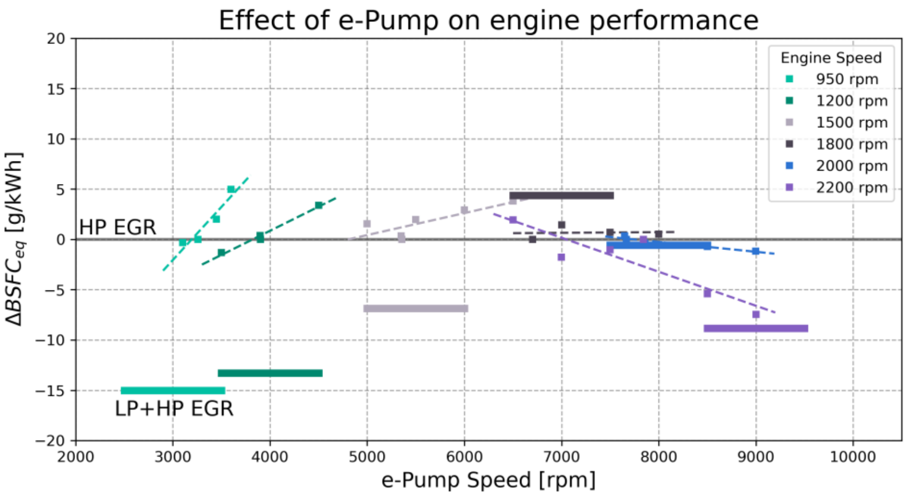


Figure 13. Effect of the EGR e-pump on engine performance depending on the engine working regime.

3.3. Full map calibration using the EGR e-pump system

Considering the previous conclusions, the assessment of the EGR e-pump was extended to a full map calibration addressing 30 operating conditions (10%, 25% ,50% ,75% ,100% at 950 rpm, 1200 rpm, 1500 rpm, 1800 rpm, 2000 rpm and 2200 rpm). For each operating condition, the best equivalent fuel consumption value was aimed while fulfilling the restrictions in terms of NO_x and soot from the previous calibration as always as possible. Figure 14 presents the $BSFC_{eq}$ map in absolute difference basis compared to the previous HP EGR calibration. It can be inferred that the use of the EGR e-pump system allows to improve the fuel consumption in a wide extent of the calibration map. In addition is interesting to note that the improvement zone is in regions widely used during conventional driving conditions. Finally, the higher benefits are also found in high engine speeds corroborating with the results presented in Figure 13.

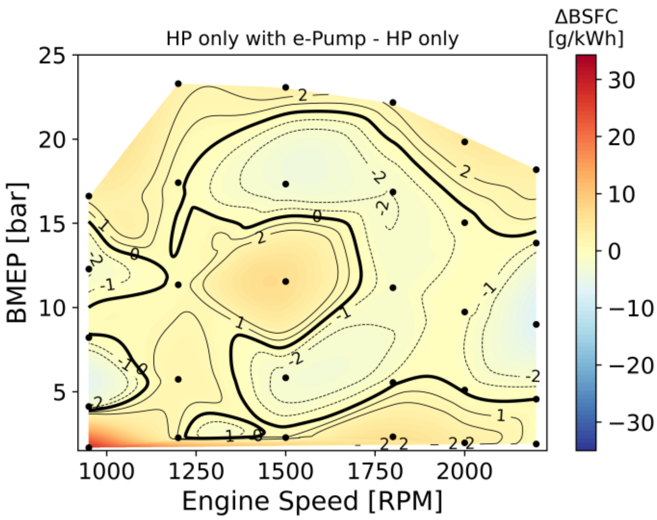
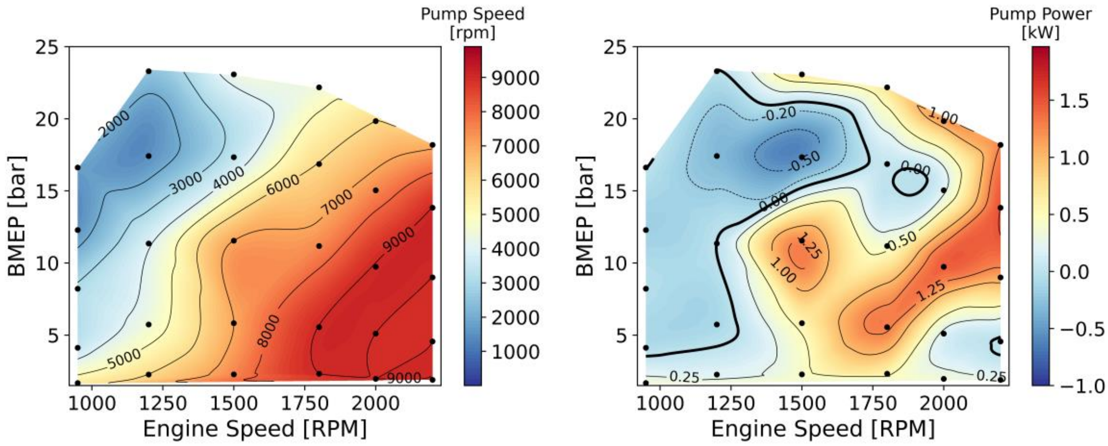


Figure 14. Differences in $BSFC_{eq}$ between the HP EGR architecture with and without the EGR e-pump.

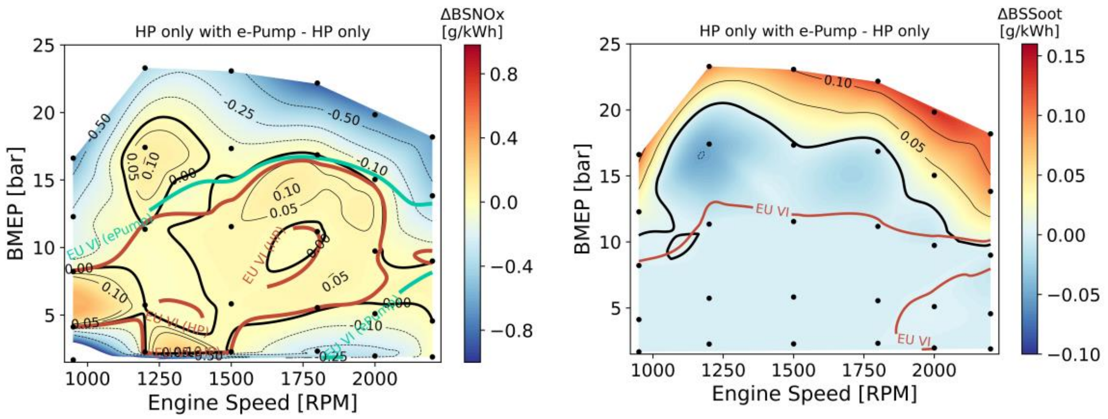
Figure 15 depicts important parameters regarding the EGR e-pump speed operation such as the pump speed and the energy flow in the pump (regeneration in negative and consumption in positive). From the analysis of Figure 15 (a), it can be inferred that the e-pump velocity is scaled directly with the engine speed. Additionally, as previously commented, the powertrain benefits from higher EGR e-pump speeds as the engine speed increases. However, as shown in the Figure 15 (a), there is a wide range in the operating map where the pump works close to or in the maximum allowable speed. This allows to state that the benefits of using the pump could be extended in case of having a bigger EGR e-pump system. The analysis of the EGR e-pump power also allows to identify the possible zones where regeneration is possible. As shown in Figure 15 (b), the e-pump works regeneration mode is enhanced at low engine speeds and higher loads where the flow velocities are low, and the pumping work does not play a dominant role on the efficiency. However, as previously discussed, at high engine speeds, the e-pump is used to decrease the pumping work of the engine, providing flexibility to the turbine, and assisting to drive the EGR flow towards the intake manifold.



(a) (b)

Figure 15. Operating conditions of the e-pump within the engine map in terms of pump speed (a) and power flow (b).

The use of the EGR e-pump system also allows to attain benefits on NO_x emissions as presented in Figure 16. The first noticeable change is the increase of the limits where engine-out tailpipe EUVI compliant NO_x emissions are obtained. Nonetheless, benefits are also evidenced at high to full load conditions. As previously stated, the use of higher EGR e-pump speeds leads to a decrease of the temperature and pressure at the inlet valve close, affecting the combustion process evolution. The simultaneous analysis of NO_x and soot emission maps allows to garner relevant information about the benefits and limitations of the EGR e-pump regarding these pollutants. As it is shown in the soot maps, the apparent benefits in NO_x emissions near to full load operation has as consequence a further increase on soot emissions compared to the original HP EGR calibration. This may be justified due to the worst conditions that are attained during the combustion to oxidize the soot emission that is generated. This mechanism (soot-NO_x trade-off) is widely addressed in the literature [40]. In this sense, it can be argued that the benefits of the EGR e-pump regarding soot and NO_x emissions are limited in these zones.



(a) (b)

Figure 16. Differences in $BSNO_x$ (a) and BSSoot (b) between the HP EGR architecture with and without the EGR e-pump including EUVI compliant regions.

3.4. Driving cycle assessment using the EGR e-pump system

Since different trends were observed in which concerns performance and emissions values, a driving cycle analysis was proposed to identify the overall performance of the EGR e-pump system on homologation and real driving conditions considering the methodological approach presented in section 2.3.2. Figure 17 presents the results of the driving cycle, comparing the results from the EGR e-pump in a mild hybrid vehicle with thosee from the original calibration with LP EGR and only HP EGR system in a conventional powertrain. In addition, the P0 mild hybrid vehicle using the HP EGR calibration maps was also added to the system to provide a reference for the electrified version of the truck. All the results are referred to the conventional engine operating with conventional diesel combustion.

As Figure 17 shows, the driving cycle evaluation highlights the penalizations in $BSFC_{eq}$ that are obtained when the LP EGR system is removed. The highest penalizations occur in urban driving conditions, where the energy requirements are low, leading to a frequent operation in low to medium load conditions. These conditions were demonstrated to be the most penalized when the LP EGR system is removed in the steady state maps. A further step was done, by including a P0 MHEV architecture coupled with the HP EGR system. This architecture comprehends one of the lowest levels of electrification but allows to have electric devices such as the e-pump. As it can be seen, the electrification of the HP EGR system platform allows $BSFC_{eq}$ improvements from ≈-9% to ≈ -18%, depending on the driving condition evaluated. The lowest benefits obtained for the MHEV architecture are found for driving conditions which contains highway phases. Finally, the EGR e-pump architecture is included in the model and assessed in the different driving conditions. The results allow to infer that the combination of MHEV and e-pump allows to reduce the overall vehicle equivalent fuel consumption in more than 4% for urban applications compared to the MHEV with only HP EGR. This is a direct consequence of the increase of the benefits attained in the steady state maps with the e-pump at low to medium engine load conditions. As the truck payload is increased, lower are the benefits for the MHEV architecture, independently of using or not the EGR e-pump, since most of the points are displaced towards high load operation. In this sense, it can be concluded that the proposed MHEV with EGR e-pump architecture is more suitable for urban applications rather than highway, enabling significant energy savings for this driving condition.

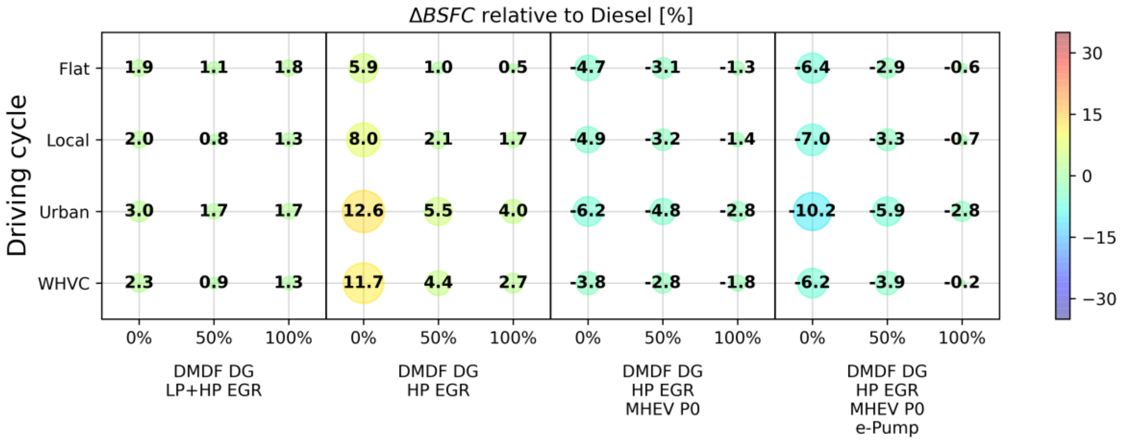


Figure 17. Difference in BSFC of different engine architectures compared to CDC under different driving conditions and payload.

The impact of the EGR e-pump on NO_x and soot emissions was also assessed considering its coupling with the MHEV architecture. Despite of the general reduction of, at least, 77% in NO_x emissions due to the low temperature combustion concept, it can be evidenced that the use of the e-pump provides an overall improvement, independently on the operating condition assessed. Nonetheless, as it is shown in Figure 19, the NO_x reduction comes at the cost of increasing the soot emissions. This is enhanced as the payload increases, since the operating conditions are shifted towards high load conditions, where the soot production is significantly higher compared to the HP EGR calibration and the LP+HP EGR.

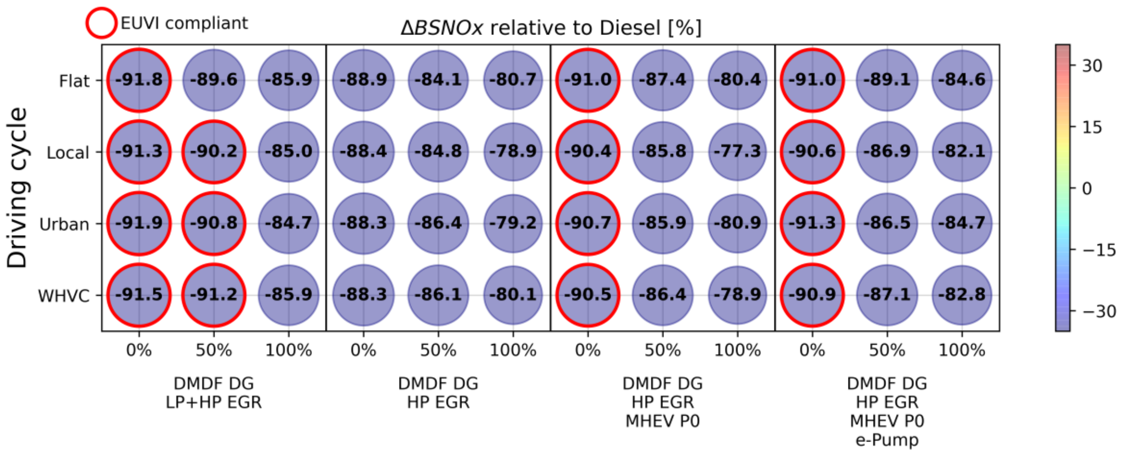


Figure 18. Difference in $BSNO_x$ of different engine architectures compared to CDC under different driving conditions and payload.

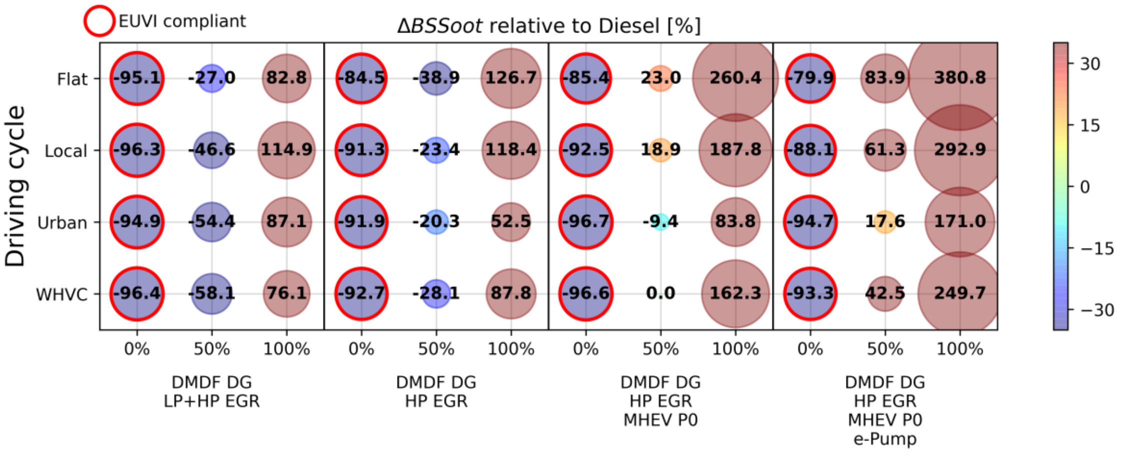


Figure 19. Difference in BSSoot of different engine architectures compared to CDC under different driving conditions and payload.

Finally, the impact of the EGR e-pump on the tank to wheel CO_2 emissions was assessed. As presented in Figure 20, the combination of mild hybridization and the EGR e-pump can provide CO_2 savings of up to 12.3%, considering applications in urban zones and with 0% of payload. Despite the decreasing trend of the TTW CO_2 savings with respect to the truck payload, the benefits can range from 12.3% up to 8. 5% for 0% to 50% of payload, which is representative of the truck payloads in urban applications. In this sense, it can be concluded that the use of the EGR e-pump system is an effective solution for advanced combustion concepts offering a pathway of saving fuel consumption and CO_2 emissions.

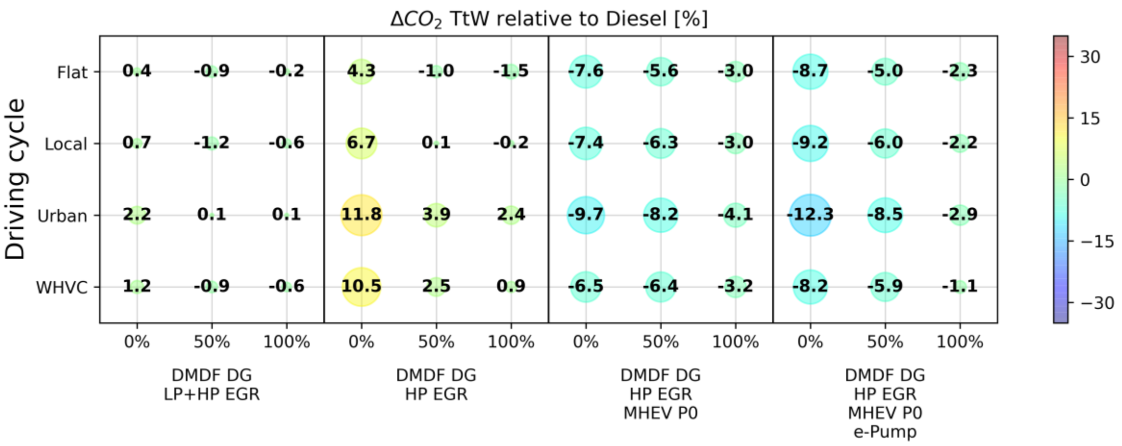


Figure 20. Difference in CO_2 emissions of different engine architectures compared to CDC under different driving conditions and payload.

4. Conclusions

This work has performed a detailed investigation regarding the application of the EGR e-pump system combined with an advanced low temperature combustion concept as a media to improve the energy conversion and management of the system. The main findings of this investigation allowed to conclude that the electrification of the air management system by means of the addition of the EGR e-pump provided an extra degree of freedom to the system. Additionally, the combination with MHEV architectures enhanced the benefits in terms of CO_2 emissions. The main conclusions can be summarized as:

- The EGR e-pump impacts the overall system performance, modifying not only the air management settings by also has secondary impacts on the fuel injection system. Moreover, two main pathways of efficiency improvement were verified: i) the EGR e-pump can allow lower pumping losses and ii) the pump can extract energy from the exhaust gases and convert it into electrical energy to feed the battery.
- The zones where each efficiency improvement is attained is highly dependent on the engine speed. Low engine speeds allow operation in regeneration mode while higher engine speed benefits the system by means of pumping losses reduction.
- Despite the modifications verified in the air management system, the emission values were not heavily impact by the introduction of the EGR e-pump. Engine loads from low to medium load remained under the same constraints, while the high to full load operation was governed by the soot-NO_x trade-off.
- The combination of the EGR e-pump system with MHEV has allowed to obtain efficiency increments and TTW CO_2 reductions higher than 20% compared with the conventional architecture running with HP only EGR system in urban applications. These benefits are a direct consequence of avoiding the low load zones and the improvements of fuel consumption in this region.

Considering the results obtained in this investigation, it is evident that the EGR e-pump system is an effective way to reduce the fuel consumption and CO2 emissions. The highest benefits are attained at urban driving conditions. This is an important conclusion since it is aligned with eco-innovation technologies that aims to reduce the impact of transportation inside of the cities, i.e., in urban circuits. Therefore, it can be suggested that the combination of EGR e-pump in DMDF combustion with mild hybridization and eco-driving techniques may offer an effective way to achieve future CO2 targets.

It can be concluded that the use of electrified air management systems and its combination with advanced combustion concepts are a feasible pathway to fulfill with the future regulations concerning criteria pollutants and CO_2. The use of this systems with MHEV can be a cheaper and direct solution to decarbonize the transportation sector and must be accounted in the mix of solution that is proposed for a carbon neutral transportation scenario since it can also run synthetic fuels.

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Notation

Abbreviatures

BAS: Belted Alternator Starter

BEV: Battery Electric Vehicle

BSFC: Brake Specific Fuel Consumption

$BSFC_{eq}$: Equivalent Brake Specific Fuel Consumption

$BSNO_x$: Brake Specific NO_x emissions

BSSoot: Brake Specific Soot emissions

CDC: Conventional Diesel Combustion

CO_2: Caron dioxide

DMDF: Dual Mode Dual Fuel combustion mode

EGR: Exhaust Gas Recirculation

EUVI: EURO VI legislation

FGT: Fixed Geometry Turbine

HP EGR: High pressure EGR circuit

ICE: Internal Combustion Engine

LP EGR: Low Pressure EGR circuit

LP+HP EGR: EGR architecture with both LP and HP circuits

MHEV: Mild Hybrid Electric Vehicle

NO_x: Nitrogen Oxides

RCCI: Reactivity Controlled Compression Ignition

RoHR: Rate of Heat Release

TRL: Technology Readiness Levels

VGT: Variable Geometry Turbine

WHVC: World Harmonized Vehicle Cycle