2025 Passenger Car and Light Commercial Vehicle Powertrain Technology Analysis

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Project Report

Project: 2025 Passenger Car and Light Commercial Vehicle Powertrain Technology Analysis	
Project for: ICCT ***Project Manager:*** David Blanco-Rodriguez	***Project No.:*** P33597 ***Report No.:*** ***Date:*** 18.11.2015
Subject/Objectives: The target of the project is the study on passenger cars and light commercial vehicle performance until 2025, by considering: • Simulation of drive cycles with different powertrain technologies • Assessment of emission compliance, fuel consumption and costs for various selected technology combinations	
Method/Solution: The study considers the results obtained by simulation of the different vehicle segments, technologies and cycles; as well as the cost analysis of all the technical solutions. The Simulation tools include in-house built models for the longitudinal simulations of powertrains and emissions.	
Summary/Results: The technology potentials are different depending on the analyzed segments. Moreover, the trend changes depending on the cycle to consider. The current study underlines these differences.	
Conclusions/Recommendations: The selection of technologies and powertrain calibration will differ depending on the final segment to consider, apart from the legislation steps and cycles to cover.	
Keywords: [CO_2] [Vehicle Simulation] [Technologies] [WLTP] [NEDC] [Diesel] [Gasoline] ***Contents:*** 341 Pages	***Project supervising:*** Dr.- Ing. Bastian Holderbaum

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1 Introduction

The International Council on Clean Transportation (ICCT) is an independent nonprofit organization founded to provide first-rate, unbiased research and technical and scientific analysis to environmental regulators.

The mission is to improve the environmental performance and energy efficiency of road, marine, and air transportation, in order to benefit public health and mitigate climate change. In such a context, ICCT is interested in the analysis of the CO_2 emissions for different passenger cars and segments, and by considering different technologies. In addition, different cycles are considered in the study: NEDC (New European Driving Cycle - as emission cycle for EU6b legislation), WLTP (Worlwide-harmonized Test Procedure - emissions procedure for EU6c legislation); and the US cycles FTP75 and HWFET (Highway Fuel Economy test).

1.1 Project objective

The target of the project is the study of passenger cars and light commercial vehicle performance until 2025, by considering:

- Simulation of drive cycles with different powertrain technologies
- Assessment of emission compliance, fuel consumption and costs for various selected technology combinations

The following boundary conditions should be mentioned:

- While the original contract was written as an assessment of 2030 technology and performance, the non-consideration of the RDE and the difficulties to find available data for all components to do assessments that far in the future, advice to shift the technology report to reflect technology likely to be available around 2025.
- The primary focus of the study is the European Market according to the EU6b regulation as well as the consideration of emissions under both the NEDC and WLTP test procedures. Nevertheless, the powertrains are designed with focus on the WLTP cycles.
    - Real Driving Emissions (RDE) are not covered within the current project. It may be possible that the selected powertrain technologies, especially the aftertreatment systems in the diesel segments, might not be compliant with RDE (legislation and guidelines not fixed at the date of starting the simulation work of the current project). One clear example is the installation of LNT as the only $deNO_x$ aftertreatment system for compact cars, it is possibly not a feasible solution for Segment C and even Segment B cars within the future RDE legislation. Nevertheless and when possible, a certain engineering margin in terms of emissions is targeted for the WLTP cycle.
    - The variants are also simulated within the US cycles in order to assess the CO_2 emissions under such cycle conditions. Nevertheless, the configuration of the vehicles was designed in order to fulfill EU6b legislation and not US standards, i.e. the vehicles are not representative of the US market needs and trends. For example, more stringent heating strategies or optimized aftertreatment systems are required for such market. This analysis is out of the scope of the present project.

- o Engines were not downsized for the analyses of weight reduction and hybrids. This will result in partially increased performance, the benefits of which were not analyzed or included in this project. However, the transmission gear ratios and the shifting strategy (for automatic transmission vehicles) were optimized in order to get the maximum possible benefits, equalizing the performance of all vehicles within the segment as much as possible. The impacts of weight and load reduction on hybrid system size and cost were also not assessed in this project.
- o Improvements in hybrid battery power density were not considered and current hybrid battery specifications were used for hybrid cost assessments.
- o CO2 simulations on the NEDC included additional flexibilities allowed for coastdown, which are not allowed on the WLTC and US cycles, and lower test mass than on the WLTC and US cycles. The CO2 comparisons between the NEDC and WLTC are affected by the different loads, not just the test cycles.
- o No increases in compression ratio for gasoline engines were included in the analyses, except for the variable compression ratio (VCR) and Miller cycle assessments.

1.2 Project defined tasks

The project considers the following tasks:

1. Vehicle Simulation:
    a. Simulation of NEDC and WLTP procedures, according to the last information (January 2015).
    b. Additional cycles for vehicle simulation by considering the US cycles FTP75 and US EPA Highway (HWFET).
    c. Sensitivity analysis on the driving resistance.
    d. Vehicle Performance (only for Gasoline).
2. Cost Analysis:
    a. Definition of “baseline” & “advanced” technology (analog to task 1, excluding “vehicle weight” and “driving resistance” variations).
    b. Definition of reference hardware or description made by experience of development and design engineers as well as additional research as base for cost analysis (no purchase of hardware)
    c. Definition of costing methodology for each component
    d. Direct manufacturing delta-cost analysis for advanced technologies (excluding “vehicle weight” and “driving resistance” variations), covering all relevant system parts
    e. Scaling of direct manufacturing delta cost for different vehicle classes (analog to task 1)
    f. This budgetary proposal considers the cost estimation of all technologies, covered in the simulations for base year 2014
3. Assessment and reporting
    a. Assessment matrix summarizing simulation results and cost analysis results
    b. Project meetings every 2-3 weeks or under demand; status report in MS PowertPoint
    c. Final report (MS Word format)

2 Main technologies considered in the project

This chapter summarizes all the main technologies that were considered in the project and highlights their advantages and some of their main drawbacks.

2.1 Air Management technologies – Exhaust gas recirculation (EGR)

By means of the EGR, a portion of the exhaust gas is extracted from the exhaust engine side and recirculated to the intake system in order to reduce NO_x for Diesel engines; and/or to reduce knock limitation or de-throttling for gasoline engines. The main effects of the EGR are:

- Reducing the effective fresh air mixture into the cylinder (lambda is reduced), thus reducing the combustion efficiency. Both reductions of the effective air (less available oxygen) and peak temperature decrease the rate of NO_x formation (mainly driven by the thermal mechanism described by the Extended-Zeldovych equations).
- The mixture of fresh air and EGR (inert gases) present a higher heat capacity, thus also reducing the peak temperature.

There are different EGR layouts with various advantages and disadvantages.

2.1.1 Cooled High Pressure EGR & Uncooled High Pressure EGR

Diesel engines:

The exhaust gas recirculation (EGR) reduces NO_X due to lower peak combustion temperatures through different effects: gas dilution through reduced oxygen concentration and higher specific heat capacity of EGR compared to air.

Cooled EGR leads to improved cylinder filling and thus O_2-mass in the cylinder, with respect to an uncooled system. The EGR cooling is especially important if the EGR is used at higher engine loads; in order to be able to keep high EGR rates without compromising the minimum required oxygen. Maximizing the A/F-ratio is important to sustain good combustion efficiency and low PM emissions. For most Diesel engines, cooled HP EGR is already standard; but in the recent years the combination of HP EGR with cooled LP EGR systems is growing rapidly (it is a well-known technology, its application is now becoming useful due to the more stringent emissions limits).

The uncooled EGR instead give some benefits in contrast to cooled EGR, by reducing total cost and supporting faster engine warm-up. Nevertheless, bypass valves can be installed to deactivate the cooling effect into cooled systems for heating or other needs. Non-cooled EGR is used for example in some medium duty (MD) or heavy duty (HD) application in combination with high-efficient SCR.

Cooled high pressure EGR has the advantage of allowing higher EGR rates at lower temperature, thus higher density of the recirculated gas. Due to that, the NO_x aftertreatment system can be dimensioned smaller and cheaper. Uncooled high pressure EGR in comparison offers improved combustion stability at low loads, faster engine warm-up, prevention of condensation in the intake manifold, no EGR cooler thus a better package and no risk of EGR cooler clogging. Alternatively, systems which mount cooled HP EGR may have a valve and an extra piping in order to bypass the cooler.

The coolant system requires an accurate control strategy to avoid significant PM disadvantages. Further additional costs and packaging limitations have to be considered for the EGR valve, piping and EGR cooler. In addition, an increased boost pressure is required to keep the intake fresh air set-point. The risk of EGR cooler fouling and/or clogging should also be mentioned. Uncooled EGR instead has worse air/fuel ratio and less cylinder filling efficiency, increased charge air temperature, higher fuel consumption and increased PM emissions. Nevertheless, it can support faster heating.

Gasoline engines:

Cooled and uncooled high pressure EGR is used in gasoline engines to reduce throttling losses. The main disadvantage of high pressure EGR is the reduction of the cylinder filling at high loads in NA engines and therefore the reduction of the full load. Due to that, high pressure EGR cannot be used in gasoline engines to reduce the knock limitation in full load condition.

The EGR actuation should be coordinated with the boosting system and the aftertreatment management.

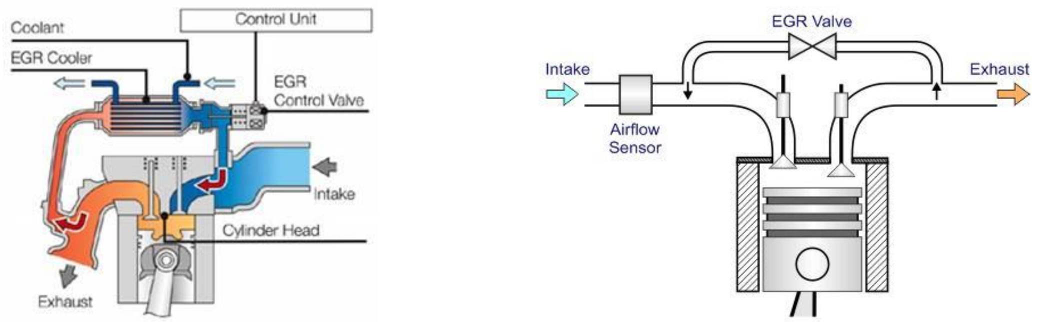


Figure 2.1 Cooled HP EGR setup (left) [Hitachi]; Uncooled HP EGR setup [Dieselnet] (right)

2.1.2 High Pressure (HP) & Low Pressure (LP) EGR

The HP-EGR loop is usually recirculated upstream of the turbine to downstream of the charge air cooler; while the LP-EGR is usually recirculated downstream DPF (to avoid any particles from entering into the compressor) and through the intake compressor. A combination of HP-EGR and LP-EGR can easily be found in the market today.

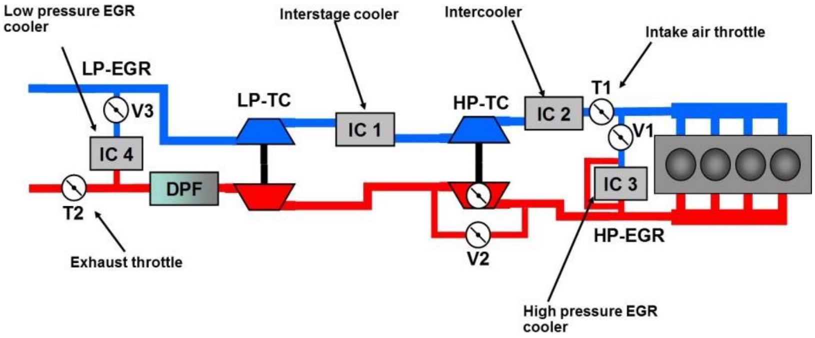


Figure 2.2 Possible configuration of a cooled HP EGR and cooled LP EGR. The HPEGR system presents a bypass valve to the high pressure EGR cooler [FEV].

The LPEGR gives the possibility of recirculating exhaust gas flow at lower temperatures, and hence supporting the turbocharger to work at more efficient areas, supporting the filling efficiency at a good range. The system allows therefore to keep high EGR rates with an appropriate fuel consumption while reducing the EGR cooling demands (the gas is recirculated downstream of the turbine and some aftertreatment components). Furthermore, LP-EGR supplies filtered exhaust gas to the intake since the exhaust gas is subtracted downstream DPF or catalyst. The combination of both LP- and HP EGR is an advisable system in order to reduce the engine-out NO_x emissions, while still getting some fuel consumption benefits with respect to use only HP EGR systems. The base EGR can be handled by LP EGR thus transient response can be optimized (compared to LP-EGR) by controlling also the HP-EGR (the pressure difference is usually higher and the route shorter). Further LP-EGR can be used at high engine loads and reduced speed, while HP-EGR may be used at low load and/or during engine warm up. At medium-high speeds and loads the combination of HP and LP EGR can be optimized in order to minimize emissions while still keeping the turbocharger and the filling efficiency at appropriate levels.

Main LP-EGR drawbacks are dependency of the packaging on the vehicle, transient response due to long EGR routing, compressor wheel erosion and deposits, condensate formation, increasing exhaust volume flow going through all aftertreatment components upstream DPF or catalyst (increasing the space velocity and thus reducing overall efficiency of the reactions) and lower exhaust temperatures compared to HP-EGR (due to a higher combustion efficiency) which also affects the afterteatment systems efficiency. The combined HPLP system presents logical drawbacks due to cost, packaging as well as a more complex strategy to control the EGR split and coolant bypasses (if any). Interdependencies with the boosting and aftertreatment systems are worth to mention.

2.1.3 Internal EGR

Internal EGR is an alternative technique of achieving NO_x-reduction by recirculating a portion of residual exhaust gas back to the engine cylinders. There are different EGR-Methods to recirculate residual exhaust gas; these usually consider variable valve timing technology for control. One possibility is to open the outlet valve during aspiration stroke and/or the inlet valve during outlet stroke. Other methods use electronic control of the outlet valve threshold. The internal EGR drawbacks (mainly fuel penalty) can be partly compensated by countermeasures such as variable charge motion or variable valve timing.

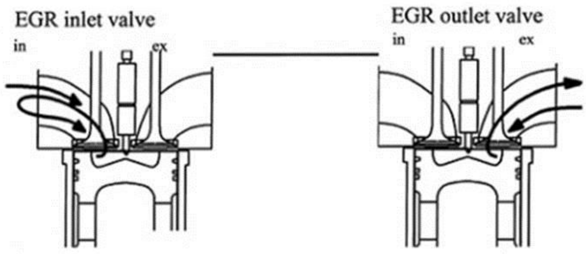


Figure 2.3 Internal EGR processes [Deutz]

The main advantages can be summarized in getting EGR effects without the need of an EGR control valve and EGR piping, lower costs, less package volume and an improved cold start performance due to the higher temperature compared to external EGR. However, the internal EGR is not enough for the current diesel engines in order to meet the stringent emissions regulations. Internal EGR reduces the burn velocity not as strong as external EGR and that leads to a better combustion stability in part load in gasoline engines. Also the de-throttling effect is stronger compared to external EGR due to the higher temperatures.

As disadvantage, the accurate control of the quantities is not straightforward, since the intake mass flow cannot be easily measured or estimated. Other main drawbacks for Diesel engines are still a worse fuel consumption and PM emissions, deposits at the intake system as well as reduction of the volumetric efficiency due to increase the overall temperature. In gasoline engines, the lower oxygen content in the combustion chamber leads to higher PM and PN emissions.

The internal EGR can be controlled by means of the variable valve timing and lifting technologies and is considered within this project for both diesel and gasoline.

2.2 Air Management technologies – Boosting systems

The key contributor to reach a higher specific power output is increasing boost pressure. This allows to increase the effective fresh air (oxygen) into the cylinders. Furthermore, increased boost pressure is necessary in combination with increased EGR to avoid reduced process efficiency and reduced full load torque. A turbocharger uses the waste energy of the exhaust gas to increase the gas pressure at the intake manifold. Therefore it consists of a turbine driven by the exhaust-gas stream and a compressor. There are different kinds of turbochargers depending on the type of application. In the following, the main layouts using within this project are discussed. In all cases, the interdependency with the EGR systems for the air path management should be considered.

2.2.1 Variable Geometry Turbocharger

By varying the geometry of the turbine the turbine's power and boost pressure can be varied. There are two main designs. The control can be achieved by rotating the vanes to change the geometry (typically for light-duty applications), or by varying the cross-section of the inlet (typical for heavy-duty applications).

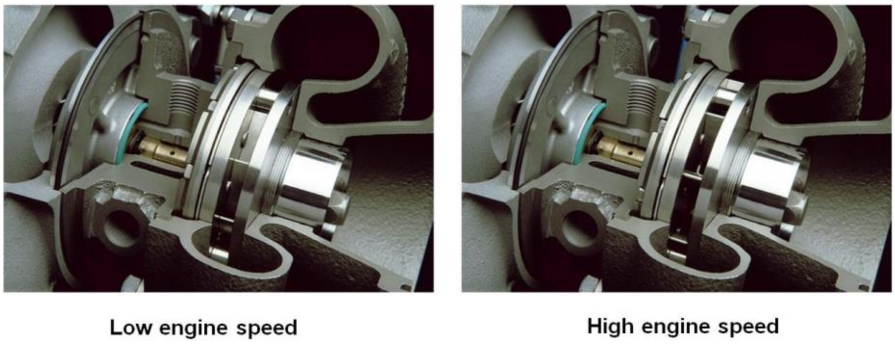


Figure 2.4 Section through a VGT [Porsche]

The turbocharger is very efficient at higher engine speeds. It presents an improved load response and reduced conflict between low-end torque and rated power requirements. It requires no waste-gate and can also be used in 2-stage turbocharging concepts.

Nevertheless this concept has to deal with lower reliability due to moving parts at high temperatures, which also limits its use at very high power stages. Further issues are the risk of variable vanes blocking as well as increased costs for gasoline engines due to high exhaust gas temperatures which as a result requires an expensive heat resistant material.

2.2.2 Waste-gate turbocharger

At high engine speeds, the waste-gate valve diverts part of the exhaust flow away from the turbine. This reduces the exhaust flow through the turbine, decreases the exhaust back pressure and limits the turbine speed. At low engine speeds and full load, the waste-gate is closed and the entire exhaust flow drives the turbine and thus the compressor. Primary function of the waste-gate is to regulate the maximum boost pressure in turbocharger systems. Waste-gate actuation can be either pneumatic or electric.

Figure 2.5 Waste-gate turbocharger [Bosch Mahle TurboSystems]

The main advantages are the regulation of the maximum boost pressure to protect the engine and the turbocharger. It further allows smaller turbocharger layout with better boost pressure built-up and has lower costs compared to Variable Turbine Geometry (VGT) or 2-Stage-Turbocharger.

However, the boost pressure control cannot be optimized as with a VGT system.

2.2.3 2-Stage-Turbocharger

Two differently sized turbochargers might be used in order to allow both systems to work at the best efficiency areas of the map. There exist different possible layouts: serial, parallel and these can be sequentially activated, among other configurations. For example and as explored in this project, a serial 2-stage turbocharger configuration can be actuated sequentially. During low to middle engine speed, the entire exhaust gas expands through the High-Pressure (HP) turbine. With increased engine speed, a HP turbine bypass valve is opened, progressively shifting more of the expansion work to the Low-Pressure (LP) turbine.

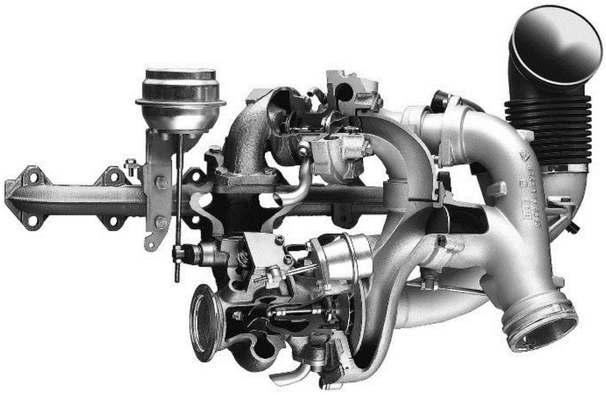


Figure 2.6 Section through a 2-Stage-Turbocharger [BMW]

This 2-Staged-Setup leads to higher torque at low engine speeds but still keeping an appropriate efficiency at both systems. The improved boosting can be used to increase torque and/or maximal power performance usually in combination with downsized engines and/or for emission reduction (PM, Diesel).

The complexity of design and packaging increases due to the additional components including bypass valve. Furthermore, the costs increase compared to a single-stage-turbocharger.

2.3 Air Management technologies – Valve train

A valve train is a mechanical system that controls operation of the valves in an internal combustion engine, [Source: Brain, Marshall (5 April 2000). "How Car Engines Work". HowStuffWorks. Retrieved 29 January 2014]. Valve train opening/closing and duration, as well as the geometry of the valve train, controls the amount of intake mass flow entering the combustion chamber at any given point in time [Source: Scraba, Wayne (October 2000). "Camshaft Tips & Definitions". Hot Rod. Retrieved 29 January 2014]. Timing for open/close/duration is controlled by the camshaft that is synchronized to the crankshaft by a chain, belt, or gear.

Valve trains are built in several configurations. Each varies slightly in layout but still performs the task of opening and closing the valves at the time necessary for proper operation of the engine. The most important configurations for the project are described in this chapter.

2.3.1 Variable Valve Lift (VVL)

VVL is a mechanism which switches the intake and/ or exhaust valve lift. There are 2-step or fully variable systems available. It can be implemented in different ways for example mechanical, electro-mechanical and hydraulic. There are various technologies where variable valve lift is combined with variable valve timing.

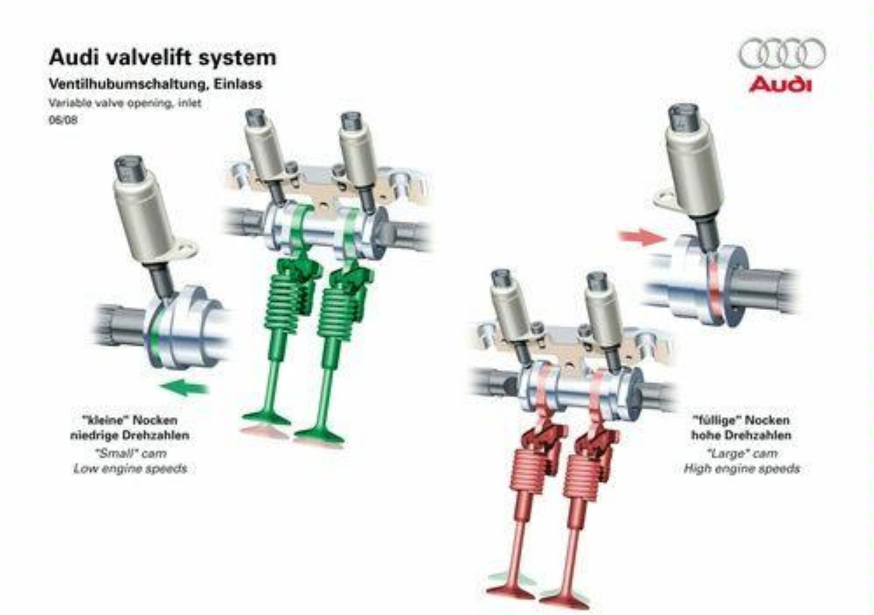


Figure 2.7 switchable VVL principle [Audi]

Through VVL throttling losses are reduced and the cylinder air mass can be controlled. It also can be used to fine tune the swirl level and for thermal management or adapted swirl levels in Diesel engines. The interdependencies can be used in combination with Variable Valve Timing.

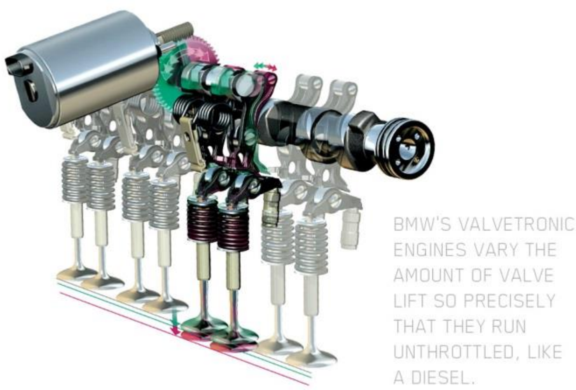


Figure 2.8 continuous VVL principle [BMW]

Negative consequences are more complex design of the valve train, increase of cost depending on technology and additional devices to realise control systems and required strategies.

2.3.2 Variable Valve Timing (VVT)

VVT overs flexible intake and/or exhaust valve timing. It can be implemented mechanically by modifying the camshaft (different cam profiles), hydraulically, electro-mechanically or pneumatically. VVT is a state-of-art technology in particular for gasoline engines; recently also introduced for Diesel engines. It is applicable for/with Miller/ Atkinson Cycle, Internal EGR, thermal management of exhaust gas temperature and swirl level adaption.

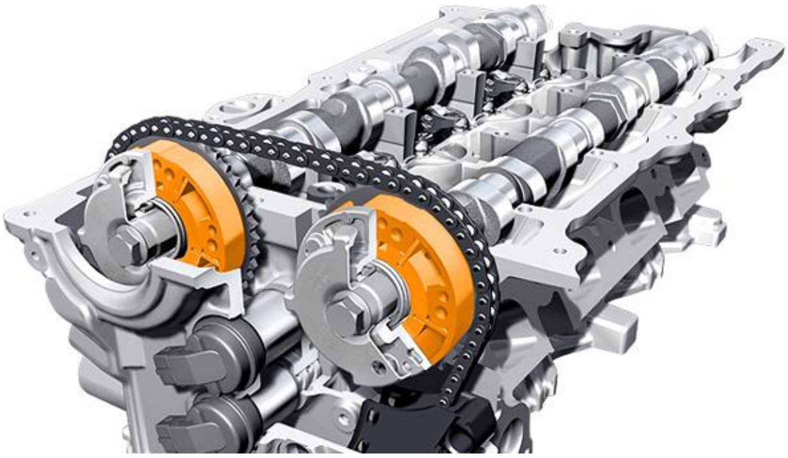


Figure 2.9 Variable Valve Timing [BMW]

Main advantages are increased torque at low engine speeds and higher maximum performance or less fuel consumption at high engine speeds. Furthermore, exhaust valve variability can be used for thermal management by increasing the exhaust temperature and thus accelerating the light-off of the catalysts (heating strategy) and varied Internal EGR to reduce NO_x emissions in Diesel engines. For Gasoline applications, VVT increases efficiency by using internal EGR and dethrottling. A VVL-system enables scavenging at low engine speed and high loads for TC DI engines to increase the low-end torque. VVT and VVL might be used and interact together.

Nevertheless, there are different disadvantages such as increasing complexity in design of the valve train, a required double overhead camshaft (DOHC) valve train and required additional devices, control systems and strategies. Moreover increased cost depending on technology and impacts on the selection of suitable valve arrangement.

2.3.3 Cam profile switching

Cam profile switching is one possibility to realize Variable Valve Timing Variable, Valve Lift and cylinder deactivation. It requires a camshaft with two different cam profiles. One cam profile provides low valve lift which is used at low to middle engine load. The other cam profile has a high valve lift and is used when the engine is spinning at mid to high engine speeds. A switch from one profile to the other is easily possible.

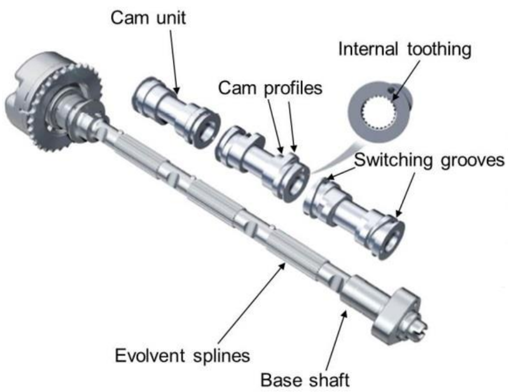


Figure 2.10 Camshaft components [Audi]

The technique improves peak performance and torque and reduces emissions and fuel consumption at low engine speeds due to lower throttling losses. There are no significant disadvantageous interdependencies with other technologies.

However additional devices, control systems and strategies are required thus slightly increased costs. Further control system for the switching procedure required.

2.4 Combustion System - Alternative Engine Cycles

Apart from the Otto and Diesel cycles, alternative engine cycles might be applied. From those, the following are considered in the project.

2.4.1 Miller/Atkinson Cycle

The Miller-Cycle describes a combustion system with early or late intake valve closing, which is here applied to gasoline engines. The highest benefits can be realized in combination with boosting and charge air cooling. The intake valve closing (IVC) is done later than usual, producing a recirculation of the available gas into the cylinder through the intake valve. The Miller cycle can be seen as an outsourcing of compression from the cylinder to the external charger. The main benefits are coming from a similar or higher compression ratio (which increase the part load operation efficiency), a lower peak pressure and temperature; and hence lower knock tendency. The Atkinson-Cycle is a combustion system with increased expansion stroke realized by an alternative crank train, which involves a system redesign.

Both Miller and Atkinson cycles have a lower effective compression ratio compared to the expansion ratio. Depending on the gasoline engine configuration, a turbocharger upgrading or installation (if NA engines) might be necessary to ensure the specific power and the low-end torque. In naturally aspirated engines, the Atkinson or Miller cycle leads directly to lower engine power and torque and this might also be compensated by an increased displacement.

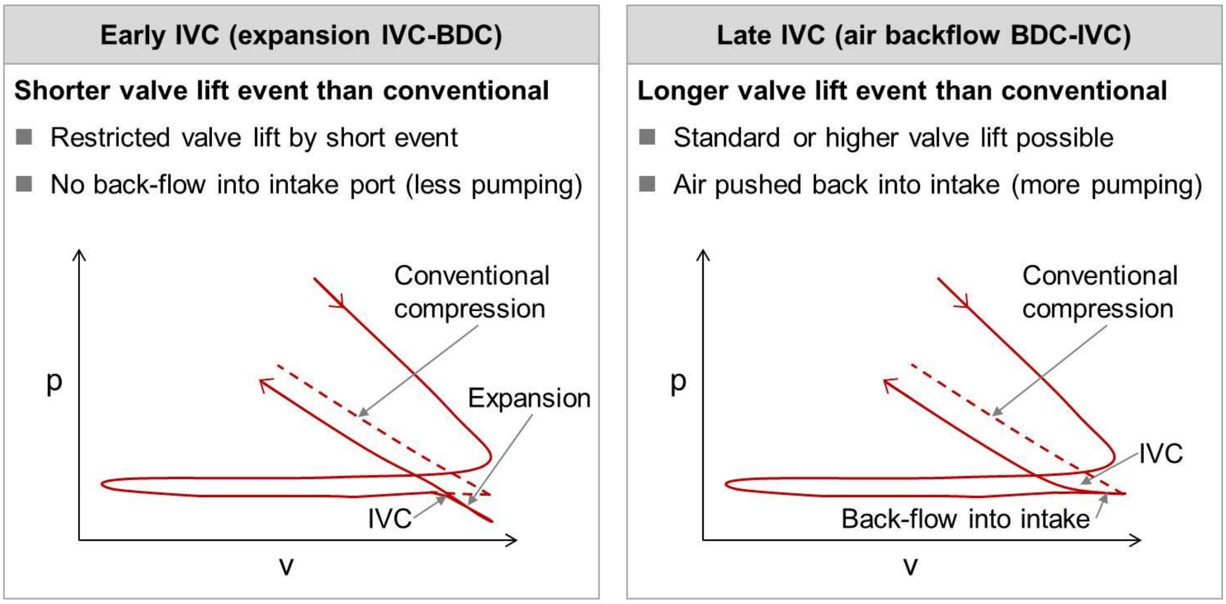


Figure 2.11 Early and Late intake valve closing in Miller Cycles [Source: FEV].

2.4.2 Two-Stage Variable Compression Ratio

The increase of the compression ratio is a natural measure in order to increase the combustion efficiency. However, this is limited by high peak cylinder pressures and temperatures, which affect to the powertrain design (friction and materials) as well as to the emissions formation (overall NO_x in diesel) and knocking tendency in gasoline engines.

Alternatively, the reduction of compression ratio helps on solving the problems mentioned before, allowing to reduce the injection pressures and reducing the overall friction and emissions, but also getting some improved powertrain efficiency at full load operation. However, this leads to cold start problems (hence requiring aggressive heating strategies), combustion stability problems and worse efficiency at part load. In the market today, there can be found in the market both tendencies of increasing and decreasing the compression ratio, requiring different side measures.

To get the best of both trends, a 2-stage compression ratio can be used for working with a higher CR at low and part load operation; and a lower CR at higher loads. With such system, an improved full load performance, reduced emissions and friction might be realized with lower compression ratios, while at lower loads the higher compression ratio allows to increase the combustion efficiency. Moreover, the compression ratio can be maximized at lower loads without redesigning the different systems which are subjected to high stresses, e.g. piston or crankshaft.

Furthermore, the cost of the 2-step VCR is much lower compared to a fully variable system and at the same time, more than the 80% of the fuel consumption reduction potential of the continuous system can be achieved in gasoline engines /Balazs, Podworny et.all “Increasing Efficiency in Gasoline Powertrains with a Variable Compression Ratio (VCR) System” . 2. Internationaler Motorenkongress 2015, Baden Baden/ In addition, there are no significant disadvantageous interdependencies with other technologies.

Although a development effort towards series application/production is required, additional oscillating masses are needed and a control strategy needs to be implemented. In addition, the increasing cost of the system should be mentioned. There are different patented methods to achieve the 2-stage VCR, the one proposed and patented by FEV for both diesel and gasoline engines is described in the Chapter 3.

2.4.3 Fully Variable Compression Ratio (VCR)

Even though not considered within the project, a fully variable VCR allows the engine to be operated at part load with high compression ratio and at high load with low CR. Possible realizations are variation of "conrod length", variable position of crank shaft etc.

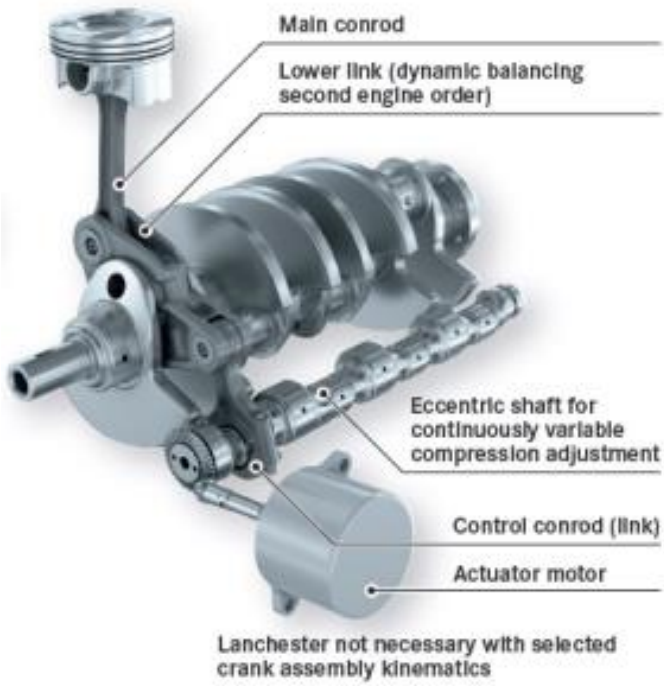


Figure 2.12 Fully variable VCR concept [Daimler]

Main advantages of this technique are further optimization potential for combustion system compared to 2-step system. The compression ratio can be completely optimized within the cycle starting with higher compression ratios at lower loads up to much lower ratios at full load. The potential is similar to the one commented before for the 2-stage system.

The fully variable CR has the possibility to adjust the CR optimal for each operation point and uses the full potential of this technology while the 2-step various system is a good compromise that shows around 80% of the potential.

However, the complexity of this system is much higher: packaging requirements, additional oscillating masses, higher friction, additional costs, higher weight, etc. These limitations also depend on the chosen configuration, since different ways of achieving variable compression ratio might be used, but these are out of the scope of this project.

2.5 Engine base technologies – Engine design

2.5.1 Downsizing

Downsizing aims to maintain power output of an engine at the same level while the displacement is reduced which means the power density increases. The result is a shift of the engine operation area towards higher specific load. This can be achieved by reducing the cylinder displacement or the numbers of cylinders. The increase of the power density may be realized by different ways: increasing the boosting level, improving the engine control and/or increasing peak firing pressure by means of higher fuel injection pressures.

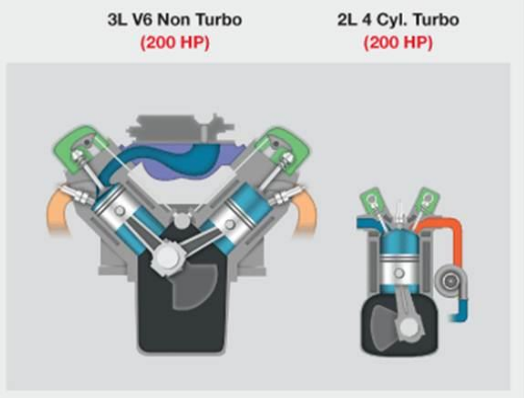


Figure 2.13 Principle of engine downsizing [Honeywell]

The main downsizing advantage is the loadpoint shift at given driving cycles to usually higher engine loads, allowing the engine to run at areas with improved thermodynamic efficiency and therefore reducing fuel consumption. The friction of the engine might also be reduced by reduction of the cylinder number. Moreover, the reduced weight and volumes accelerates the warm-up by reducing the friction in the cold phase of the legislation cycles.

This technology requires to increase the peak cylinder capability, improve the boosting to obtain higher specific power, while still assessing the durability and robustness aspects: the powertrain should work at higher pressures and support higher stresses. In gasoline engines the extended full load and operation at higher load leads to a stronger knock limitation. Countermeasures like improved charge air motion and turbulence in the combustion chamber might avoid the necessity to reduce the CR. Within the field of gasoline engines, the combination of downsizing with direct injection and boosting has improved the fuel consumption heavily in the last years.

Diesel engines reach higher specific PM/NOx-Emissions since the system needs to work at higher loads, limiting the downsizing step, i.e. at higher loads the specific NO_x emissions are higher, supported by the fact that the EGR rates are also much lower. Therefore, the downsizing require the upgrade of the deNO_x techniques: higher EGR rates up to higher loads, higher efficiency aftertreatment systems. Anyway, the average exhaust temperatures tend to increase by usually supporting higher efficiency of the aftertreatment systems and faster light-off. Nevertheless, the overall increase of the engine-out NO_x is usually not compensated by the increased efficiency, as also analysed during this project. Apart from that, an upgrade of the boosting system or redesign is required.

2.5.2 Downspeeding

Usually in combination with downsizing and transmission upgrading, the downspeeding has been established as an effective method to reduce friction by shifting the engine operation point towards lower engine speed and higher loads. This will reduce the specific fuel consumption. Downspeeding increases the demand on the maximum gear spread of the transmission and usually leads to higher demanded number of gears, especially when combined with downsizing. Today manual transmissions, automatic transmissions (AT) or dual clutch transmissions (DCT) with 7 gears or more use nearly the full potential for downspeeding. Another alternative solution for improving the engine operation point is the continuously variable transmission (CVT). Main disadvantage of the CVT is the lower transmission efficiency compared to manual transmissions, DCT and AT. The main advantage of downspeeding is a significantly reduction of the specific fuel consumption which is mainly caused by the reduced friction and improved combustion efficiency.

Key issues of this technology concentrate on lack of torque rise, requirement of adapted transmission and the trade-off between emissions and fuel consumption, especially in diesel engines. The downspeeding is also limited due to noise (NVH), and comfort reasons (drivability).

2.6 Engine base technologies – Low Friction Design

The following technologies present several benefits which mainly reduce the average friction of the engine, especially in the legislative cycles when the system starts from cold conditions. The electrification approaches optimize the pumping requirements for the coolant and oil depending on the real needs of the engine, in contrast with non-variable mechanical pumps. The combination of the electrification with split cooling, allows the system to optimize the coolant requirements through the different engine systems: the cylinder block oil and coolant circuits should reach the warm temperature as soon as possible in order to reduce the friction, while the charge air cooler is kept colder in order to keep the filling efficiency at an appropriate level. In addition, other basic measures such as crankshaft offset adjustments, bearings design or steel pistons (instead of aluminium) also helps reducing the average engine friction.

2.6.1 Electrical Water Pump

The installation of an electrical water pump allows to completely manage the pumping (to the coolant mass) needs depending on the engine conditions. If the engine requires less cooling, the pump operation is reduced, thereby lowering power consumption (demand-based flow rate control). At the same time and in combination with the split cooling, lower coolant masses and no pumping (or low pumping power) support a much faster warming up of the system, thus reducing the overall friction.

Key advantages are: Significant potential for reduction of fuel consumption (depending on operation cycle), increased component lifetime and more flexibility in component arrangement if realized as electrified auxiliary. In addition there are no significant disadvantageous interdependencies with other technologies. Nevertheless, the cost of the electrification may be mentioned.

2.6.2 Split cooling

In a standard cooling system, the coolant goes through all the system coolers (charge air and EGR), cylinder heads and engine block circuits. The separation or split of the circuits allows to optimize the coolant needs for every circuit. In combination with a fully variable water pump (as described in the previous subsection) the pumping can be adjusted for all the different circuits, even getting higher fuel reduction potential.

In the market, different splits can be found. For example, it can be found two circuits: one for the cylinder block and one for the rest of the systems (mainly air charge cooler, oil circuits, EGR coolers and cylinder heads). In such system, the pumping through the cylinder blocks when the system is cold is switched off since it is beneficial to the system reaching the warm temperature as soon as possible, thus reducing the overall friction over the system (the oil circuit is also affected). However, for the other systems, some cooling is required. The system might be realized by introducing a second pump in order to fully control both circuits, or can also be realized by using a valve which is closed when the system is cold, but later when it is warmed, it is open to have only one circuit. This is controlled by means of a thermostat.

Further benefits but with a higher cost can still be realized by splitting the system into 3 circuits as shown in the Figure 2.14: one micro-circuit for the EGR cooler with one electrical coolant pump; one connected to the cylinder heads, oil cooled and gearbox with a thermostat; and a third circuit at lower temperature for the charge air cooler with an electrical coolant pump.

When the temperature in the cylinder head reaches a critical level the coolant pump is activated with an optimized coolant volume flow (red line in Figure 2.14). The cylinder block is bypassed and only the cylinder head is steamed with coolant until the block temperature reaches a critical level and a second thermostat opens. After the complete engine is cooled by the coolant pump the coolant volume flow can be optimized continuously to reduce the electric energy consumption to a minimum (blue line in Figure 2.14). At normal operating temperature, the thermo-stat (3/2 way valve) enters the regulating mode and opens the coolant circuit in the direction of the front radiator.

The low-temperature circuit for indirect charge-air cooling (orange line in Figure 2.14) is operated independently of the high-temperature circuit, hence supporting to keep the filling efficiency at an appropriate level.

The benefit is double: a faster warm-up and lower auxiliaries power requirements. Both contribute to reduce the average friction of the engine, especially when this is cold. Moreover, keeping an appropriate coolant mass flow through the charge air cooler allows to keep the filling efficiency at a good range.

In this project, the split cooling with a separation of two circuits with one electrical water pump and one thermostat is considered as part of the optimized friction technologies.

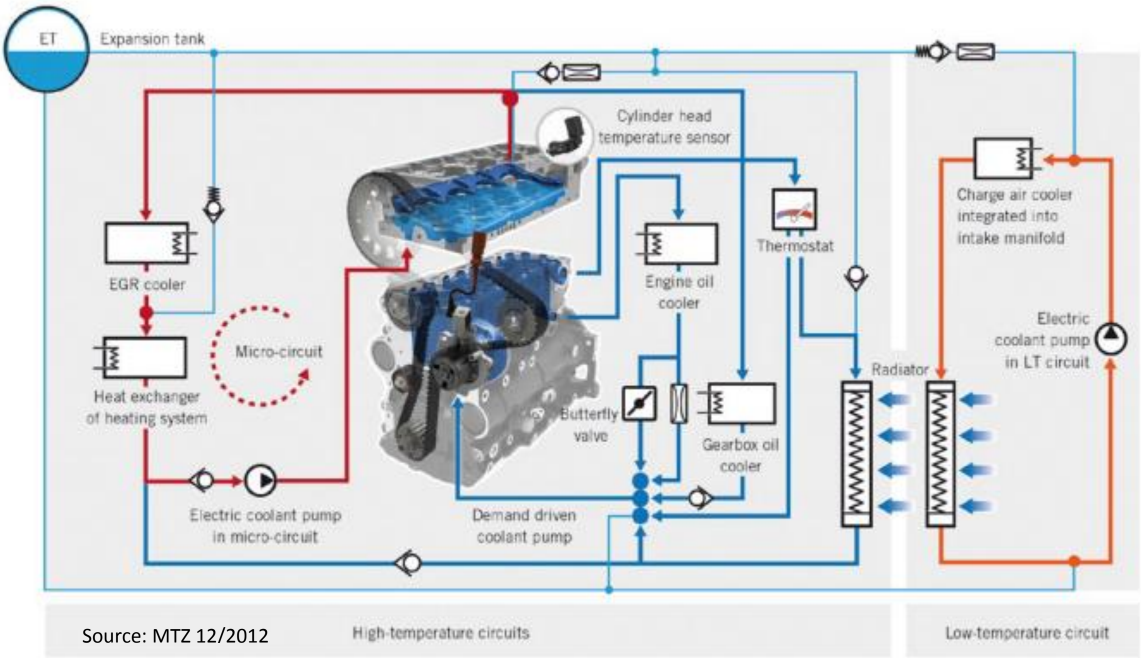


Figure 2.14 Thermal management and coolant circuit

2.6.3 Controlled Oil Pump

As with the fully variable coolant pump, the oil pumping is adapted to the oil requirements of the engine. If the engine requires lower oil through the system (at lower power stages), the pump operation is reduced and thereby lowering power consumption. There are different designs possible e.g. exterior gear pump, vane pump or pendulum-slider pump.

Figure 2.15 Section through an oil pump [BMW]

The main advantages are the reduction of fuel consumption especially at low speeds and loads, more flexibility in component arrangement if realized as electrified auxiliary and no significant disadvantageous interdependencies with other technologies. The extra cost with respect to a mechanical pump should be mentioned.

2.6.4 Other Low Friction Designs

Other basic measures in order to reduce the friction are in the direction of optimized materials and coatings, introduction of different materials and the optimization of the design. Some examples are:

- i. Crankshaft & Camshaft
  1. Advanced Crankshaft Material
  2. Camshaft with Roller Bearings
  3. Crankshaft with Roller Bearings
- ii. Steel Pistons
- iii. Coating
  1. Coated Tooth Chain
  2. Coating of Cylinder Liner
  3. Coating of Piston Ring
  4. Coatings for Camshaft and Cam Follower
- iv. FRED - Friction Reducing Sealing System
- v. Smaller bearing sizes

Nevertheless, the engine robustness and aging with such measures should be assessed. The modernization and the higher capacities of the companies in terms of computer aided engineering (CAE) allows to further optimize the final powertrain design.

2.7 Emission Management – Aftertreatment Systems

Due to the progressive introduction of more stringent regulations in terms of emissions, the development of new and/or optimized aftertreatment systems in order to reduce the tailpipe emissions is required. The most of the technologies exist in the market since years, e.g. the DPF or the DOC are standard in diesel passenger cars, the three-way catalyst in gasoline, or the SCR in heavy duty trucks. Moreover, optimized management strategies or the combination and renewal of these technologies are being applied nowadays, e.g. SCR catalysts on diesel particulate filters (SCRoF or SDPF). Hereinafter, some of the main basic technologies for aftertreatment are briefly described.

2.7.1 SCR Technology

The selective catalyst reduction (SCR) technology represents an efficient technology for NO_x reduction in diesel engines, and will be applied in the short future to lean gasoline engines (due to the progressive increase of NO_x emissions. Actual systems use urea (obtained from an aqueous solution, with commercial name AdBlue®) as reductant. Urea-SCR systems include following components: SCR catalyst, auxiliary oxidation catalysts, urea injection system, urea tank, urea refill tube and optional ammonia slip catalyst. High NO_x reductions strongly depend on the current catalyst temperature, NO_2 share and on the urea injection control strategy.

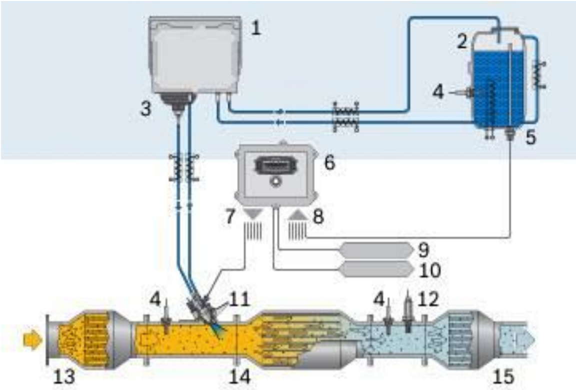


Figure 2.16 SCR System [Bosch]

Through long-term experience in commercial vehicle applications, especially in Europe, SCR systems have become highly developed. The system leads to high NO_x-conversion efficiency especially at high temperatures (higher than 250 °C). The system also presents low sensitivity to the sulphur poisoning and ageing compared to the Lean NO_x Trap (LNT) technology. This quality might be interesting in the developing countries, where the fuel quality is lower and thus the sulphur levels higher. However, the system is also considerably more expensive than the LNT systems. Moreover, the required infrastructure affects not only the cost but the packaging; the following systems are required: On-board urea storage tank, urea dosing infrastructure. Furthermore, there are some related problems which should be considered, such as urea deposit formation at medium and low temperatures, freezing of reductant and the complex transient control. Further interdependencies with the remaining exhaust aftertreatment system have also to be kept in mind.

An SCR on filter (SCRoF), also known as SDPF, introduces catalyst material over a typical DPF. This system can be located right downstream the DOC, in close-coupled position which permits to increase the average temperature of the system, hence increasing significantly the deNO_x efficiency. The urea uniformity over the system (it might be required some mixer to enhance the uniformity and hydrolysis), the robustness and the ageing should be assessed for such system.

2.7.2 Lean NOx Trap Technology (LNT)

The basic principle operation is the storage of the NO_x species into the LNT by adsorption principles which highly depends on the exhaust temperature. At the same time, desorption activities are existing in parallel. Therefore, the system presents a temperature window where the physical deNO_x trapped capacity is maximum. Nevertheless, and in order to avoid reaching the capacity limit of the LNT, a regular purging by means of running the engine at rich conditions is mandatory. The control and management of such rich regeneration modes consider different variables: storage level of the LNT, current operating point conditions, aimed deNO_x efficiency, exhaust temperature or drivability conditions, among others. The management of the rich modes is relatively complex and should be optimized depending on the final objective NO_x efficiency: higher frequency of rich modes lead to higher deNO_x efficiency but also higher fuel penalty; and the opposite. The duration of these modes lasts approximately 3-10 s and occur with a relative high frequency (between 4 and 10 times in the WLTP depending on the powertrain and required efficiencies). The LNT regeneration is achieved by adjustment of air path and injection parameters (post injection or external fuel doser).

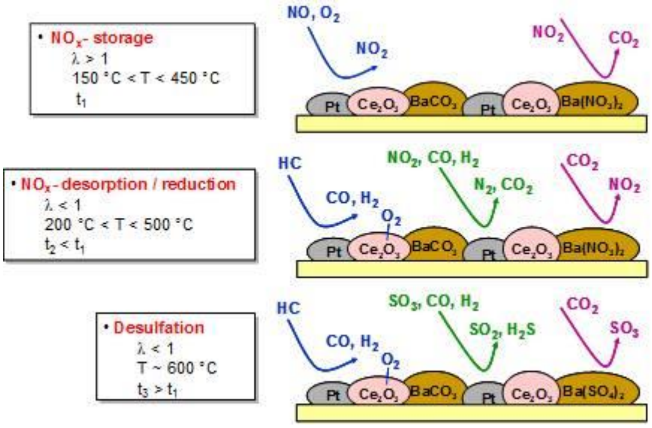


Figure 2.17 LNT processes

The main benefit of the LNT is that it requires no urea dosing infrastructure, supporting lower packaging penalties than the SCR systems. In addition, the LNT efficiency at lower temperatures is even higher than the SCR one. This system might get higher efficiencies than the SCR in cycles which present a long cold phase, such as the NEDC. Therefore, the total cost of the LNT systems with respect to SCR ones is considerably lower.

The main drawbacks are the limited deNO$_x$ efficiency at high temperatures and space velocities (SV) or sensitivity to thermal ageing. Moreover, the sulphur poisoning affects heavily the LNT activities (critical in developing countries, where cheaper deNO$_x$ solutions are preferable). The calibration of the rich modes is also worth to mention. Finally, the required PGM (platinum group metals) loading increase the cost of the LNT brick itself.

The LNT is only advisable for the smaller diesel segments; but the combination of this system with others, such as passive or active SCR solutions might be necessary for future RDE legislation when considering real driving cycles. In such conditions, the LNT efficiency drops heavily.

2.7.3 Combined LNT and SCR Systems

Another technique combines a close-coupled LNT and a downstream SCR catalyst with active urea injection. The close-coupled LNT is used for NOx storage and NOx conversion during warm-up and low load / low exhaust temperature operation. The SCR system instead offers high NOx conversion efficiency during high-load engine operation with sufficient exhaust temperature. By this setup the DOC functionalities are taken over by the LNT.

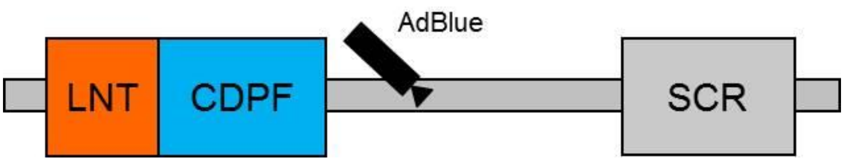


Figure 2.18 Structure of the Combined LNT and SCR System

The system offers excellent high temperature efficiency in combination with good low temperature NOx conversion efficiency, less demands for SCR heating during warm-up and low load/ low exhaust temperature operation and less urea consumption compared to pure SCR concepts.

Some negative implication are additional costs for LNT, a larger packaging due to higher LNT volume compared to DOC, rich engine operation for LNT regeneration, additional fuel consumption for LNT regeneration however mostly compensated by reduced heating demands for SCR. The combined system is interdependent with fuel injection system (necessity for LNT regeneration). This technology is only applied to diesel engines.

2.7.4 Gasoline Particulate Filter with Integrated TWC

The 3-way catalyst (3WC) coating is integrated into the gasoline particulate filter. This is a similar approach as used state-of-the-art with coated diesel particulate filters (CDPF) for diesel applications.

Figure 2.19 Gasoline Particulate Filter [NGK]

Key advantages of the technique are lower costs compared to separate bricks, package benefits compared to separate bricks and OBD benefits.

Key issues are the coating process, back pressure performance and the 3WC conversion efficiency.

The technology is only applicable for gasoline automobiles.

2.8 Electrification - Energy Storage & Hybrid Classification

With respect to increasing substitutional potential of existing fossil technologies also the automobile sector has expanded its fleet to the electrical mobility. Key technologies and storage systems are introduced in this chapter.

2.8.1 Battery Technology

To use brake energy recuperation as one of the key features of hybrid powertrains, an electrical energy storage is required. A high voltage (HV) traction battery is able to store more electrical energy than a conventional 12V battery. Most common battery types are nickel metal hydride batteries and Li-Ion batteries. A Li-Ion battery has a power density of more than 1 kW / l and an energy density of more than 60 Wh / l. The HV battery must be connected thru a high-voltage, high-amperage cable to the converter.

Figure 2.20 Electric Vehicle Battery [Toyota Prius]

Due to the available energy of the HV battery, auxiliaries can be supplied with high voltage power so that they can be driven electrical and thus demand-oriented.

As negatives consequences mechanical impact protection is required, much lower energy density compared to liquid fuels is provided, weight is increased and cooling of the battery might be necessary. Interdependencies with the vehicle electric system are also to be considered.

2.8.2 48V On-Board Power Supply

The 48V/12V dual voltage system substitutes the 12V On-board power supply. It enables mild hybrid system and supports the handling of higher power demands of a growing list of vehicle features and functionality including heated seats and steering wheels, active chassis systems, and electric power steering and air conditioning. The system also supports usage of electric components such as electric boosting systems or electric heated catalysts. The 12-V (nominal) circuit is used to retain a conventional lead-acid battery to handle engine starting and the vehicle’s equipment including the lighting and infotainment systems.

The system allows more recuperation of braking energy, higher efficiency because it can use a lighter-gauge wiring harness and lower cost due to less copper usage.

2.8.3 Full Hybrid & Mild Hybrid

A hybrid vehicle is a vehicle that uses two different traction systems which means it consists of at least two different energy storage systems and two energy converters. Usually the traction systems are a combustion engine with fuel tank and an electric motor with battery. A Full Hybrid is a hybrid vehicle which is able to drive pure electric using the electric motor. Imaginable system structures are serial, parallel or power-split. The Mild Hybrid is a hybrid vehicle with an electric motor which is not able to drive the vehicle pure electric by itself but supports the combustion engine. Imaginable system structures are parallel or power-split. In hybrid vehicles the electric machines can be mounted at different positions to allow various functionalities.

Mild hybrid powertrains are usually realized by means of P0 and P1 configurations, even though other layouts like P2 are possible. In P2 configuration the integration effort and the costs are higher. The electric machine is in the P0 configuration coupled to the crankshaft via a belt. This configuration is also known as Belt Starter Generator (BSG). In the P1 configuration the electric machine is directly coupled to the crankshaft by a tooth wheel or a chain. The P1 configuration allows the transmission of higher maximum torque compared to the belt driven electric machine. Due to that fact the maximum power for recuperation is higher.

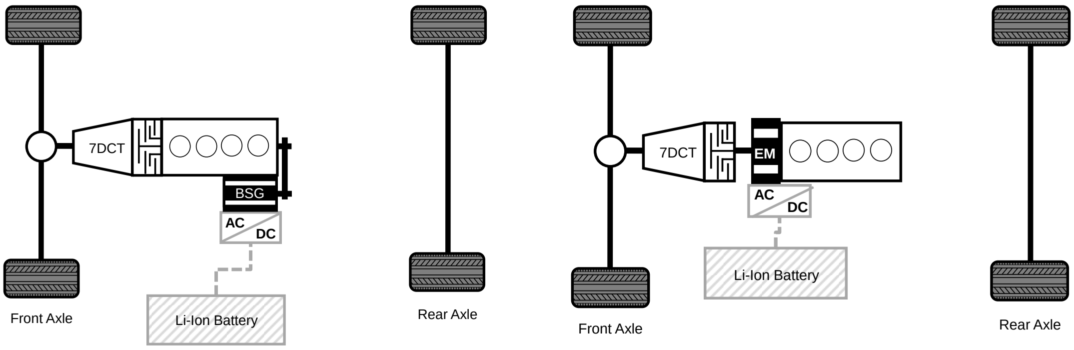


Figure 2.21 Mild Hybrid setup (left P0 BSG, right P1)

The best cost to efficiency ratio of full hybrid architectures has the P2 parallel hybrid. In the P2 configuration is the electric machine located in the transmission between two clutches. That allows the functionality of pure electric driving with the electric machine that has higher fuel consumption saving potential than boosting due to substitution of the worst efficiency regions of the combustion engines in low part load.

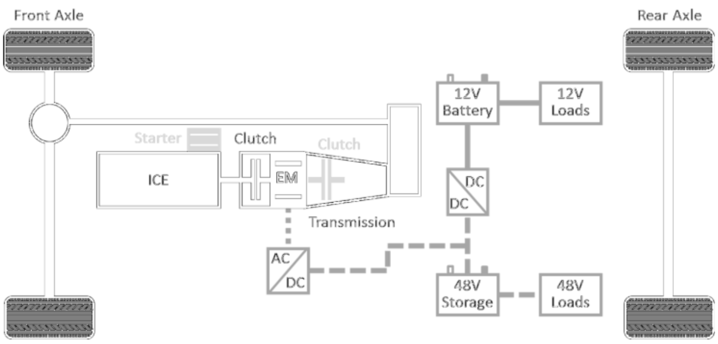


Figure 2.22 Full Parallel Hybrid setup (P2)

In both configurations disadvantageous engine operating points can be avoided which leads to a lower fuel consumption. It also offers less noise emissions, extended range compared to an electric vehicle due to the combustion engine range.. Another advantage is the possibility to recuperate breaking energy.

Key issues are much higher costs compared to a vehicle powered only by a combustion engine, influence of packaging/periphery due to the increased required space, increased weight and interdependencies with the combustion system and exhaust aftertreatment system.

2.8.4 Other configurations not considered within the project

2.8.4.1 Electric Vehicle

An electric vehicle has a completely electrified drive train. The required components are a HV battery, a high voltage electric system, an inverter and at least one electric motor. The battery is charged through an external electric-power supply. Several models are in series production, e.g. Renault Fluence Z.E., BMW ActiveE, Ford Focus Electric.

Figure 2.23 Electric Vehicle [Renault Fluence Z.E.]

The electric vehicle offers no local emissions, less noise emissions, low operating costs assuming durable batteries, a potential of decreasing greenhouse gas emissions when renewable energy is used to charge the battery and no significant interdependencies with other technologies.

However the electric vehicle has higher costs compared to a vehicle powered by a combustion engine. It has a short range due to the much lower energy density of the battery compared to liquid fuels. Furthermore, the electric vehicle has long charging duration and increased weight.

2.8.4.2 Plug-In Hybrid (parallel)

A Plug-In Hybrid has a similar component setup as the full and mild hybrid. It is a hybrid vehicle which can be charged by an external power source and uses mainly the electric motor. The combustion engine is used to extend the range ("Range Extender").

Figure 2.24 Plug-In Hybrid [Suzuki Swift Plug-in Hybrid]

Main advantages are low operating costs (assuming durable batteries are used), less local emissions and less noise emissions, decrease of the greenhouse gas emission when renewable energy is used to charge the battery as well as an extended range compared to an electric vehicle.

Main disadvantages are the same as for the full and mild hybrid.

2.8.4.3 Range Extender (serial)

A Range Extender is additional aggregate in an electric vehicle for cruising range elongation as a modular component of a serial hybrid concept. It offers the possibility to on-board recharge the battery via generator. A range extender is typically installed with small gasoline engine or alternatively a fuel cell system. The cost and weight compared to enlarge the battery are normally lower by installing the range extender.

Figure 2.25 Range Extender [KPSG, FEV]

Main drawbacks of this technology are NVH performance, range extender module costs, packaging and possibly the interdependencies with the combustion system and exhaust aftertreatment system.

2.9 Transmission Types

The task of the transmission is to transfer the torque of the internal combustion engine to the drive wheels at a proper speed. The transmission reduces the high engine speed to the lower wheel speed by increasing the torque. Two technologies are described in this chapter the Dual Clutch Transmission (DCT) and the Automatic Transmission (AT).

2.9.1 Automatic Transmission (AT)

The AT is a (globally) wide-spread solution for the pickup and SUV segment e.g. due to torque overshoot at start-up. The gear ratios of an automatic transmission are realized via planetary gears. Synchronization is realized via multi-plate clutch, multi-disk brakes, etc. The start-up element is usually a torque converter.

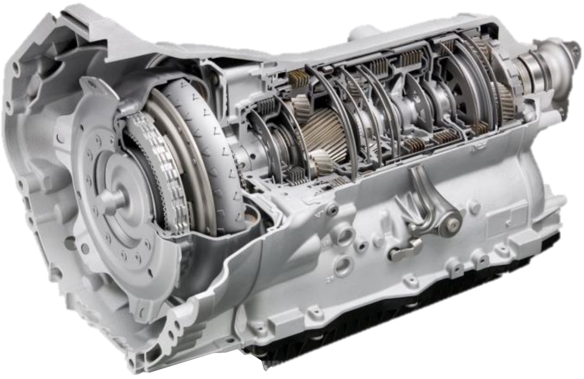


Figure 2.26 Section through an Automatic Transmission [ZF]

The main benefits are: good launch and engine take-off and the comfort (by means of a smooth ride). These transmissions are widespread for the heavier vehicles, especially in the US-market. It also enhances hybridization potential.

However, a system for supplying oil for the actuators and torque converter is required. In the future, it is expected to get enhanced and optimized control of the transmissions to optimize the oil needs, especially at lower loads, where the torque converter gets lower efficiencies.

Other future technology trends of AT are:

- Improved gearing leads to increased ratio spread, smaller ratio steps, less open shift elements and less mechanics.
- Enhanced feedback control leads to less calibration effort and self-adapting and torque sensors to improve the robustness and shift quality.
- Reduced friction is realized through new bearing concepts, low-loss gearing, bearings, seals and drag loss reduction.
- Future actuation focuses on a 2 stage/variable pump, hydraulics optimization (which can lead to significant fuel reductions), electro-mechanic and shift by wire.
- Improved coupling elements are Zero slip torque converters, damping, dog clutches, one way elements and active disconnection of friction elements.

2.9.2 Dual Clutch Transmission (DCT)

DCT combines almost the comfort of a conventional AT with the dynamics of an MT. DCT comprises two independent and separate transmissions. The dual clutch connects both transmissions with the engine via two driving shafts in a force-locking manner. Furthermore, shift events are realized without interruption of traction. The TM is operated via a mechatronic module (incl. ECU, sensors, etc.). For the smaller segments which require less power needs, a dry dual clutch is enough (no oil requirements), while higher power versions require a hydraulic system realizing a “wet” dual clutch.

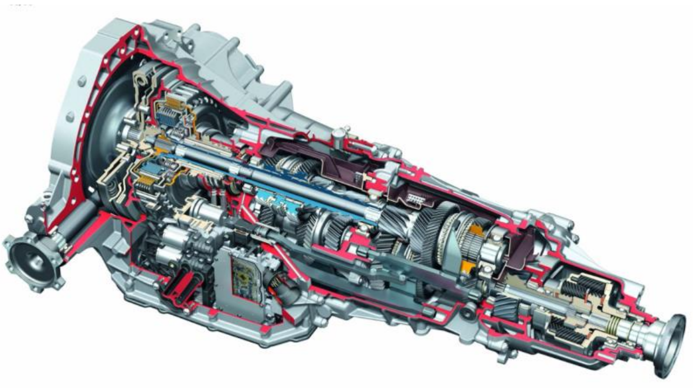


Figure 2.27 Section through a Dual Clutch Transmission [Audi]

The main benefits are modern and sportive reputation, increased fuel efficiency, less emission, high potential for hybridization and the combination of fuel economy, sport and automatic.

The challenges and potential of improvement for such transmissions are similar to those of the ATs: still becoming more competitive regarding costs, the shift quality improvement, better launch performance and ongoing fuel economy enhancement.

2.9.3 Continuous Variable Transmission (CVT)

Continuous variable transmissions use a variator in combination with an torque converter to realize fully variable gear ratios in a wide range. A torque converter is required for the drive away.

The advantage of the CVT is the optimization of the engine operation points for optimal efficiency. With the CVT the engine can be operated on the maximum efficiency curve with the highest efficiency for each output power.

The disadvantage is the relative low efficiency of the transmission in comparison to manual transmissions or dual clutch transmissions (DCT).

CVT were used mainly in gasoline powertrains in US and Japanese market und have low market penetration in Europe. Therefore the CVT is not further investigated in this study.

3 Component and Cost Analysis

This Chapter provides accurate and up-to-date powertrain assessment through a detailed cost calculation analysis that compares the main technologies and variants considered in the project. The Figure 3.1 shows an overview of the analysed technologies within all segments and both for Gasoline and Diesel. Hereinafter these technologies are described in detail by considering a general description, analysed components, scaling methodology and final cost estimation. For an interpretation of the cost estimations it is important to consider that the project approach focuses the year 2030 as scenario for the cost analysis. Especially for the technics VCR and hybridization which don't reach the level of series production yet. The cost estimation approach is based on a determination of considered hardware for the different technologies. Based on that, cost estimations for the different vehicle segments are executed with corresponding scaling methodologies.

		Engine size	EGR	T/C variation	Valve-train	Compr.-ratio	Engine friction	Hybridi-zation	Trans-mission	After-treatment
Diesel	B	✓	✓		✓		✓	✓	✓	✓
	C	✓	✓		✓	✓	✓	✓	✓	✓
	D	✓	✓		✓	✓	✓	✓	✓	
	E	✓	✓	✓	✓	✓	✓	✓	✓	
	SUV	✓	✓		✓	✓	✓	✓	✓	
	LCV	✓	✓	✓	✓	✓	✓	✓	✓	
Gasoline	B	✓	✓	✓	✓		✓	✓	✓	✓
	C	✓	✓	✓	✓	✓	✓	✓	✓	✓
	D	✓	✓	✓	✓	✓	✓	✓	✓	✓
	E	✓	✓	✓	✓	✓	✓	✓	✓	✓

Figure 3.1 Overview of analysed technologies.

3.1 Cost estimation EGR Diesel engines

3.1.1 Description and hardware determination

For Diesel engines two base systems have been selected.

For segments B & C, an uncooled high pressure EGR (HPEGR) in combination with a cooled low pressure EGR is base system. The high-pressure EGR route starts at the upstream TC turbine to downstream intercooler body (In some engines, it can end right downstream the throttle body depending on the configuration). The low-pressure EGR (LPEGR) route depends on the final aftertreatment configuration:

- In engines with LNT+CDPF, the LPEGR usually starts right downstream the DPF system.
- In engines with DOC+SDPF (close-coupled installation), the LPEGR starts right downstream the SDPF system.
- In general, there exist many other possible aftertreatment configurations, where the EGR can be located in between the aftertreatment systems. For example, in engines with DOC + DPF + underfloor SCR systems and depending on the configuration, the LPEGR route can start downstream the DPF and upstream the SCR. Nevertheless, the two layouts depicted above have been selected and used for the project

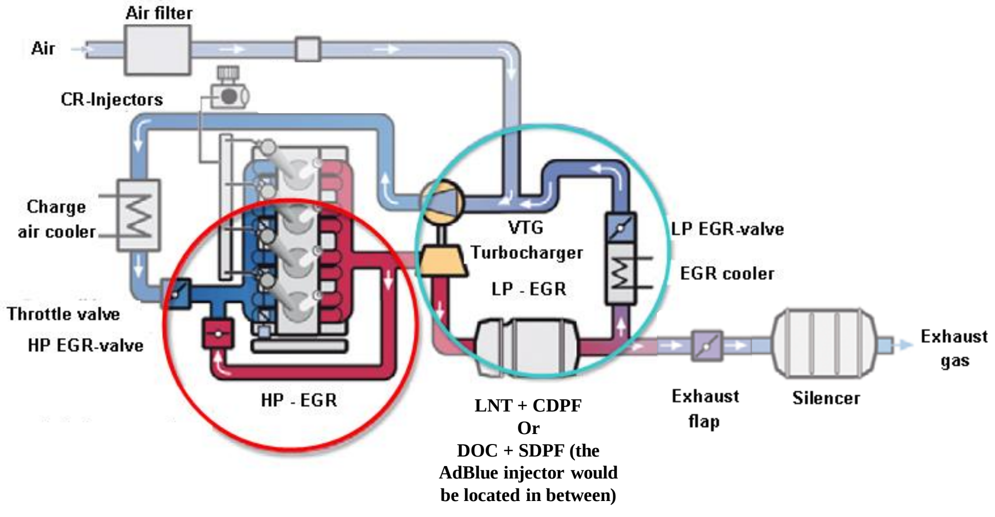


Figure 3.2 Uncooled high-pressure EGR & cooled low-pressure EGR

For segments D, E, SUV & LCV, a cooled high-pressure EGR is base system. The route starts upstream of the turbine to downstream charge air cooler TC compressor. This EGR system is shown in Figure 3.3

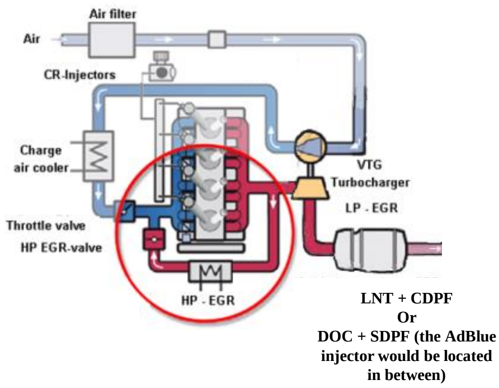


Figure 3.3 Cooled high-pressure EGR

As EGR variant for all segments a combined cooled high-pressure EGR & cooled low-pressure EGR is defined. The route remains the same as for the uncooled HP-EGR/cooled LP-EGR system. A high-pressure cooler is added if necessary. The system is shown in Figure 3.4

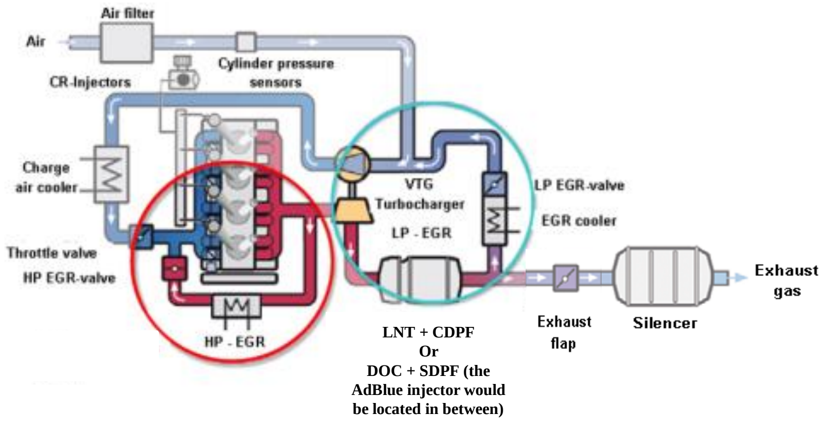


Figure 3.4 Cooled high-pressure EGR & cooled low-pressure EGR

Table 3.1 shows the considered EGR variants for Diesel vehicle classes:

Segment	P [kW]	Base	Variant
B	60	Uncooled HP/cooled LP	Cooled HP/cooled LP
C	80	Uncooled HP/cooled LP	Cooled HP/cooled LP
D	110	Cooled HP	Cooled HP/cooled LP
E	150	Cooled HP	Cooled HP/cooled LP
SUV	150	Cooled HP	Cooled HP/cooled LP
LCV	120	Cooled HP	Cooled HP/cooled LP

Table 3.1 Overview of selected EGR systems for Diesel engines

3.1.2 Components of EGR systems and cost influencing parameter

The EGR systems are split into four groups for presenting the system cost of each segment: the EGR cooler (high-pressure, low-pressure), the EGR valve (high-pressure, low-pressure), the pipes and miscellaneous. The components of each group and the cost influencing parameter for each part are shown in Table 3.2.

Part or component group	Components	Cost influencing parameter
EGR cooler high-pressure/low-pressure	EGR water-cooled cooler (high pressure/low pressure)	Cooler housing and flanges are assumed as not variable cooling elements are scaled by engine power
EGR valve high-pressure/low-pressure	EGR Valve (high pressure/low pressure)	Cost for high-pressure and low-pressure valve are assumed as not variable
EGR pipes	EGR exhaust gas pipes (inlet, outlet) Water pipes for cooling (inlet, outlet)	Cost for EGR pipes are assumed as not variable
EGR misc.	Gaskets, screws Assembly Modifications at waterpump Modifications at watercooler Modifications at intercooler Modifications at ECU	Modifications at water pump, watercooler and intercooler are scaled by engine power. Cost for others are assumed as not variable

Table 3.2 EGR component groups and cost influencing parameter

3.1.3 Vehicle segment scaling methodology

At the EGR system the majority of the components are assumed to be independent from engine power or other technical parameters and therefore the cost for these parts are assumed to be constant for all variants. Otherwise, there are some component costs which are influenced by the required cooling power for the recirculated exhaust gas, which is depended on engine power. The cost of these parts - especially the cooling elements of the EGR-cooler, the water pump, the water cooler and the intercooler – are proportionally scaled by the power of the vehicle of each segment.

Table 3.3 gives an overview of the scaling factors for Diesel vehicle segments:

Part	B [60kW]	C [80kW]	D [110kW]	E [150kW]	SUV [150kW]	LCV [120kW]
Cooler housing & flange	100%	100%	100%	100%	100%	100%
Cooler cooling elements	55%	73%	100%	136%	136%	109%
EGR valve	100%	100%	100%	100%	100%	100%
EGR pipes	100%	100%	100%	100%	100%	100%
Water pump, water cooler, intercooler	55%	73%	100%	136%	136%	109%
Others misc.	100%	100%	100%	100%	100%	100%

Table 3.3 Overview of the EGR technology scaling factors for Diesel segments

3.1.4 Cost estimation result EGR systems Diesel segments

Figure 3.5 shows the delta cost estimation of 39€ for upgrading a uncooled HPEGR/cooled LPEGR with a cooled HPEGR/ cooled LPEGR for Diesel engines in segment B. Main cost driver is the additional cooler. Other costs result from additional water pipes and a slightly higher effort in assembly.

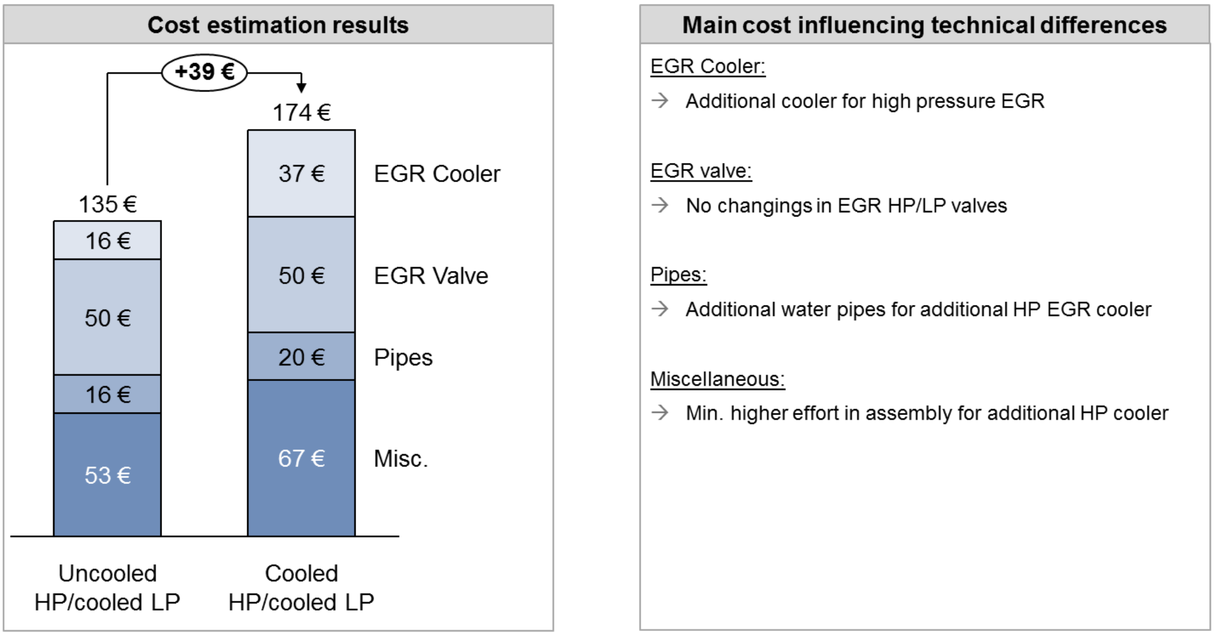
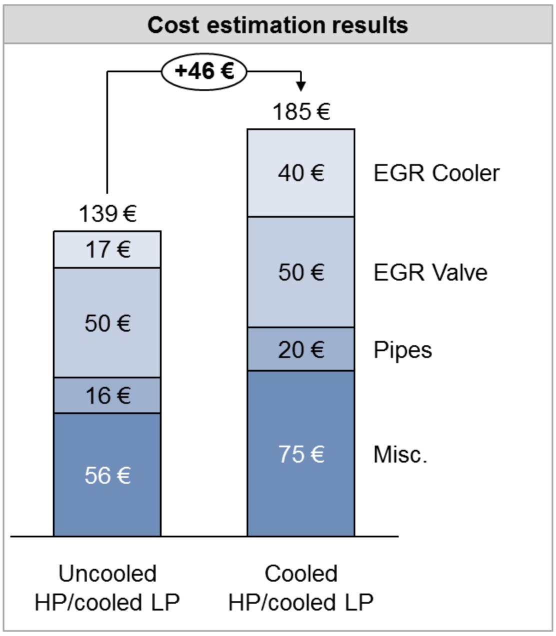


Figure 3.5 Cost estimation, EGR technology, Diesel segment B

For segment C the cost difference is slightly higher (46€) than for segment B caused by the higher power of the engines.



Main cost influencing technical differences

EGR Cooler:
→ Additional cooler for high pressure EGR

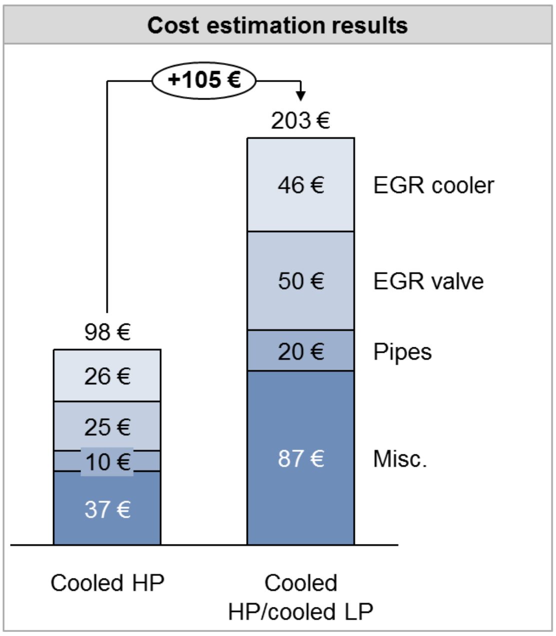
EGR valve:
→ No changings in EGR HP/LP valves

Pipes:
→ Additional water pipes for additional HP EGR cooler

Miscellaneous:
→ Min. higher effort in assembly for additional HP cooler

Figure 3.6 Cost estimation, EGR technology, Diesel segment C

In Segments D, E, SUV and LCV, a cooled high-pressure EGR is replaced by a cooled high-pressure/cooled low-pressure EGR. In this case an additional low-pressure EGR cooler is needed as well as a low-pressure EGR valve. Further parts are pipes, gaskets, an exhaust flap and modifications at the water pump, the water cooler and the intercooler due to the higher cooling effort. Together these parts lead to higher cost of 105€ at Diesel segment D, as shown in Figure 3.7.



Main cost influencing technical differences

EGR cooler:
→ Additional LP EGR cooler

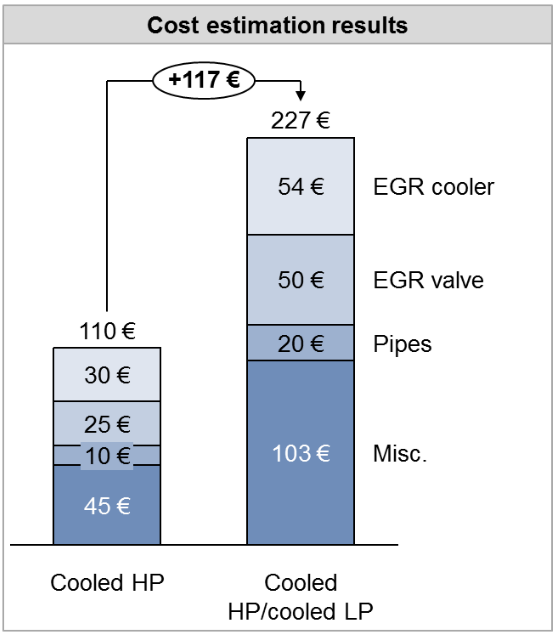
EGR valves:
→ Additional EGR valve for low pressure

Pipes:
→ Additional pipes for LP water and EGR flow

Miscellaneous:
→ Additional exhaust flap, additional gaskets, screws etc. for LP, additional assembly for LP, modifications at water pump, water cooler, intercooler

Figure 3.7 Cost estimation, EGR technology, Diesel segment D

For the segment E and SUV, the cost difference for changing the EGR from cooled high-pressure to cooled high-pressure/cooled low-pressure is slightly higher than for segment D due to higher engine power. Figure 3.8 shows the delta cost for these segments.



Main cost influencing technical differences

EGR cooler:
→ Additional LP EGR cooler

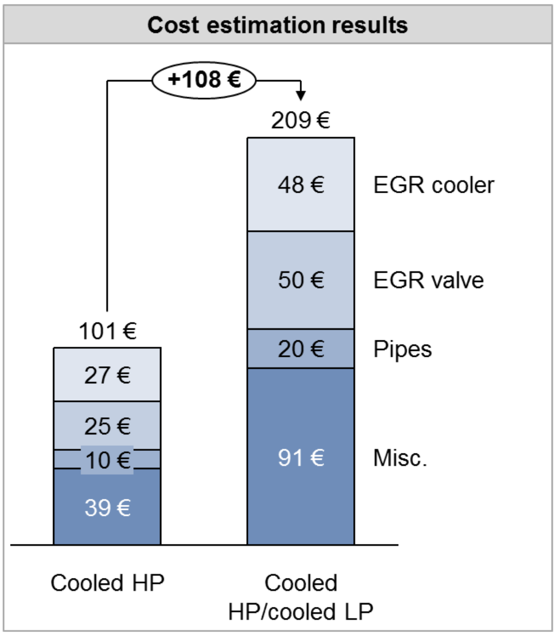
EGR valves:
→ Additional EGR valve for low pressure

Pipes:
→ Additional pipes for LP water and EGR flow

Miscellaneous:
→ Additional exhaust flap, additional gaskets, screws etc. for LP, additional assembly for LP, modifications at water pump, water cooler, intercooler

Figure 3.8 Cost estimation, EGR technology, Diesel segments E & SUV

For segment LCV the delta costs are a little bit lower compared to E and SUV because of the lower engine power. Figure 3.9 shows the delta cost for segment LCV.



Main cost influencing technical differences

EGR cooler:
→ Additional LP EGR cooler

EGR valves:
→ Additional EGR valve for low pressure

Pipes:
→ Additional pipes for LP water and EGR flow

Miscellaneous:
→ Additional exhaust flap, additional gaskets, screws etc. for LP, additional assembly for LP, modifications at water pump, water cooler, intercooler

Figure 3.9 Cost estimation, EGR technology, Diesel segment LCV

An overview for EGR technology changings at Diesel engine is given in Table 3.4

Segment	P [kW]	Base	Variant	Delta cost
B	60	Uncooled HP/cooled LP	Cooled HP/cooled LP	39 €
C	80	Uncooled HP/cooled LP	Cooled HP/cooled LP	46 €
D	110	Cooled HP	Cooled HP/cooled LP	105 €
E	150	Cooled HP	Cooled HP/cooled LP	117 €
SUV	150	Cooled HP	Cooled HP/cooled LP	117 €
LCV	120	Cooled HP	Cooled HP/cooled LP	108 €

Table 3.4 Cost estimation, EGR technology, Diesel segments, overview

3.2 Cost estimation EGR gasoline engines

3.2.1 Description and hardware determination

For all gasoline segments the base engines are not equipped with an external EGR system, while the modified engines of all segments consider a cooled low-pressure EGR.

The route starts downstream catalyst to upstream TC compressor. The cooled low-pressure system is shown in Figure 3.10.

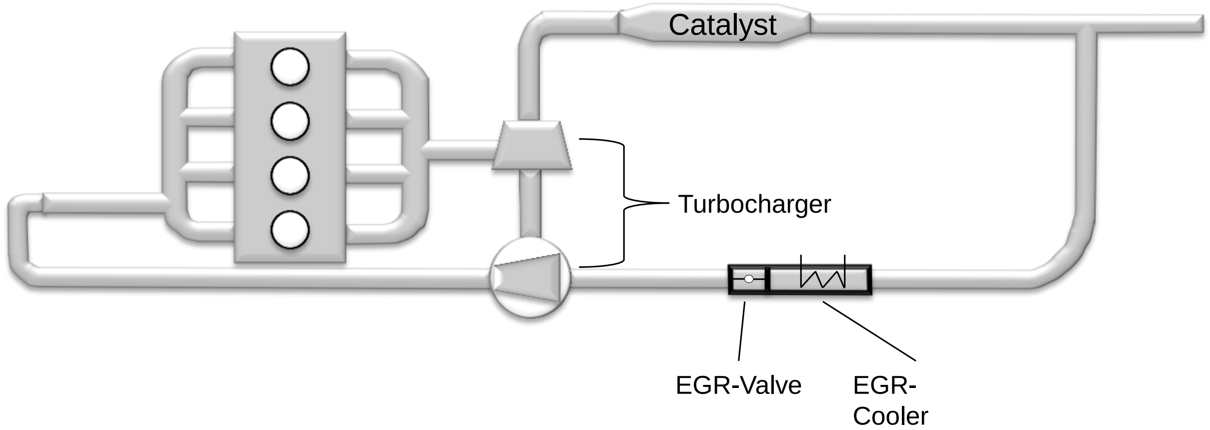


Figure 3.10 Cooled low-pressure EGR

As an additional variant for segment D and E a dedicated EGR was selected instead of the low-pressure EGR. In this technology one cylinder is producing the EGR for all four cylinders of the engine. This happens with a split exhaust manifold. In addition to the high-pressure EGR loop and EGR cooling a supercharger with a bypass valve and an after-cooler is used. A cold start valve was installed as alternative path for exhaust gases when the EGR valve is closed. A PFI injector is added to the intake manifold which allows flexibility in how the extra fuel to the 4th cylinder is delivered. Figure 3.11 shows the schematic of the dedicated EGR system.

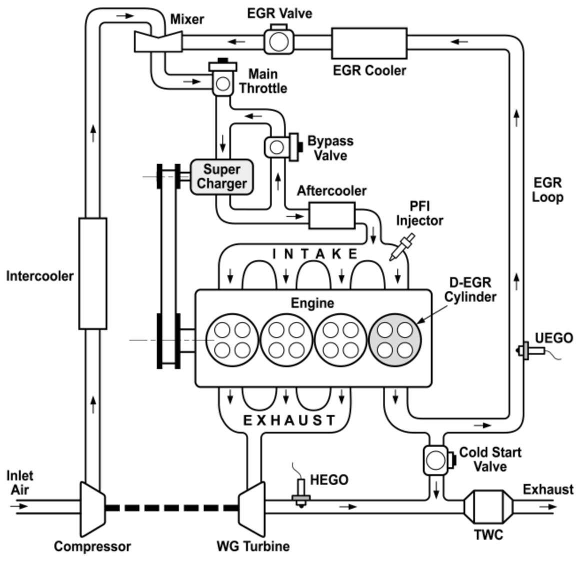


Figure 3.11 Schematic of the dedicated EGR system (Source: SwRI)

Table 3.5 gives an overview of the EGR system variants for gasoline engines:

Segment	P [kW]	Base	Variant 1	Variant 2
B	65	w/o EGR	Cooled LP	-
C	95	w/o EGR	Cooled LP	-
D	135	w/o EGR	Cooled LP	Dedicated EGR
E	180	w/o EGR	Cooled LP	Dedicated EGR

Table 3.5 Overview of selected EGR systems for gasoline engines

3.2.2 Components of EGR systems and cost influencing parameter

The EGR systems are split into four groups for presenting the system cost of each segment: the EGR cooler (HP/LP), the EGR valve (HP/LP), the pipes and miscellaneous. Differences to the Diesel technology are an additional modification of the crankcase (water channel) due to a not existing EGR at the base engine. Furthermore is the effective cooler length assumed to be 50% longer than the Diesel EGR cooler. The components of each group and the cost influencing parameter for each part are shown in Table 3.6.

Part or component group	Components	Cost influencing parameter
EGR cooler high-pressure/low-pressure	EGR water-cooled cooler (high pressure/low pressure)	Cooler housing and flanges are assumed as not variable Cooling elements are scaled by engine power
EGR valve high-pressure/low-pressure	EGR Valve (high pressure/low pressure)	Cost for high-pressure and low-pressure valve are assumed as not variable
EGR pipes	EGR exhaust gas pipes (inlet, outlet) Water pipes for cooling (inlet, outlet)	Cost for EGR pipes are assumed as not variable
EGR misc.	Gaskets, screws Assembly Modifications at water-pump Modifications at watercooler Modifications at intercooler Modifications at ECU Modifications at crankcase	Modifications at water-pump, watercooler and intercooler are scaled by engine power. Cost for other parts are assumed as not variable

Table 3.6 EGR component groups and cost influencing parameter

The dedicated EGR system was split into six components groups. The groups and the components of each group are shown in Table 3.7.

Component group	Components
Supercharger	Roots-supercharger
EGR control	EGR Valve Cold start valve
Boosting system	Supercharger drive Bypass valve Piping bypass After-cooler
EGR cooler	EGR cooler
Piping	EGR loop piping Mixer Modification at exhaust manifold
Others	PFI injector Modification at intake manifold Assembly Modification at water-pump Modification at water-cooler Modification at intercooler

Table 3.7 Dedicated EGR component groups and components

3.2.3 Vehicle segment scaling methodology

At the gasoline EGR system, like at the Diesel EGR calculation, most of the components are assumed to be independent from engine power. Other components (especially the cooler systems), which are affected by the engine power, are calculated according to Diesel EGR calculation. Table 3.8 gives an overview of the scaling factors for gasoline segments:

Part	B [65kW]	C [95kW]	D [135kW]	E [180]
Cooler housing & flange	100%	100%	100%	100%
Cooler cooling elements	59%	86%	123%	164%
EGR valve	100%	100%	100%	100%
EGR pipes	100%	100%	100%	100%
Water pump, water cooler, intercooler	59%	86%	123%	164%
Others misc.	100%	100%	100%	100%

Table 3.8 Overview of EGR technology scaling factors for gasoline segments

3.2.4 Cost estimation result EGR systems gasoline segments

For segment B, the cost difference between an engine without EGR and with a cooled low-pressure EGR is 93€. The main cost influencing technical differences are the additional cooler for LP-EGR, the additional EGR valve, the additional pipes and the modifications for the water-cooler, the water-pump and the intercooler. Higher engine power leads to higher cooling capacity and therefore to higher part cost. Hence the gasoline LP EGR cost increases from segment B to segment E. The EGR cooler for gasoline engines needs higher cooling capacity due to the higher exhaust gas temperatures than Diesel EGR cooler and that leads directly to higher cost compared to the Diesel engines. Figure 3.12 shows the cost split for all gasoline segments.

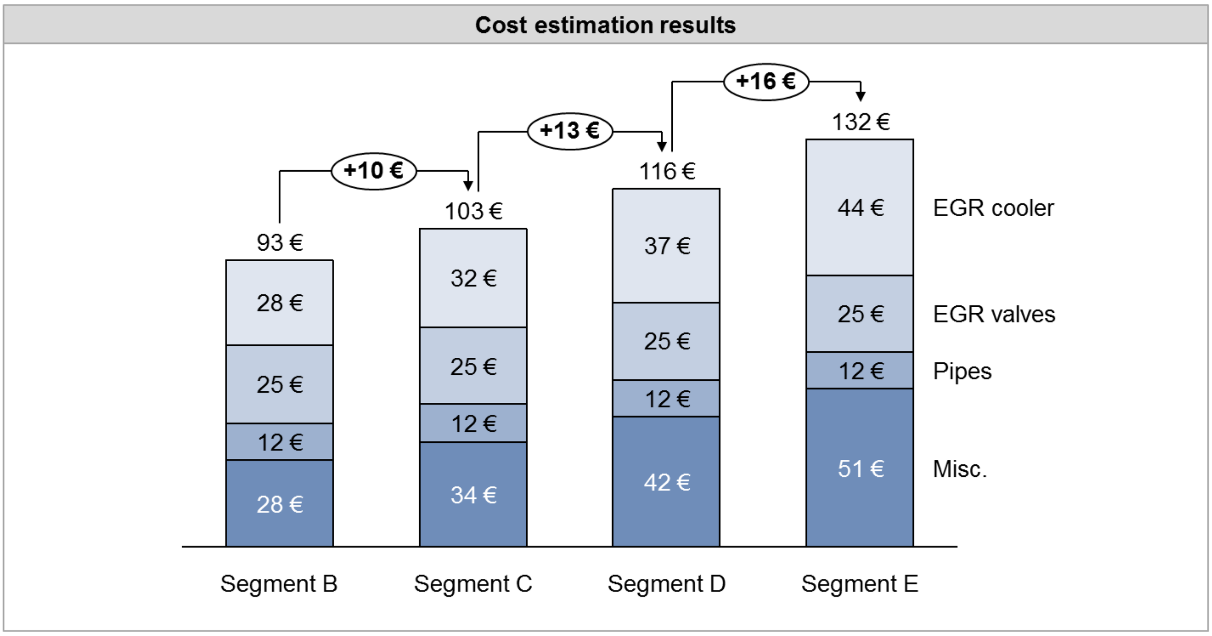
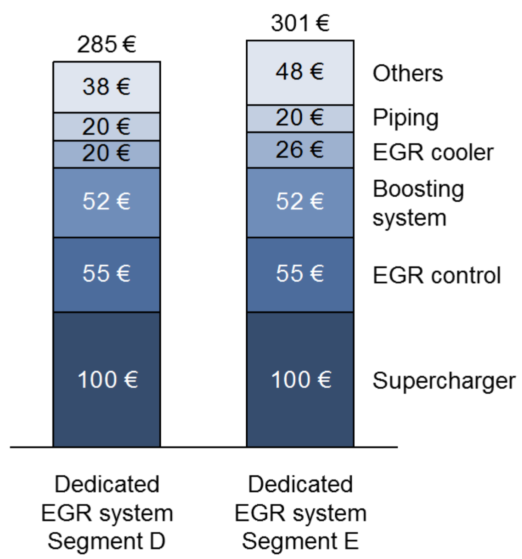


Figure 3.12 Cost estimation, EGR technology, gasoline segments. The differences are driven by the total engine power, which affects the miscellaneous and EGR cooler costs.

At the dedicated EGR system the roots-supercharger is the main cost driver. The EGR control parts, consisting of EGR valve and cold start valve, amount to the next highest cost part. The boosting system which includes the supercharger drive, the bypass valve and piping and the extra after-cooler leads to additional 52€. Costs for the EGR cooler and the modifications at water-pump, water-cooler and intercooler rise slightly from segment D to E due to the higher power, similar to the LP-EGR scaling. An overview of the dedicated EGR costs for segments D and E at gasoline engines is shown in Table 3.9.

Cost estimation results



Main cost influencing parameter

Supercharger:
- → Roots-supercharger

EGR control:
- → EGR valve
- → Cold start valve

Boosting system:
- → Supercharger drive
- → Bypass valve and piping
- → Aftercooler

Others:
- → Modifications at manifolds, water pump, water cooler, intercooler
- → PFI injectors

Table 3.9 Cost estimation, EGR technology dedicated EGR, gasoline engines

An overview for EGR technology changings at gasoline engine is given in Table 3.10

Segment	P [kW]	Base	Variant 1	Delta cost	Variant 2	Delta cost
B	65	w/o EGR	Cooled LP	93 €	-	-
C	95	w/o EGR	Cooled LP	103 €	-	-
D	135	w/o EGR	Cooled LP	116 €	Dedicated EGR	169 €
E	180	w/o EGR	Cooled LP	132 €	Dedicated EGR	169 €

Table 3.10: Cost estimation, EGR technology, gasoline segments, overview

3.3 Cost estimation transmission

For the cost estimation of the various vehicle segments three kinds of transmissions are considered: manual transmission (MT), automatic transmission (AT) and dual clutch transmission (DCT). In addition, the number of selected gears ranges from 5 and 6 in the case MT, up to 7, 8 or 10 for the AT and DCT. Hereinafter a short description of all transmissions is provided.

3.3.1 Description and hardware determination

In this section the hardware and technical realization of the different transmissions is described. Parts of this description are the number and order of shafts and gears or the design of the synchronization. Also the design of the clutch is considered since in some variants two different clutch systems are considered.

5-speed MT
For the 5-speed concept on the input shaft the idler gears of the 3rd, 4th and 5th speed step are engaged. On the output shaft the loose gear wheels of the 1st and 2nd speed step as well as the fixed gear wheels of the 3rd, 4th, 5th step are located. The reverse gear is realized by a separate intermediate shaft between the input and output shaft which changes the direction of rotation of the output shaft in case of engagement. Furthermore, the 1st and 2nd gear step are double synchronized, the 3rd, 4th, 5th gear step are single synchronized. The actuation system contains a clutch pedal, a master cylinder and hydraulic lines.

6-speed MT
To realize an additional gear step, it is assumed that the loose gear wheel of the 6th gear step is added on the input shaft and the corresponding fixed gear wheel is added on the output shaft. In this context, the loose gear wheel of the 6th gear step can be shifted by a double-sided synchronizer unit which is responsible for the 5th gear step at once. The actuation system contains a clutch pedal, a master cylinder and hydraulic lines.

7-speed DCT
The front-transverse 7-speed DCT has a common DCT-design with two input shafts, a dual dry clutch, three output shafts and a two-part housing (bell & gearbox housing). On input shaft 1 the fixed gear wheels of the 1st and 3rd gear step are engaged with their corresponding loose gear wheels on output shaft 1 and the fixed gear wheels of 5th and 7th gear step are engaged with their corresponding loose gear wheels on output shaft 2. The common fixed gear wheel of the 4th (loose gear wheel: output shaft 2) and the 6th gear step (loose gear wheel: output shaft 2) and the common fixed gear wheel of 2nd gear step (loose gear wheel: output shaft 1) and the intermediate fixed gear wheel of the reverse gear are located on input shaft 2. The reverse gear is realized by an intermediate fixed gear wheel on output shaft 2 which is connected to input shaft 2 and the reverse loose gear wheel on output shaft 3. The synchronization units are actuated by a common mechatronic system which has an integrated transmission control unit (TCU).

10-speed DCT front-transverse
The front-transverse 10-speed DCT is constructed with a dual wet clutch, two input shafts, two output shafts, one hollow shaft and a housing divided into two parts. On the one hand, there are four loose gear wheels located on output shaft 1. On the other hand, 4 loose gear wheels are engaged on the hollow shaft which can be connected to

output shaft 2 by a separate single-sided synchronization unit. Furthermore, there are three fixed gear wheels on input shaft 1, two fixed gear wheels and one loose gear wheel located on input shaft 2. In order to realize 10 gear steps an advanced connection scheme of gear wheels and synchronization units is applied.

10-speed DCT inline

The inline 10-speed DCT is constructed with a dual wet clutch, two input shafts, one output shafts one hollow shaft and a housing divided into three parts. On the one hand, there are two loose gear wheels and one single-sided synchronizer unit located on the output shaft. On the other hand, 4 loose gear wheels are engaged on the hollow shaft which can be connected to the output shaft by a separate single-sided synchronization unit. Furthermore, there are three fixed gear wheels on input shaft 1, one fixed gear wheel and three loose gear wheels located on input shaft 2. In order to realize 10 gear steps an advanced connection scheme of gear wheels and synchronization units is applied as well.

7-speed AT:

For the 7-speed AT a multi-ratio transmission with 5 switching elements, without one-way clutches and 4 planetary gear sets is assumed. Responsible for the change of gears the switching elements are controlled by a hydraulic control unit which is oil-provided by a separate oil pump driven by chain from the input shaft. Furthermore, the AT is designed with on input and one output shaft and one-piece housing according to common AT inline concepts.

8-speed AT:

The 8-speed automatic transmission is based on the 7-speed AT. The additional gear step can be enabled with the same hardware configuration. 7- or 8-speed functionality of the transmission only depends on intelligent interconnection of the planetary sets. No additional costs are considered.

For the full hybrid P2 variants in the segments D, E and LCV, a modified 10-speed DCT is necessary. For the SUV, the configuration is analogue to the one of the Segment E. It contains an additional clutch between the electric machine and the engine. The housing is also modified. Furthermore the mechatronic actuation is modified because the additional clutch has to be operated. The influence on the costs is displayed in the section of hybridization. In Table 3.11 an overview of the selected transmissions in the different Diesel vehicle segments is provided.

Segment	P [KW]	Input Torque [Nm]	Transmission mounting	Base variant	Downsizing
B	65	220	front-transverse	5-speed MT	6-speed MT
C	95	260	front-transverse	6-speed MT	7-speed DCT
D	135	350	front-transverse	6-speed MT	10-speed DCT
E	180	530	inline	8-speed AT	10-speed DCT
SUV	180	530	inline	8-speed AT	10-speed DCT
LCV	120	400	front-transverse	6-speed MT	10-speed DCT

Table 3.11 Overview of selected transmissions, Diesel vehicle segments

In Table 3.12 the transmission variants within the gasoline vehicle segments are shown.

Segment	Power [kW]	Input Torque [Nm]	Transmission mounting	Base	Variant 1	Variant 2
B	65	up to 160	front-transverse	5-speed MT	6-speed MT	
C	95	up to 220	front-transverse	5-speed MT	6-speed MT	7-speed DCT
D	135	up to 315	inline	8-speed AT	10-speed DCT	
E	180	up to 350	inline	8-speed AT	10-speed DCT	

Table 3.12 Overview of selected transmissions, gasoline vehicle segments

3.3.2 Components of transmissions and cost influencing parameter

In this section an overview about component groups and components of the different transmissions is provided. The names of the component groups are identical in every kind of transmission but the content differs. For example in the component group clutch a dry clutch is considered for the manual transmissions whereas for the DCT transmissions either a dual dry clutch or a dual wet clutch is selected.

Component group	Figure	Components
Clutch		• Dry clutch, miscellaneous*
Housing		• Housing, miscellaneous*
Wheel set		• Input- & output shafts, loose & fixed gear wheels, differential, miscellaneous*
Bearings		• Different kind of bearings, miscellaneous*
Shift elements		• Synchronizer rings, synchronizer units, shift forks, miscellaneous*
Actuation		• Clutch pedal, master cylinder, hydraulic lines, miscellaneous*
Assembly & EOL test		• Assembly and End-of-Line-Test
* bolts, nuts, washers, sealings		

Table 3.13 Component groups and components, manual transmissions

In Table 3.13 the component groups and components of the manual transmissions are displayed. To visualize the particular component groups a small figure is shown too. Corresponding to the manual transmission the component groups of the automatic transmissions and dual clutch transmissions are provided in Table 3.14 and Table 3.15.

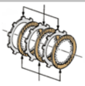
Component group	Figure	Components
Clutch		• Hydrodynamic torque converter, miscellaneous*
Housing		• Housing, miscellaneous*
Wheel set		• Input- & output shafts, planetary gear sets, miscellaneous*
Bearings		• Different kind of bearings, miscellaneous*
Shift elements		• Brakes, clutches
Actuation		• Mechatronics module including oil supply
Assembly & EOL test		• Assembly and End-of-Line-Test
* bolts, nuts, washers, sealings		

Table 3.14 Component groups and components, automatic transmissions

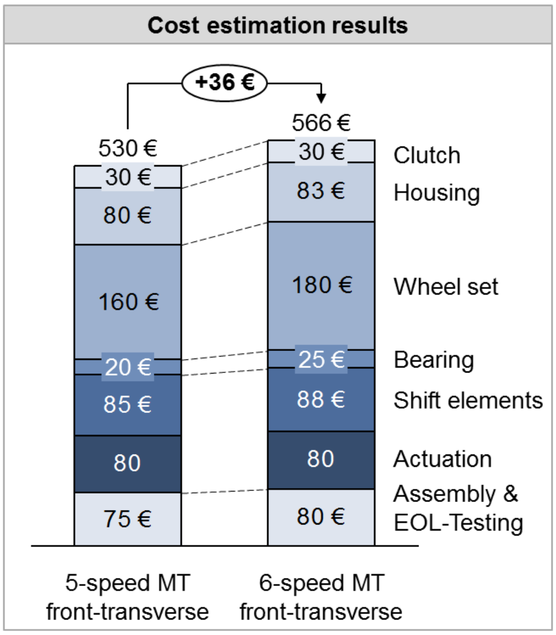
Component group	Figure	Components
Clutch		• Dual dry clutch vs. dual wet clutch
Housing		• Housing, miscellaneous*
Wheel set		• Input- & output shafts, loose & fixed gear wheels, differential, miscellaneous*
Bearings		• Different kind of bearings, miscellaneous*
Shift elements		• Synchronizer rings, synchronizer units, shift forks, miscellaneous*
Actuation		• Mechatronics module including oil supply
Assembly & EOL test		• Assembly and End-of-Line-Test
* bolts, nuts, washers, sealings		

Table 3.15 Component groups and components, dual clutch transmissions

3.3.3 Scaling methodology

Since the selected transmissions are not scaled by the different vehicle segments no scaling methodology is provided in this section.

3.3.4 Cost estimation results Diesel vehicle segments



Main cost influencing technical differences

Clutch:
→ ./.

Housing:
→ Enlarged gearbox housing cover with extended machining effort due to additional bearing positions for in- and output shaft

Wheel set:
→ Extended input and input and output shaft; additional pair of gear wheels due to 6th gear step

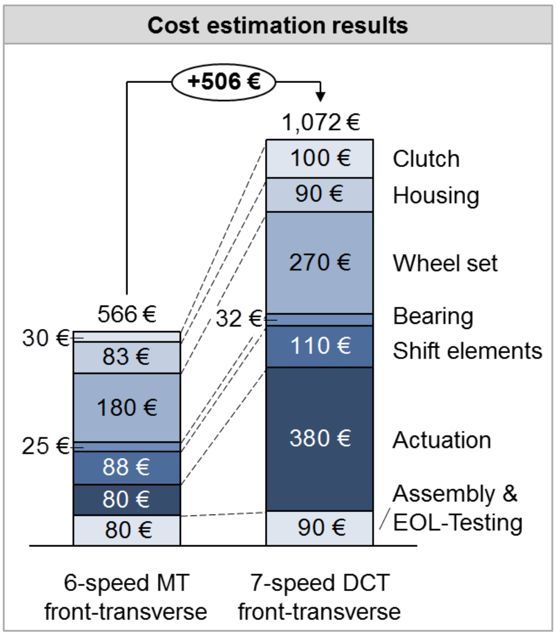
Bearing:
→ Additional bearings (gearbox housing cover & gear wheels)

Shift elements:
→ Additional synchronizer ring (double-sided synchronizer unit instead of single-sided synchronizer unit

Assembly & EOL testing:
→ Increased assembly effort (wheel set, gearbox housing cover) due to additional gear step

Figure 3.13 Cost estimation, transmission, Diesel segment B

Figure 3.13 presents the incremental transmission cost estimates for Diesel engines in segment B. The delta costs of 36 € are mainly driven by the additional costs of the modified wheel set concept. An additional pair of gear wheels is necessary due to the 6th gear step. Furthermore the component groups assembly, shift elements, housing and bearings are also affected by the change from 5-speed MT to 6-speed MT. In Figure 3.14 the cost estimation results of Diesel segment C are displayed.



Main cost influencing technical differences

Clutch:
→ Dual dry clutch

Housing:
→ Enlarged housings with extended machining effort

Wheel set:
→ Two axial arranged input shafts; three instead of one output shaft; additional 7th gear step

Bearing:
→ Additional bearings due to higher amount of shafts and gear steps

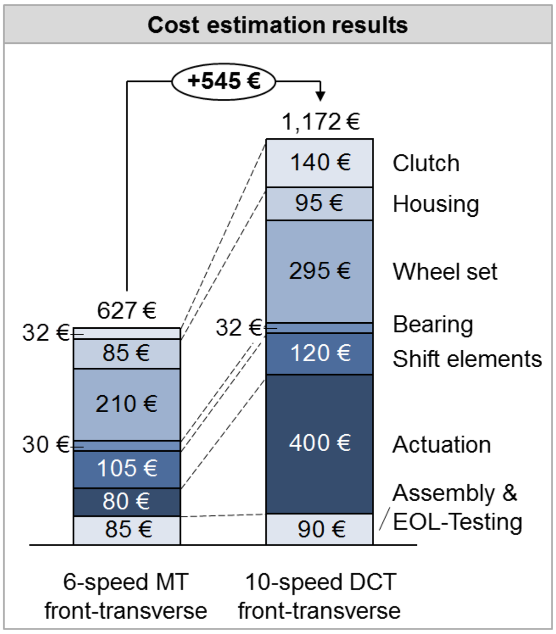
Shift elements:
→ Additional synchronizer with synchronizer ring for 7th and reverse gear step

Actuation:
→ Mechatronic system instead of MT actuation

Assembly & EOL testing:
→ Increased assembly effort due to more complex and extended amount of wheel set components

Figure 3.14 Cost estimation, transmission, Diesel segment C

The 7-speed DCT is operated by a complex mechatronic system. It is estimated with 380 € and causes approximately 35% of the total cost and 59% of the delta cost. The rest of the delta costs can mainly be explained by the more complex dual dry clutch and the extended wheel set. In Figure 3.15 the cost estimation results of Diesel segment D are displayed.



Main cost influencing technical differences

Clutch:
→ Dual wet clutch

Housing:
→ Extended housing, integration of oil supply of dual wet clutch (e.g. oil cooler)

Wheel set:
→ 2 additional shafts , 2 additional gear wheels

Bearing:
→ Additional needle bearings due to additional loose gear wheels and shafts

Shift elements:
→ Additional synchronizer units

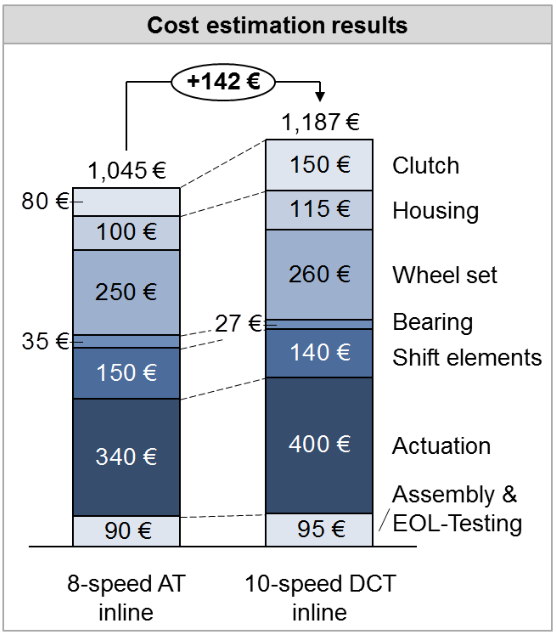
Actuation:
→ Mechatronic system instead of MT actuation

Assembly & EOL testing:
→ Increased assembly effort due to more complex wheel set

Figure 3.15 Cost estimation, transmission, Diesel segment D

Similar to segment C, in segment D the delta costs are primarily caused by the mechatronic actuation system of the 10-speed DCT. Also the Dual wet clutch leads to higher cost of 108€.

Figure 3.16 presents the incremental transmissions cost estimates for diesel segment E vehicles and SUVs. The delta cost of 142 € mainly comes of the dual wet clutch and the more complex actuation system of the 10-speed DCT. The cost estimation for segment SUV is the same as for segment E since the technical specifications are the same.



Main cost influencing technical differences

Clutch:
→ Hydrodynamic torque converter vs. dual wet clutch

Housing:
→ One-part design vs. three-part design

Wheel set:
→ Planetary gear set vs. advanced wheel set concept

Bearing:
→ ./.

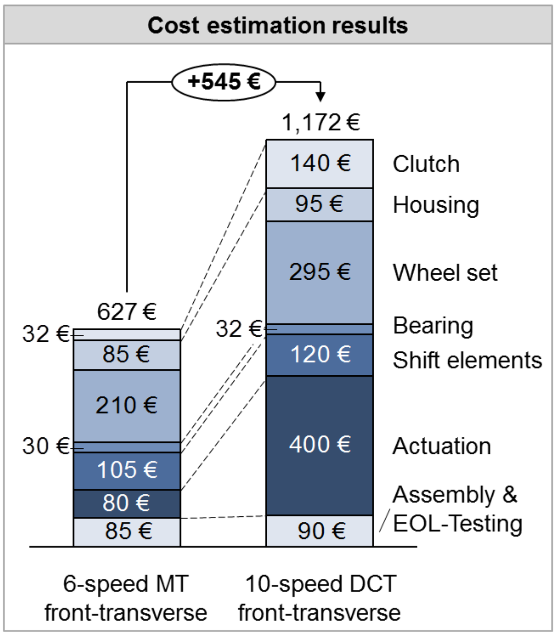
Shift elements:
→ Brakes, clutches vs. synchronizer units

Actuation:
→ More complex system architecture (more complex solenoid valves, additional sensors, actuators)

Assembly & EOL testing:
→ Extended assembly effort due to wheel set architecture

Figure 3.16 Cost estimation, transmission, Diesel segment E and SUV.

Finally in Figure 3.17 the cost estimation of Diesel segment LCV is shown. A 6-speed MT and 10-speed DCT are selected which differ especially in the kind of actuation. The MT actuation contains of a clutch pedal, master cylinder and hydraulic lines whereas the DCT actuation is based on a mechatronic system which leads to delta cost of 320 €.



Main cost influencing technical differences

Clutch:
→ Dual wet clutch

Housing:
→ Extended housing, integration of oil supply of dual wet clutch (e.g. oil cooler)

Wheel set:
→ 2 additional shafts , 2 additional gear wheels

Bearing:
→ Additional needle bearings due to additional loose gear wheels and shafts

Shift elements:
→ Additional synchronizer units

Actuation:
→ Mechatronic system instead of MT actuation

Assembly & EOL testing:
→ Increased assembly effort due to more complex wheel set

Figure 3.17 Cost estimation, transmission, Diesel segment LCV

Finally in Table 3.16 an overview on all delta cost of the different variants is shown.

Segment	Power [kW]	Input Torque [Nm]	Transmission mounting	Base	Variant 1	Delta cost
B	65	220	front-transverse	5-speed MT	6-speed MT	+36€
C	95	260	front-transverse	6-speed MT	7-speed DCT	+506€
D	135	350	front-transverse	6-speed MT	10-speed DCT	+545€
E	180	530	inline	8-speed AT	10-speed DCT	+142€
SUV	180	530	inline	8-speed AT	10-speed DCT	+142€
LCV	120	400	front-transverse	6-speed MT	10-speed DCT	+545€

Table 3.16 Cost estimation, transmissions, Diesel vehicle segments, overview

3.3.5 Cost estimation results gasoline vehicle segments

Basically the structure of the selected transmissions within the gasoline vehicle segments doesn't differ from the corresponding Diesel vehicle segments. In the following figures the cost estimation results of the gasoline vehicle segments are provided.

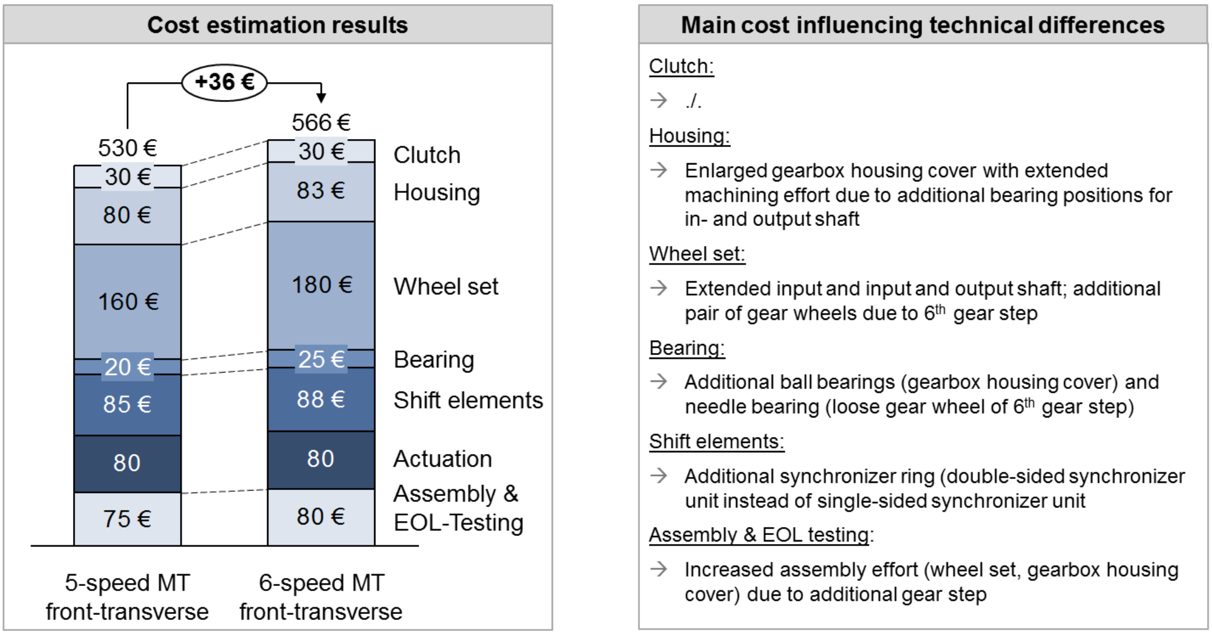
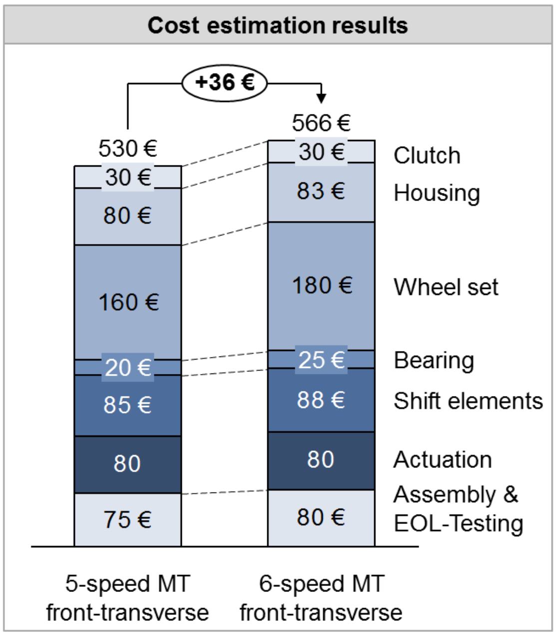


Figure 3.18 Cost estimation, transmission, gasoline segment B

Figure 3.18 presents the incremental transmissions cost estimates for the gasoline class B vehicle. Like in the corresponding Diesel segment B the delta cost of 36 € are mainly caused by the additional costs of the modified wheel set concept.



Main cost influencing technical differences

Clutch:
→ ./.

Housing:
→ Enlarged gearbox housing cover with extended machining effort due to additional bearing positions for in- and output shaft

Wheel set:
→ Extended input and input and output shaft; additional pair of gear wheels due to 6th gear step

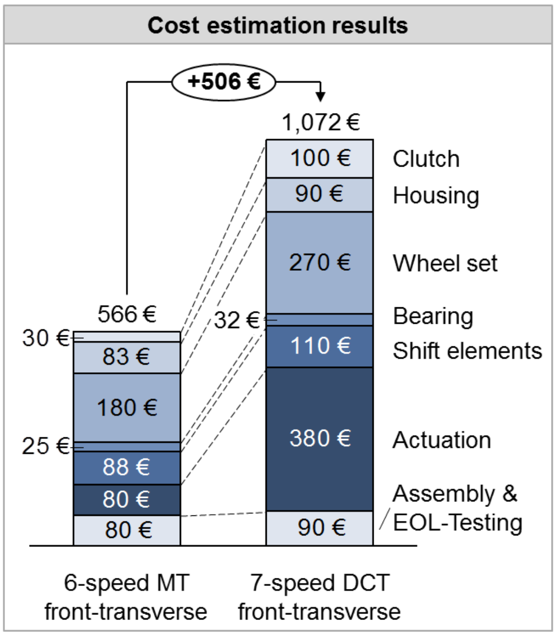
Bearing:
→ Additional ball bearings (gearbox housing cover) and needle bearing (loose gear wheel of 6th gear step)

Shift elements:
→ Additional synchronizer ring (double-sided synchronizer unit instead of single-sided synchronizer unit

Assembly & EOL testing:
→ Increased assembly effort (wheel set, gearbox housing cover) due to additional gear step

Figure 3.19 Cost estimation, transmission, gasoline segment C, 5-speed MT vs. 6-speed MT

Gasoline segment C is the only segment where two variants are added to the base. So in Figure 3.19 the 6-speed MT (variant 1) is compared to the 5-speed MT (base). In Figure 3.20 a 7-speed DCT (variant 2) is compared to the 6-speed MT (variant 1). The delta cost between the base and variant 1 are the same as in segment B because the same transmissions are selected.



Main cost influencing technical differences

Clutch:
→ Dual dry clutch

Housing:
→ Enlarged housings with extended machining effort

Wheel set:
→ Two axial arranged input shafts; three instead of one output shaft; additional 7th gear step

Bearing:
→ Additional bearings due to higher amount of shafts and gear steps

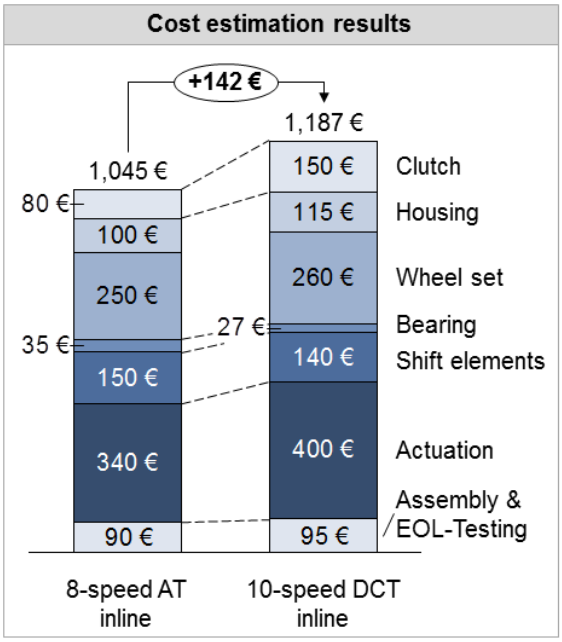
Shift elements:
→ Additional synchronizer with synchronizer ring for 7th and reverse gear step

Actuation:
→ Mechatronic system instead of MT actuation

Assembly & EOL testing:
→ Increased assembly effort due to more complex and extended amount of wheel set components

Figure 3.20 Cost estimation, transmission, gasoline segment C, 6-speed MT vs. 7-speed DCT

The delta cost of 506 € between variant 2 and variant 1 are mainly caused by the mechatronic actuation system of the 7-speed DCT. Remaining delta cost can be explained by an additional dual wet clutch and modifications in other component groups.



Main cost influencing technical differences

Clutch:
→ Hydrodynamic torque converter vs. dual wet clutch

Housing:
→ One-part design vs. three-part design

Wheel set:
→ Planetary gear set vs. advanced wheel set concept

Bearing:
→ ./.

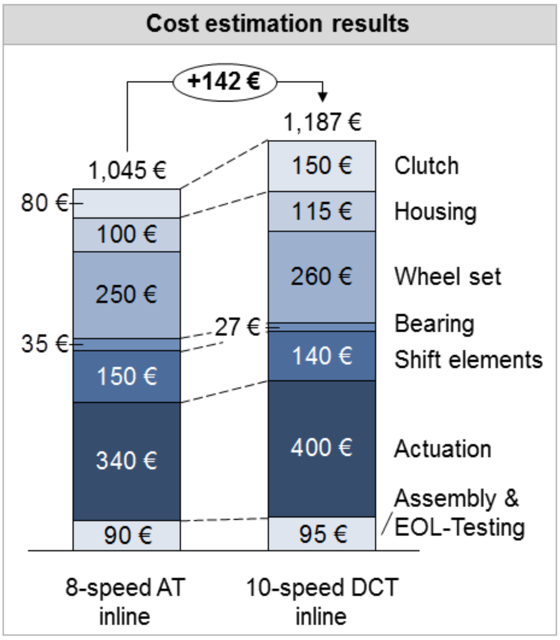
Shift elements:
→ Brakes, clutches vs. synchronizer units

Actuation:
→ Solenoid ball seat valves [AT] vs. pressure regulating solenoid valves [DCT]

Assembly & EOL testing:
→ ./.

Figure 3.21 Cost estimation, transmission, gasoline segment D

The cost estimation results of segment D and E are displayed in Figure 3.21 and Figure 3.22. Because the selected transmissions are the same the segments can be analysed together. The delta cost of 142 € is mainly driven by the higher cost of the dual wet clutch and required actuators for the DCT-10 when comparing with the torque converter of the AT transmission. The rest is separated between the other component groups.



Main cost influencing technical differences

Clutch:
→ Hydrodynamic torque converter vs. dual wet clutch

Housing:
→ One-part design vs. three-part design

Wheel set:
→ Planetary gear set vs. advanced wheel set concept

Bearing:
→ ./.

Shift elements:
→ Brakes, clutches vs. synchronizer units

Actuation:
→ Solenoid ball seat valves [AT] vs. pressure regulating solenoid valves [DCT]

Assembly & EOL testing:
→ ./.

Figure 3.22 Cost estimation, transmission, gasoline segment E

Table 3.17 summarizes the delta cost of all considered variants. For the segment C, two variants are compared with the base powertrain. The delta cost of variant 2 refers to variant 1.

Segment	Power [kW]	Input Torque [Nm]	Transmission mounting	Base	Variant 1	Delta cost	Variant 2	Delta cost
B	65	up to 160	front-transverse	5-speed MT	6-speed MT	+36€	-	-
C	95	up to 220	front-transverse	5-speed MT	6-speed MT	+36€	7-speed DCT	+506€
D	135	up to 315	inline	8-speed AT	10-speed DCT	+142€	-	-
E	180	up to 350	inline	8-speed AT	10-speed DCT	+142€	-	-

Table 3.17 Cost estimation, transmissions, gasoline vehicle segments, overview

3.4 Cost estimation hybridization

A technical solution to reduce emissions and the fuel consumption is to electrify the powertrain. There are different technical solutions ranging from micro hybrid concepts to full electric vehicles. Within this project, different variants are considered under the generic term hybridization. They range from a common start/stop system as entry-level variant to an advanced start/stop system, a P0 48V hybrid system or a P2 full hybrid system.

Hereinafter the four different technical concepts are described. This description comprises the determination of functionality and considered hardware including component groups and components, necessary technical modifications and the scaling methodology through different segments.

3.4.1 Description and hardware determination

Start/stop system

The considered start/stop system is realized as an enhanced start/stop system with a modified starter to meet the requirement of multiple starts compared to conventional starter. In addition, the battery is realized as an Absorbent Glass Mat battery (AGM). The standard alternator is also modified. Apart from that, different kinds of sensors are included in the start/stop system: Intelligent battery sensor, brake pressure sensor and for manual transmissions also a neutral gear position sensor. In Figure 3.23, the basic structure of the system is shown.

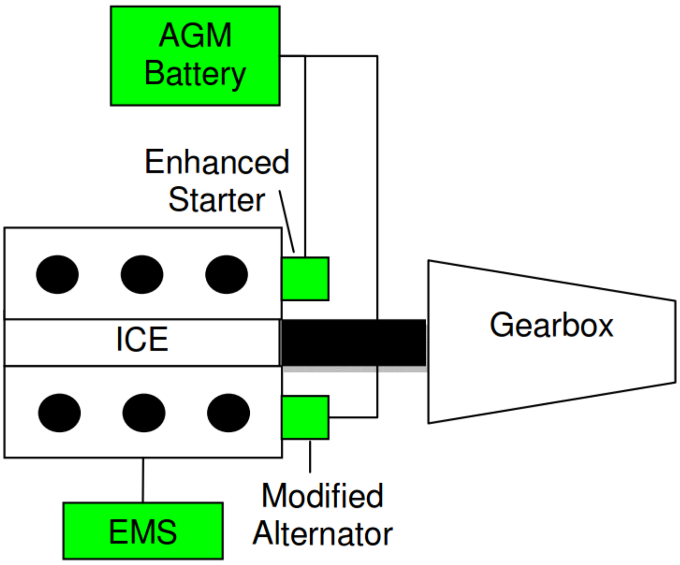


Figure 3.23 Overview enhanced start/stop system

Advanced start/stop system

The advanced start/stop system is based on the enhanced start/stop system. To provide the functionality of shutting the engine down at higher velocity compared with the enhanced start/stop system, a modification of the Engine Management System (EMS) is necessary. Besides this modification no other parts are affected.

P0 48V hybrid system

The P0 48V hybrid system uses a Belt-Starter-Generato (BSG) as combined starter and alternator. A 48V Li-Ion battery is connected with the BSG via an AC/DC inverter. In addition, further components are affected by using a BSG instead of a conventional starter. Due to higher loads the crank pulley and the belt have to been modified. An additional belt tensioner is added to the P0 48V system. Compared to the start/stop systems considered before, the conventional alternator is needless. All functions are taken over by the BSG. In Figure 3.24 an overview on the P0 48V system is provided.

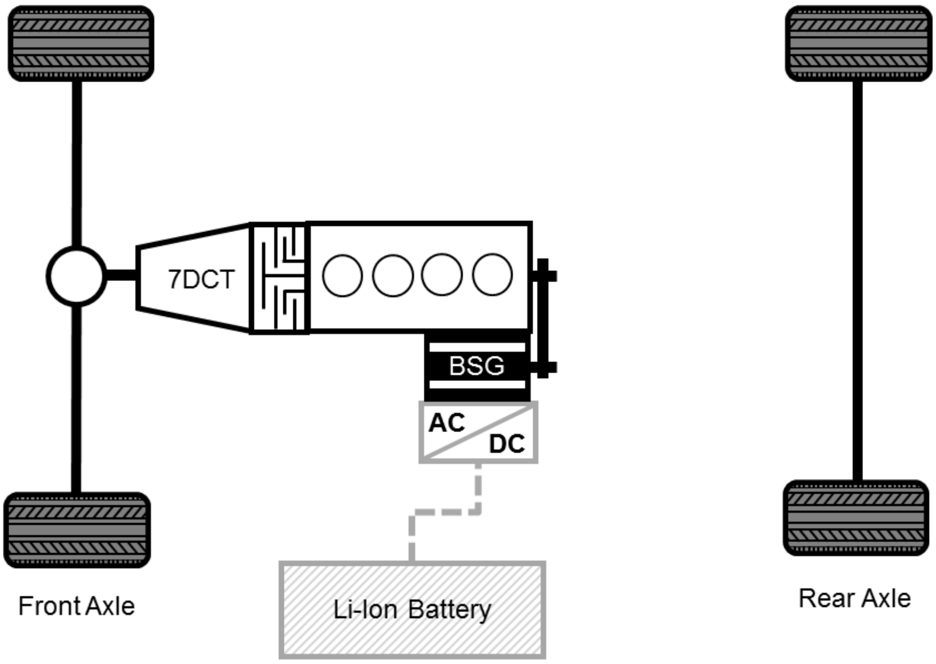


Figure 3.24 Overview P0 48V hybrid system

P2 hybrid system

The P2 hybrid system is designed as a parallel full hybrid. The integrated starter generator (ISG) is located between the transmission and an additional clutch, which is considered in the section of transmission. The high-voltage battery operates at a nominal voltage level of 350V. Different operating modes are possible e.g. driving under purely electric power or the coasting mode. Besides ISG and battery, the power unit is another important part of the hybrid system. Compared with the other hybrid variants, a belt drive is needless, the A/C compressor works electrically and is connected to the 350 V power supply. In Figure 3.25 an overview of the P2 hybrid system is provided.

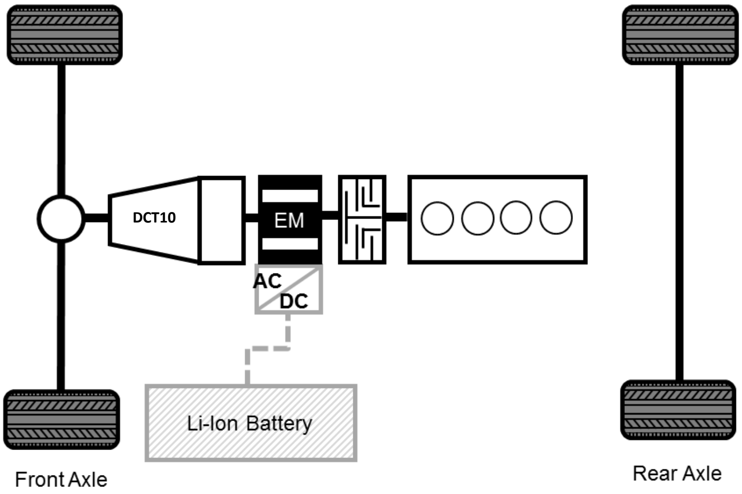


Figure 3.25 Overview of P2 hybrid system.

In Table 3.18 the allocation of the hybrid systems to the Diesel segments is provided. The technical specifications of the variants are shown in the following section.

Segment	Power [kW]	Base variant	Variant 1	Variant 2	Variant 3
B	60	Without	Start/stop	P0 48V	-
C	80	Without	Start/stop	Advanced Start/stop	P0 48V
D	110	Without	Start/stop	Advanced Start/stop	P2
E	150	Without	Start/stop	Advanced Start/stop	P2
LCV	120	Without	Start/stop	Advanced Start/stop	P2
SUV	150	Without	Start/stop	Advanced Start/stop	P2

Table 3.18 Overview of selected hybrid systems for Diesel engines

For the gasoline engines a corresponding overview is displayed in Table 3.19.

Segment	Power [kW]	Base variant	Variant 1	Variant 2	Variant 3
B	60	Without	Start/stop	P0 48V	-
C	80	Without	Start/stop	Advanced Start/stop	P0 48V
D	110	Without	Start/stop	Advanced Start/stop	P2
E	150	Without	Start/stop	Advanced Start/stop	P2

Table 3.19 Overview of selected hybrid systems for gasoline engines

3.4.2 Components of hybridization systems and cost influencing parameter

In the following tables an overall view of considered components groups, components and cost influencing parameter of the different hybrid systems are shown. Table 3.20 provides an overview of the enhanced start/stop system.

Component group		Components	Cost influencing parameter
Battery		• AGM-battery	• Scaled by engine power
Alternator		• Modified Alternator	• Scaled by engine power
Starter		• Modified Starter	• Scaled by engine power
Sensors		• Intelligent battery sensor • Neutral position sensor (only for MT) • Brake pressure sensor	• Cost are assumed as non-variable • Cost are assumed as non-variable • Cost are assumed as non-variable

Table 3.20 Overview of component groups enhanced start/stop system

In Table 3.21 the component groups, components and cost influencing parameter of the advanced start/stop system are displayed. The structure is almost identical with the enhanced start/stop system, only the modified EMS due to advanced functionality is considered in the component group electronics.

Component group		Components	Cost influencing parameter
Battery		• AGM-battery	• Scaled by engine power
Alternator		• Modified Alternator	• Scaled by engine power
Starter		• Modified Starter	• Scaled by engine power
Sensors		• Intelligent battery sensor • Neutral position sensor (only for MT) • Brake pressure sensor	• Cost are assumed as non-variable • Cost are assumed as non-variable • Cost are assumed as non-variable
EMS		• Modified EMS due to advanced functionality	• Cost are assumed as non-variable

Table 3.21 Overview of component groups advanced start/stop system

In Table 3.22 the different component groups and considered components of the P0 48V hybrid system are displayed.

Component group		Components	Cost influencing parameter
Battery		• 48V Li-Ion-battery, AGM-battery	The specifications are identical for all segments, so no scaling is required
BSG		• BSG • Inverter	
Electronics		• DC/DC • Engine Management System	
Miscellaneous		Miscellaneous components: • Tensioner • Modified flywheel • Modified belt • Pedal coupled brake system	

Table 3.22 Overview component groups P0 48V hybrid system

In Table 3.23 an overview of the component groups of the P2 hybrid system is displayed.

Component group		Components	Cost influencing parameter
ISG + PowerUnit	 	• ISG • PowerUnit	• Scaled by power EM • Scaled by power EM
Battery		• Battery module • Cooling, Housing • Battery Management System • Wiring • Miscellaneous • Assembly	• Scaled by stored energy battery • Assumed as non-variable • Assumed as non-variable • Assumed as non-variable • Assumed as non-variable • Assumed as non-variable
Miscellaneous		• HV-Wiring • Electric A/C compressor • HCU/VCU	• Assumed as non-variable • Assumed as non-variable • Assumed as non-variable
Transmission		• Modified Housing • Additional Clutch • Modified Actuation • More complex assembly	• No Scaling necessary since transmissions are calculated separately
Omitted components		• Auxiliary drive • Alternator • Starter	• Scaled by power engine • Scaled by power engine • Scaled by power engine

Table 3.23 Overview component groups P2 hybrid

The technical specifications of the Li-Ion battery and the BSG/ISG of the P0 and the P2 hybrid system are displayed in the following tables.

P0 48V hybrid system			
Technical specifications of battery and ISG in P0 48V hybrid system		Diesel	Gasoline
		B, C	B, C
Battery	Type [-]	Li-Ion	Li-Ion
	Nominal. Voltage [V]	48	48
	Total Energy [kWh]	0.53	0.53
	Max Power [kW]	16	16
	SOC Operation Window [%]	80-30	80-30
BSG	Type [-]	PMSM	PMSM
	Max. Torque [Nm]	44	44
	Max. Power [kW]	15	15

Table 3.24 Overview technical specifications P0 48V hybrid system

The specifications of the P2 hybrid system are displayed in Table 3.25 below.

P2 hybrid system						
Technical specifications of battery and ISG in P2 hybrid system		Diesel			Gasoline	
		D	E, SUV	LCV	D	E
Battery	Type [-]	Li-Ion	Li-Ion	Li-Ion	Li-Ion	Li-Ion
	Nominal. Voltage [V]	350	350	350	350	350
	Total Energy [kWh]	1.1	1.3	1.8	1.1	1.3
	Max Power [kW]	40	45	50	40	45
	SOC Operation Window [%]	80-40	80-40	80-40	80-40	80-40
ISG	Type [-]	PMSM	PMSM	PMSM	PMSM	PMSM
	Max. Torque [Nm]	210	250	300	210	250
	Max. Power [kW]	35	40	45	35	40

Table 3.25 Overview technical specifications P2 hybrid system

3.4.3 Cost estimation results - hybridization

In the following section the different hybrid variants of each segment are analysed with regards to the delta cost. The section is separated by fuel type so at the beginning the Diesel variants are displayed, the gasoline variants follow afterwards.

3.4.3.1 Cost estimation results Diesel segments

Figure 3.26 shows the cost estimation results of Diesel segment B. Some explanations about the costs of the component groups are provided in the right section “Main cost influencing technical differences”. First the variant start/stop is compared to the base. In the next step the hybrid variant P0 48V is compared to the variant start/stop.

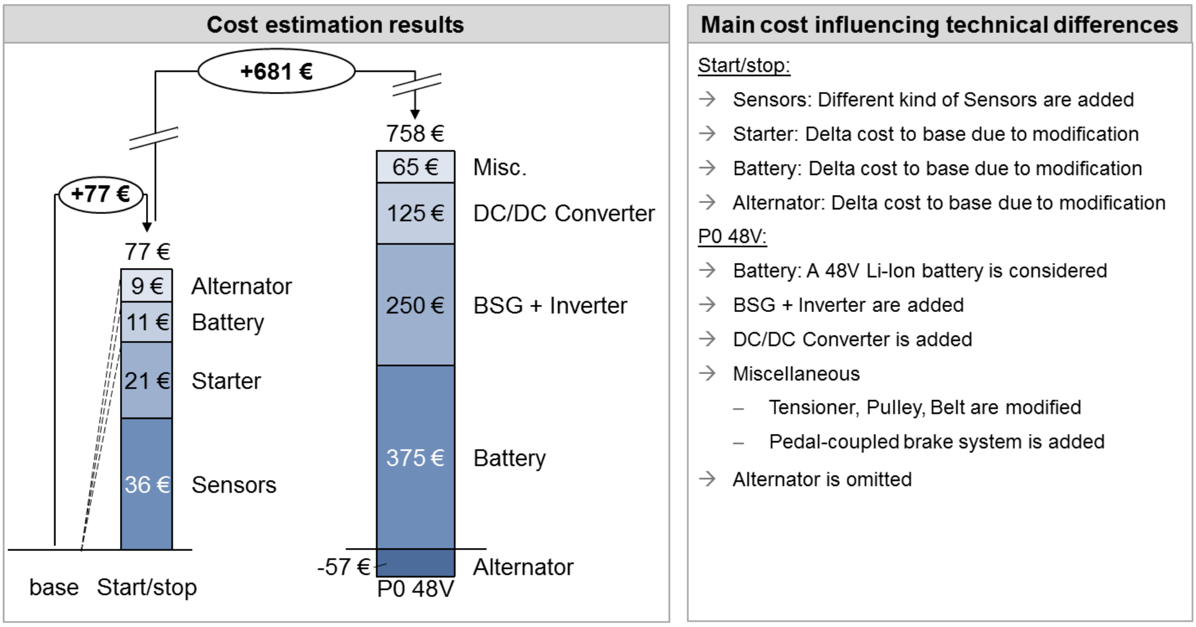


Figure 3.26 Cost estimation, hybridization, Diesel segment B

In Figure 3.27 the results of Diesel segment C are displayed. Differently from segment B, the advanced start/stop system costs -12 € with respect to the baseline. This lower cost is virtual, since the neutral position sensor cost for this variant is already included into the 7-speed DCT transmission cost. By still considering 5 € for the EMS, the final cost is negative due to the explanation above.

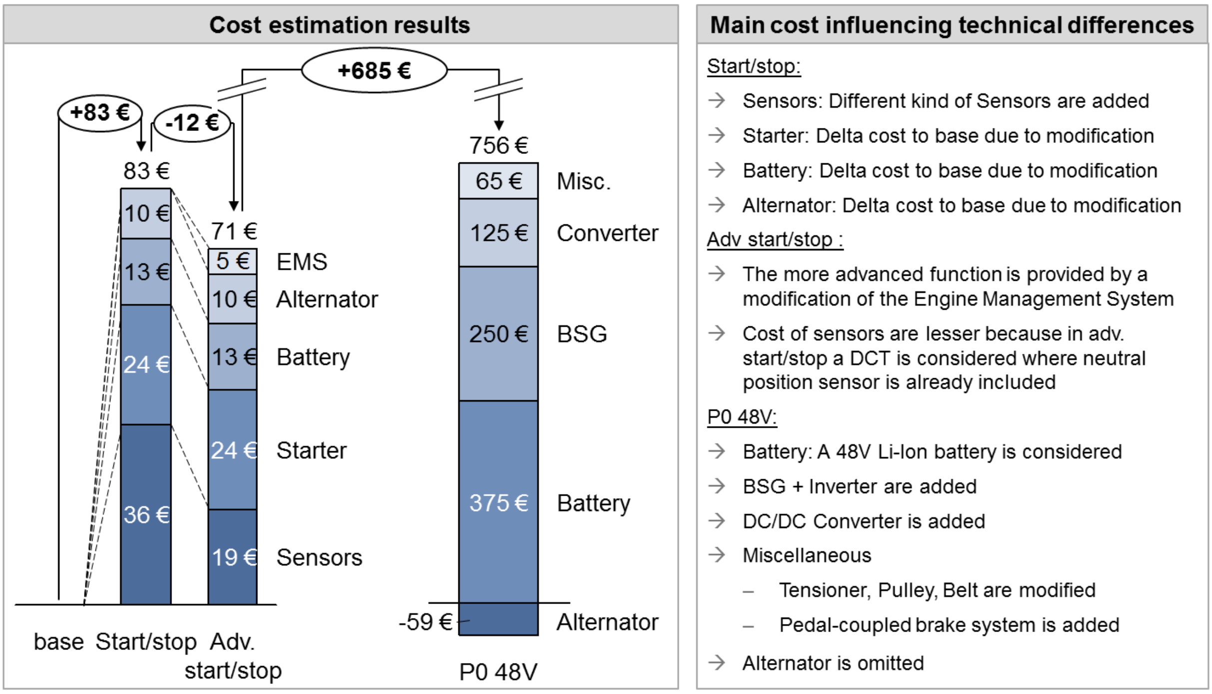


Figure 3.27 Cost estimation, hybridization, Diesel segment C

For the following segments D, E, SUV and LCV a P2 full hybrid system is considered instead of a P0. Figure 3.28 provides an overview of the cost estimation of Diesel segment D. The delta cost of +1865 € are mainly driven by the high-voltage battery and the power unit. A full parallel hybrid version mounted on a 10-speed DCT is also considered. In this respect, it is assumed that a common connection system between the combustion engine and transmission is added. This connection system mainly contains a clutch which can separate the combustion engine and the the electric motor from each other. Thereby, the electric motor is mounted on the input shaft of the transmission. Furthermore, technical modifications of the mechatronic unit are necessary to control the separating clutch.

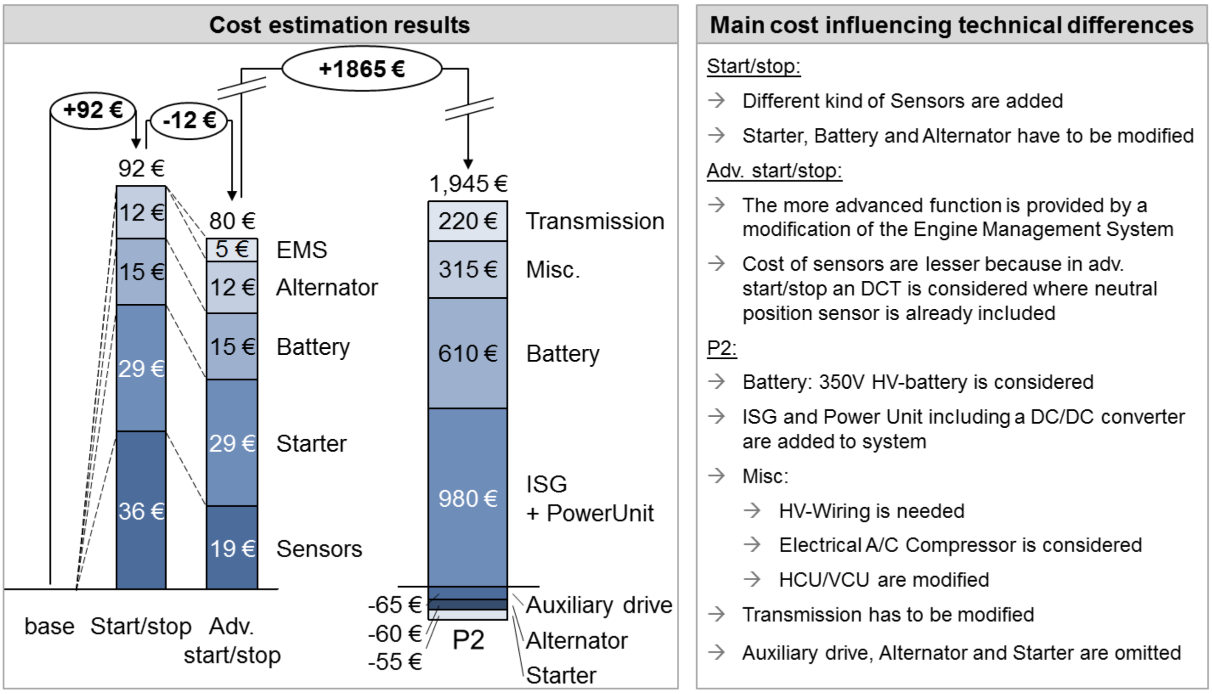


Figure 3.28 Cost estimation, hybridization, Diesel segment D

For the segments E, SUV and LCV, the corresponding cost estimation results are shown below in Figure 3.29 and Figure 3.30. The structure of the cost estimations is mainly the same. They vary in terms of different specifications of the full parallel hybrid system e.g. different batteries which cause a cost difference.

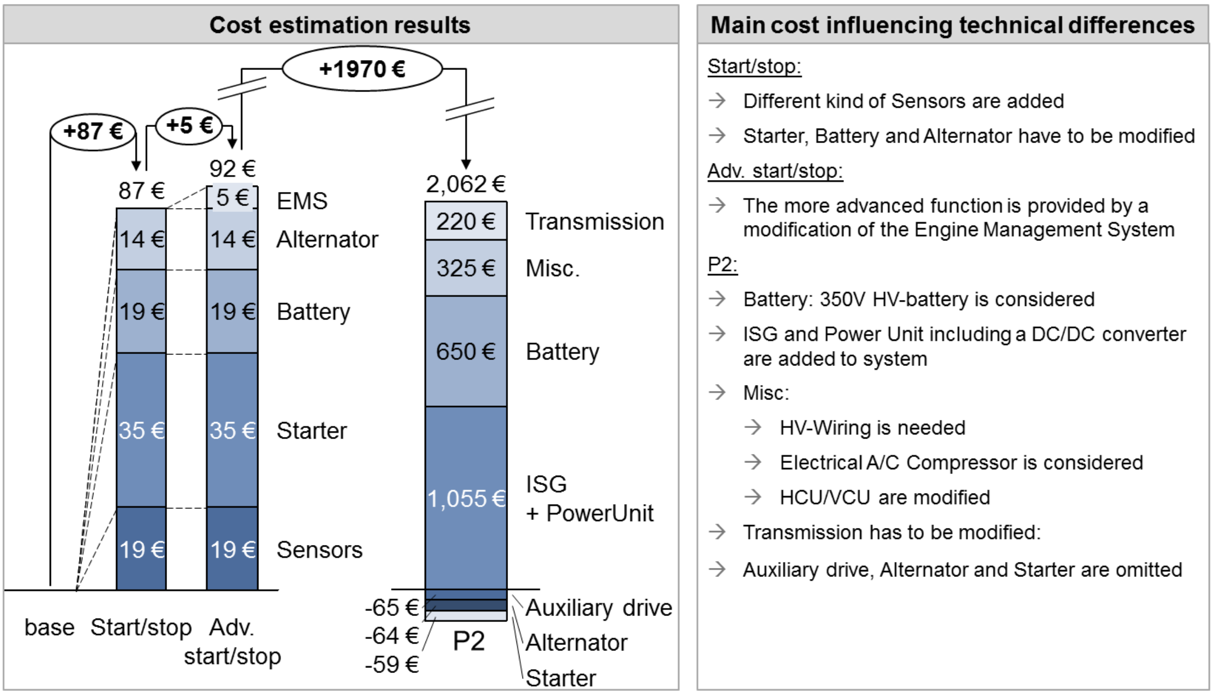


Figure 3.29 Cost estimation, hybridization, Diesel segment E and SUV.

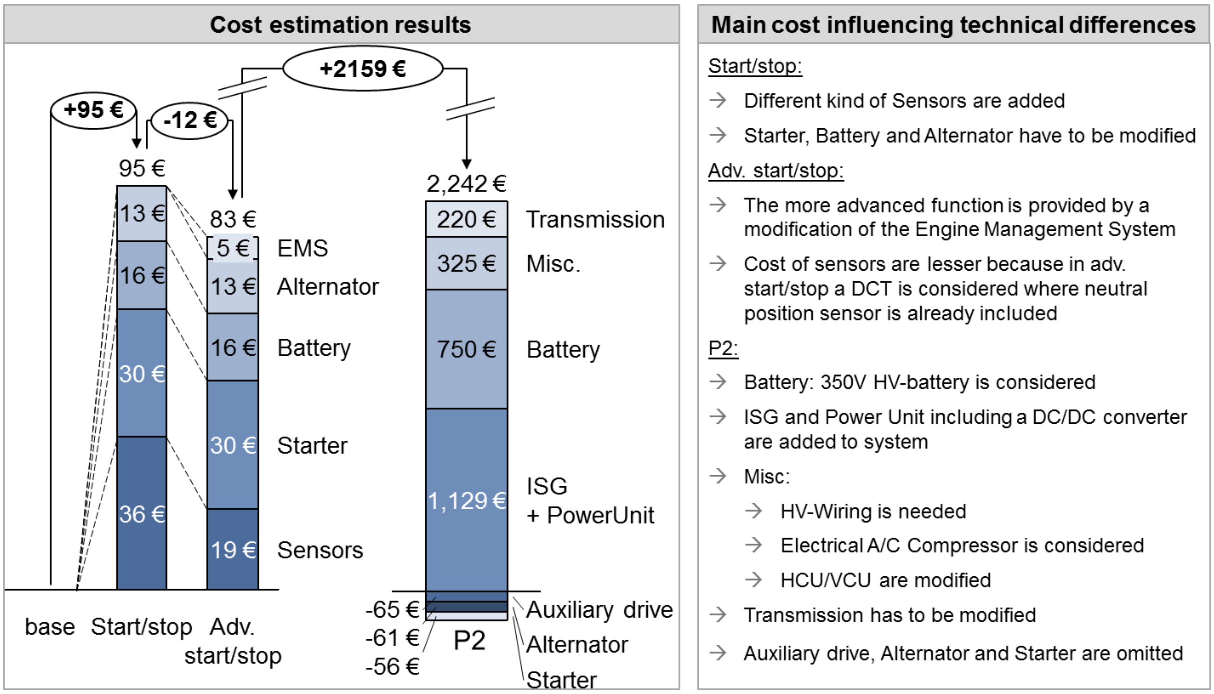


Figure 3.30 Cost estimation, hybridization, Diesel segment LCV

Closing up this section the cost estimation results of all Diesel segments are provided in Table 3.26

Segment	Power [kW]	Base variant	Variant 1	Delta cost	Variant 2	Delta cost	Variant 3	Delta cost
B	60	Without	Start/stop	+ 77 €	P0 48V	+ 681 €	-	-
C	80	Without	Start/stop	+ 83 €	Advanced Start/stop	- 12 €	P0 48V	+ 685 €
D	110	Without	Start/stop	+ 92 €	Advanced Start/stop	- 12 €	P2	+ 1.865 €
E	150	Without	Start/stop	+ 77 €	Advanced Start/stop	+ 5 €	P2	+ 1.970 €
SUV	150	Without	Start/stop	+ 87 €	Advanced Start/stop	+ 5 €	P2	+ 1.970 €
LCV	120	Without	Start/stop	+ 95 €	Advanced Start/stop	- 12 €	P2	+ 2.159 €

Table 3.26 Cost estimation results, Diesel vehicle segments, summary

3.4.3.2 Cost estimation results gasoline segments

In the following section the cost estimation results for the gasoline variants are displayed. Since the technical specifications of the different hybrid variants do not depend on the fuel type, the structure of the cost estimations is similar to the Diesel section. Figure 3.31 presents cost estimates for the gasoline B class.

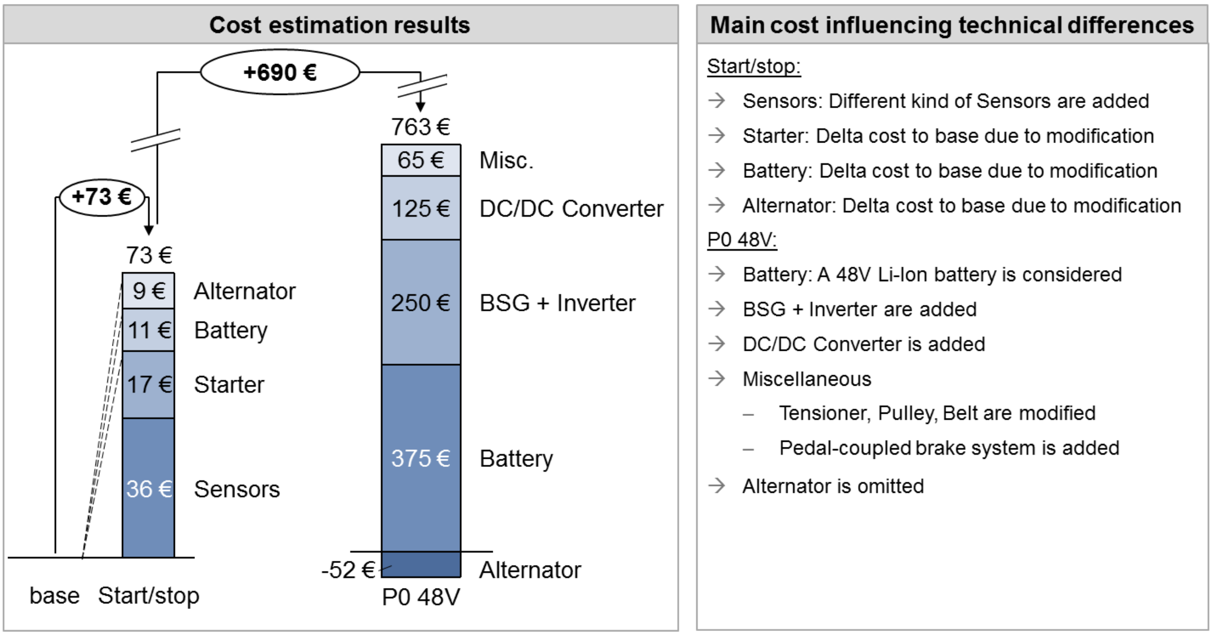


Figure 3.31 Cost estimation, hybridization, gasoline segment B

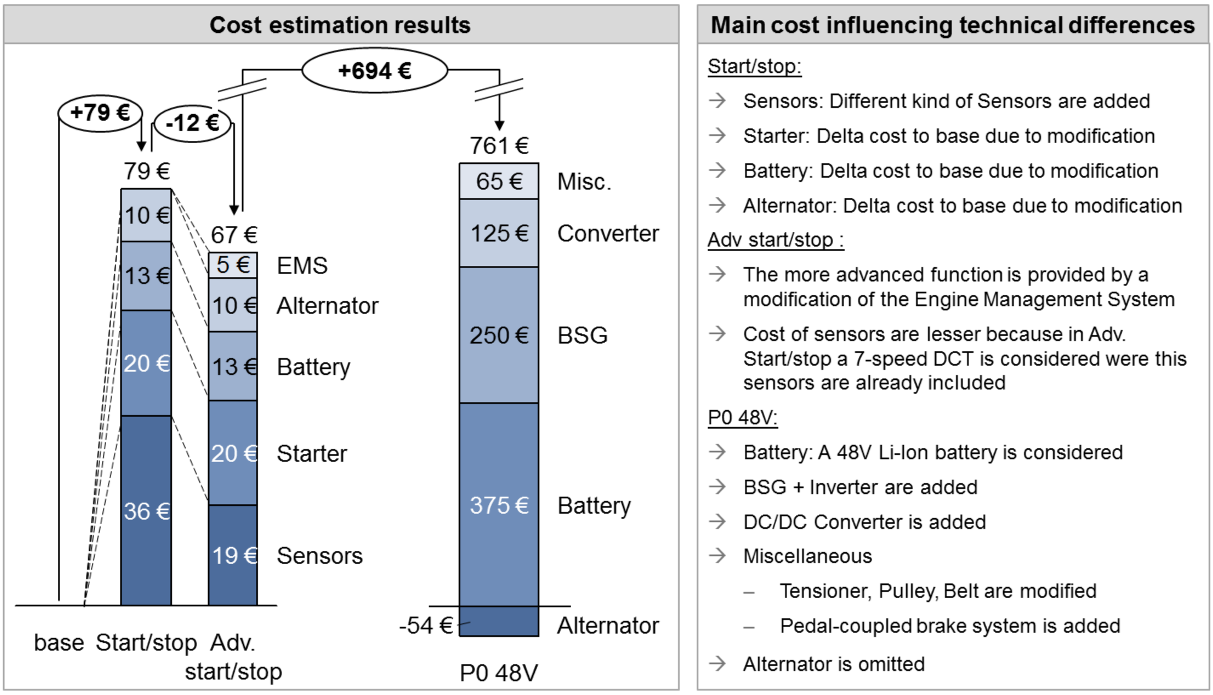
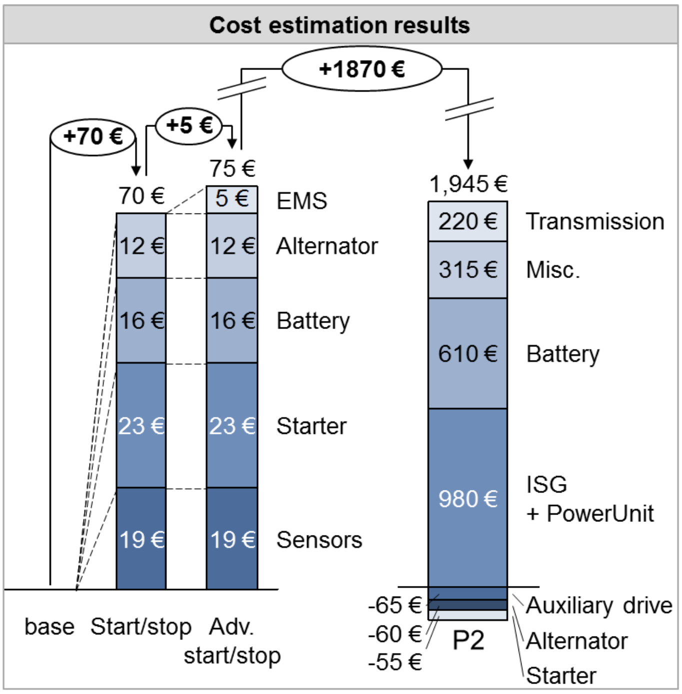


Figure 3.32 Cost estimation, hybridization, gasoline segment C

The Figure 3.32 shows an overview of the cost estimation of gasoline segment C. For the segments D and E, a full parallel hybrid system P2 is considered. As mentioned before the structure resembles the Diesel variants. Since no manual transmission is selected in the segments D and E, the cost difference between start/stop and advanced start/stop can be attributed to the modification of the EMS.



Main cost influencing technical differences

Start/stop:

- → Different kind of Sensors are added
- → Starter, Battery and Alternator have to be modified

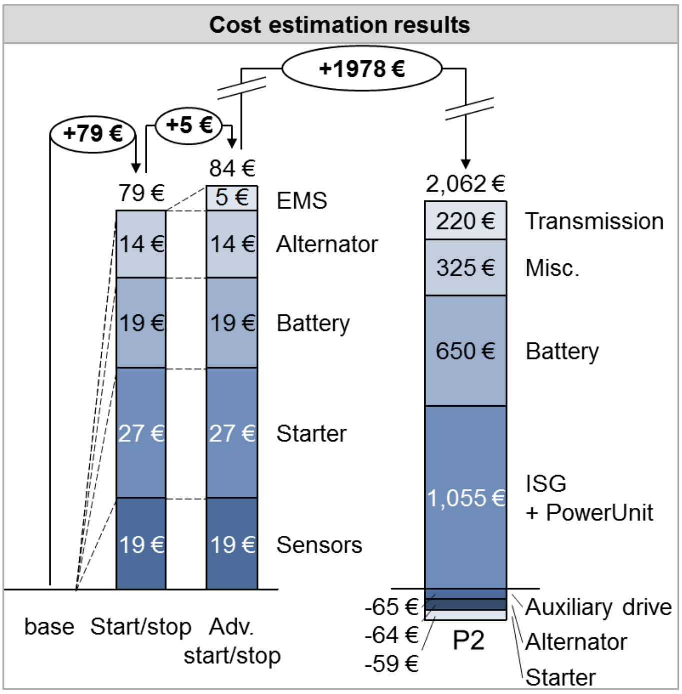
Adv. start/stop:

- → The more advanced function is provided by a modification of the Engine Management System

P2:

- → Battery: 350V HV-battery is considered
- → ISG and Power Unit including a DC/DC converter are added to system
- → Misc:
  - → HV-Wiring is needed
  - → Electrical A/C Compressor is considered
  - → HCU/VCU are modified
- → Transmission has to be modified
- → Auxiliary drive, Alternator and Starter are omitted

Figure 3.33 Cost estimation, hybridization, gasoline segment D



Main cost influencing technical differences

Start/stop:

- → Different kind of Sensors are added
- → Starter, Battery and Alternator have to be modified

Adv. start/stop:

- → The more advanced function is provided by a modification of the Engine Management System

P2:

- → Battery: 350V HV-battery is considered
- → ISG and Power Unit including a DC/DC converter are added to system
- → Misc:
  - → HV-Wiring is needed
  - → Electrical A/C Compressor is considered
  - → HCU/VCU are modified
- → Transmission has to be modified
- → Auxiliary drive, Alternator and Starter are omitted

Figure 3.34 Cost estimation, hybridization, gasoline segment E

To sum up, the cost estimation results of all gasoline segments are provided in Table 3.27.

Segment	Power [kW]	Base variant	Variant 1	Delta cost	Variant 2	Delta cost	Variant 3	Delta cost
B	60	Without	Start/stop	+ 73 €	P0 48V	+ 690 €	-	-
C	80	Without	Start/stop	+ 79 €	Advanced Start/stop	- 12 €	P0 48V	+ 694 €
D	110	Without	Start/stop	+ 70 €	Advanced Start/stop	+ 5 €	P2	+ 1870 €
E	150	Without	Start/stop	+ 79 €	Advanced Start/stop	+ 5 €	P2	+ 1978 €

Table 3.27 Cost estimation results, gasoline vehicle segments, summary

3.5 Cost estimation Turbocharger Diesel engines:

This section covers the cost calculation when upgrading the single stage turbochargers (TCs) to two-stage in Diesel engines. In the case of downsizing steps, the costs coming from the TC adaptation are covered within the cost estimation of different engine sizes. Therefore, and for Diesel, only at 3 segments, i.e. E, SUV and LCV, the TC is upgraded from single to double-stage and the related costs are depicted hereinafter.

3.5.1 Description and hardware determination

For turbochargers, technology state of the art techniques and hardware are assumed. In the case of Diesel, all engines mount at least a single-stage turbocharger (there is no NA engine). For some of the segments, the downsizing step requires to upgrade the turbochargers from 1 stage to double stage due to the higher specific required power and in order to keep the performance at a reasonable level for the driver. This happens in the segments E/SUV (both mount the same engine) and LCV. Therefore and starting from the downsized version, two-stage turbochargers are serial connected, in contrast with the single-stage TC used for the base variants of these segments. For all two-stage turbochargers mounted in the diesel engines, one of the TCs is a variable geometry turbine (VGT or VNT indistinctively), managed by an electrical actuator, while the other is typically controlled by a waste-gate valve if necessary.

Therefore, the upgrading of the TC occurs in the Diesel engine only at segments E, SUV and LCV. Finally, the cost of the adaptation of the single-stage TCs when modifying the engine size is assumed negligible, and therefore not calculated.

Segment	Base	Variant
E/SUV	Single-stage TC with VGT	Two-stage TC with VGT
LCV	Single-stage TC with VGT	Two-stage TC with VGT

Table 3.28 Overview of selected Turbocharger systems for Diesel engines

3.5.2 Components of Turbocharger systems and cost influencing parameter

The turbocharger components are assigned to different component groups (Table 3.29). Costs for compressor, turbine and bearing housing (housings) are influenced by their weight as well as by the material (depending on engines exhaust temperature).

Costs for the core unit are also depending on the engine power. Three different sizes of core units with various shaft diameters are assumed for all turbocharges. The higher the engine power, the bigger the assumed shaft diameter. The bearing housing is also scaled in discrete steps, since the standard today is purchased the bearing housing and core unit as a unit, depending on the engine rated power. At the same time, the turbine and compressor wheel are scaled also depending on the engine power. All other groups are assumed as non-variable.

	Components	Cost influencing parameter
Turbine housing	Turbine housing	Weight and exhaust temperature
Bearing housing	Bearing housing	Weight and exhaust temperature
Compressor housing	Compressor housing	Weight and exhaust temperature
Core unit	Turbine wheel Turbine shaft Flinger sleeve Compressor wheel Bearing Nut	Engine power Engine power Engine power Engine power Engine power Engine power
VGT/ Waste gate	VGT unit/ Waste gate mechanism Actuator	Non-variable Non-variable
Others TC	Heat shield Back plate Sealing's Bolts	Non-variable Non-variable Non-variable Non-variable
Assembly and overhead	Assembly Overhead	Non-variable Non-variable
Supplied parts	Tubes Clamps Oil supply Water supply*	Non-variable Non-variable Non-variable Non-variable

Table 3.29 Turbocharger component groups and cost influencing parameter

3.5.3 Vehicle segment scaling methodology

Detailed costs for all cost variable components have been scaled based on benchmark costs. Table 3.30 shows the scaling factors for each component group.

Segment	Turbine housing	Bearing housing	Compressor housing	Core unit	VGT/ Waste gate	Others	Assembly & OH	Supplied parts
LCV diesel	100%	100%	100%	100%	100%	100%	100%	100%
E/SUV diesel	105%	107%	121%	107%	100%	100%	104%	100%

Table 3.30 Overview of the turbocharger technology scaling factors for Diesel engines

3.5.4 Cost estimation result - turbocharger system Diesel segments

To upgrade a single-stage turbocharger with VGT to a two-stage turbocharger with VGT, an additional turbocharger has to be considered for segment E. Furthermore tubes and clamps for serial connection as well as an additional oil supply are included in the calculation. The assumptions for the LCV segment are equivalent. Figure 3.35 shows the cost estimation results for segments LCV and E of Diesel engines:

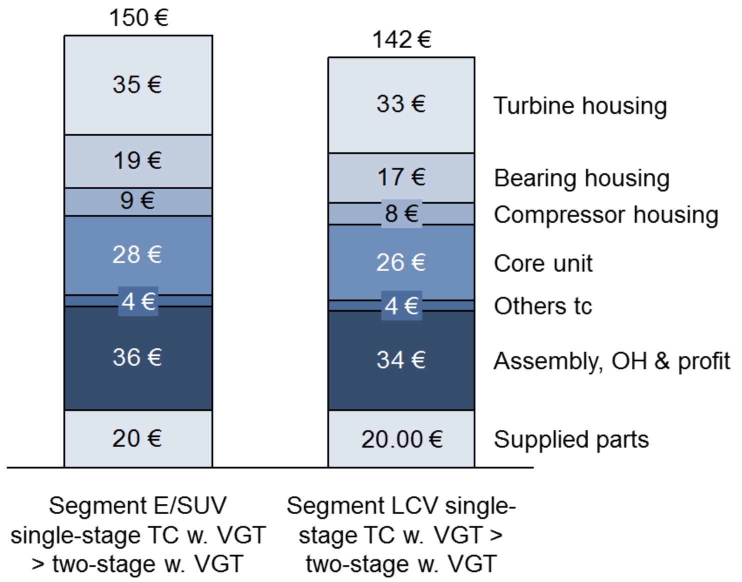


Figure 3.35 Cost estimation turbocharger technology, Diesel engines

Delta costs for changing a single-stage turbocharger with VGT to a two-stage turbocharger with one VGT controller in segment E are 150 €. The corresponding delta costs for LCV segment are 142 €. An overview is shown in Table 3.31.

Segment	Base	Variant	Delta cost
E/SUV	Single-stage TC with VGT	Two-stage TC with VGT	+150 €
LCV	Single-stage TC with VGT	Two-stage TC with VGT	+142 €

Table 3.31 Cost estimation, Turbocharger technology, Diesel engines, overview

3.6 Cost estimation Turbocharger gasoline engines:

Similarly as for Diesel engines, cost coming from installing turbochargers and/or upgrading from single-stage TCs to two-stage TCs in gasoline engines is described in this section. The related costs to TC adaptations or installation due to engine size changes are depicted in the Section 3.8.

3.6.1 Description and hardware determination

For gasoline engines, the technical variants are shown in Table 3.32; where the base TCs install a single-stage TC with a waste-gate control. In Segments C and D, the technical difference between base and variant 1 is an additional serial connected turbocharger, keeping the waste-gate control. In Segment E variant 1 is realized by an additional VGT (Variable Turbine Geometry) unit and variant 2 by a 2nd turbocharger with waste gate. The VGT of the 1st variant is not installed anymore in the variant 2.

Segment	Base	Variant 1	Variant 2
B	Single-stage TC	-	-
C	Single-stage TC	Two-stage TC	-
D	Single-stage TC	Two-stage TC	-
E	Single-stage TC	Single-stage TC with VGT	Two-stage TC w/o VGT

Table 3.32 Overview of selected turbocharger for gasoline engines

3.6.2 Components of Turbocharger systems and cost influencing parameter

The cost influencing parameters for gasoline are the same as for Diesel (Table 3.29). For gasoline turbochargers, an additional water supply in group “supplied parts” has been considered.

3.6.3 Vehicle segment scaling methodology

Detailed costs for all cost variable components have been scaled based on benchmark data. Table 3.33 shows the results for each component group. Percentages for component group “VGT/ Waste gate” cannot be calculated because the base (segment C) has no delta costs in this component group. The waste gate costs for segment C are considered in the engine delta cost estimation.

Segment	Turbine housing	Bearing housing	Compressor housing	Core unit	VGT/ Waste gate	Others	Assembly & OH	Supplied parts
C gasoline	100%	100%	100%	100%	-	100%	100%	100%
D gasoline	113%	125%	127%	115%	-	100%	110%	100%
E gasoline (V1)	0%	0%	0%	0%	-	0%	5%	0%
E gasoline (V2)	127%	133%	157%	133%	-	100%	120%	100%

Table 3.33 Overview of Turbocharger technology scaling factors for gasoline engines

Bear in mind that for segment E variant 1, the additional costs of the VGT are considered, while the waste-gate cost is subtracted from the engine side.

3.6.4 Cost estimation result - turbocharger system, gasoline segments

Figure 3.36 shows an overview of the gasoline results. Main cost influencing parameters are further examined in the following: for each segment C and D another complete turbocharger has to be considered. In addition to compressor, bearing, and turbine housing; an extra core unit (w/o VGT) has to be calculated. Moreover, tubes and clamps for serial connection as well as an additional oil and water supply plus miscellaneous components e.g. additional sealings are the main cost influencing parameters and have to be considered.

For segment E V1, the waste gate (including a pneumatic actuator) is replaced by a VGT unit with an electrical actuator. For segment E V2, a complete 2nd turbocharger (including a waste gate with a pneumatic actuator) has been considered. The elimination of the electrical driven VGT allows saving around 58 €.

In total, an upgrade from a single-stage TC to a double-stage TC in the segment C in gasoline costs 178 € extra. For the D segment, the same change costs 200 €. While in the segment E, the VGT single-stage TC costs around 59 € with respect to a single-stage TC controlled by a wastegate; at the same time, the installation of a double-stage TC (with waste-gate control) cost 163 € more than the single-stage TC with VGT, and 222 € more with respect to the single-stage with waste-gate.

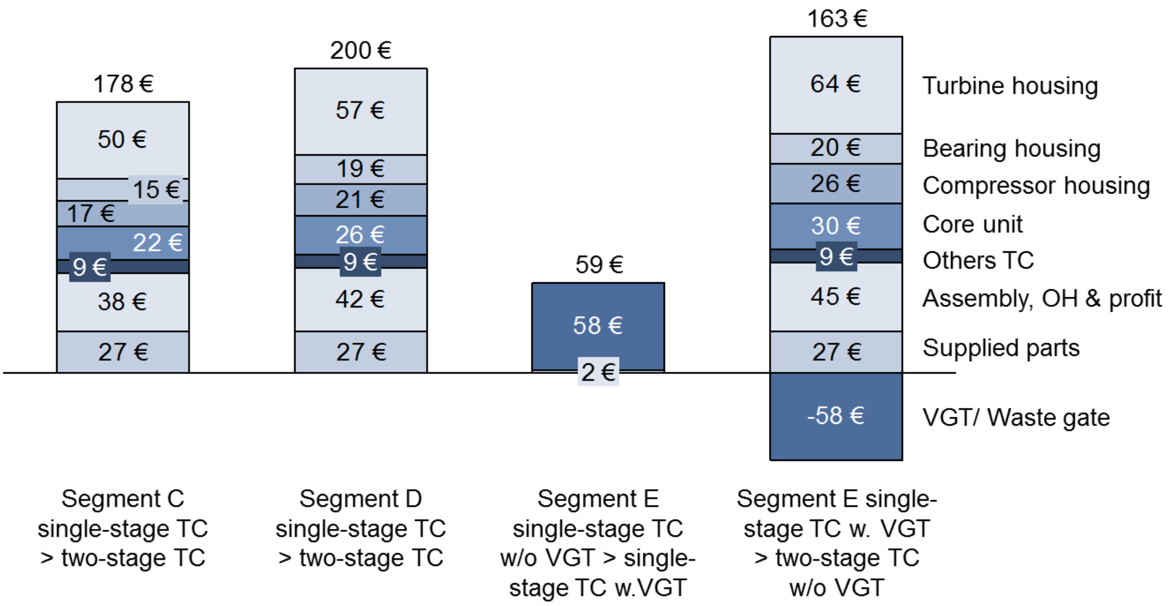


Figure 3.36 Cost estimation - turbocharger technology, gasoline engines

An overview for Turbocharger technology cost (gasoline engine) is shown in Table 3.34.

Segment	Base	Variant 1	Delta cost	Variant 2	Delta cost
B	Single-stage TC	-	-	-	-
C	Single-stage TC	Two-stage TC	+178 €	-	-
D	Single-stage TC	Two-stage TC	+200 €	-	-
E	Single-stage TC	Single-stage TC with VGT	+59 €	Two-stage TC w/o VGT	+163 €

Table 3.34 Cost estimation, Turbocharger technology, gasoline engines, overview

3.7 Cost estimation engine size Diesel engines:

This section shows the cost calculation for the Diesel downsizing versions.

3.7.1 Description and hardware determination

For the cost estimation of the selected Diesel engines the cost structure of a standard 2.0 L 4 cylinder (single) turbocharged Diesel benchmarked engine was used as reference for the base engine. Components of this engine are e.g. a cast iron crankcase, an aluminium cylinder head, a forged crankshaft, a built camshaft, solenoid injectors and a VGT turbocharger. The component cost for the 3 and 4 cylinder engines with a different displacement are scaled from this selected engine. For the base powertrains of segments E and SUV, the cost structure of a 3.0 L 6 cylinder Diesel engine with comparable technology was used. Table 3.35 shows the considered engine variants for Diesel vehicle classes:

Segment	Base variant	Variant
B	3 cylinder, 1.4l	3 cylinder, 1.2l
C	4 cylinder, 1.6l	3 cylinder, 1.4l
D	4 cylinder, 2.0l	4 cylinder, 1.6l
E	6 cylinder, 3.0l	4 cylinder, 2.0l
SUV	6 cylinder, 3.0l	4 cylinder, 2.0l
LCV	4 cylinder, 2.2l	4 cylinder, 1.8l

Table 3.35 Overview of selected engine sizes for Diesel engines

3.7.2 Components of different Diesel engines and cost influencing parameter

The first step for the engine cost calculations was a breakdown of the benchmark engines to different functional groups like valve train, crank train, cooling system or engine electronics. In a second step all engine components are allocated to one of this groups, e.g. the group “Crankcase” contains the components crankcase, main bearing caps, main bearings, a mass balancing system (for 3 cylinder engines) and some smaller parts like bolts. Table 3.36 shows the breakdown to functional groups and the components which belong to each group.

Component group	Components
Crankshaft	Crankshaft, Con-rod, Pistons, Piston-rings, misc.
Crankcase	Crankcase, Main bearing caps, main bearings, mass balancing system (3 cylinder), misc.
Cylinder head	Cylinder head, cylinder head cover, gasket, misc.
Valve train	Valves, camshafts, cam bearing frame, camp haser, misc.
Timing drive	Tensioner, idler, timing belt/chain, guides, misc.
Accessory drive	Belt tensioner, idler, belt
Intake system	Intake manifold, gaskets, misc.
Fuel injection system	Injection pump, rail, injectors, pipes, misc.
Lubrication system	Oil pan, oil pump, dipstick, oil filter, tubes, pipes, misc.
Cooling system	Water pump, thermostat, water pipes, misc.
Induction air charging	Turbo charger, intercooler, gaskets, misc.
Breather system	Oil separator, back pressure valve, misc.
Exhaust system	Catalyst, exhaust manifold, gasket, misc.
Engine electronics	ECU, wiring harness, sensors, spark plugs, ignition coils, throttle valve, misc.
Auxiliaries	Starter, generator, vacuum pump, air-conditioning compressor
Flywheel	Flywheel (one-mass, dual-mass or dual-mass with centrifugal pendulum)
Engine assembly	Engine assembly (3,4,6 cylinder)

Table 3.36 Diesel engines component groups and components

3.7.3 Vehicle segment scaling methodology

In the next step costs for each single engine components were estimated. Based on the costs from used benchmark engines the costs for main components have been scaled for each engine. For this approach the most cost influencing parameter for each component were defined. For example the changing in displacement has a big influence on crankcase cost while valves have been scaled by the specific power per displacement. Table 3.37 shows this scaling parameter for the main components.

Components	Scaling parameter
Crankshaft	Power, no. of cylinder
Con-rod	Specific torque per cylinder, no. of cylinder
Piston	Bore (depending on displacement)
Crankcase	Displacement, power, no. of cylinder
Cylinder head	Power, no. of cylinder
Camshaft	No. of cylinder
Valves	Specific power per displacement
Oil pump	Engine power, no. of bearings
Oil cooler	Power, injection kind (NA, DI)
Waterpump	Small influence of engine power
ECU	No. of cylinder, turbocharged (yes/no)
Generator	Small influence of engine power

Table 3.37 Diesel engines components and scaling parameter for cost estimation

By using the influence of the scaling parameter on the component cost different scaling factors have been calculated. For example cost for a piston change from the 100% at base engine (2.0l, 110kW) to 95% for the 1.6l engine at segment D variant1 with the same power. The scaling factors are shown in Table 3.38.

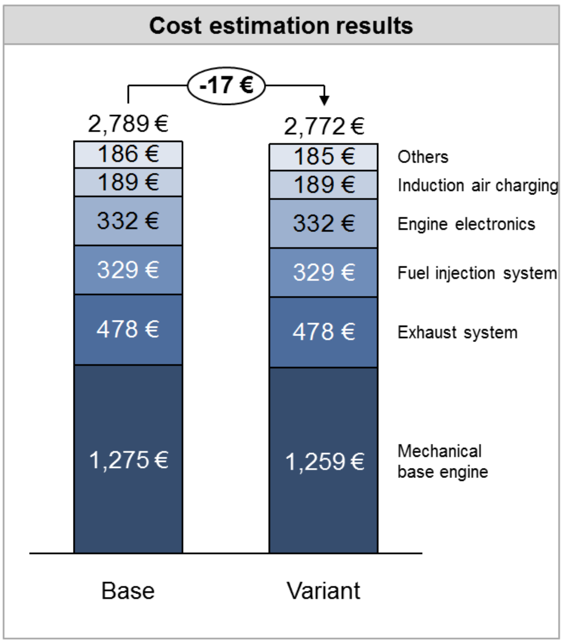
Segment	B	B	C	C	D	D	E/SUV	LCV	LCV
Cylinder	3	3	4	3	4	4	4	4	4
Displacement [Liters)	1.4	1.2	1.6	1.4	2	1.6	2	2.2	1.8
Power [kW]	60	60	80	80	110	110	150	120	120
Crankshaft	77%	77%	92%	84%	100%	100%	111%	103%	103%
Crankcase	78%	74%	92%	80%	100%	95%	104%	103%	98%
Cylinder head	80%	76%	92%	82%	100%	95%	104%	103%	98%
Piston	98%	95%	95%	98%	100%	95%	100%	103%	98%
Con-rod	95%	95%	95%	99%	100%	100%	107%	102%	102%
Valves	100%	100%	100%	100%	100%	100%	100%	100%	100%
Camshaft	85%	85%	100%	85%	100%	100%	100%	100%	100%
Oil pump	95%	95%	100%	95%	100%	100%	100%	100%	100%
Oil cooler	90%	90%	100%	90%	100%	100%	100%	100%	100%
Waterpump	95%	95%	95%	99%	100%	100%	107%	102%	102%
ECU	90%	90%	100%	90%	100%	100%	100%	100%	100%
Generator	95%	95%	95%	99%	100%	100%	107%	102%	102%

Table 3.38 Overview of the Diesel engines scaling factors for single components for all scaled vehicle segments (except 6 cylinder)

In the next step the quantity of each component per engine needs to be multiplied by the scaled component cost, e.g. all 4-cylinder engines require 4 pistons, all 3-cylinder engines require 3 pistons. With this approach costs for each component group for all selected engines have been estimated. The results compared to the respective group costs of the base engines lead to the engine delta costs for each variant per segment.

3.7.4 Cost estimation result Diesel engines

At segment B, the displacement changes from 1.4l to 1.2l with the same number of cylinders (3). Therefore, the cost savings are estimated to be 17 €. This delta cost results from the minimal changings of bore and stroke which have a small cost influence on crankcase, cylinder head and piston costs. Figure 3.37 shows the delta costs for segment B. Figure 3.38 shows the delta cost for the mechanical base engine which contains all the mechanical parts.



Main cost influencing technical differences

Mechanical base engine

→ Bore and stroke of the crankcase are a little bit smaller due to the displacement decrease

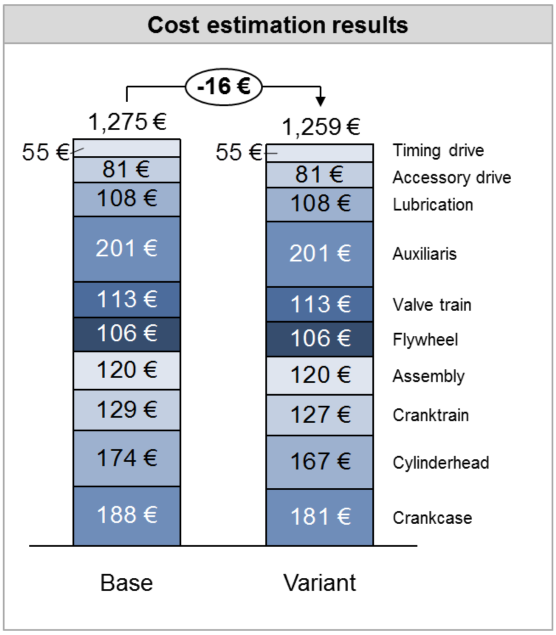
→ The cylinder head is a little bit smaller due to the bore decrease

→ The piston is a little bit smaller due to the bore decrease

Other systems

→ The other system costs are not influenced by the displacement decrease

Figure 3.37 Cost estimation, engine size, segment B, divided by component groups



Main cost influencing technical differences

Crankcase

→ Bore and stroke of the crankcase are a little bit smaller due to the displacement decrease

Cylinder head

→ The cylinder head is a little bit smaller due to the bore decrease

Cranktrain

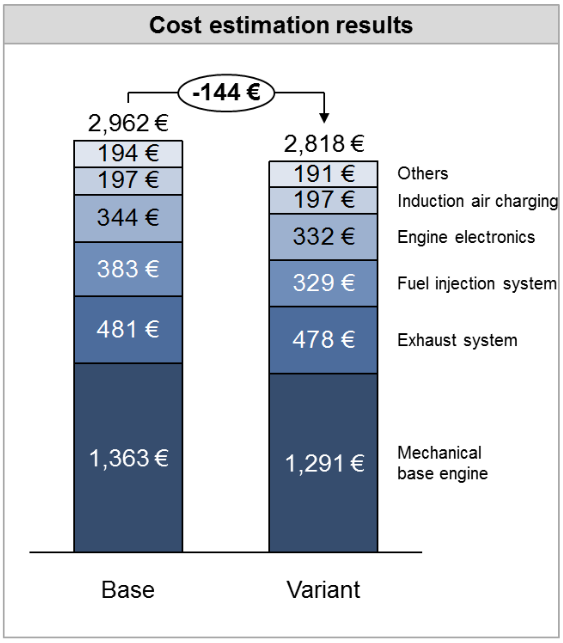
→ The piston is a little bit smaller due to the bore decrease

Other parts

→ Costs for other parts are not influenced by the displacement decrease

Figure 3.38 Cost estimation, engine size, segment B, mechanical base engine, divided by component groups

At segment C the engine changes from a 4 cylinder 1.6l engine to a 3 cylinder 1.4l engine. This leads to a higher cost reduction at crankcase, cylinder head and crank train than in segment B. However, a mass balancing system for the 3 cylinder engine has to be added to the crankcase group. Small cost reductions originate from a reduced fuel injection system, a smaller exhaust manifold and reduced engine electronics. The delta cost for segment C are shown in Figure 3.39. The delta cost for the base engines of segment C are shown in Figure 3.40.



Main cost influencing technical differences

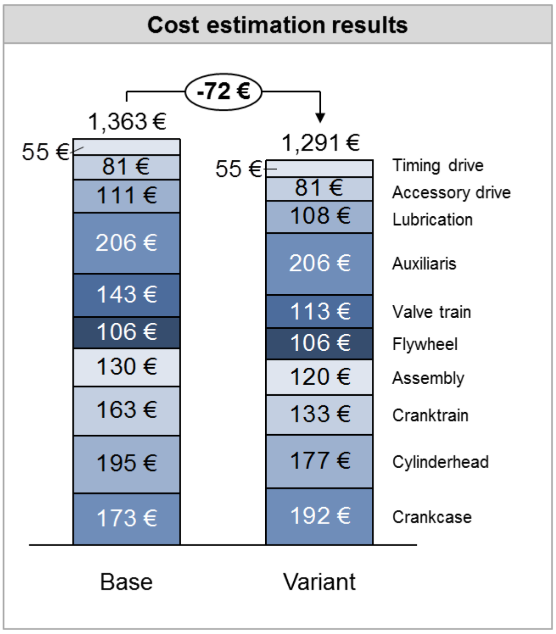
Mechanical base engine
- → The crankcase is smaller due to the cylinder decrease
- → The cylinder head is smaller due to the cylinder decrease
- → 3 instead of 4 pistons and con-rods
- → Additional mass balancing system

Fuel injection system
- → 3 instead of 4 injectors
- → Smaller rail necessary

Exhaust manifold
- → Smaller exhaust manifold

Engine electronics
- → ECU and wiring harnesses for 3 cylinder engine instead for 4 cylinder engine

Figure 3.39 Cost estimation, engine size, segment C, divided by component groups



Main cost influencing technical differences

Crankcase
- → Crankcase is smaller due to 3 cylinder engine
- → Additional mass balancing system for 3 cylinder engine

Cylinder head
- → Smaller cylinder head due to 3 cylinder engine

Cranktrain
- → Smaller crankshaft
- → 3 Pistons and Con-rods instead of 4

Assembly
- → Assembly of 3 cylinder engine instead of 4 cylinder engine

Valve train
- → Shorter camshaft, valves etc. only for 3 cylinders

Figure 3.40 Cost estimation, engine size, segment C, mechanical base engine, divided by component groups

The differences at segment D are comparable to segment B: only the displacement has changed. This leads to small delta costs of -23€. The cost estimation results are shown in Table 3.41. The results for the mechanical base engine costs are shown in Figure 3.42.

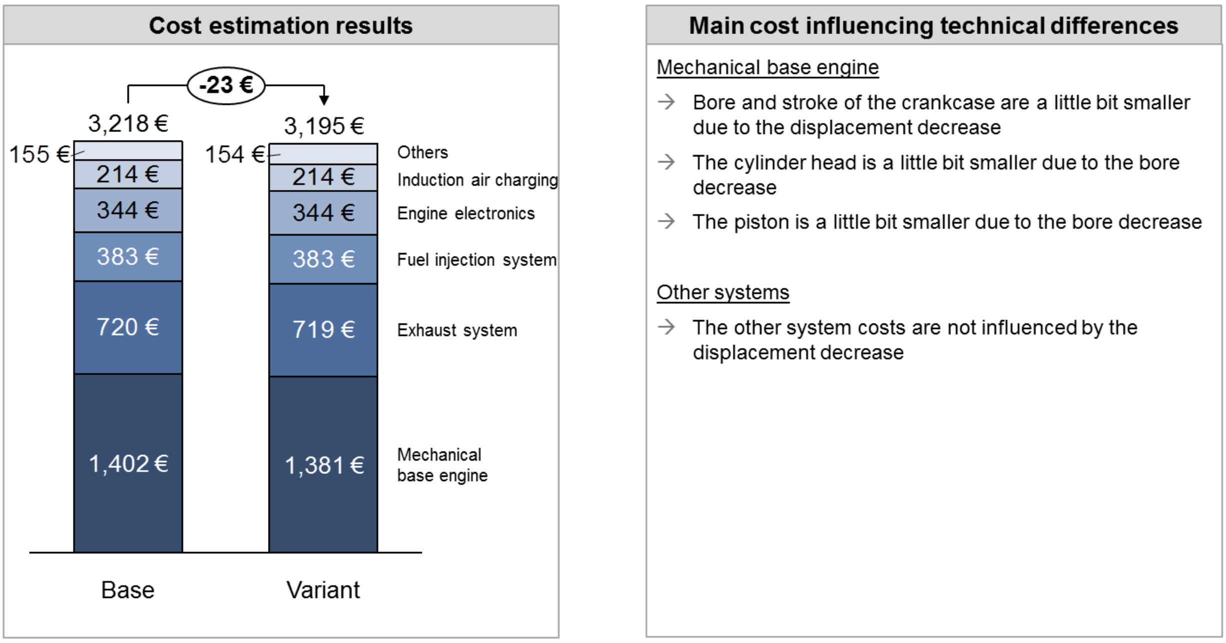


Figure 3.41 Cost estimation, engine size, segment D, divided by component groups

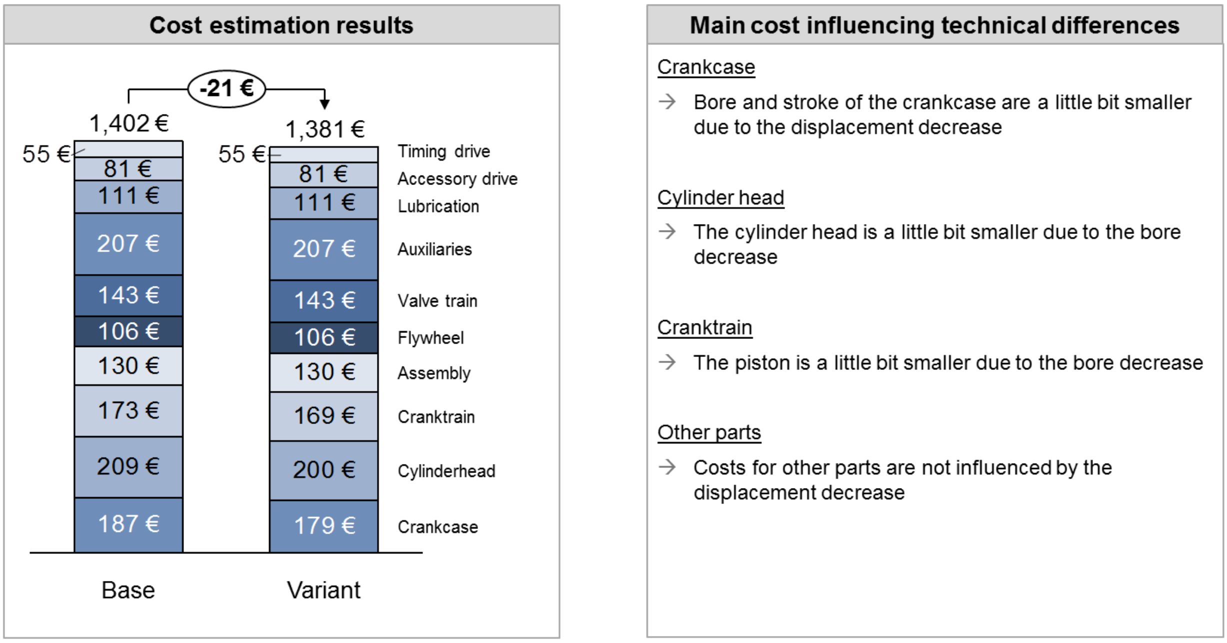


Figure 3.42 Cost estimation, engine size, segment D, mechanical base engine, divided by component groups

At segment E and SUV a 6 cylinder inline engine is reduced to a 4 cylinder inline engine. The cost saving is estimated to be 412€. At the mechanical base engine crankcase, cylinder head and crankshaft are smaller. The crank train contains only 4 instead of 6 pistons which allow to save several engines parts and material, e.g. con-rods, valves, among others. The fuel injection system is reduced to 4 injectors and a smaller rail. The engine electronic costs decrease due to a less complex ECU and wiring harness. The 4 cylinder engine at the variant contains two turbocharger instead of one at the base engine. The cost reduction for other component groups is small. Both engines mount an automatic transmission. Therefore they do not use a flywheel like the selected engines in other segments but a flex-plate only. This results in cost of only 17€ in “group flywheel”. Figure 3.43 shows the cost comparison of segment E and SUV. Figure 3.44 shows the cost comparison of the mechanical base engines for segment E and SUV.

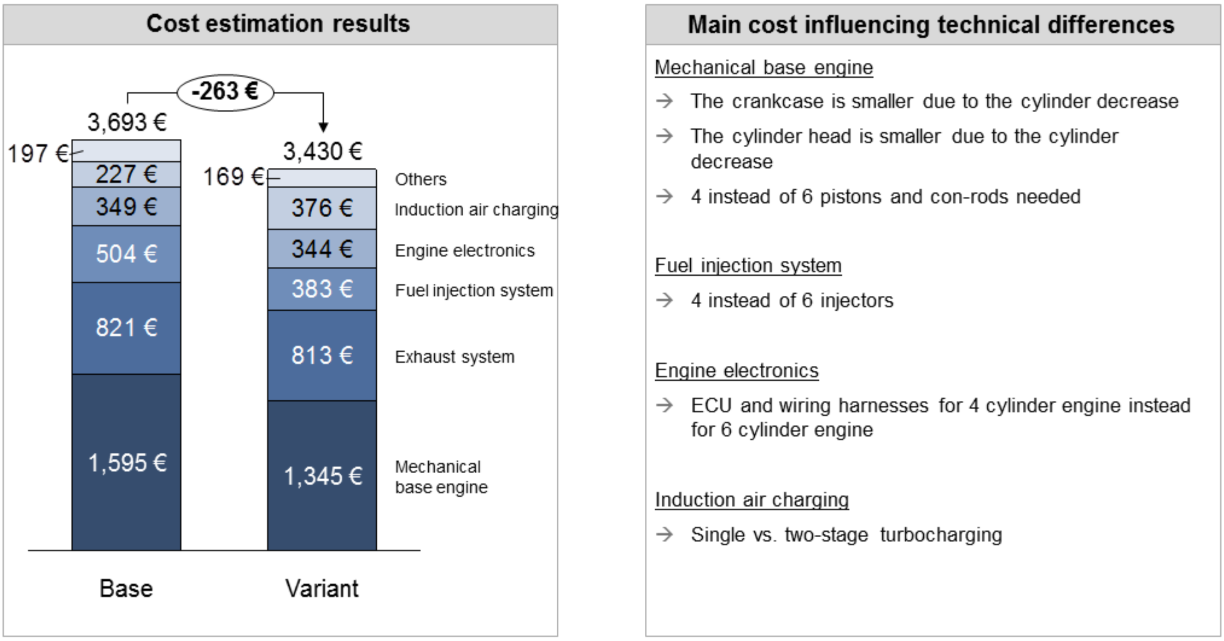


Figure 3.43 Cost estimation, engine size, segment E and SUV, divided by component groups

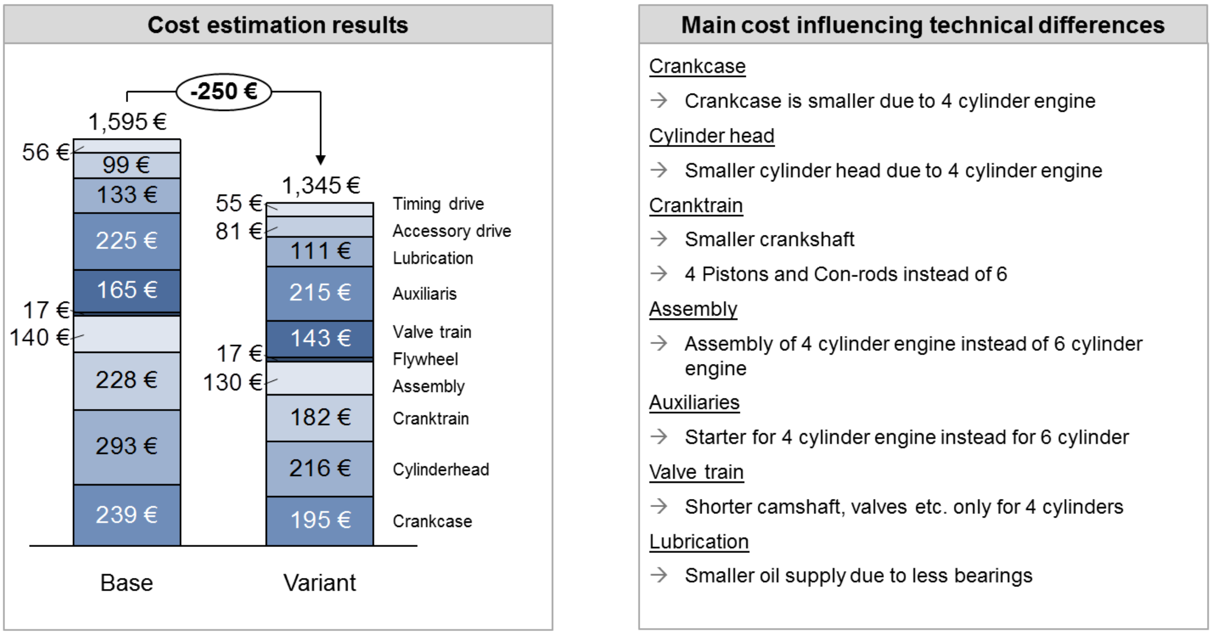


Figure 3.44 Cost estimation, engine size, segment E and SUV, mechanical base engine, divided by component groups

At Segment LCV, the displacement of the engine changes from 2.2l to 1.8l. The delta costs are resulting from smaller crankcase, cylinder head and pistons. The engine with less displacement at the variant contains two turbocharger instead of one at the base engine. Table 3.45 shows the delta costs for segment LCV. Figure 3.46 shows the delta cost of the mechanical base engines for segment LCV.

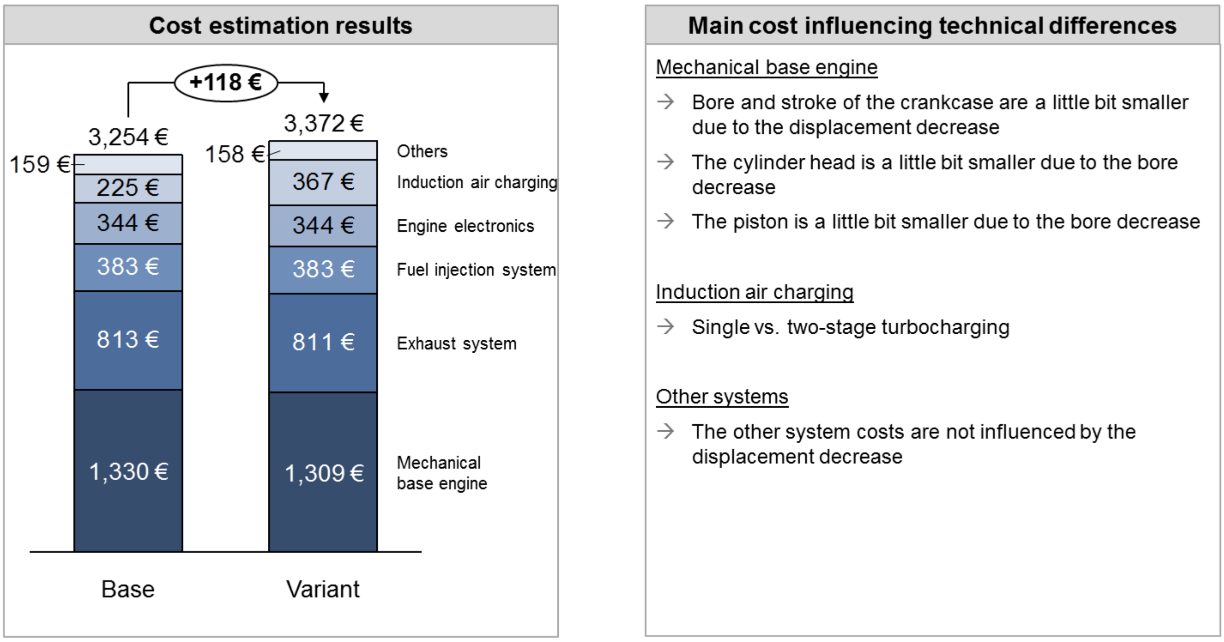


Figure 3.45 Cost estimation, engine size, segment LCV, divided by component groups

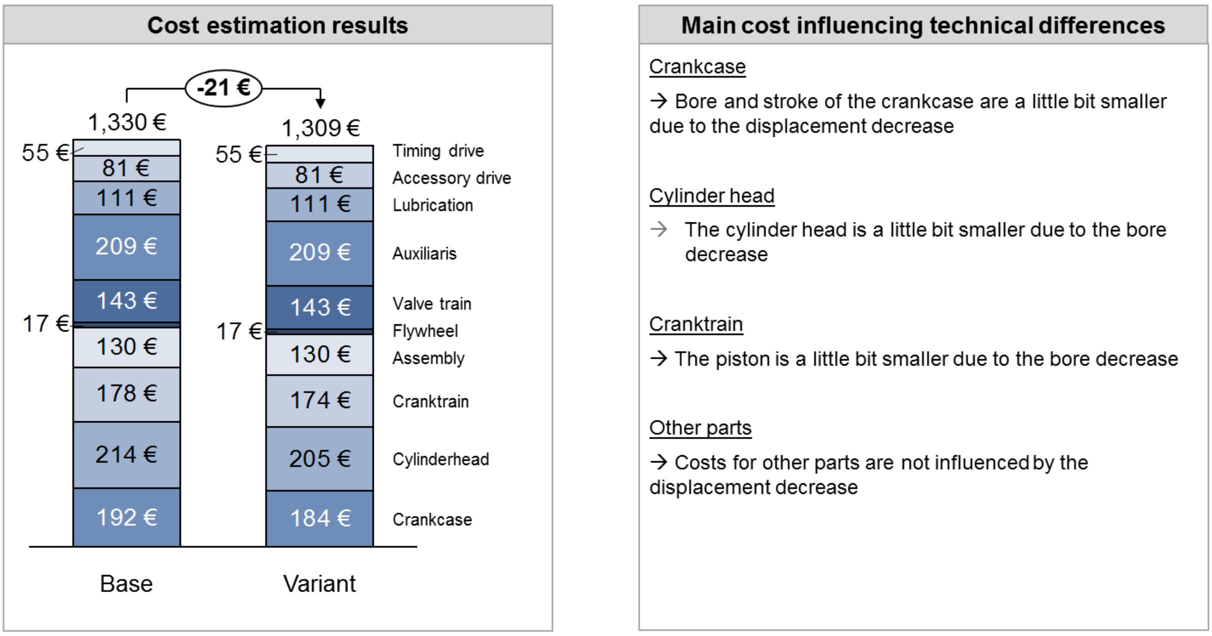


Figure 3.46 Cost estimation, engine size, segment LCV, mechanical base engine, divided by component groups

An overview of the estimated delta costs at Diesel engines for engine downsizing including changes for additional turbochargers (double stage turbochargers required for the downsized versions of the Diesel E, SUV and LCV classes) as well as downsizing-based changes at the after-treatment system is shown in Table 3.39.

Segment	Base	Variant	Delta cost
B	3 cylinder, 1.4l	3 cylinder, 1.2l	-17 €
C	4 cylinder, 1.6l	3 cylinder, 1.4l	-144 €
D	4 cylinder, 2.0l	4 cylinder, 1.6l	-23 €
E	6 cylinder, 3.0l	4 cylinder, 2.0l	-263 €
SUV	6 cylinder, 3.0l	4 cylinder, 2.0l	-263 €
LCV	4 cylinder, 2.2l	4 cylinder, 1.8l	+118 €

Table 3.39 Cost estimation, engine size, Diesel engines, overview.

3.8 Cost estimation engine size gasoline engines:

This section shows the cost calculation for the gasoline downsizing versions.

3.8.1 Description and hardware determination

As base engine for of the segments B and C a typical natural aspirated 4 cylinder inline engine was used. This engine includes an aluminium crankcase and aluminium cylinder head, double overhead camshafts and a multipoint port fuel injection.

A natural aspirated V-engine was used as base engines for segments D and E. Due to the V-concept the engine contains a two-step timing drive with timing chains.

For the direct injected turbocharged engines a typical 4 cylinder inline DI engine was used and scaled to the different displacements and power variants. This engine contains an aluminium crankcase and cylinder head, double overhead camshafts and direct injection with solenoid valves. The parts of the 3 cylinder DI engines have also been scaled from this 4 cylinder DI engine.

Table 3.40 shows an overview of the selected gasoline engines for all segments.

Segment	Base variant	Variant 1	Variant 2	Variant 3	Variant 4
B	4 cylinder, 1.3l, NA	3 cylinder, 1.0l, DI	3 cylinder, 0.8l, DI	-	-
C	4 cylinder, 1.8l, NA	4 cylinder, 1.4l, DI	3 cylinder, 1.0l, DI	3 cylinder, 0.8l, DI	-
D	6 cylinder, 2.4l, NA	4 cylinder, 1.8l, DI	4 cylinder, 1.4l, DI	3 cylinder, 1.0l, DI	-
E	6 cylinder, 2.6l, NA	4 cylinder, 2.4l, DI	4 cylinder, 2.0l, DI	4 cylinder, 1.6l, DI	3 cylinder, 1.2l DI

Table 3.40 Overview of selected engine sizes for gasoline engines

3.8.2 Components of gasoline engines and cost influencing parameter

The approach for gasoline engines was similar to the Diesel engines: The engines have been split into the same functional groups. The groups are shown in Table 3.41, while the components of each group and the most cost influencing scaling parameter for each part are shown in Table 3.42.

Component group	Components
Crankshaft	Crankshaft, Con-rod, Pistons, Piston-rings, misc.
Crankcase	Crankcase, Main bearing caps, main bearings, mass balancing system (3 cylinder), misc.
Cylinder head	Cylinder head, cylinder head cover, gasket, misc.
Valve train	Valves, camshafts, cam bearing frame, camp haser, misc.
Timing drive	Tensioner, idler, timing belt/chain, guides, misc.
Accessory drive	Belt tensioner, idler, belt
Intake system	Intake manifold, gaskets, misc.
Fuel injection system	Injection pump, rail, injectors, pipes, misc.
Lubrication system	Oil pan, oil pump, dipstick, oil filter, tubes, pipes, misc.
Cooling system	Water pump, thermostat, water pipes, misc.
Induction air charging	Turbo charger, intercooler, gaskets, misc.
Breather system	Oil separator, back pressure valve, misc.
Exhaust system	Catalyst, exhaust manifold, gasket, misc.
Engine electronics	ECU, wiring harness, sensors, spark plugs, ignition coils, throttle valve, misc.
Auxiliaries	Starter, generator, vacuum pump, air-conditioning compressor
Flywheel	Flywheel (one-mass, dual-mass or dual-mass with centrifugal pendulum)
Engine assembly	Engine assembly (3,4,6 cylinder)

Table 3.41 Gasoline engines component groups and components

Components	Scaling parameter
Crankshaft	Power, no. of cylinder
Con-rod	Specific power per cylinder, no. of cylinder
Piston	Bore (depending on displacement)
Crankcase	Displacement, power, no. of cylinder
Cylinder head	Power, no. of cylinder
Camshaft	No. of cylinder
Valves	Specific power per displacement
Oil pump	Engine power, no. of bearings
Oil cooler	Power, injection kind (NA, DI)
Waterpump	Small influence of engine power
ECU	No. of cylinder, turbocharged (yes/no)
Generator	Small influence of engine power

Table 3.42 Gasoline engines component groups and cost influencing parameter

3.8.3 Vehicle segment scaling methodology

The cost of the naturally aspirated engines (base engines for all segments) has been calculated taken as reference representative benchmark engines with comparable technology. For the turbocharged DI-engines the calculation was done like for the Diesel vehicles, all costs have been scaled from a 4 cylinder 1.4l benchmark engine. Table 3.43 gives an overview of the scaling factors for the turbocharged engines at gasoline segments.

Segment	B	B	C	C	C	D	D	D	E	E	E	E
Cylinder	3	3	4	3	3	4	4	3	4	4	4	3
Displacement [Liters]	1	0.8	1.4	1	0.8	1.8	1.4	1	2.4	2	1.6	1.2
Power [kW]	65	65	95	95	95	135	135	135	180	180	180	180
Crankshaft	81%	80%	100%	97%	97%	121%	121%	119%	134%	145%	144%	144%
Crankcase	79%	74%	100%	81%	76%	112%	102%	83%	129%	120%	109%	90%
Cylinder head	76%	73%	100%	78%	74%	111%	104%	81%	120%	117%	111%	87%
Piston	98%	91%	100%	98%	91%	109%	100%	98%	121%	113%	105%	105%
Con-rod	97%	97%	100%	112%	112%	113%	113%	131%	124%	128%	128%	153%
Valves	100%	100%	100%	100%	140%	100%	100%	140%	100%	100%	140%	140%
Camshaft	85%	85%	100%	85%	85%	100%	100%	85%	100%	100%	100%	85%
Oil pump	95%	95%	100%	95%	95%	100%	100%	95%	100%	100%	100%	95%
Oil cooler	90%	90%	100%	90%	90%	100%	100%	90%	100%	100%	100%	90%
Waterpump	94%	94%	100%	100%	100%	107%	107%	107%	113%	116%	116%	116%
ECU	90%	90%	100%	90%	90%	100%	100%	90%	100%	100%	100%	90%
Generator	94%	94%	100%	100%	100%	107%	107%	107%	113%	116%	116%	116%

Table 3.43 Overview of the turbocharged gasoline engines scaling factors for single componets for all vehicle segments

Additionally to the cost scaling per single component the quantity of each component per engine was considered to get to the complete engine cost for the 3-cylinder and 4-cylinder engines, e.g. 4 pistons at 4-cylinder engines and 3 pistons at 3-cylinder engines.

3.8.4 Cost estimation result gasoline engines

At Segment B, the base engine changes from a natural aspirated 4 cylinder inline engine to a turbocharged 3 cylinder DI engine at variant 1. At the mechanical base engine costs for crankcase, cylinder head, crank train and valve train are less due to the cylinder reduction. The costs for a mass balancing system and a valve timing system on exhaust side are additional. The costs for the exhaust system decrease due to displacement changings. Costs for a turbocharger are added at variants 1 and 2 which makes the cost of the downsized engine higher than base engine cost. At the fuel injection system a high-pressure pump and rail and DI-injectors replace the PFI-system. Additionally at variant 2 the DI-system works with 350 bar which also leads to higher costs. Figure 3.47 shows the delta cost of segment B. Figure 3.48 shows the cost split for the mechanical base engines of segment B.

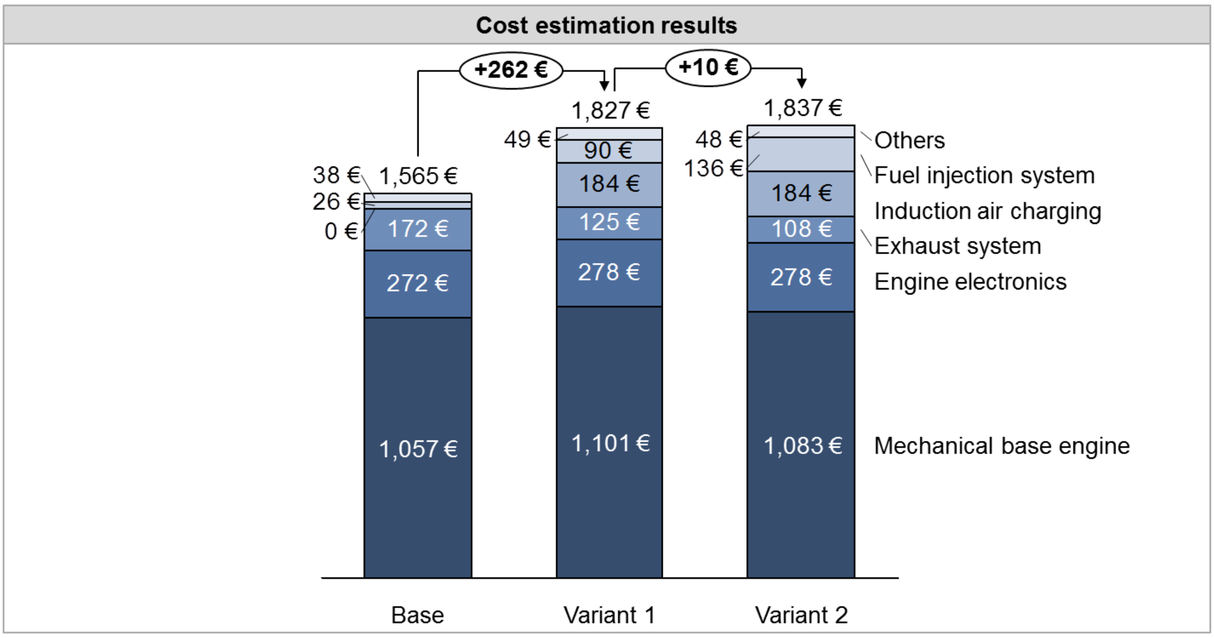


Figure 3.47 Cost estimation, engines sizes, segment B, divided by component groups

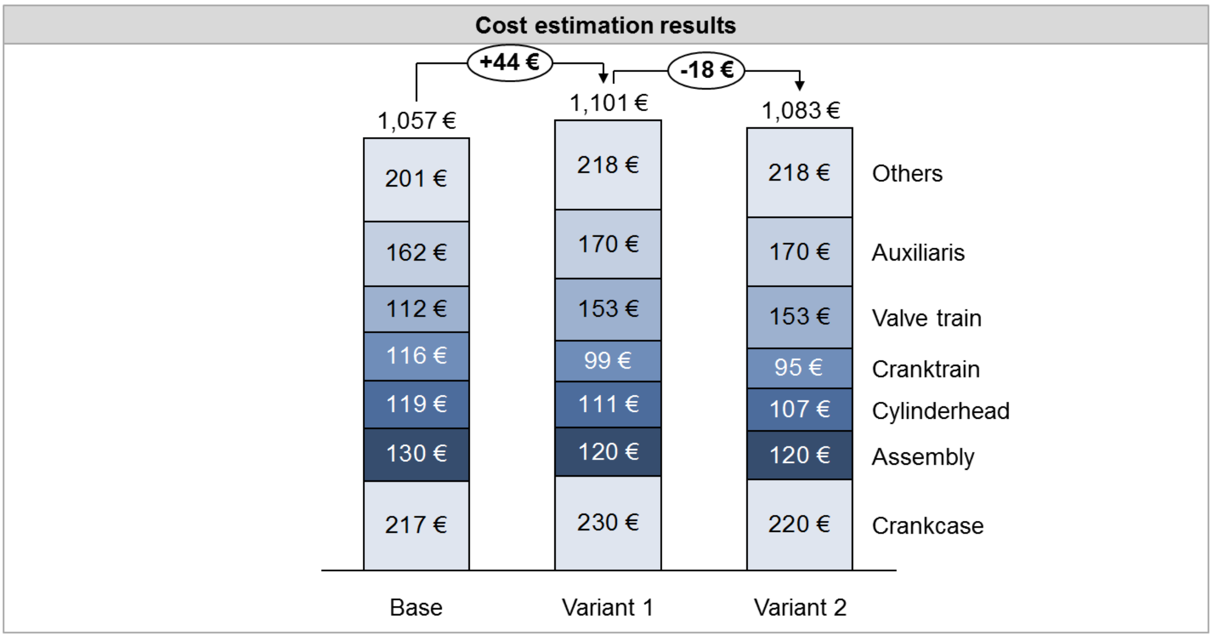


Figure 3.48 Cost estimation, engines sizes, segment B, mechanical base engine, divided by component groups

Main cost influencing technical differences
Mechanical base engine
→ The crankcase is smaller due to reduced number of cylinders (4 at base engine, 3 at variant 1 & 2)
→ Additional mass balancing system at 3 cylinder engines → total cost of group crankcase are higher for variant 1 & 2
→ The cylinder head is smaller due to reduced number of cylinders (4 at base engine, 3 at variant 1 & 2)
→ Crank train and valve train reduced from 4 to 3 cylinder (at all variants)
→ N/A engines: valve timing only at intake side → total cost of valve timing are higher for all variants due to additional cam phaser
Induction air charging
→ Additional Turbocharger for downsized engines (all variants)
Fuel injection system
→ At all variants: high-pressure fuel pump, high-pressure rail and piping, DI-injectors instead of PFI-injector
→ 350 bar system at 0.8l 3 cylinder engine (variant 2)
Others
→ Includes breather system, timing drive, accessory drive, flywheel, lubrication: small changings for DI engines (all variants)

Figure 3.49 Cost estimation, engines sizes, segment B, main cost influencing technical differences

At segment C the base engine is similar to segment B, a natural aspirated 4 cylinder inline engine with a slightly higher displacement of 1.8liter. The first variant is a 4 cylinder turbocharged engine. Mainly due to the additional turbocharger and the high-pressure fuel injection the costs for variant 1 are higher than for the base engine.

The step to the 3 cylinder engines follows in variant 2 and 3. With one cylinder less the costs for the mechanical base engine drop as well as the cost for the engine electronics. The fuel injection system changes to a 350 bar system which lead to higher cost at the 3 cylinder engines. Due to the displacement reduction the cost for the exhaust systems are sinking. Figure 3.50 shows the cost estimation results for segment C.

Figure 3.51 shows the cost split for the mechanical base engines of segment C.

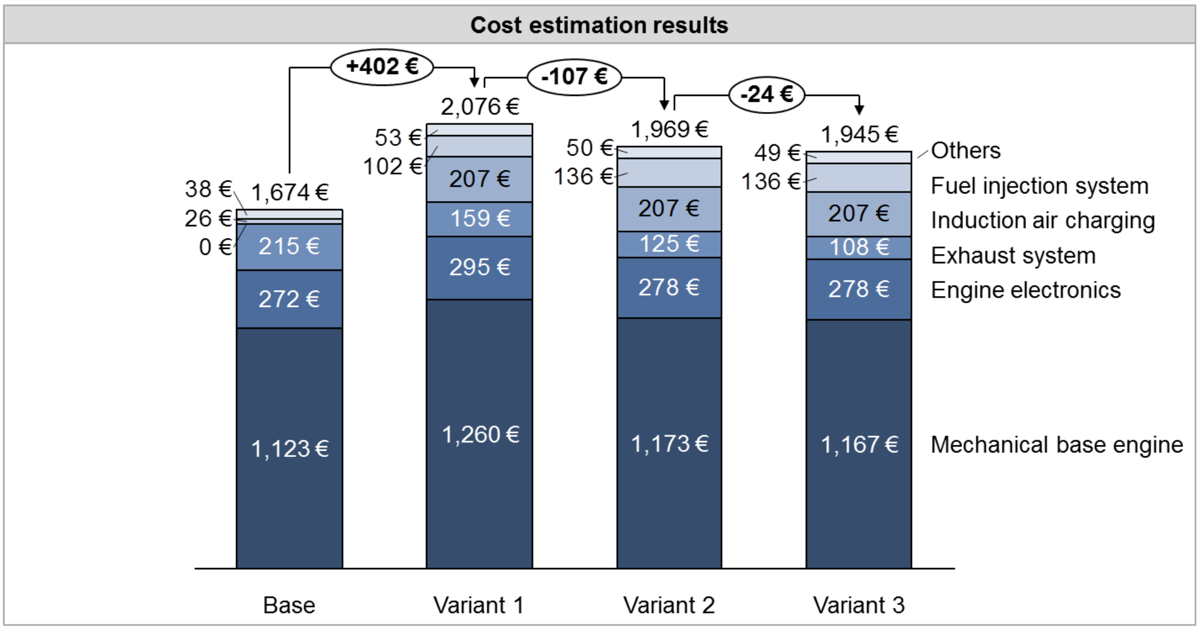


Figure 3.50 Cost estimation, engines sizes, segment C, divided by component groups

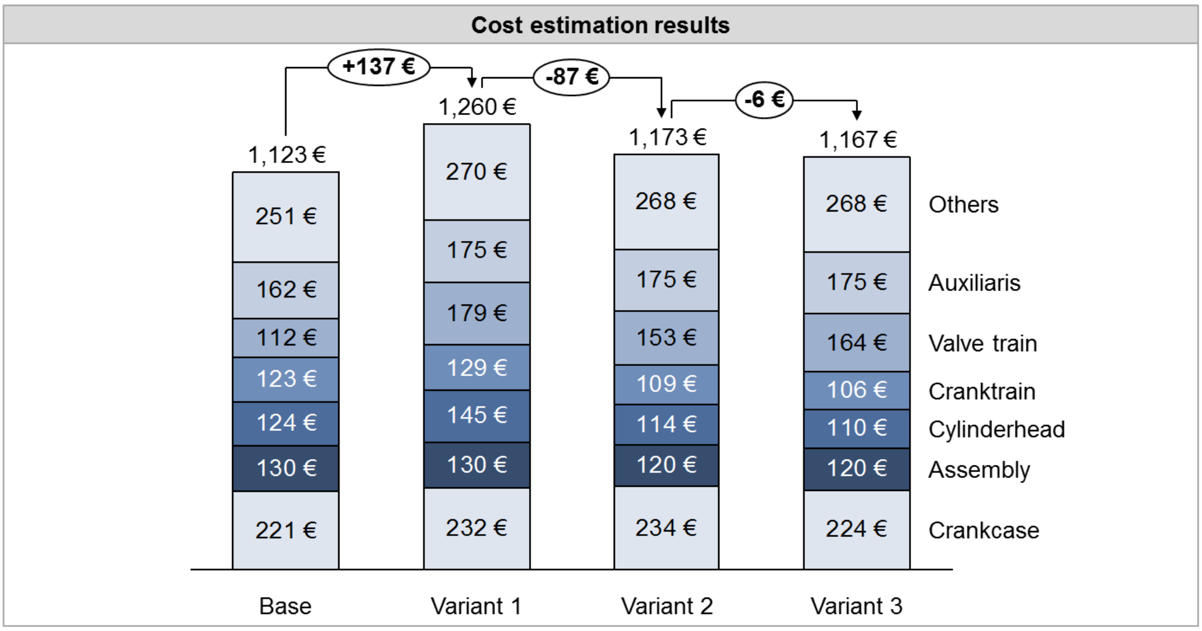


Figure 3.51 Cost estimation, engines sizes, segment C, mechanical base engine, divided by component groups

Main cost influencing technical differences
Mechanical base engine
→ The crankcase is smaller due to reduced number of cylinders for variants 2 & 3
→ Additional mass balancing system at 3 cylinder engines → total cost of group crankcase are higher for variants 2 & 3
→ The cylinder head is smaller due to reduced number of cylinders
→ Crank train and valve train reduced from 4 to 3 cylinder at variants 2 & 3
→ N/A engines: valve timing only at intake side → total cost of valve timing are higher for all variants due to additional cam phaser
Induction air charging
→ Additional Turbocharger for downsized engines (all variants)
Fuel injection system
→ At all variants: high-pressure fuel pump, high-pressure rail and piping, DI-injectors instead of PFI-injector
→ 350 bar system at 3 cylinder engines (variants 2 & 3)
Others
→ Includes breather system, timing drive, accessory drive, flywheel, lubrication: small changings for DI engines (all variants)

Figure 3.52 Cost estimation, engines sizes, segment C, main cost influencing technical differences

Base engine for segment D is a natural aspirated 6 cylinder V-engine. At variant 1 the mechanical base engine changes to a 4 cylinder inline engine. Therefore costs for crankcase, cylinder head, crank train, valve train and timing drive are highly reduced. Exhaust system cost are assumed to fall in relation to engine displacement. In the opposite way the fuel injection system cost rise due to direct injection with piezo injectors. The costs for the turbochargers are added at the turbocharged variants.

From variant 2 to variant 3, there is only a small step in displacement changing. Although the piezo injectors are not within variant 3 anymore, cost for fuel injection rise due to an additional port fuel injection system.

From variant 3 to variant 4, the costs drop due to the change from a 4 cylinder engine to a 3 cylinder engine. Variant 4 contains a second turbocharger for the extreme downsized 3 cylinder engine. This additional turbocharger nearly compensates the cost reduction from other groups.

Figure 3.53 shows the cost estimation for the different engine sizes of segment D. Figure 3.54 shows the cost estimation for the mechanical base engines of segment D.

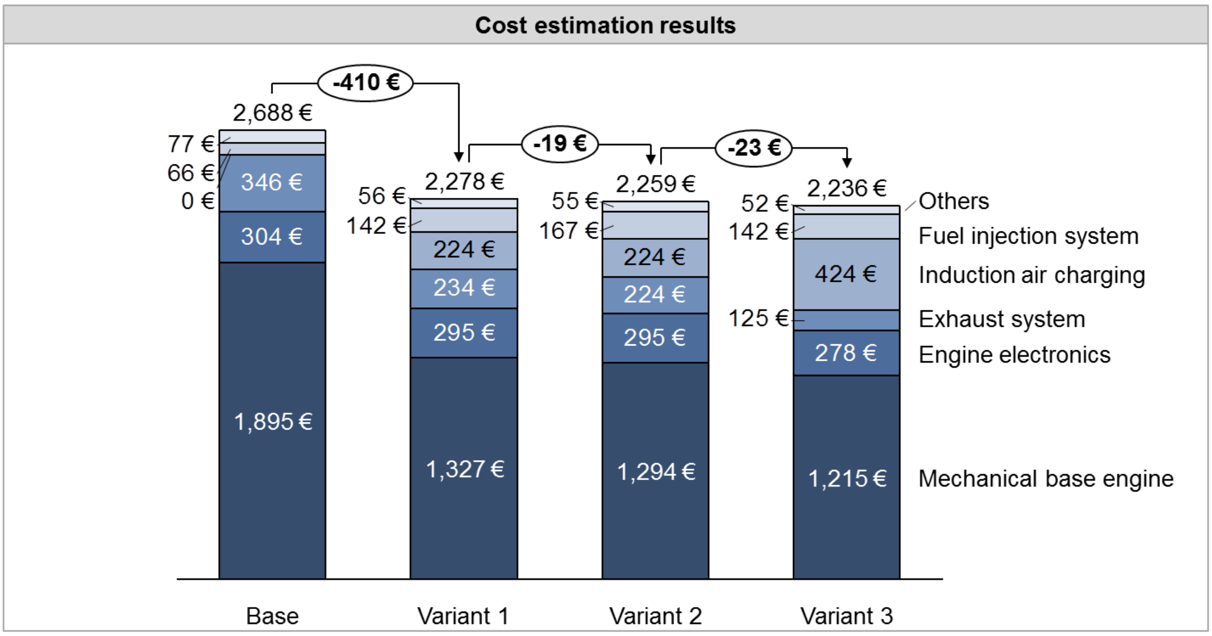


Figure 3.53 Cost estimation, engines sizes, segment D, divided by component groups

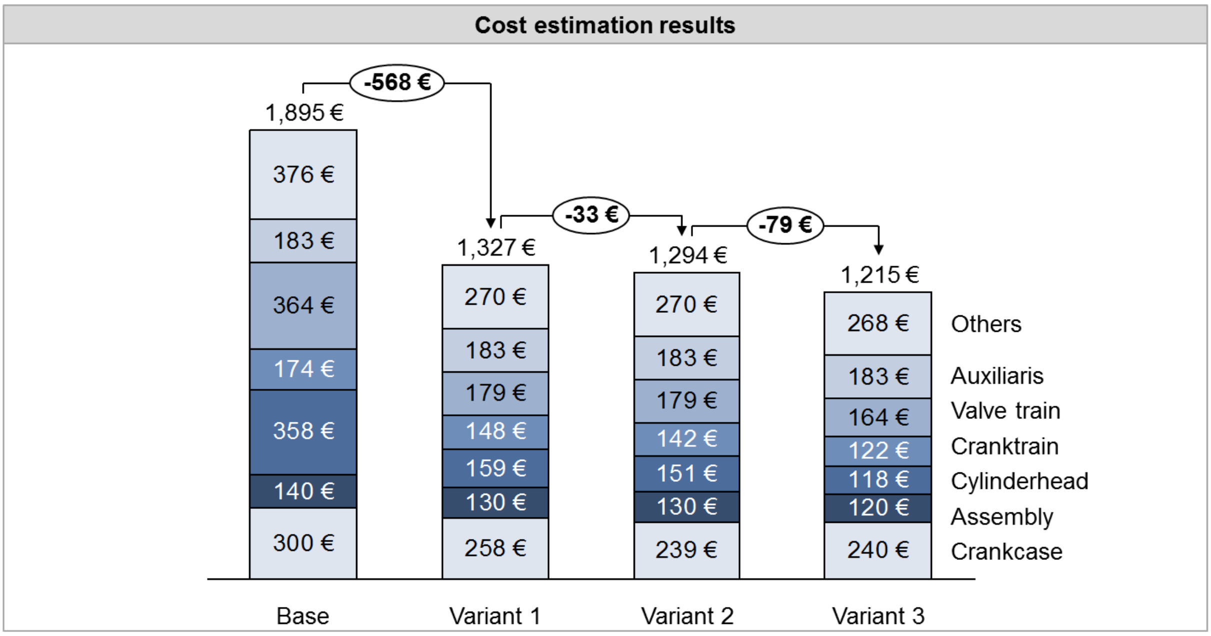


Figure 3.54 Cost estimation, engines sizes, segment D, mechanical base engine, divided by component groups

Main cost influencing technical differences

Mechanical base engine

→ The crankcase is smaller due to reduced number of cylinders (6 at base engine; 4 at variant 1 & 2; 3 at variant 3)

→ The cylinder head is smaller due to reduced number of cylinders (6 at base engine; 4 at variant 1 & 2; 3 at variant 3)

→ Crank train reduced from 6 to 4 cylinder (variant 1 & 2) or 3 cylinder (variant 3)

→ Valve train reduced from V-engine (base) to inline engine (all variants)

→ Timing drive changes from V-engine (base) to inline engine (all variants)

→ Additional mass balancing system for 3 cylinder engine (variant 3)

Induction air charging

→ Additional turbocharger for downsized engines (variant 1& 2), two-stage turbocharger at 3 cylinder engine (variant 3)

Engine electronics

→ 6 ignition coils and spark plugs for 6 cylinder engine (base) vs. 4 (variant 1 & 2) and 3 (variant 3)

→ ECU for turbocharged engines has more functions to control, more sensors (at all variants)

Fuel injection system

→ At all variants: high-pressure fuel pump, high-pressure rail and piping, DI-injectors instead of PFI-injector

→ 3 instead of 4 DI-injectors, smaller rail for 3 cylinder engine (variant 3)

→ Piezo injectors at variant 1, MHI + additional PFI injection at variants 2 and 3 (due to changings in aftertreatment system)

Figure 3.55 Cost estimation, engines sizes, segment D, main cost influencing technical differences

At segment E the base engine is also a 6 cylinder V-engine but with slightly higher power (180kW) which leads to slightly higher cost compared to base engine in segment D.

Variants 1, 2, and 3 are 4-cylinder engines whereby the high cost decrease at the mechanical base engine can be explained. Again, the turbochargers are added as well as the direct injection system. Costs for engine electronics drop marginally due to the cylinder reduction. Costs for the exhaust system decrease with the lower engine displacement. The extreme downsized 3 cylinder engine at variant 4 contains a mass balancing system and an additional second turbocharger. It is assumed that this engine also uses a more expensive flywheel with a centrifugal pendulum. The result is only a small cost increase from variant 3 to variant 4 of 31 €, although cylinder quantity is reduced. Figure 3.56 shows the cost estimation result for gasoline engines at segment E. Figure 3.57 shows the cost estimation for the corresponding mechanical base engine costs.

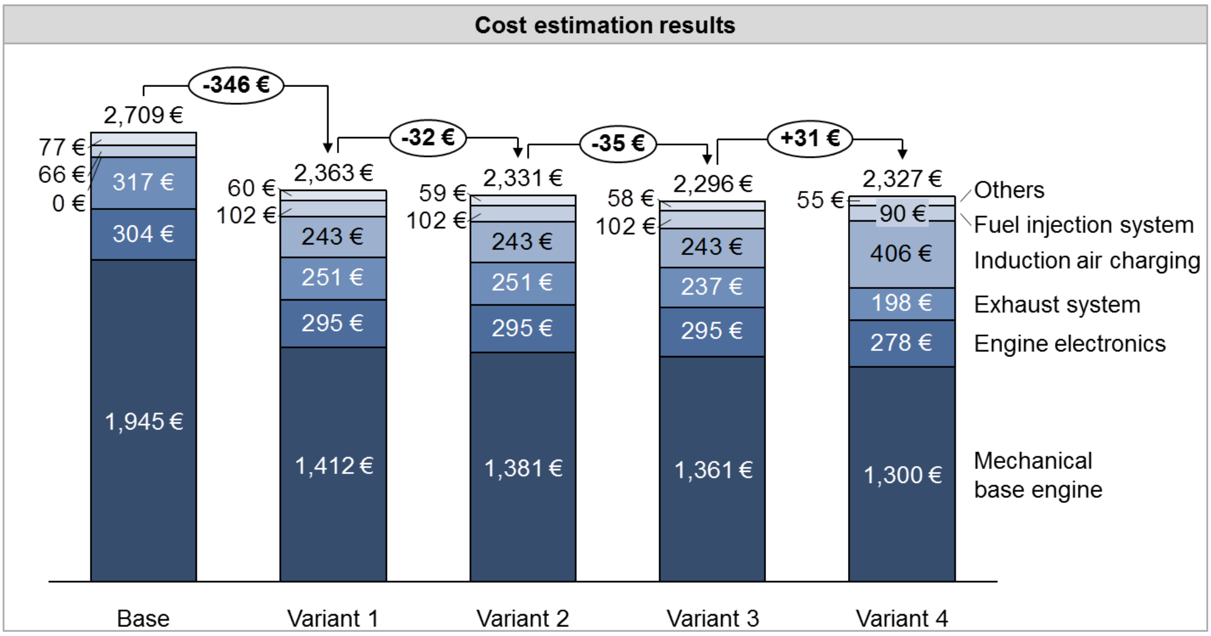


Figure 3.56 Cost estimation, engines sizes, segment E, divided by component groups

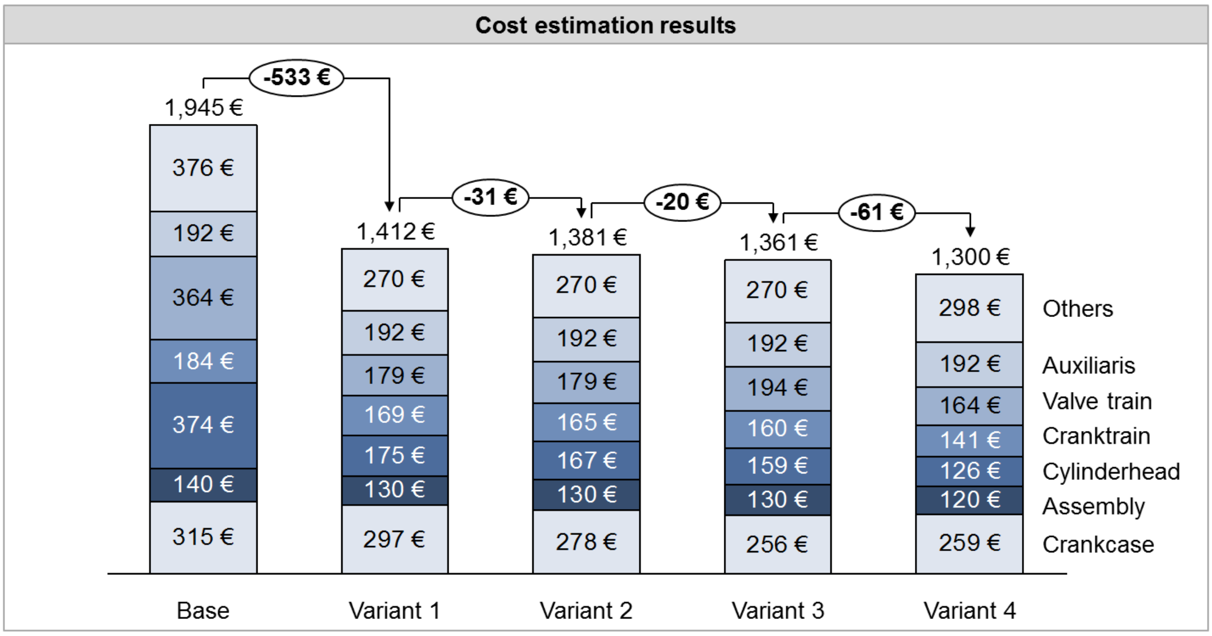


Figure 3.57 Cost estimation, engines sizes, segment E, mechanical base engine, divided by component groups

Main cost influencing technical differences
Mechanical base engine
→ The crankcase is smaller due to reduced number of cylinders (6 at base engine; 4 at variant 1, 2 & 3; 3 at variant 4)
→ The cylinder head is smaller due to reduced number of cylinders (6 at base engine; 4 at variant 1, 2 & 3; 3 at variant 4)
→ Crank train reduced from 6 to 4 cylinder (variant 1, 2 & 3) or 3 cylinder (variant 4)
→ Valve train reduced from V-engine (base) to inline engine (all variants)
→ Timing drive changes from V-engine (base) to inline engine (all variants)
→ Additional mass balancing system for 3 cylinder engine (variant 4)
Induction air charging
→ Additional Turbocharger for downsized engines (variants 1, 2 & 3), two-stage turbocharger at 3 cylinder engine (variant 4)
Engine electronics
→ 6 ignition coils and spark plugs for 6 cylinder engine (base) vs. 4 (variant 1, 2 & 3) and 3 (variant 4)
→ ECU for turbocharged engines has more functions to control, more sensors (at all variants)
Fuel injection system
→ At all variants: high-pressure fuel pump, high-pressure rail and piping, DI-injectors instead of PFI-injector
→ 3 instead of 4 DI-injectors, smaller rail for 3 cylinder engine (variant 4)
→ Piezo injectors at variant 1, MHI + additional PFI injection at variants 2, 3 and 4 (due to changings in aftertreatment system)

Figure 3.58 Cost estimation, engines sizes, segment E, main cost influencing technical differences

An overview of the estimated delta costs at gasoline engines for engine downsizing including the changes for additional turbochargers as well as downsizing-based changes at the after-treatment system is shown in Table 3.44.

Segment	Base variant	Variant 1	Delta cost	Variant 2	Delta cost	Variant 3	Delta cost	Variant 4	Delta cost
B	4 cylinder, 1.3l, NA	3 cylinder, 1.0l, DI	+ 262€	3 cylinder, 0.8l, DI	+10 €	-	-		
C	4 cylinder, 1.8l, NA	4 cylinder, 1.4l, DI	+402€	3 cylinder, 1.0l, DI	-107 €	3 cylinder, 0.8l, DI	-24 €		
D	6 cylinder, 2.4l, NA	4 cylinder, 1.8l, DI	-410 €	4 cylinder, 1.4l, DI	-19 €	3 cylinder, 1.0l, DI	-23 €		
E	6 cylinder, 2.6l, NA	4 cylinder, 2.4l, DI	-346 €	4 cylinder, 2.0l, DI	-32 €	4 cylinder, 1.6l, DI	-35 €	3 cylinder 1.2l, DI	+31 €

Table 3.44 Cost estimation, engine size, gasoline engines, overview

3.9 Cost estimation valvetrain Diesel engines:

The installation of variable valve timing systems in the Diesel engines which mount SDPF as an aftertreatment system is depicted hereinafter.

3.9.1 Description and hardware determination

The Diesel base engine uses a standard valvetrain without variable valve timing or variable valve lift. The cost of the base valvetrain system is part of the base engine cost at engine size calculation.

The valvetrain variant for Diesel engines is a variable valve timing at the exhaust side only. This valve timing technology is realized with a camphasing system which rotates the angle of the camshaft relative to the crankshaft. Main element of this technology is a cam phaser on the end of the camshaft. The cam phaser consists of an inner rotor which is directly joined to the camshaft and an housing with outer rotor which is directly joined to the timing chain. A control valve regulates the oil pressure for the oil which flutes the cam phaser and thereby changes the angle of the camshaft relative to the crankshaft. Figure 3.59 illustrated the function of the variable valve timing system.

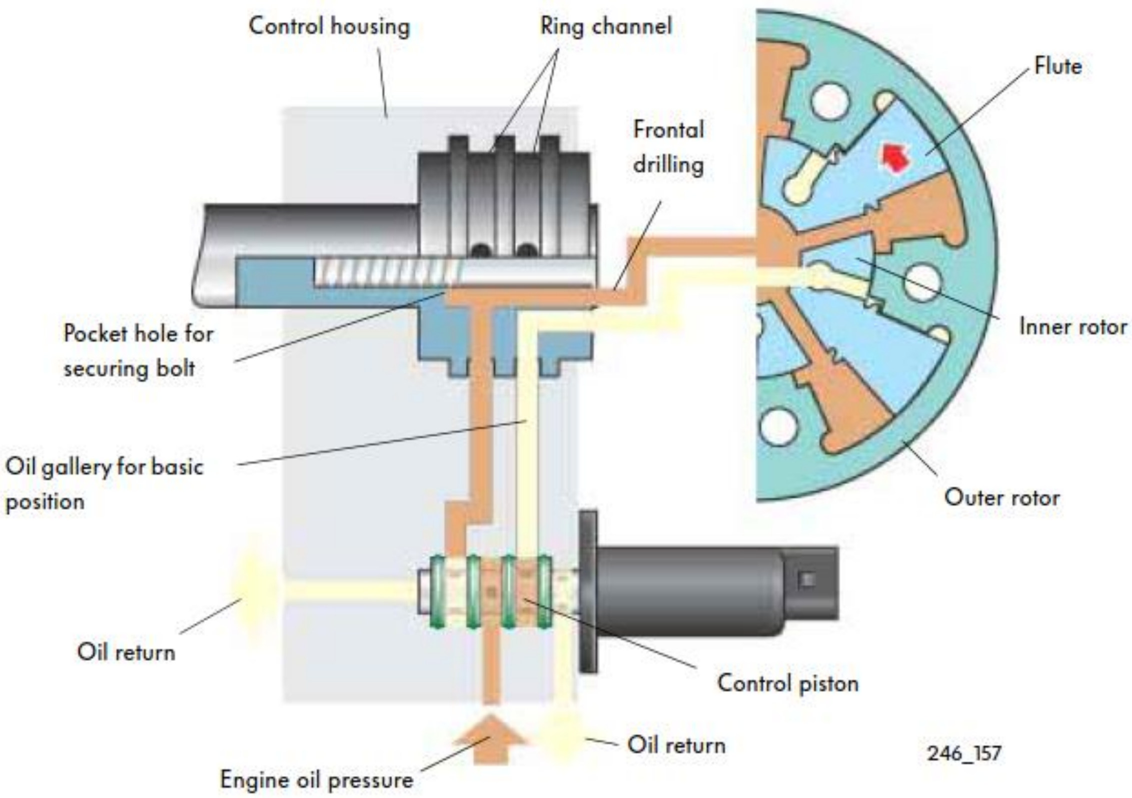


Figure 3.59 Functionality of the variable valve timing technology (Source: Volkswagen)

For all Diesel segments, the same updated valvetrain system was chosen. Therefore the technology changing is from a base valvetrain system to a variable valve timing at exhaust side for all Diesel engines. Table 3.45 shows the chosen technologies for Diesel engines.

Segment	Cylinder	Base	Variant
B	3	Not variable	Variable
C	3	Not variable	Variable
D	4	Not variable	Variable
E	4	Not variable	Variable
SUV	4	Not variable	Variable
LCV	4	Not variable	Variable

Table 3.45 Overview of selected valvetrain systems for Diesel engines (for the variable it is applied only to the exhaust).

3.9.2 Components of valvetrain systems and cost influencing parameter

The variable valve timing system was split into three groups for the cost estimation. Main part of the system is the cam phaser. The second group contains the oil control valve, the camshaft position sensor and wiring. The last group contains the design modifications that have to be done to use the variable valve timing. These are an extension of the cylinder head on the front end, a modification of the sensor trigger wheel, a flange for the position sensor, a flange for the control valve and an oil supply in the camshaft. Table 3.46 shows the component groups and the respective components.

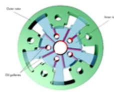
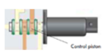
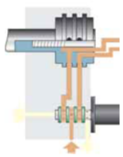
Component group		Components
Cam phaser		Cam phaser
Valve, sensor & wiring		Control valve Camshaft position sensor Wiring for new electrical components
Design modifications		Extension of cylinder head on front end for cam phaser Sensor trigger wheel at camshaft Flange for position sensor Flange for control valve Oil supply for cam phaser in camshaft

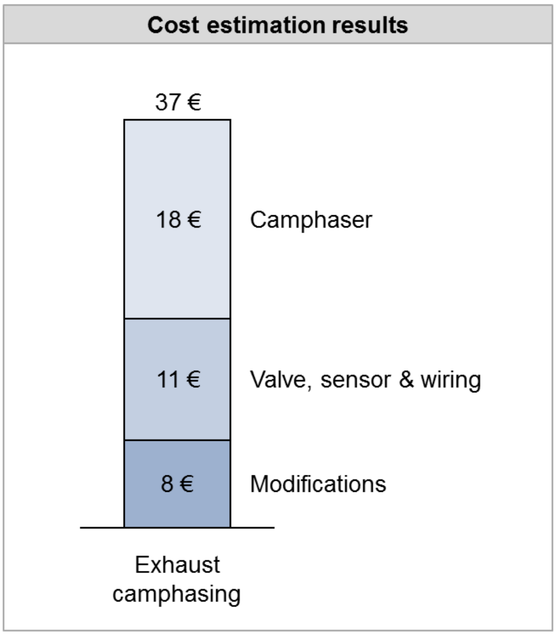
Table 3.46 Variable valve timing component groups and components

3.9.3 Vehicle segment scaling methodology

The cost for each part is not related to any specific engine characteristic. There is no direct relation between valve timing cost and number of cylinders or engine power or some other engine characteristic. Thus, the cost for “standard” variable valve timing components can be used for each Diesel vehicle segment. No scaling factors are necessary.

3.9.4 Cost analysis result Valvetrain systems Diesel segments

The total delta cost for changing a Diesel engine with a standard valvetrain to a Diesel engine with a variable valve timing at the exhaust camshaft are 37€. Main cost driver is the cam phaser with 18€. The additional control valve, the position sensor and the wiring cost 11€. The design modifications sum up to 8€. Figure 3.60 shows the delta cost estimation split by component groups for the valvetrain technology at Diesel engines.



Main cost influencing technical differences

Cam phaser:
- → Additional cam phaser at exhaust side

Other additional components:
- → Control valve
- → Camshaft position sensor
- → Wiring for new electrical components

Design modifications:
- → Extension of cylinder head on front end for cam phaser package
- → Sensor trigger wheel at camshaft
- → Flange for position sensor
- → Flange for control valve
- → Oil supply for cam phaser in camshaft

Figure 3.60 Cost estimation, variable valve train technology, Diesel engines

Table 3.47 shows an overview of the selected Diesel valvetrain variants and delta cost for adding an exhaust variable valve timing.

Segment	Cylinder	Base	Variant	Delta cost
B	3	Not variable	Variable	37 €
C	3	Not variable	Variable	37 €
D	4	Not variable	Variable	37 €
E	4	Not variable	Variable	37 €
SUV	4	Not variable	Variable	37 €
LCV	4	Not variable	Variable	37 €

Table 3.47 Cost estimation, variable valve train technology, Diesel engines, overview (for the variable it is applied only to the exhaust).

3.10 Cost estimation valvetrain gasoline engines

The installation of variable valve timing systems in the gasoline engines is depicted hereinafter.

3.10.1 Description and hardware determination

The gasoline base engines use a double variable valve timing (DVVT): At both camshafts, intake and exhaust side, a cam phaser can rotate the angle of the camshaft relative to the crankshaft. For gasoline engines two variants for the valvetrain technology have been chosen:

The first variant is an upgrade with a variable valve lift system. With this system it is possible to switch between to different cam lobes heights. Therefore the basic camshaft is replaced by a special variable valve lift (VVL) camshaft. This VVL camshaft contains a base shaft on which one cam lobe shaft per cylinder is placed. The cam lobe shafts have two different cam lobe heights. An actuator is used to move the cam lobe shaft relative to the base shaft. Therefore the valve opening height can be changed. Figure 3.61 shows how the actuator moves the cam lobe shaft.

Figure 3.61 Functionality of the variable valve lift (Source: Volkswagen)

The second variant is the usage of the Miller-cycle. In this thermodynamic cycle the intake valves are left open longer than in a normal gasoline engine. For this cycle no additional components are needed and no parts have to be adjusted. Therefore no delta cost is necessary.

Table 3.48 shows an overview of the selected gasoline valvetrain variants

Segment	Cylinder	Base	Variant 1	Variant 2
B	3	DVVT	DVVT + VVL	DVVT + VVL + Miller
C	3	DVVT	DVVT + VVL	DVVT + VVL + Miller
D	4	DVVT	DVVT + VVL	DVVT + VVL + Miller
E	4	DVVT	DVVT + VVL	DVVT + VVL + Miller

Table 3.48 Overview of selected valvetrain systems for gasoline engines

3.10.2 Components of valvetrain systems and cost influencing parameter

Main component of the variable valve lift is the VVL camshaft which is built of a base shaft, one movable cam lobe shaft per cylinder and a lock system existing of springs and balls. One actuator per cylinder is needed to move the cam lobes. Some modifications on the cylinder head cover need to be done to mount the actuators.

Component group		Components
Camshaft		VVL camshaft existing of a base shaft, one cam lobe shaft per cylinder, springs, balls
Actuation		One double pin actuator per cylinder
Modifications		Modifications at cylinder head cover to mount the actuators

Table 3.49 Variable valve lift component groups and components

3.10.3 Vehicle segment scaling methodology

The camshaft cost is related to the number of cylinder per engine. The length and the number of cam lobe shafts increases with each cylinder. The cost per actuator stay the same, the total cost of actuation is related to the number of cylinders as one actuator is needed per cylinder. The costs for the cylinder head cover modifications change in a similar way.

3.10.4 Cost analysis result valvetrain systems gasoline segments

The delta cost for changing the valvetrain system of 3 cylinder gasoline engines from a DVVT valvetrain to a variable valve lift system are 85€. The main cost driver is the replacement of the standard camshaft by a camshaft for variable valve lift. This new camshaft costs 64€ instead of the 14€ for the base camshaft. The cost for the additional electrical actuators sum up to 24€. The additional modifications that have to be done at the cylinder head cover to mount the actuators are 11€. Figure 3.62 shows the overview of the delta cost for the variable valve lift for gasoline engines with 3 cylinders.

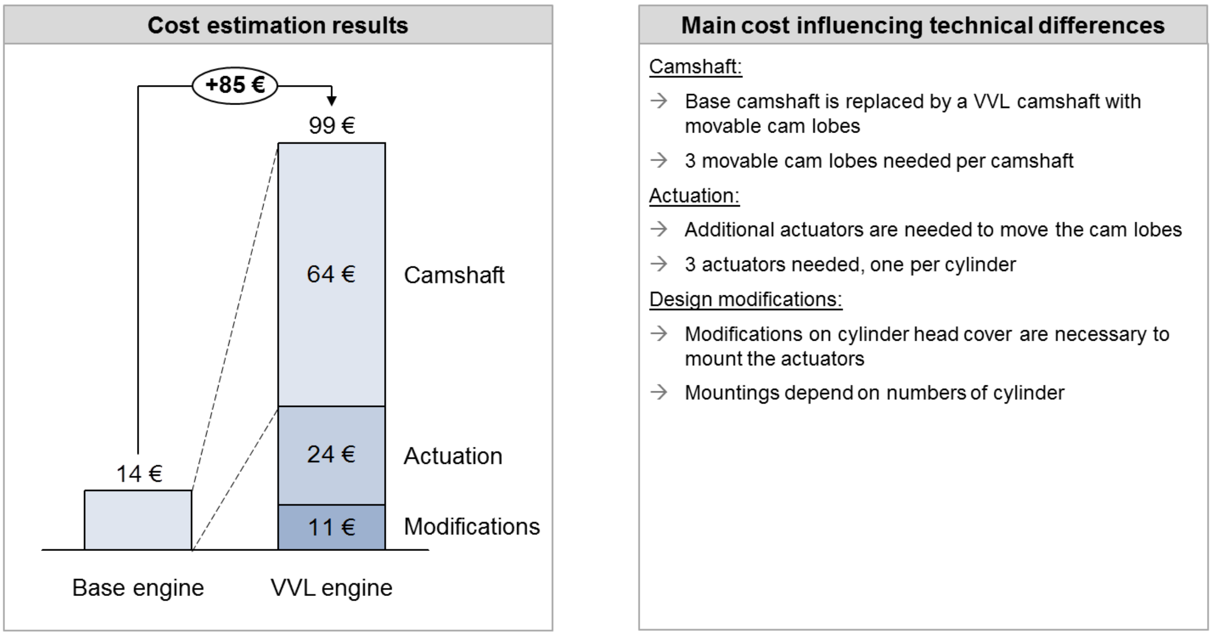


Figure 3.62 Cost estimation, variable valve lift technology, 3 cylinder gasoline engines

The delta cost for changing the valvetrain system of 4 cylinder gasoline engines from a DVVT valvetrain to a variable valve lift system are 110€. Because of the higher number of cylinder comparing to the 3 cylinder engines, longer camshafts and one additional movable cam lobe shaft are needed. Therefore the cost of the VVL camshaft sum up to 82€. The electrical actuator cost 32€. The design modifications are 13€. Figure 3.63 shows the overview of the delta cost for the variable valve lift for gasoline engines with 4 cylinders.

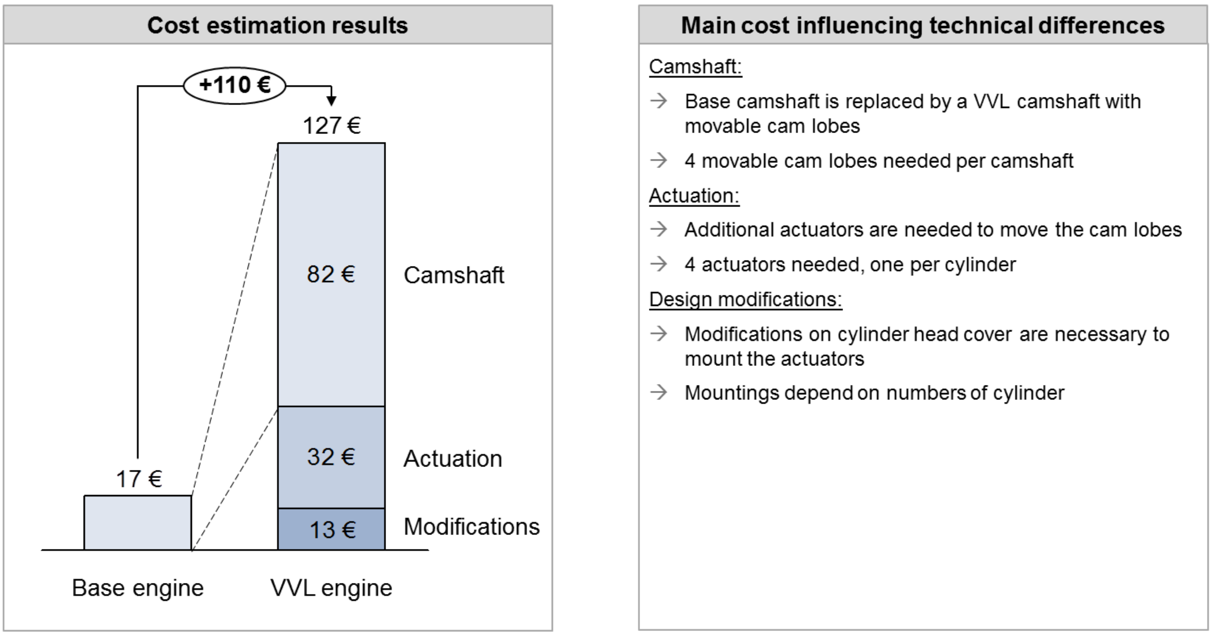


Figure 3.63 Cost estimation, variable valve lift technology, 4 cylinder gasoline engine

Table 3.50 shows an overview of the selected gasoline valvetrain variants and delta cost for adding a variable valve lift at gasoline engines.

Segment	Cylinder	Base	Variant 1	Delta cost	Variant 2	Delta cost
B	3	DVVT	DVVT + VVL	85 €	DVVT + VVL + Miller	-
C	3	DVVT	DVVT + VVL	85 €	DVVT + VVL + Miller	-
D	4	DVVT	DVVT + VVL	110 €	DVVT + VVL + Miller	-
E	4	DVVT	DVVT + VVL	110 €	DVVT + VVL + Miller	-

Table 3.50 Cost estimation, variable valve train technology, gasoline engines, overview

3.11 Cost estimation VCR technology Diesel and gasoline engines:

3.11.1 Description and hardware determination

A variable compression ratio (VCR) can be realized in many ways. In this study a variable compression is achieved by a connection rod that can switch between two different eye-to-eye-lengths due to an eccentric in the small eye. All components that are necessary to realize a VCR-con-rod incl. their functions are listed in Table 3.51.

For an interpretation of the cost estimations for this VCR system it is important to consider that the cost calculation is based on current knowledge, design, and manufacturing techniques, but under a high production scenario.

A VCR-con-rod consists of a forged body and several mechanical subcomponents that enable an adjustment of the compression rate in two stages. Therefore the following components including the corresponding functions are mandatory.

Component group	Component	Function
Body (con-rod)	Body	Besides the conventional functions a VCR-body includes extra functions that are incurred by the following additional subcomponents.
Mechanics (con-rod)	Eccentric	By turning the eccentric the distance between pin center and big eye is changed with a direct influence on the engine pistons stroke.
	Lever	The lever is directly connected to the eccentric and the push rods. It enables an adjustment of the eccentric position by the position of the VCR-pistons.
	Push rod	The push rods connect the lever to the VCR-pistons.
	VCR-piston	As a result of a changing volume flow in the pressure chambers (controlled by the valve) the VCR-pistons change their position and thus the position of push rods, lever and eccentric.
	Valve	The valve controls inlet and outlet flow of the pressure cambers and is switchable by cam disc and activator.
Actuation	Cam disc	By moving the cam disc along the cylinder length the valve can be switched in the lower dead center. The cam disc is connected to the crankcase by an inlay.
Actuation	Actuator	The switchable valve is adjusted by the cam disk, which is itself driven by the electrical actuator.

Table 3.51 VCR- components and their functions

For each variant affected by VCR-technology all conventional baseline con rods are replaced by two-stage VCR-con-rods, a cam-disc and an actuator unit (shown in Figure 3.64). All considered gasoline variants are shown in Table 3.52.

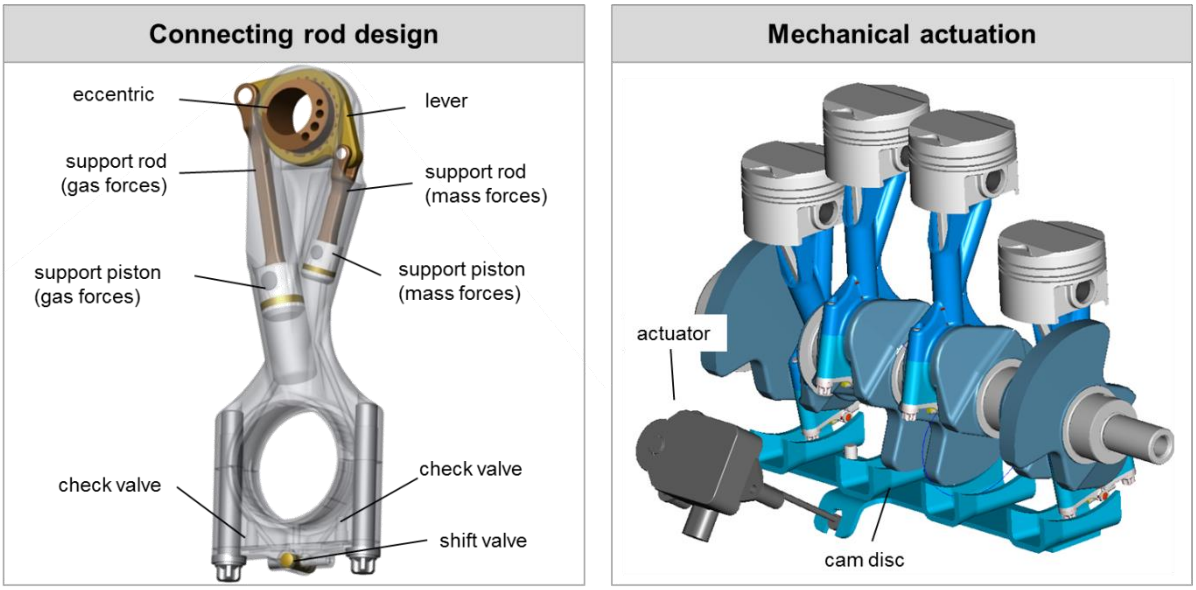


Figure 3.64 Cost estimation, VCR conrod parts

Segment	Specific torque [Nm]	Base	Variant
C	58.33	Not variable	two-step VCR
D	60.00	Not variable	two-step VCR
E	79.50	Not variable	two-step VCR

Table 3.52 Overview of selected VCR systems for gasoline engines. Specific torque is referred as per cylinder.

Table 3.53 shows the segments for Diesel engines where the VCR technology is used.

Segment	Specific torque [Nm]	Base	Variant
C	90.00	not variable	two-step VCR
D	87.50	not variable	two-step VCR
E	132.50	not variable	two-step VCR
LCV	92.50	not variable	two step VCR

Table 3.53 Overview of selected VCR systems for Diesel engines. Specific torque is referred as per cylinder.

3.11.2 Components of VCR systems and cost influencing parameter

In order to estimate total VCR technology costs for each segment the following boundaries are assumed and methodologies are used. Based on a complete bill of material for each segment the VCR-technology-costs are scaled by different parameters. An overview is shown in Table 3.54. The main difference between the con-rods for each segment is the weight. Connectingrods of higher segments have to deal with higher torque load and therefore have to be higher dimensioned. All con-rods consist of the same components.

Body: Most of the matters of expenses are assumed as variable and scaled by the net mass of the body. Only the manufacturing processes washing, cracking, assembly and testing are supposed to be non-variable.

Moving mechanics have been assumed as variable and scaled by the specific torque, except the eccentric, which is considered as non-variable. Other mechanic components e.g. valve and springs are assumed as non-variable.

In addition to the VCR-con-rod, the actuation also has been taken into account. Costs for cam-disc, inlay and bearing cover are dependent of the quantity of the cylinder. The costs for the electrical actuation engine are assumed as non-variable as well as an additional crankcase machining to connect the cam-disc to the crankcase.

Costs for assembly and overheads have been assumed as non-variable.

Table 3.54 shows an overview of the VCR technology components and the cost influencing parameter.

Part or component group	Components	Cost influencing parameter
Body (con-rod)	Raw material Machining (base body)	Raw part mass: Specific torque Length, diameter;
Mechanics (con-rod)	Lever Eccentric Push rods Pistons Valve	Specific torque Specific torque Specific torque Specific torque Non variable
Actuation	Inlay Cam-disc Actuator Miscellaneous	Quantity of cylinder Quantity of cylinder Non variable Quantity of cylinder
Assembly & overhead	Assembly Overhead	Non variable Non variable [%]

Table 3.54 VCR component groups and cost influencing parameter

3.11.3 Vehicle segment scaling methodology

Detailed costs have been scaled for all cost variable components based on available benchmark data. Table 3.55 and

Table 3.56 show the resulting scaling values. The VCR component cost for gasoline engines in segment C are set as 100 %, since the benchmark base VCR is installed in a similar vehicle.

Segment	Body	Mechanics	Actuation
C	100%	100%	100%
D	134%	135%	108%
E	144%	153%	108%

Table 3.55 Overview of the VCR technology scaling factors for gasoline engines

Segment	Body	Mechanics	Actuation
C	113%	122%	100%
D	147%	160%	108%
E	168%	202%	108%
LCV	152%	165%	108%

Table 3.56 Overview of the VCR technology scaling factors for Diesel engines

3.11.4 Cost estimation result – VCR technology Diesel engines

Figure 3.65 shows the cost estimation result for Diesel engines. The VCR technology costs are higher in segment D then in segment C although the specific torque per con-rod is lower for segment D compared to segment C. This results from the higher number of cylinders. The costs for one con-rod of segment D segment are indeed a bit cheaper than for a con-rod of segment C.

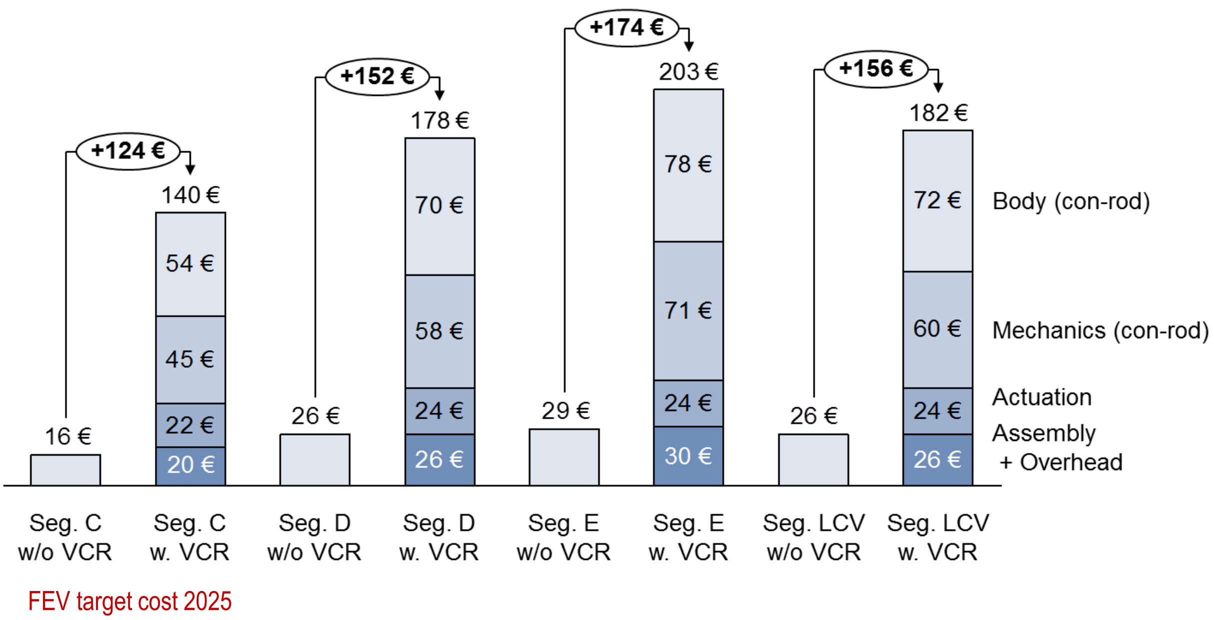


Figure 3.65 Cost estimation VCR technology, Diesel engines

To summarize, the delta cost for adding an VCR technology at Diesel engines in C segment is 124 €. For the D segment D, the cost increases 152 € for the VCR version. For the segment E, the delta costs are 174 € and for LCV segment 156 €.

3.11.5 Cost estimation result – VCR technology gasoline engines

Figure 3.66 shows the cost estimation result for gasoline engines. Upgrading segment C gasoline with VCR technology costs 110 €. Although the cost difference for a VCR-con-rod between segment C and D is almost identical (they have almost the same specific torque per con-rod), nevertheless the total difference is 32 €, due to a larger number of cylinder (4 compared to 3). Extra costs for upgrading segment D to VCR technology are 142 €. The extra costs of 150 € for segment E are the result of the higher specific torque.

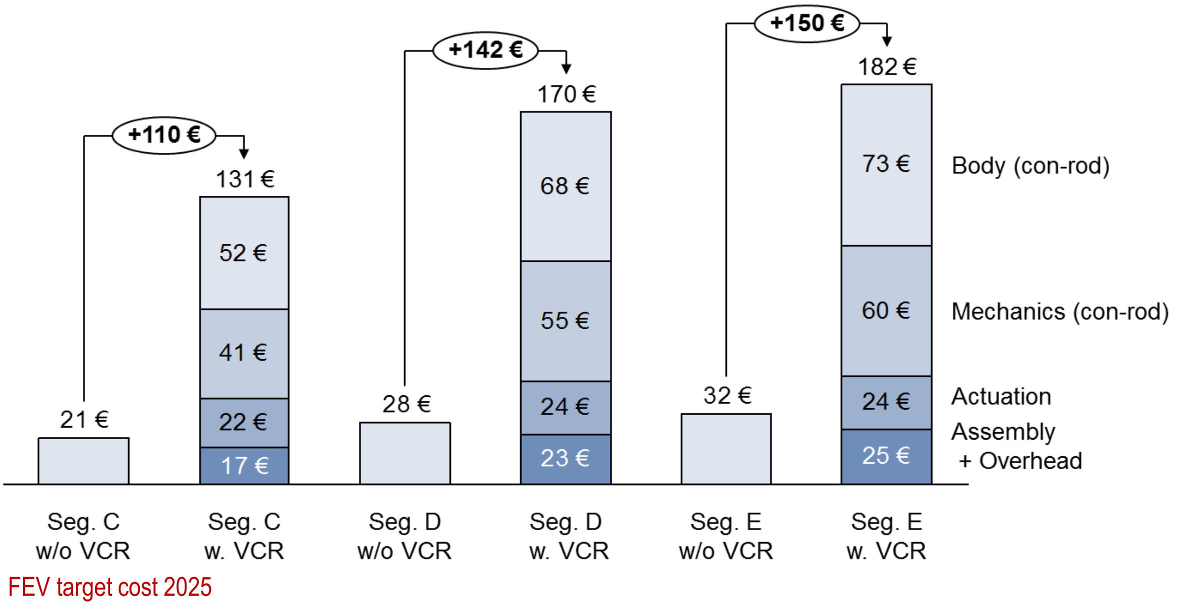


Figure 3.66 Cost estimation - VCR technology, gasoline engines

3.12 Cost estimation aftertreatment system Diesel engines

Faced with increasing emission standards the exhaust aftertreatment system plays are more important role in future considerations. The technical realization depends to a large extent on the fuel type, so the description of component groups, components and technical specifications is split into a Diesel and a gasoline part.

3.12.1 Description and hardware determination

Two different aftertreatment systems are considered within the vehicle segments. The first one consists of a Lean NO_x Trap (LNT) and a catalyzed Diesel Particulate Filter (CDPF). The second variant consists of a Diesel Oxidation Catalyst (DOC) and an SCR-coated Diesel Particulate Filter (SDPF). In both systems additionally a Slip Catalyst (SC) is mounted downstream to reduce the sulphur emission. In case of the SCR-system the slip cat should also reduce the ammonia slip. A short introduction in the both systems follows in the next section:

a) LNT/CDPF

- **LNT**
  The LNT is a discontinuous working catalyst which reduces the emissions of nitrogen oxides. The brick material is considered as cordierite. Its layout is similar to a three-way-catalyst. Only an element (e.g. barium) to store the NO_x is added. Because the reactants are used up during the reduction process, the catalyst has to be regenerated from time to time.

- **CDPF**
  The CDPF is a coated Diesel particulate filter which is designed to remove particles or soot from the exhaust gas. The brick material is considered as silicon carbide (SiC). The Coating of the CDPF contains metals of the platinum-group metals (PGM) to provide a catalytic function. These metals are palladium, platinum and rhodium. Due to their high and fluctuate price level, the loading has a significant impact on the cost structure of the CDPF. A structural layout is displayed in Figure 3.67.

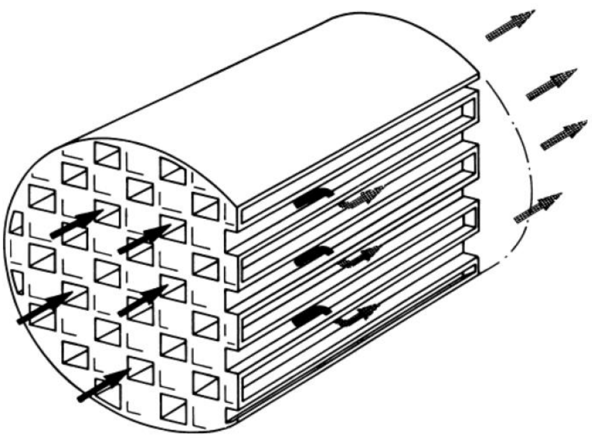


Figure 3.67 Layout DPF Source: FEV GmbH

In the second variant the LNT is replaced by a DOC and the CDPF is replaced by a SDPF:

b) DOC/SDPF

- **DOC**
  In contrast to the LNT the DOC works in a continuous way. It's designed to remove CO and HC out of the exhaust gas. Therefore a coating of PGM is necessary. The brick is also considered as cordierite.

- **SDPF**
  For the SDPF SiC is selected as brick material just as for the CDPF. The coating of the SDPF contains Cu-zeolites to provide the selective catalytic reduction of NO_x. The layout, which is shown in Figure 3.67 fits for the SDPF too.

Additionally in this aftertreatment system an SCR-system is considered to comply with the current emission standards. Therefore Urea is injected between the DOC and the SDPF to enable the conversion of NO_x. The injected urea is being transformed into NH_3 inside the SDPF where the NH_3 serves as a reducing agent. To provide a homogeneaous distribution of urea a mixer is mounted between DOC and SDPF. The SCR-system causes additional costs for the urea tank, injection unit, piping etc. In Figure 3.68 an overview of a close coupled aftertreatment system including a SDPF in provided.

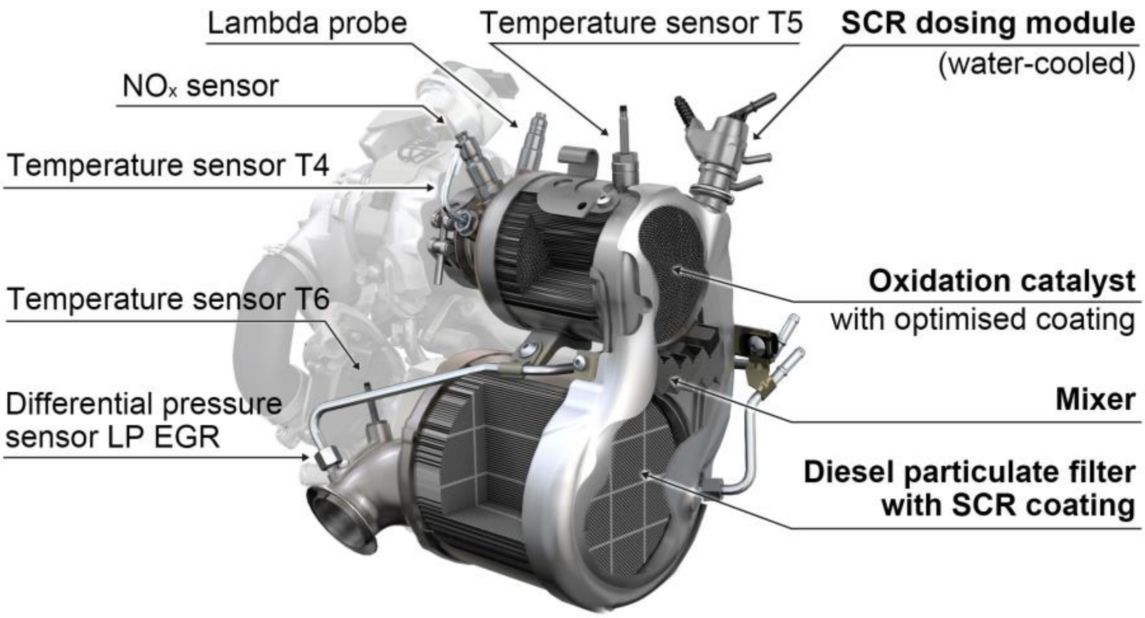


Figure 3.68 Overview close coupled aftertreatment system including SDPF (Source: VW, 22nd Aachen Colloquium Automobile and Engine Technology 2013)

A Slip-catalyst is added downstream. Its brick material is considered as cordierite.

In Table 3.57 you can see what kind of aftertreatment system is considered in which specific vehicle segment. In the segments B and C both aftertreatment systems are included whereas in the segments D, E, SUV and LCV only the system DOC+SDPF is considered as base. Therefore only the segments B and C are analyzed in the cost estimation section later on.

Segment	Displacement [l]	Base variant	Variant 1
B	1.2-1.4	1.6l LNT + 3.2l CDPF	1.31l DOC + 3.5l SDPF + 1.0l SlipCat
C	1.4-1.6	1.6l LNT + 3.2l CDPF	1.31l DOC + 3.5l SDPF + 1.0l SlipCat
D	1-6 - 2.0	1.31l DOC + 3.5l SDPF + 1.0l SlipCat	-
E	2.0 - 3.0	1.6l DOC + 5.0l SDPF + 1.0l SlipCat	-
SUV	2.0 - 3.0	1.6l DOC +5.0l SDPF + 1.0l SlipCat	-
LCV	1.8 - 2.2	1.6l DOC + 5.0l SDPF + 1.0l SlipCat	-

Table 3.57 Overview of considered aftertreatment systems in different segments, Diesel

3.12.2 Components of aftertreatment systems and cost influencing parameter

In the following tables an overall view of considered components groups, components and cost influencing parameter of the different aftertreatment systems are shown. Table 3.58 provides an overview of aftertreatment system LNT/CDPF. The components of each catalyst are brick, washcoat, coating and canning. Since the specifications of the segments B and C are the same, no scaling is necessary.

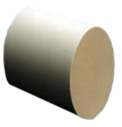
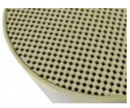
Component group		Components	Cost influencing parameter
LNT		• Brick • Washcoat • Coating • PGM loading • Canning	• Scaled by volume of catalyst • Scaled by volume of catalyst • Scaled by volume of catalyst • Calcutated by considered PGM loading • Scaled by volume of catalyst
CDPF		• Brick • Washcoat • Coating • PGM loading • Canning	• Scaled by volume of catalyst • Scaled by volume of catalyst • Scaled by volume of catalyst • Calcutated by considered PGM loading • Scaled by volume of catalyst

Table 3.58 Overview of component groups aftertreatment system 1 LNT/CDPF, Diesel variants

In Table 3.59 the component groups, components and cost influencing parameter of the aftertreatment system DOC/SDPF are displayed. Compared to the aftertreatment system LNT/CDPF a SCR-system and a SlipCat are considered as additional component groups.

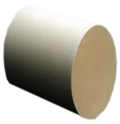
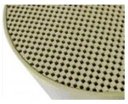
Component group		Components	Cost influencing parameter
DOC		• Brick • Washcoat • Coating • PGM loading • Canning	• Scaled by volume of catalyst • Scaled by volume of catalyst • Scaled by volume of catalyst • Calcutated by considered PGM loading • Scaled by volume of catalyst
SDPF		• Brick • Washcoat • Coating • PGM loading • Canning	• Scaled by volume of catalyst • Scaled by volume of catalyst • Scaled by volume of catalyst • Calcutated by considered PGM loading • Scaled by volume of catalyst
Slip Cat		• Brick • Washcoat • Coating • PGM loading • Canning	• Scaled by volume of catalyst • Scaled by volume of catalyst • Scaled by volume of catalyst • Calcutated by considered PGM loading • Scaled by volume of catalyst
SCR system		• Injection valve • Tank • Pump and valve unit • Sensor unit, control unit • Piping • Bolts, Brackets • Mixer	• Cost are assumed as non-variable • Scaled by volume of SDPF • Cost are assumed as non-variable • Cost are assumed as non-variable • Cost are assumed as non-variable • Cost are assumed as non-variable • Cost are assumed as non-variable

Table 3.59 Overview of component groups aftertreatment system 2 DOC/SDPF, Diesel segments

3.12.3 Vehicle segment scaling methodology

As mentioned above only the segments B and C are analysed in the cost estimation. Since the specifications of these segments are the same no scaling methodology is provided in this section. In the Table 3.60 and the Table 3.61 the volumes of the considered catalysts are displayed.

Part	Scaling	Volume of catalyst/filter [l]	
		B	C
LNT	Catalyst volume	1.6	1.6
CDPF	Filter volume	3.2	3.2

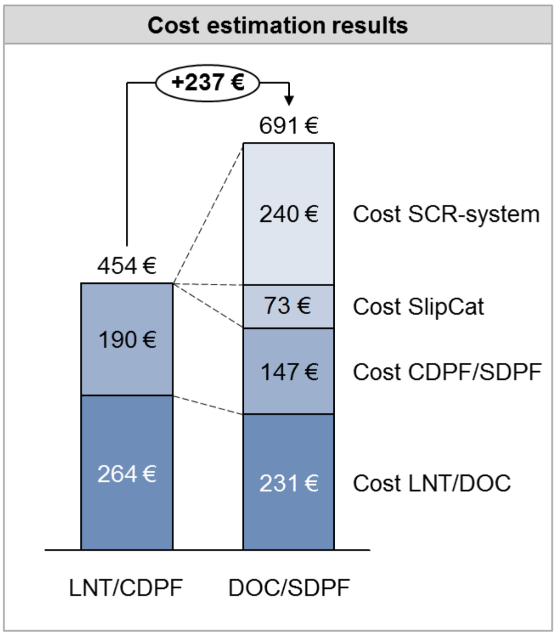
Table 3.60 Overview of volumes catalysts and filters aftertreatment system LNT/CDPF

Part	Scaling	Volume of catalyst/filter [l]	
		B	C
DOC	Catalyst volume	1.31	1.31
SDPF	Filter volume	3.5	3.5
SlipCat	Catalyst volume	1	1

Table 3.61 Overview of volumes catalysts and filters aftertreatment system DOC/SDPF

3.12.4 Cost estimation results aftertreatment systems Diesel segments

In the following figures the cost estimation results of the different segments are provided.



Main cost influencing technical differences

LNT/DOC:

→ Cost difference between LNT and DOC due to varying PM Ratio and higher coating cost of LNT

CDPF/SDPF:

→ Majority of cost difference due to different PGM Loading

SlipCat:

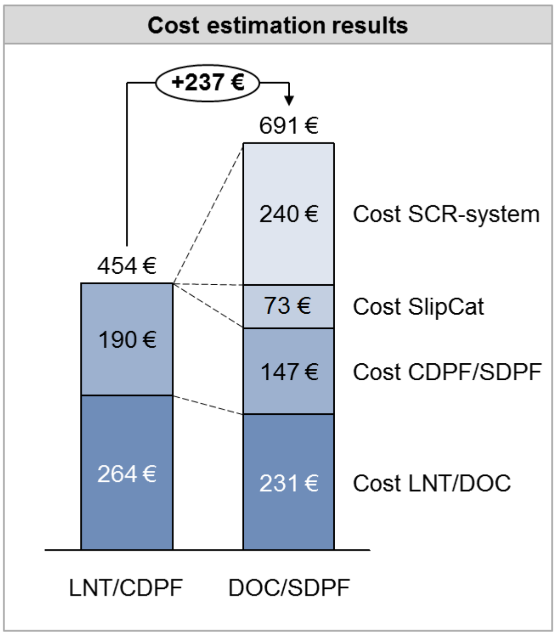
→ A slip catalyst is considered in the aftertreatment system DOC/SDPF

SCR-system:

→ In the aftertreatment system DOC/SDPF an additional SCR-system is considered

Figure 3.69 Cost estimation, aftertreatment technology, Diesel segment B

The difference between the LNT/CDPF and the DOC/SDPF system in segment B is 237 €. On the one hand the DOC is cheaper as the LNT due to varying PGM ratio. On the other hand a SCR-system and a SlipCat are added.



Main cost influencing technical differences

LNT/DOC:

→ Cost difference between LNT and DOC due to varying PM Ratio and higher coating cost of LNT

CDPF/SDPF:

→ Majority of cost difference due to different PGM Loading

SlipCat:

→ A slip catalyst is considered in the aftertreatment system DOC/SDPF

SCR-system:

→ In the aftertreatment system DOC/SDPF an additional SCR-system is considered

Figure 3.70 Cost estimation, aftertreatment technology, Diesel segment C

Because in segment C the same hardware is considered as in segment B, the costs remain the same.

Cost summary aftertreatment system Diesel segments

This section closes with a cost summary of both aftertreatment systems within the different segments. The primary reason for the difference lies in the cost of the PGM loading.

Segment	Displacement [l]	Base variant	Variant 1	Δ cost
B	1.2	1.6l LNT + 3.2l CDPF	1.31l DOC + 3.5l SDPF + 1l SlipCat	+ 237 €
C	1.4	1.6l LNT + 3.2l CDPF	1.31l DOC + 3.5l SDPF + 1l SlipCat	+ 237 €

Figure 3.71 Cost summary, aftertreatment, Diesel segments

3.13 Cost estimation aftertreatment system gasoline engines

3.13.1 Description and hardware determination

Two different aftertreatment systems are considered within the gasoline variants. The first one is a Three-Way-Catalyst (3WC) with an optional modified injection system. The second variant is a Four-Way-Catalyst (4WC), which is only relevant for segment E.

3WC + injection system (optional)
The 3WC is the common solution in the gasoline aftertreatment system. Technically it's an oxidation catalyst which operates at the stoichiometric point of the combustion. Different kind of injection systems are considered in different variants and segments. The brick could be compared with the cordierite bricks of the LNT, but with a higher quantity of cells per square-inch.

4WC
In segment E of the gasoline variants a 4WC is needed to comply with the emission standards. Technically it's a SiC brick as used as a DPF in the Diesel aftertreatment systems. It's coated like a 3WC.

In Table 3.62 you can see what kind of aftertreatment system is considered in which specific segment of the gasoline variants. In the segments B, C and D the 3WC aftertreatment system is included with different optional injection variants. In segment E the 4WC aftertreatment system is considered as variant 2.

Segment	Displacement [l]	Base variant	Variant 1	Variant 2
B	0.8-1.0	1.0l 3WC	0.8l 3WC + DI 350 bar	-
C	1.0-1.4	1.4l 3WC	1.0l 3WC + DI 350 bar	-
D	1.6-2.6	2.6l 3WC	2.0l 3WC + Piezo	1.6l 3WC + DI + PFI
E	1.6-3.0	3.0l 3WC	2.0l 3WC + Piezo	1.6l 4WC

Table 3.62 Overview of considered aftertreatment systems in different segments of gasoline variants

3.13.2 Components of aftertreatment systems and cost influencing parameter

In Table 3.63 the different component groups are displayed. The Injection System is optional, not in every variant a modified injection system is considered.

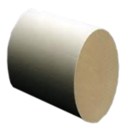
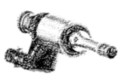
Component group		Components	Cost influencing parameter
3WC		• Brick • Washcoat • Coating • PGM loading • Canning	• Scaled by volume of catalyst • Scaled by volume of catalyst • Scaled by volume of catalyst • Calcutated by considered PGM loading • Scaled by volume of catalyst
Injection System		Depending on kind of injection, considered kinds are: • Direct Injection • Piezo Injection • Port Fuel Injection	• Cost are assumed as non-variable • Cost are assumed as non-variable • Cost are assumed as non-variable

Table 3.63 Overview of component groups aftertreatment system 3WC gasoline segments

As mentioned above the 4WC is technically a SiC brick which is coated as a 3WC. Its components are shown in Table 3.64.

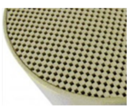
Component group		Components	Cost influencing parameter
4WC		• Brick • Washcoat • Coating • PGM loading • Canning	• Scaled by volume of catalyst • Scaled by volume of catalyst • Scaled by volume of catalyst • Calcutated by considered PGM loading • Scaled by volume of catalyst

Table 3.64 Overview of component groups aftertreatment system 4WC gasoline segments

3.13.3 Scaling methodology

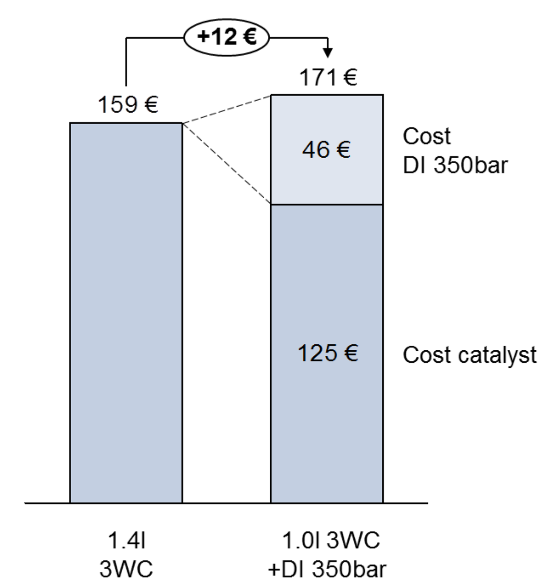
In the different vehicle segments different catalyst volumes are considered. The costs were estimated on the base of a scaling methodology. As base a 1.0l 3WC were selected. In Table 3.65 an overview about the different scaling factors is displayed.

Segment	Part	Scaling	Volume of catalyst/filter [l]	Scaling factor
B	**3WC**	Catalyst volume	1.0	100%
B	**3WC**	Catalyst volume	0.8	80%
C	**3WC**	Catalyst volume	1.4	127%
C	**3WC**	Catalyst volume	1.0	100%
D	**3WC**	Catalyst volume	2.6	210%
D	**3WC**	Catalyst volume	2.0	169%
D	**3WC**	Catalyst volume	1.6	142%
E	**3WC**	Catalyst volume	3.0	238%
E	**3WC**	Catalyst volume	2.0	169%
E	**4WC**	Catalyst volume	1.6	-

Table 3.65 Scaling factors 3WC, gasoline vehicle segments

3.13.4 Cost estimation results aftertreatment systems gasoline segments

Cost estimation results



Main cost influencing technical differences

3WC:

- → In terms of better comparability the same catalyst volume is considered

Direct Injection 350 bar:

- → Higher cost due to modified injection system. Following parts are modified:
  - → Pump
  - → Rail
  - → Pipes
  - → Injectors

Figure 3.72 Cost estimation, aftertreatment technology, gasoline segment B

The technology leap in segment B is from a 1.4l engine with 4 cylinders to a 1.0l engine with 3 cylinders. The aftertreatment system changes from a 1.4l 3WC to a 1.0l 3WC with Direct Injection 350bar. Because the volume drops from 1.4l to 1.0l the costs of the TWC also is lower due to the lower volume. In addition the cost of the Direct Injection system is included so the total cost increase by 12 €. By the installation of a DI of high pressure (350 bar), and apart from the higher fuel economy, the PN is reduced; and therefore the requirements from the 3WC are lower (the volume is reduced).

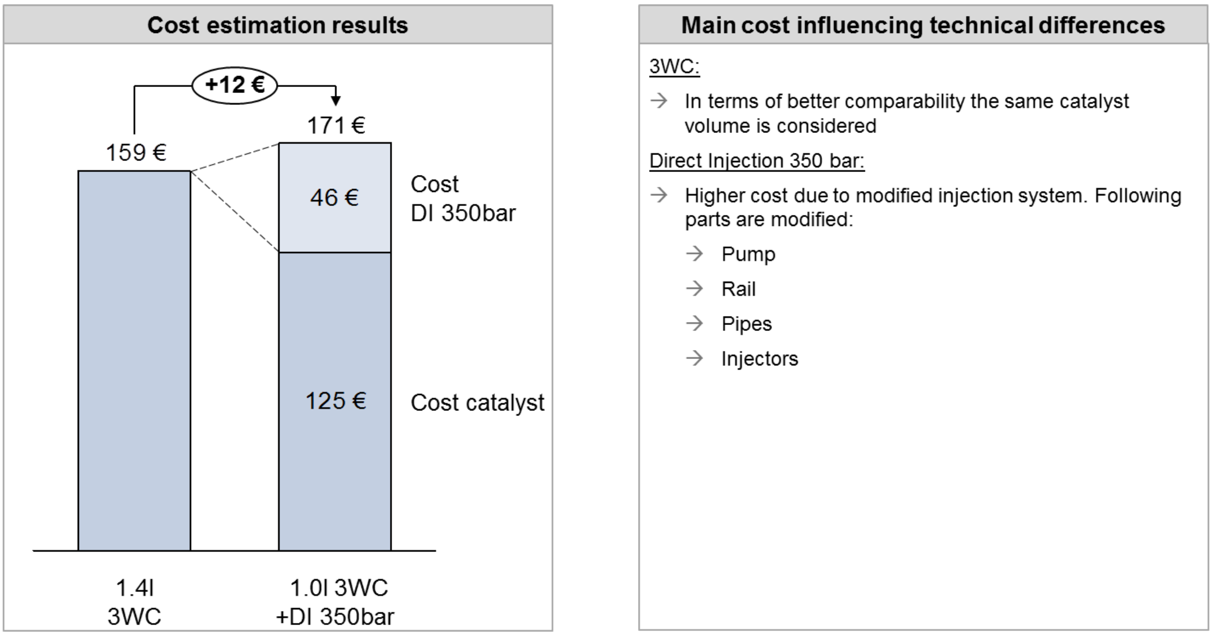


Figure 3.73 Cost estimation, aftertreatment technology, gasoline segment C

The cost estimation for segment C (Figure 3.73) is the same one as in segment B.

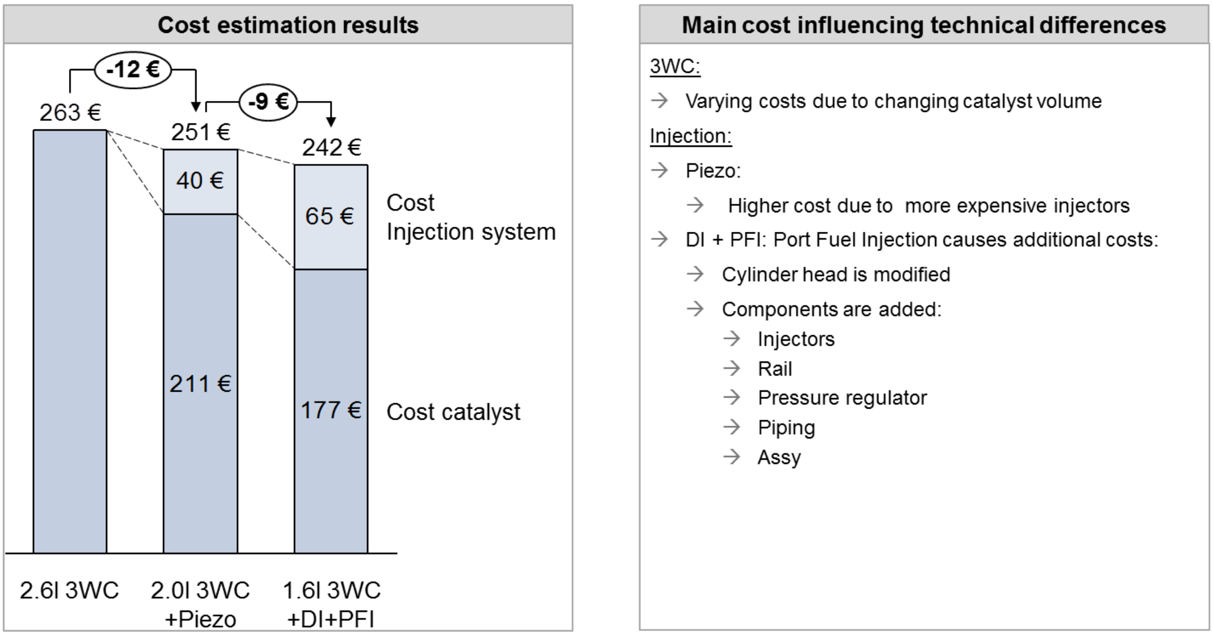


Figure 3.74 Cost estimation, aftertreatment technology, gasoline segment D

In segment D the base is a 2.6l 3WC. In the first technology step the engine changes from a naturally aspirated engine to a turbocharged engine. In the aftertreatment section only the additional Piezo injectors are considered. The rest of the changes are summed up in the engine section. In variant 2 a Port Fuel Injection system is considered in addition. The volume of the 3WC varies between all different variants because of the varying displacement volume of the engines.

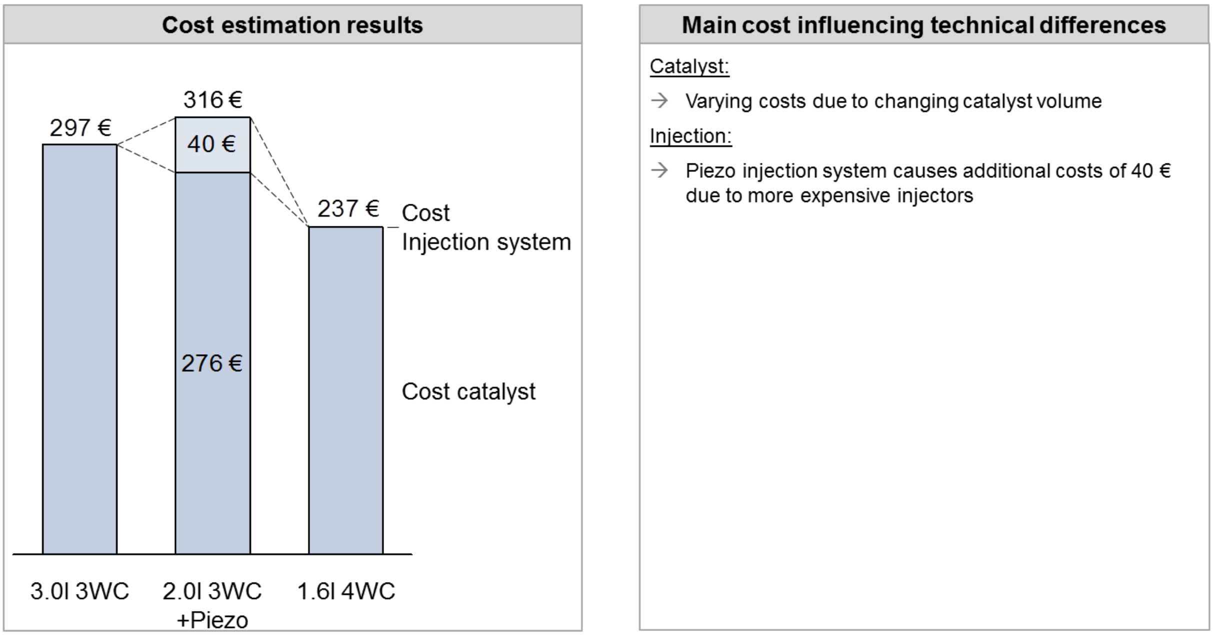


Figure 3.75 Cost estimation, aftertreatment technology, gasoline segment E

In segment E (Figure 3.75) the base is a 3.0l 3WC. As explained for segment D in the first technology step the engine changes from a naturally aspirated engine to a turbocharged engine. In the injection system Piezo injectors are considered. This leads to higher costs in the amount of 40€. In the next technology-step a 1.6l 4WC is implemented. Compared to the base no changes in the injection system are taken into account in this variant.

In Table 3.66 an overview of the cost estimation of all gasoline vehicle segments is provided.

Segment	Displacement [l]	Base variant	Variant 1	Δ cost	Variant 2	Δ cost
B	0.8-1.0	1.0l 3WC	0.8l 3WC + DI 350 bar	29 €	-	
C	1.0-1.4	1.4l 3WC	1.0l 3WC + DI 350 bar	12 €	-	
D	1.6-2.6	2.6l 3WC	2.0l 3WC + Piezo	-12 €	1.6l 3WC + DI + PFI	-9 €
E	1.6-3.0	3.0l 3WC	2.0l 3WC + Piezo	19 €	1.6l 4WC	-79 €

Table 3.66 Cost summary, aftertreatment technology 3WC, gasoline

3.14 Cost estimation engine friction

3.14.1 Description and hardware determination

In this chapter different technical solutions are summed up under the generic term engine friction. This includes an engine friction reduction by 20%, a split cooling design and a substitution of the water-pump by an electrical water-pump. In the gasoline vehicle segments a variable oil pump is additionally included. All considered solutions aim to achieve emission reduction goals.

To achieve a friction reduction of 20% it is assumed to use roller bearings at the camshafts. The crankshaft bearing diameter will be reduced. For the gasoline engines the oil pump will be modified to a controlled oil pump.

Split cooling describes an approach of the engine-cooling system where the head and the block of the engine are cooled by independent circuits so it's possible to regulate the specific temperature of each area independently. In general it's preferable if the block runs warmer while the head runs cooler to reduce frictional losses. Therefore an additional thermostat incl. actuation and manufacturing is needed.

In contrast to a conventional water pump the electrical water pump provides additional functionalities. Its capacity isn't depending on the engine speed since its drive is decoupled from the engine output. So it's possible to adjust the power of the water pump to specific requirements of the engine. The possibility of steady conditions is another advantage of an electrical water pump. A disadvantage of the electrical design is the cost-effectiveness since electrical pumps are more expensive than conventional ones.

Similar to the mentioned water pumps conventional unregulated oil pumps have the disadvantage that their output is coupled to the engine output. So it isn't ensured that the pump provides the corresponding output in all operating points of the engine.

Table 3.67 shows the engine friction variants for the Diesel engines, Table 3.68 shows the engine friction variants for gasoline engines.

Segment	Base	Variant
B	baseline	-20% friction, split cooling, el. coolant pump
C	baseline	-20% friction, split cooling, el. coolant pump
D	baseline	-20% friction, split cooling, el. coolant pump
E	baseline	-20% friction, split cooling, el. coolant pump
SUV	baseline	-20% friction, split cooling, el. coolant pump
LCV	baseline	-20% friction, split cooling, el. coolant pump

Table 3.67 Overview of engine friction technology variants for Diesel engines

Segment	Base	Variant
B	baseline	-20% friction, var. oil pump, split cooling, el. coolant pump
C	baseline	-20% friction, var. oil pump, split cooling, el. coolant pump
D	baseline	-20% friction, var. oil pump, split cooling, el. coolant pump
E	baseline	-20% friction, var. oil pump, split cooling, el. coolant pump

Table 3.68 Overview of engine friction technology variants for gasoline engines

3.14.2 Components of engine friction systems and cost influencing parameter

For the cost estimation of the engine friction technology the parts are split into three groups. The first group contains the components and modifications for the friction modifications. The roller bearing costs are depending on the number of bearings inside the engine and thereby resulting on the number of cylinders. The main bearing diameter reduction costs are depending on a concrete engine design. Without knowing a specific design the costs are assumed as cost neutral. Costs for oil pump modification are assumed as non-variable.

For the split cooling the costs for the additional thermostat incl. actuation and manufacturing are assumed to be constant at all segments.

The delta costs of replacing the mechanical water-pump by an electrical water-pump are also assumed to be the same for all engine segments. Table 3.69 shows an overview of the components for engine friction technology.

Part or component group	Components	Cost influencing parameter
Friction modification	Roller bearings at camshafts Main bearings diameter reduction Modification oil pump (only gasoline engines)	Scaled by no. of bearings (cylinder) Depending on concrete design, assumed as cost neutral Cost assumed as non variable
Split cooling	Additional manufacturing for split cooling Additional thermostat incl. actuation	Cost assumed as non variable Cost assumed as non variable
Substitution water-pump	Electrical water-pump substitutes mechanical water-pump	Cost assumed as non variable

Table 3.69 Engine friction technology, component groups and components

3.14.3 Cost estimation results

At all engines the main cost driver is the substitution of the mechanical water-pump by an electrical water-pump with delta costs of 50€. Additional delta cost of 15€ originate from the additional split cooling system. The friction modifications rise slightly from three to four cylinder engines. Due to the additional oil pump modification at gasoline engines the technology costs are a little bit higher than for Diesel engines. Figure 3.76 shows the delta cost results for Diesel engine. Figure 3.77 shows the delta cost results for gasoline engines.

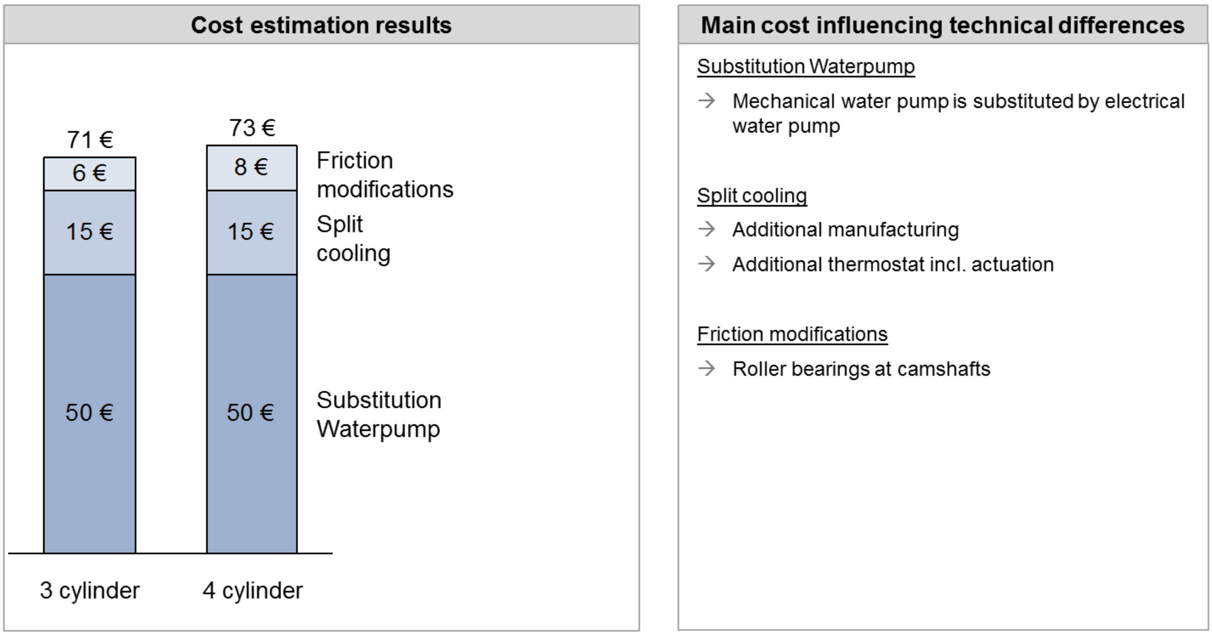


Figure 3.76 Cost estimation, engine friction technology, Diesel engines

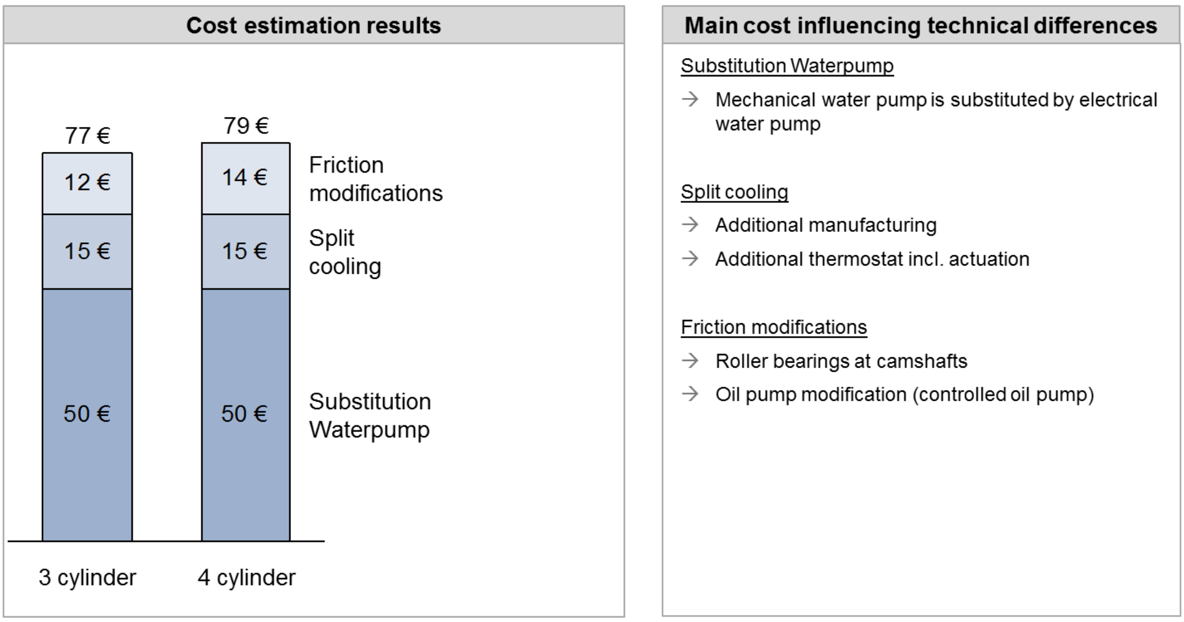


Figure 3.77 Cost estimation, engine friction technology, gasoline engines.

4 Simulation

4.1 Overview

The focus of the simulation study is the CO_2 emissions calculation for a complete selection of passenger car segments which represents a full picture of the fleet distributions for diesel and gasoline passenger cars. These include the segments B, C, D, E, SUV and LCV for diesel vehicles and B, C, D and E for gasoline vehicles. For those segments, different technologies already described in the Chapter 1 are analysed. These include:

- Start and Stop;
- Downsizing (up to 3 steps in gasoline, one for diesel, due to the design and emissions limitations);
- Different transmissions, including MT, DCT and AT;
- Mass and driving resistance reductions, considering two future scenarios;
- Engine technologies, different EGR layouts, variable valve timing, electrical water pump, split cooling, variable oil pumping, miller combustion or different turbocharging, among others;
- Various types of hybridization, including P0 and P2 configurations, mild and full hybrid.

These technologies are applied to every segment to configure future vehicles.

4.2 Boundary conditions

The model setup and calibration are based on available reference vehicle data e.g. from benchmark projects and investigations. These vehicles are representative for their segment and show an average fuel consumption level. The vehicle manufacturer and type will not be described in order to ensure confidentiality.

The cycles to consider are relevant for the final calibration. For this project, the EU market is the focus of the study, and the NEDC procedure as representative of EU6b is considered, while the WLTP as representative procedure of EU6c. There are important remarks when considering these procedures regarding test masses, vehicle configuration, flexibilities, etc. and these have been adapted for the project needs.

For gasoline vehicles the emissions are relative uncritical and can mostly be achieved by the use of the three way catalyst or the four way catalyst (by adding particulate filtering capabilities). In the case of Diesel, NO_x, HC, CO and soot emissions should be assessed because the engine configuration and calibration are strongly affected by the emission limits. Therefore, the exhaust temperature and conditions, as well as the aftertreatment systems are considered in the simulation only for the Diesel case. In the case of gasoline, the simulation of emissions and exhaust conditions is not covered; however, the engine maps have been calibrated with appropriate exhaust conditions and technologies to achieve the future emission standards were considered.

NEDC

For the case of the NEDC, the procedure establishes to test the vehicle with the corresponding inertia weight class (IWC). This means, that some changes that carry over some mass variations cannot be observed in the simulation if the IWC does not change. This might happen for example when downsizing the engines, or when including hybrid components, which will modify the vehicle mass. Therefore, the mass used for the NEDC cycle simulation is the reference mass of every vehicle, and not the inertia weight class.

WLTP

The WLTP (marked here as Procedure) considers the testing of two different vehicles, the “Vehicle H and the “Vehicle L”:

- The “Vehicle H” is the vehicle within the vehicle family with the combination of road load relevant characteristics (i.e. test mass high –TM_H-, aerodynamic drag and tyre rolling resistance) producing the highest cycle energy demand. Furthermore, this vehicle is relevant for pollutant emissions legislation.
- The “Vehicle L” is the vehicle within the vehicle family with the combination of road load relevant characteristics (i.e. test mass low (TM_L), aerodynamic drag and tyre rolling resistance) producing the lowest cycle energy demand. The CO_2 emissions in the cycle for any vehicle in between “Vehicle L” and “Vehicle H” can be calculated by interpolating the values of the CO_2 emissions in dependency of the mass of the target vehicle.

Coast down

Due to the different vehicle configurations and flexibilities in both procedures, the coast down coefficients for the NEDC and WLTP will be different. Normally, the NEDC procedure allows several flexibilities, such as increased tyre pressures, reduced auxiliaries during testing, range of ambient temperature, etc., which produces lower requirements in terms of vehicle drag. These flexibilities are limited for the WLTP and this affects the final coast down value. As a consequence, the drag force over WLTP is higher than ones over NEDC. The values provided by ICCT are based on the average running resistances of the top 10 vehicles of every segment:

	Cd*Af	RRC total	RRC Tyres	RRC Drivetrain
	/ m²	/ kg/t	/ kg/t	/ kg/t
Segment				
B	**0.748**	**11.895**	10.168	1.727
C	**0.772**	**11.408**	9.681	1.727
D	**0.721**	**11.570**	9.843	1.727
E	**0.766**	**11.814**	9.782	2.032
J	**0.979**	**12.434**	9.589	2.844
N1 III	**1.531**	**11.235**	9.508	1.727

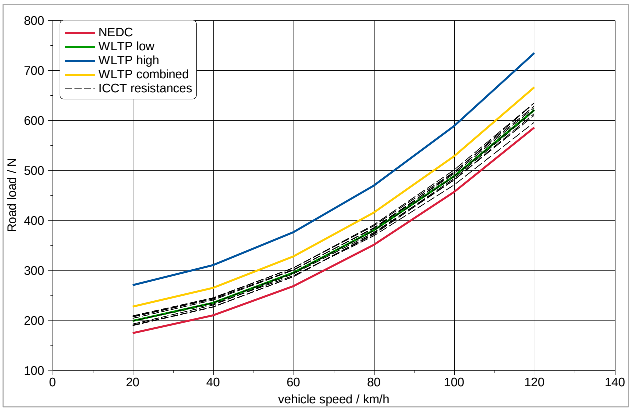
Table 4.1 Average Coast down coefficients of the top 10 most sold vehicles for every segment.

For every vehicle family, the mass and drag characteristics of the average vehicle are defined by assuming that 40% of vehicles have TM_H drag and mass characteristics and 60% of vehicles have TM_L drag and mass characteristics.

The Table 4.2 shows now the average variation of the coast down coefficients depending on the final cycle to consider, as modifying the average vehicle drag coefficients investigated by ICCT, based on FEV benchmark information. The Figure shows how the aerodynamic coefficient of the “Vehicle H” is 10% higher than the one of the “Vehicle L”, and the one of NEDC is 2.5 lower than the one of the latter. At the same time, the rolling resistance is 20 % higher for the “Vehicle H” with respect to the “Vehicle L”, and 10 % lower for the NEDC with respect to the “Vehicle L”. Assuming that the values are applied to the combined mass, the final coast down coefficients for the simulation are shown in the Table 4.2.

	NEDC		TML		combined (base)		TMH	
	RRC	Cd*Af	RRC	Cd*Af	RRC	Cd*Af	RRC	Cd*Af
	/ kg/t	/ m²	/ kg/t	/ m²	/ kg/t	/ m²	/ kg/t	/ m²
B	9.9128	0.7016	11.0143	0.7196	11.8954	0.7484	13.2171	0.7915
C	9.5065	0.7241	10.5628	0.7427	11.4078	0.7724	12.6753	0.8170
D	9.6419	0.6761	10.7133	0.6934	11.5703	0.7211	12.8559	0.7627
E	9.8451	0.7186	10.9390	0.7370	11.8141	0.7665	13.1268	0.8107
J	10.3615	0.9174	11.5128	0.9410	12.4338	0.9786	13.8153	1.0351
N1 III	9.3626	1.4350	10.4029	1.4718	11.2351	1.5307	12.4834	1.6190

Table 4.2 Final coast down coefficients considered in the simulation.



Shown data:

- Evaluation of NEDC, WLTP low and WLTP high coast down data shows
  - Cd*A
    - Low → NEDC = -2.5 %
  - Cd*A
    - Low → high = +10 %
  - RR
    - Low → NEDC = -10 %
  - RR
    - Low → high = +20 %

Figure 4.1 Coast down coefficients variation for the different cycles. The figure shows an example with the Segment C vehicle and the different drag by comparing the original values delivered by the ICCT, and the new ones applied to the different cases.

US cycles

The US cycles FTP75 and HWFET have also been simulated by keeping the EU calibration. However, the results should only be considered as indicative of what the same vehicle tested on different test cycle achieves. Therefore, the configuration of the vehicles has not been optimized for the US market and may not be representative of this market.

Baseline masses

The final definition of masses, coast down and technologies is presented and decided by ICCT. Based upon that, the following main factors are decided, and shown in the following figure. In addition, it should be noted that the change of the components of every variant is also considered in order to update the final vehicle mass.

ICCT data base 2013												
		NEDC					WLTP		WLTP			US Tests
							Max vehicle load					
	Mass iro	Ref. mass	Max. laden mass	Max. extras	Average extras	TML	TM	TMH	TML	Test mass	TMH	US Mass
Segment	[kg]	/ kg	/ kg	/ kg	/ kg	/ kg	/ kg	/ kg	/ kg	/ kg	/ kg	kg
DIESEL												
B	1224	**1249**	1658	150	60	410	350	260	**1310**	1361	**1437**	1350
C	1434	**1459**	1935	175	70	476	406	301	**1530**	1590	**1679**	1570
D	1625	**1650**	2126	225	90	476	386	251	**1721**	1797	**1912**	1781
E	1838	**1863**	2347	275	110	485	375	210	**1935**	2029	**2169**	2014
J	1688	**1713**	2195	275	110	482	372	207	**1785**	1879	**2019**	1864
N1 III	2026	**2051**	2800	220	88	749	661	529	**2261**	2324	**2419**	2180
GASOLINE												
B	1150	**1175**	1564	150	60	390	330	240	**1233**	1284	**1361**	1276
C	1345	**1370**	1834	175	70	464	394	289	**1440**	1499	**1588**	1481
D	1578	**1603**	2057	225	90	454	364	229	**1671**	1747	**1862**	1734
E	1800	**1825**	2287	275	110	463	353	188	**1894**	1988	**2128**	**1976**
J	1415	**1440**	1870	275	110	430	320	155	**1505**	1598	**1739**	**1591**

Figure 4.2 Coast down and mass reduction scenarios for simulation, as defined by ICCT.

Scenarios of mass and coast down reduction

The Figure 4.3 shows the two proposed scenarios for the mass and the coast down reductions in the simulation of the future vehicles, as investigated by ICCT.

	Baseline M1 CD1	Scenario I M2 CD2	Scenario II M3 CD3
Mass	-	-10%	-20%
RRC	-	-25%	-35%
Cd*A	-	-10%	-20%

Figure 4.3 Coast down and mass reduction scenarios for simulation, defined by ICCT.

Regarding the mass calculation:

- Every mass step will consider finally additionally the componentry (downsizing, after-treatment systems, hybrid components, etc.).
- The mass reduction step of M1 and M2 (-10% and -20%) is assumed as vehicle mass reduction
- The reductions are directly applied to the mass in running order.
- The Scenario I is assumed for 2020 and the Scenario II for 2025.

Regarding the rolling resistance:

- The reductions are applied to the Tyres associated Rolling Resistance Curve. The driveline effect will be considered constant.
- Looking at the analysis presented by ICCT, a 35% of reduction in 2025 might be feasible.

Regarding, the aerodynamic drag (Cd*A)

- According to FEV experience, reductions can be expected for the Aerodynamic factor, although the frontal Area will remain highly constant (the trend is even increasing in the last years).
- It should be considered other effects, such as the comfort in passenger cars.

- Assuming that there would be a variability upon the different segments, a maximum reduction of 20% at 2025 seems technically feasible.

Methodology for the transmission design and vehicle performance

Based on the maximum laden mass of every vehicle, the gear ratios are designed for every vehicle as follows:

- Criteria for the 1st gear
  - The launch of the vehicle must be possible with a minimum margin of 10 % with respect to the full load and with 30 % road gradient. The engine speed for the check is considered constant at 1000 rpm.
- Criteria for the maximum gear design
  - For Segments B and C, minimum power reserve of 10% at 130 kph and 3% slope at the highest gear.
  - For Segments D, E and SUV, minimum power reserve of 10% at 150 kph and 3% slope at the highest gear.
  - For Segment LCV, minimum power reserve of 10% at 100 kph and 3% slope at the highest gear.
- For all vehicles, the transmission gear ratios are adjusted to ensure the criteria mentioned above, but also considering the vehicle and powertrain properties: vehicle mass, coast down, turbocharger, hybrid systems, etc.

Shifting strategy

For the manual transmission vehicles, and depending on the cycles, the shifting strategy is as follows:

- Fixed for NEDC.
- For the WLTP, it is calculated by using the version at February 2015 of the UNECE Guidelines (Adapted from the Steven Tool).
- For the US cycles, calculated by following the standards: the gear shift depends on the vehicle speed.

For manual transmission vehicles, the shifting profile is fixed in the NEDC, while for the WLTP should be calculated by means of the guidelines established by the UNECE. Currently, the shifting profile for WLTP leads to drive on the highest possible gear while still keeping a theoretical margin of 10 % with respect to the full load (in order to consider auxiliaries, transmission efficiency, further losses, etc.). In addition, some drivability conditions are applied for correcting the final gear profile. Therefore, the shifting strategy for the WLTP leads the engine to a downspeeding strategy, usually beneficial for fuel efficiency in diesel but not always for gasoline. However, the higher emissions in diesel engines merit a specific analysis (close to the low-end torque area, the EGR calibration should be traded-off with the soot emissions). For the WLTP, the upgrading of MT5 transmissions, typical ones for the compact and small segments, to MT6 transmissions can offer more potential to drive the engine at more efficient areas. Additionally, the upgrading to automatic transmissions allows the powertrains to be further optimized.

To sum up about shifting strategies:

- In the case of diesel, the WLTP shifting strategy for manual transmission vehicles is normally in the direction of a better fuel economy but higher NO_x emissions, especially if comparing with NEDC. In the case of gasoline, the downspeeding is not always beneficial and depends upon the point.

- For the automatic transmission, the shifting strategy is optimized for the best fuel economy in the case of gasoline (as explained below). For diesel, the trade-off emissions-fuel economy should be optimized, and the shifting line tends to avoid areas of high NO_x emissions, especially with the base variants which have higher masses and coast down coefficients.
- In the following Chapters the main results and effects of the technologies are discussed for the diesel as well as for gasoline variants.

4.3 Diesel simulation

4.3.1 Model set-up and boundary conditions

The SimEx Simulation environment, as shown in Figure 4.4, is a Software solution developed by FEV in order to get a full chain tool for the calibration, simulation and validation of Diesel vehicles strategies.

The model considers different submodels for the full longitudinal simulation of the powertrain, such as transmissions, driveline, engine, after-treatment or hybrid components. The tool can be used for optimization, e.g. minimizing the total costs of the system (fuel and AdBlue® for SCR systems) while keeping the CO_2 and pollutant emissions at a certain value, defined by the considered standards. For achieving such objectives, engine cycles can be simulated with a user-friendly interface and within a short computer time, permitting to quickly set and run a big number of iterations.

The parameters for optimization can be chosen in completely free manner e.g. different combustion modes or engines, operation strategies (warming-up phases, shifting strategy, etc.), EATS parameters like size and position, among others. Furthermore different models can be considered.

If the tool is already used in a state where the exhaust aftertreatment is not fixed or defined there is the opportunity to design and optimize the aftertreatment system on the lowest total cost of ownership including the prices for precious metal and consumed fuel over life-time.

Overall, this tool allows the newly integrated optimization of the whole calibration process instead of calibrations based on individually optimized parts, the so-called holistic approach.

The vehicle model presents the following subsystems:

- Engine: the fuel efficiency, exhaust conditions as well as the engine emissions are calculated based on available maps, which have been previously calculated based on the different considered technologies and calibrations. A set of three maps (normally 30°C, 60°C and 90 °C engine coolant temperature based) is considered. Furthermore, different combustion modes, e.g. for rich combustion or variable valve timing actuations, can be simulated. Even though not considered in the project, DoE models are also available in the tool. In addition, a physics based mean value engine model (MVEM) of the air and turbocharger paths based on physical equations and maps may be integrated for a more detailed physics calculation of the air path in order to improve the transient response of the system. Other engine related submodels consider engine coolant and oil temperature models in the same way as shown in Figure 4.56 (for standard and split cooling version or the alternator).

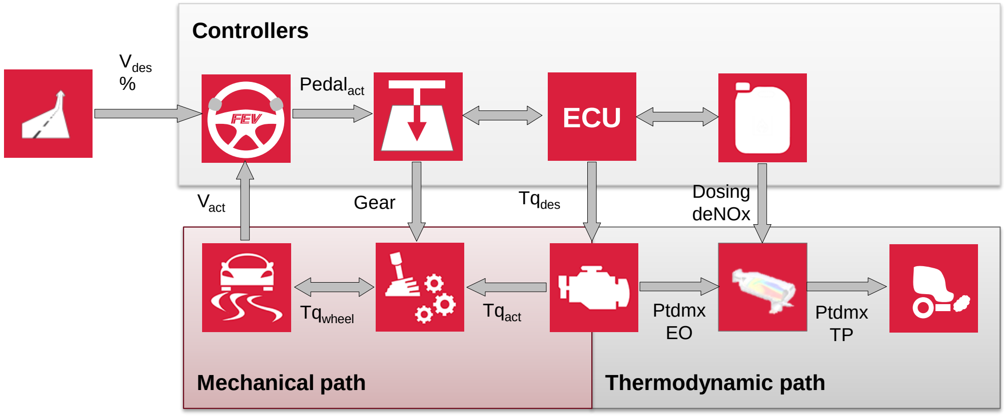


Figure 4.4 SimEx simulation environment

- Engine Aftertreatment systems: the different EATS might be modelled with the kinetic reactions and/or with grey box approaches based on look-up tables, as currently implemented in the most of the ECUs. For this project, the DOC, DPF, SCR and SDPF systems have been modelled with a set of physical equations and maps. The coordinators for the AdBlue® injection or regeneration of the DPF and LNT systems are also considered when necessary.

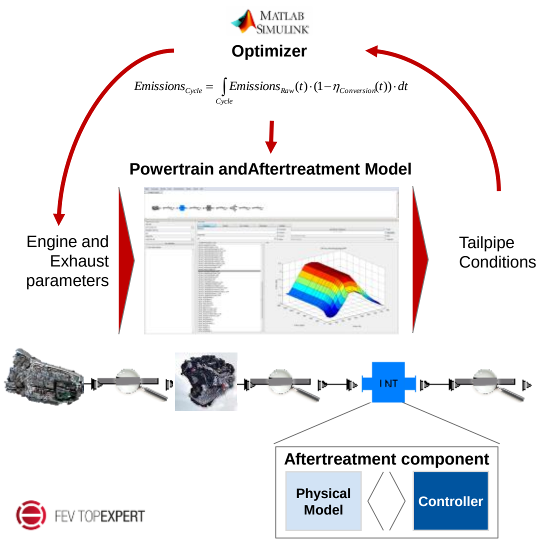


Figure 4.5 SimEx integrated Powertrain and Aftertreatment simulation with an user friendly interface

- Transmission: different transmission concepts (MT, DCT or AT) are also available. The optimization of the shift strategy as well as the gear ratio design may also be performed in order to reduce fuel consumption and emissions, while keeping the drivability and vehicle performance as required.
- Hybrid components might also be modelled, e.g. belt starter generator with 48 V (P0 solution) or full parallel hybrids (P2 solution). Other configurations not considered in the project are also possible.
- Driver and Vehicle model: The driver influence and the vehicle physics (inertia, mass and coast down) are considered jointly with the transmission model for defining the

engine speed and load, which will be further used in the engine model. A standard driver for all cases is selected for the current project, provided that his effect might be more relevant under real driving conditions, not considered in the current study. The driver influence is therefore neglected when comparing different vehicles and cycles.

Due to the relevance in the current project, the calibration and selection of the main components for the simulations are briefly described:

- Engine calibration
- Aftertreatment configuration and strategy
    - LNT regeneration
    - SCR systems and AdBlue dosing strategy
- Shifting strategy selection
- Hybrid strategy

4.3.1.1 Engine configuration

The Figure 4.6 shows an example of different advanced engine technologies (e.g. variable valve train, variable compression ratio), whose impact can be simulated on the basis of using data from FEV internal investigations on research engines.

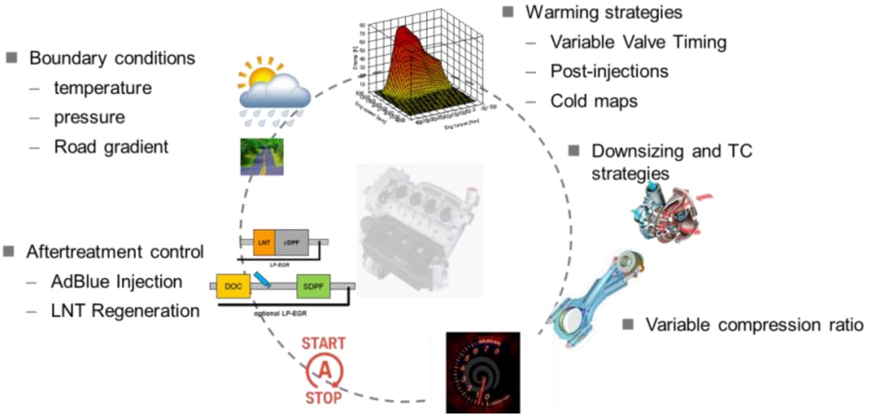


Figure 4.6 Overview of different technologies that can be simulated for Diesel Engines.

The calibration is made by means of map based modifications as shown in the example of Figure 4.7. In this example, starting from a real measured map, this is adapted to a specific engine: the new full load, engine friction or the new engine size are considered. At this step, the new full load is achieved by means of analysing similar benchmark vehicles. The engine friction is recalculated based on internal tools which analyse the friction of different engine configurations considering power, number of cylinders, cylinder displaced volume, applied technologies and the most relevant properties.

In subsequent steps, other technologies might also be analysed. For example, the installation of an electrical water pump, or a split cooling system jointly with some basic friction reduction measures are measures that are helping to reduce the overall engine friction. At the same time, the two first technologies also aid on reducing the warmup phase, which will be covered by the specific engine coolant model. For this, the coolant and oil temperature models are recalibrated to consider this behaviour while engine measurements are used in order to get the new factors properly. The influence over the engine maps is collected and build through delta maps which are obtained by means of the experience gained through actual engine testing, the most of the cases by correctly deriving these maps from global DoE campaigns of the considered technologies.

In the last step shown in the example, different compression ratios can also be simulated or even variable compression ratio technologies. For the latter example, the strategy for the CR step selection might also be tuned; for the current project and for diesel, a 2-stage operation has been selected. Apart from this, different warming strategies (electrical heating, post-injections or VVT), aftertreatment layouts, and other conditions might also be covered.

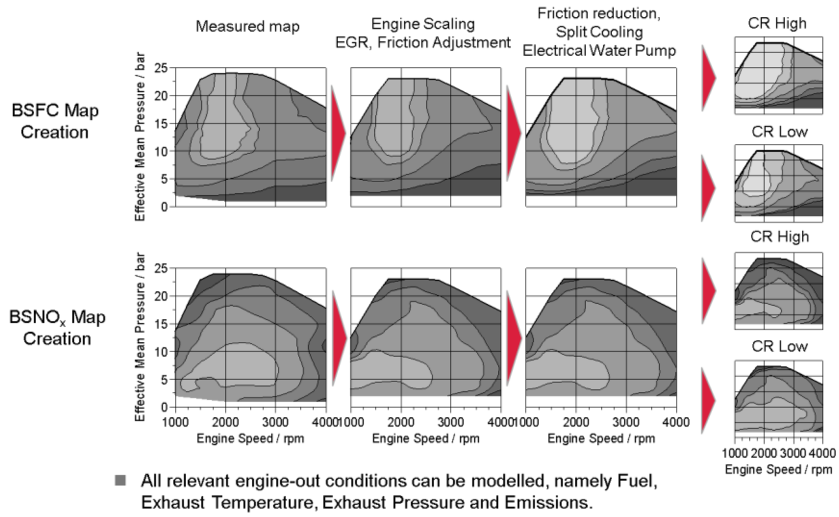


Figure 4.7 Design of Engine Maps.

4.3.1.2 Aftertreatment configuration and strategy

The aftertreatment configuration starts with the selection of appropriate aftertreatment (EATs) layouts depending on the final application. The Figure 4.8 shows a possible selection of EATs for different applications and according to the today´s expectations of FEV for EU6c. Bear in mind that the picture for the US market would be completely different, since the limits and the cycles are more stringent; the aftertreatment layouts are limited to active solutions, and the heating strategies even play a more relevant role.

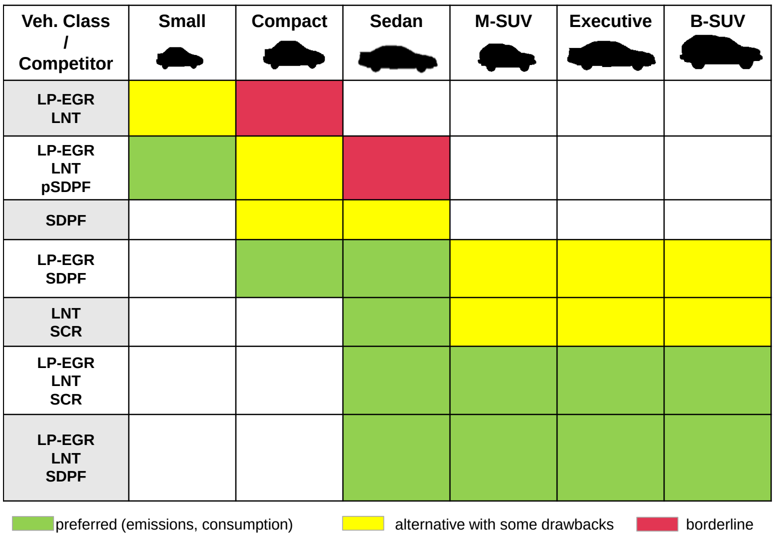


Figure 4.8 Different DeNO$_x$ solutions for different vehicle segments according to today´s expectations of FEV for EU6c. (Even if not mentioned, the DPF is installed in all subsystems where a SDPF is not installed)

Starting for with **the small segments**, there is a clear trend today to select LNT-only solutions for deNO$_x$, deHC and deCO, since it is a cost effective solution: with the LNT system helps you reduce the NO$_x$, HC, CO without the need of installing a DOC and some SCR solution; which might also require AdBlue injection. From this point of view, B and C-segments might be appropriate candidates for such configuration. However, this is valid for the current EU6b and NEDC cycle procedure for emissions. The expected application of the WLTP as procedure, and the RDE for EU6c, might become such deNO$_x$ solution obsolete, if used without any combination. The LNT is a NO$_x$ storage solution, reaching the maximum efficiency at medium-low exhaust temperature, but presenting poor performance at high exhaust temperatures, where the SCR reaches its maximum efficiency. In addition, higher LNT volumes might be more expensive than SCR ones, due to the PGM costs; at the same time, the LNT should be purged every certain time by running the engine for 1-10s at rich combustion, therefore increasing the final fuel consumption and hence the CO_2 emissions.

An **intermediate solution** is the installation of passive SDPF systems (pSDPF) as well as keeping the LNT, which takes the NH3 released during LNT regeneration in order to use it as reduction agent with the NO$_x$. This system can increase the overall LNT efficiency up to around 10%, without the need of AdBlue injection and without requiring increasing the volume of the LNT. However, the EAT strategy is becoming more complex. This is one possible combined solution, which obviously requires some packaging redefinition. For compact vehicles, such combined solution might be an appropriate alternative, since the use of only-LNT might not guarantee the emissions fulfilment in RDE.

For **heavier vehicles**, starting from the D-Segment, the use of active deNO$_x$ solutions seems mandatory, understanding for active, systems which require AdBlue injection. For such systems, there is also a range of different solutions. These cover close coupled SDPF systems (closed to the engine), LNT with an underfloor (but active) SCR and LNT with a close coupled SDPF. Even if being an engine related technology, the low pressure EGR (LPEGR) is a preferable technology with respect to the use of only high pressure EGR solutions, since the EGR rates can be increase up to much higher loads as well as the LPEGR loop influences positively the fuel economy.

The AdBlue, which is conveniently hydrolysed into NH3, is dosed into the system and stored in the catalytic in order to react with the NO_x. The efficiency of such systems is maximum from 250 °C on approximately, being lower at lower temperatures. Furthermore, more aggressive cycles with lower (or no) cold phases are the best for such systems, since the efficiency are maximised. Therefore, the efficiency tends to be lower for SCR-based systems in the NEDC than in the WLTP.

At the same time, higher efficiencies at higher temperatures allow to calibrate the engine with higher engine-out NO_x emissions, which means better fuel economy with respect to LNT only systems. Furthermore, there is no need to run rich modes for purging the system when there is no LNT system. However, the AdBlue injection and tank should be considered as well as the packaging and the OBD-related costs (sensor, calibration, etc.). For the current project, since no real driving cycle was analysed, the AdBlue injections are far from reaching the worst-case scenario. As an example, these can be up to 4-5 times higher in real driving cycles than in WLTP and even more with respect to the NEDC. Due to that, this value is not considered as relevant for the current project.

Due to the high number of variants to simulate and for the sake of simplicity, two main aftertreatment layouts have been considered:

- LNT + DPF
- DOC + SDPF

In the following and due to the relevance in the current project, the two main coordinator strategies are briefly described.

4.3.1.3 LNT coordinator

The LNT systems store the NO_x during normal operation. However, when such systems reach their limits, they should be cleaned or “purged” in order to empty them. This happens by running the engine at rich combustion mode (λ~0.95) during a short time (up to 10 s). The number, duration as well as the location of such rich events are properly coordinated. There are a number of certain conditions, which should be fulfilled in order to run the rich events and clean the systems. These can be summarized into:

- Exceedance of the maximum allowable load by the LNT. When this value is reached, the system is not able to store NO_x anymore.
- Drivability constraints: Minimum gear, minimum-maximum speed, Engine torque. For example, there is a window over the engine map which should be accordingly calibrated in order to run a rich event, but avoiding any drivability limitation.
- Physical limits of the exhaust systems: there exists a LNT temperature range not-exceed in order to avoid any damage over the LNT or other exhaust systems.
- Fuel consumption constraints: running rich events is not fuel efficient. Every purging increase the final fuel consumption over the cycle. This penalty may range up to 2% of higher fuel consumption in the WLTP for a C-Segment vehicle, but even higher at real driving conditions.

All these conditions should be calibrated and optimized to reach the final emission legislation but minimizing the final fuel penalty.

4.3.1.4 AdBlue Dosing strategy

The m_{NH3}, which defines the NH_3 load of the SCR, should be filled up by injecting Urea (or the commercial solution AdBlue®) to the system. This task in the model is made by the Urea Dosing Model, which is similar to the ones you can find in a standard ECU today.

The dosing model considers two main contributions in order to calculate the total AdBlue® to be injected to the system:

- **Feed-forward control:** defines the precontrol quantity based on the removed NO_x previously calculated in the SCR model, where α_{dos} is the dosing ratio which relates the real quantity of NH_3 required in order to remove the NO_x calculated by means of the SCR model. This value mainly depends on the exhaust temperature, being higher when this increase since the NH3 oxidation mechanism is more relevant. Up to EU6, the AdBlue® injections are restricted at high temperatures: this allows reducing the total NH_3 consumed in the system. However, for the simulation considered in this project, the storage mode is considered at all time and conditions; producing that the AdBlue® injections are considerably higher at higher temperatures with respect to today´s vehicles. This is in the direction of the future calibration.
- Storage level control: **The storage-level controller (load governor)** defines the required $\dot{m}_{NH_{3,SL}}$ in order to fill the catalyst at an appropriate level. This target level is defined to avoid NH_3 slip while keeping the catalyst ammonia level (m_{NH3}) at an appropriate value. Therefore, this target should be a value with some margin with respect to the maximum allowable capacity of the SCR system during temperature gradients but still being high enough to ensure a reasonable conversion efficiency η_{est}.

To sum up and for the sake of simplicity, the total $\dot{m}_{NH_3}$to be dosed in the system considers the contribution of these two quantities, as well as the injector limitations or the exhaust temperature (dosing is not allowed under 180°C and it is limited at low temperature to avoid deposits of urea).

The calibration of all aftertreatment models, including DOC, DPF, SDPF and LNT is made based on laboratory gas bench measures. These also include the piping and the packaging, by considering the heat transfer and losses. Finally the strategies for the LNT regeneration and SCR systems consider the FEV application experience as well as the model optimization; e.g. the number of $deNO_x$ events as well as the AdBlue injection will differ depending on the application.

4.3.1.5 Shifting strategy

The gear shifting is defined for the vehicles with manual transmissions; but the shifting strategy can be optimized for the automatic transmission vehicles. In this project, two main automatic transmissions are subjected to study: dual-clutch transmissions and automatic transmissions. Even if the performance or physical layout is different for these two, the shifting strategy can be optimized in the same way for both.

The Figure 4.9 shows the fuel share over the brake specific fuel consumption –BSFC- map (top plots) and engine-out brake specific NO_x share –$BSNO_x$- (bottom plots) for one engine with two different shifting strategies: left is BSFC optimized, while the right shows one for a BSFC-NO_x trade-off.

The left plots consider the best BSFC line; the optimization considers the most efficient line of the engine: at every step, it is analysed if the engine is running at the best possible area of the map. If the engine is working over the line, a downshift can increase the engine speed but reducing the load (power is constant). Alternative, if the engine is working at high engine speed and low load, it is possible that downshifting can be beneficial to increase the low and the engine speed (downspeeding). When accelerating in diesel, downspeeding is usually beneficial to reduce the fuel consumption. A certain hysteresis area is considered to avoid frequent gear shifting and to ensure the drivability and an efficient actuation. This is also the typical strategy for gasoline vehicles.

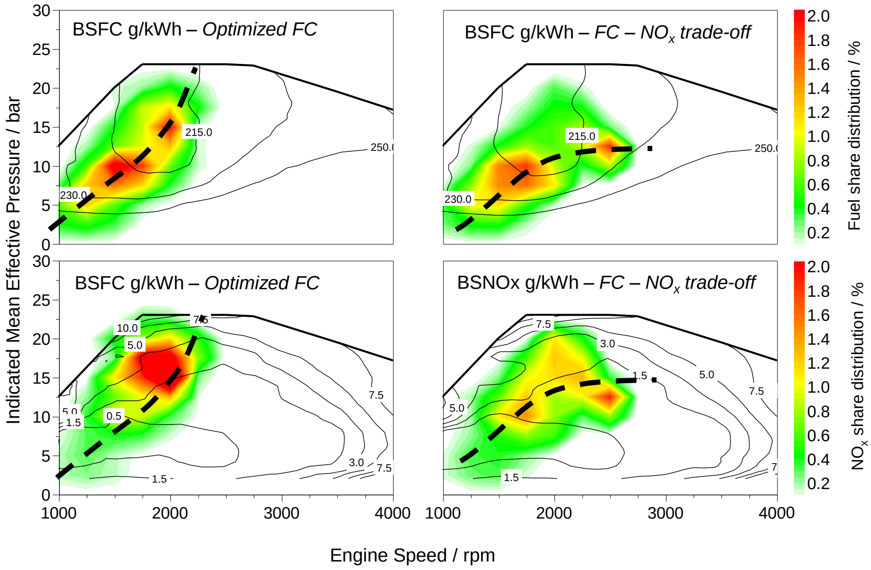


Figure 4.9 Fuel consumption optimized shift strategy for driving cycle

Nevertheless, the NO_x emissions tend to be higher when the engine is working more efficiently. It can be clearly seen how there are some red areas with values of $BSNO_x$ between 5-10 g/kWh which contribute to higher emissions. Therefore, this fact should be considered and the shifting line should be optimized in order to cut such emissions. Hence the right plots shows how the shifting line is close to the best BSFC line up to around 1800 rpm, but from that value on, it is tilted to areas with lower $BSNO_x$. This second version will give rise to lower engine NO_x emissions but lower fuel economy. At the very end, the deNOx efficiency of the installed aftertreatment systems must be considered accordingly. During this project, the shifting strategy has been modified in order to decrease the BSFC but still keeping the $BSNO_x$ under the defined limits. Typically, the LNT systems may be efficient at lower temperatures, but less efficient at higher temperatures, which coincide with the higher engine loads. Therefore, the Figure 4.9 is a good example of a vehicle mounting a LNT system, where the shifting line (right plots) is moved to areas of lower $BSNO_x$ at higher loads, but still optimized (best BSFC) for the partial and low load operation.

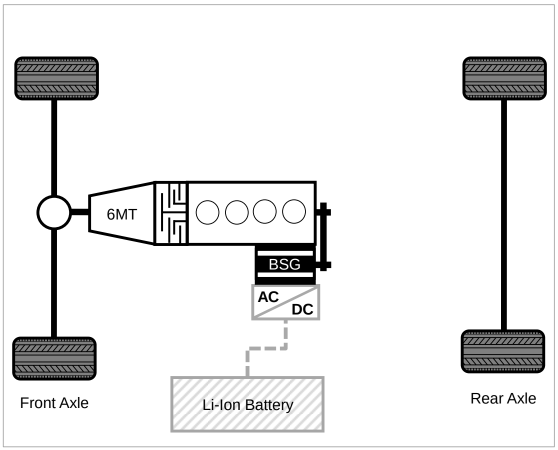
4.3.1.6 Hybrid strategy

Two different hybrid layouts have been explored in the project: P0 for the B and C segments, and P2 for D, E, SUV and LCV.

The **P0 configuration** considers the installation of a Belt Starter Generator (BSG) coupled to the engine and the AC/DC convertor, at the same time connected to the Battery. The BSG boosts the engine when necessary being able to:

- Increase the effective low-end torque and performance (Performance oriented).
- Shifting the combustion engine load in order to work at better efficiency areas of the engine (e.g. it can act in coordination with the shifting strategy of DCT vehicles).
- Reducing the combustion engine load to avoid high load engine areas where the NO_x emissions are high (e.g. no EGR areas).

- Gaining energy due recuperation to increase overall powertrain efficiency.
- The transmission gearing might optimized if considering the BSG.
- The shifting strategy might be benefited from the hybrid system, which allows reducing the effective engine-out NO_x emissions. Additionally, this fact might work as an enabler for introducing less sophisticated deNO_x aftertreatment systems.



The vehicle features a P0 configuration including:

- Diesel Engine
  - 1.2L (B-seg)
  - 1.4L (C-seg) 7 gears DCT
- 15 kW 48V BSG
- 0,53 kWh Li-Ion Battery

Figure 4.10 P0 configuration.

The Figure 4.11 shows an example of how the boosting strategy is defined. First, the electrical boost (e-Boost) is requested in order to minimize the engine-out NO_x emissions and reducing the total energy required by the engine. At the same time, the e-Boost only operates at areas where the electric machine operates at the best efficiency point. The electrical energy in order to load the system is only gained during recuperation, no additional charge is provided by the engine.

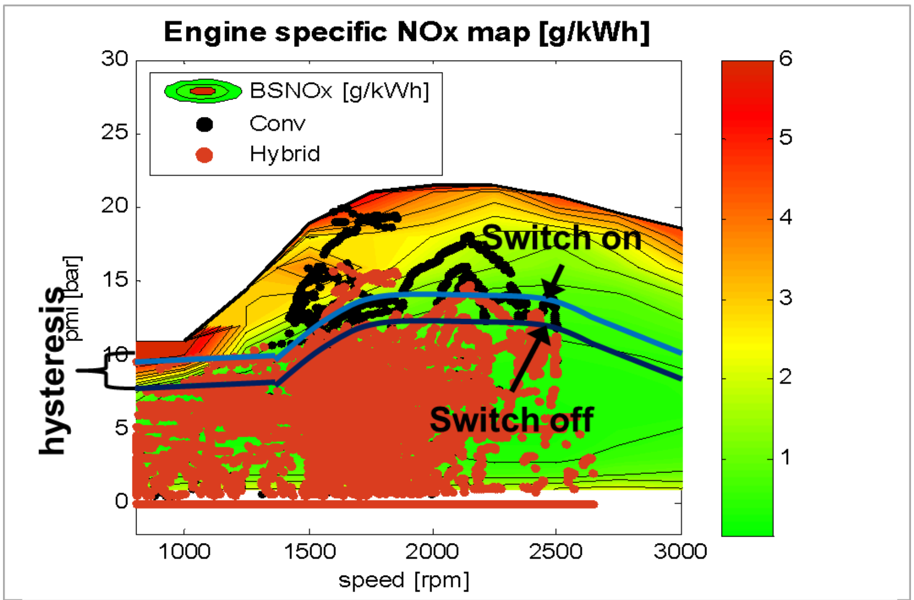


Figure 4.11 P0 boosting strategy.

The **P2 configuration** considers the installation of an electrical motor (EM) coupled between the DCT transmission and the clutch. The EM is of course connected to an AC/DC convertor and the battery. The P2 acts similarly to the P0 but presents a higher battery capacity (>1 Kwh against 0.5 kWh for P0) and higher power (~40 kW against 15 kW of the P0). Therefore, the final NO_x reduction and CO_2 benefit can be maximized with respect to the P0 configuration.

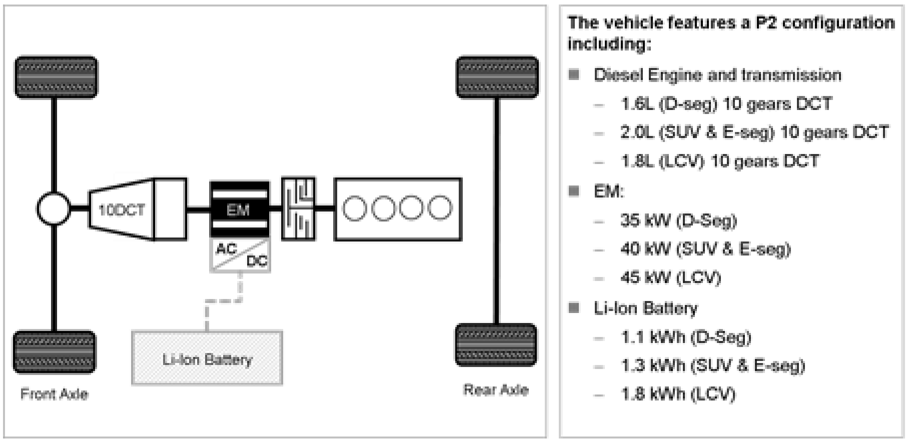


Figure 4.12 P2 configuration.

The strategy is similar to the one of the P0, a set of On/Off thresholds are defined based on the actual power demand. The hybrid support is active whenever the engine operates higher than the switch on line but it is deactivated if it is working lower than the switch-off line. Other conditions, such as the battery load, are also considered. As shown in the Figure 4.13, the lower engine efficiency areas can be avoided by running the vehicle with the EM at such points. As for the P0 case, the electric energy is only gained by recuperation, no active charging is considered. Furthermore, full electrical driving is not here considered.

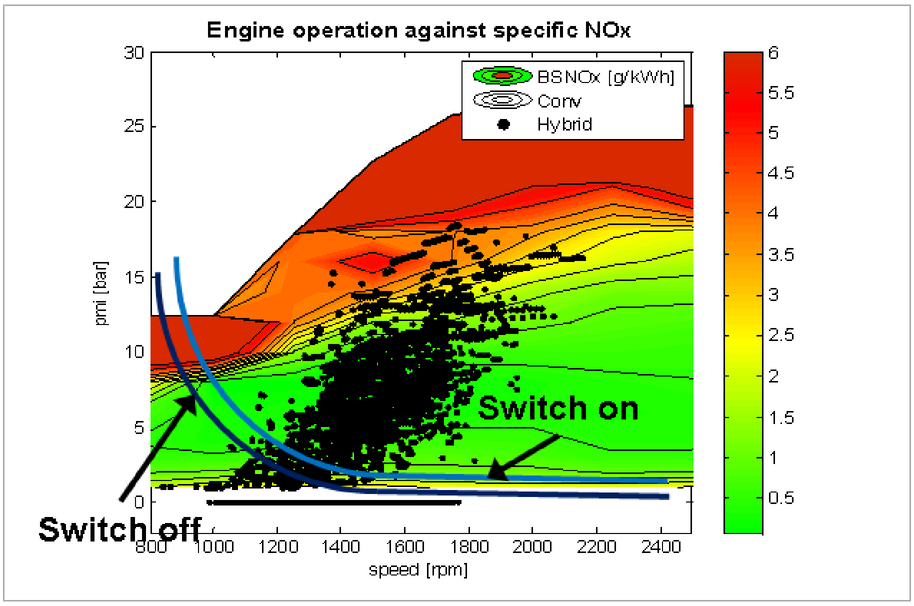


Figure 4.13 P2 electric driving strategy.

4.3.2 Tailpipe Emissions

The emissions management in diesel engines impacts directly or indirectly the final powertrain calibration and thus the fuel consumption, especially for diesel powertrains. As discussed in the boundary conditions, the emissions target considered within the project are meeting emissions standards according to EU6b limits while considering both NEDC (procedure for EU6b) and WLTP (procedure for EU6c).

In the most of the cases, the worst case scenario comes with the WLTP TM_H cycle: the engine-out NO_x emissions are considerably higher than in NEDC or WLTP TM_L, and even with a possible higher $deNO_x$ efficiency of the aftertreatment systems, the tailpipe emissions tend to increase.

One important remark should be mentioned again here: the real driving emissions (RDE) and hence, the emissions compliance of the selected powertrains with real driving cycles is out of the scope of this project. This analysis would require the selection of an appropriate real driving route with real conditionings (traffic, weather, urban/rural/highway share) and the use of the preselected evaluation tools for processing the emissions results, e.g. CLEAR, EMROAD and/or ACEA guidelines. This study might be performed by FEV but it would merit a dedicate effort.

Furthermore, two aftertreatment layouts have been selected for the project:

- LNT + cDPF for the most of the variants of Segments B and C ,
- DOC + SDPF for Segments D, E, SUV and LCV; and for some variants in the Segments B and C.

These selected aftertreatments were fixed at the beginning of project; however, this selection might be reconsidered if covering RDE conditions. For example, the base vehicle of the segment C (mounting a LNT), might require some upgrade in order to fulfil RDE conditions, by means of installing a LNT + passive SDPF or simply installing a DOC + SDPF. These is just one example, but anyway a specific study would be necessary.

After simulating the vehicles, the emission targets were fulfilled for all cases (boundary condition of the simulation). However, there are two specific variants which are fairly interesting due to have the higher risk of high tailpipe emissions: Base and downsized versions of the C segment. These variants kept a really low engineering margin, i.e. the emissions targets are fulfilled but it does not keep a sufficient margin with respect to the tailpipe emissions. These can easily be identified in the Figure 4.14, and these presents no mass or coast down reduction, mount a LNT system (with lower $deNO_x$ efficiency at higher temperatures), and for the current study they represent the worst-case scenario. Moreover, the baselines are also fitted for fulfilling EU6 emission limits within the NEDC cycle as well as in WLTP. The simulation of the WLTP procedure with higher average masses, coast down and with a more aggressive cycle will clearly introduce higher complexity, and will require of higher $deNO_x$ efficiencies.

The Figure 4.14 shows the engine-out and tailpipe NO_x for the baseline and downsized versions of all segments. Moreover, the fuel penalty coming from the LNT regeneration strategy for the segments B and C, which mount LNT, is also shown in the bottom plot. The reference cycle is the WLTP TM_H.

Regarding the **engine-out emissions,** the heavier segments present higher engine-out emissions, with a breakthrough in the LCV vehicle, which is clearly the heaviest vehicle with the highest rolling resistance and worse aerodynamics. The segments E and SUV behave quite similar as also happen with the CO_2 emissions: the segment SUV presents worse aerodynamics and rolling resistance, but the segment E is heavier. Overall, both segments are similar. The step between the segment C and D is also high, mainly due to the change of engine technology (apart from the weight and coast down): segments B and C mount a

CHPCLPEGR engine while the others mount only CHPEGR, limiting the possibility to keep high EGR rates at higher loads.

Regarding the **tailpipe emissions,** it is also clear the difference between the segments B and C, with a LNT system, with respect to the other segments, which mount a close-coupled SDPF system. Even though the engine-out emissions are considerable higher for the latter segments, the deNO$_x$ efficiency is also much higher for them.

Furthermore, **the downsizing trend in order to reduce CO_2 emissions presents clear disadvantages in terms of NO_x emissions for all the diesel segments**. The main reason is that both downsizing and downspeeding together run the engine at lower speeds and higher loads, where the EGR rates are limited and the corresponding NO_x are also higher. A possible upsizing strategy might be advisable for RDE legislation, since the benefits coming from the downsizing in terms of CO_2 can be counteracted by increase of engine and aftertreatment costs.

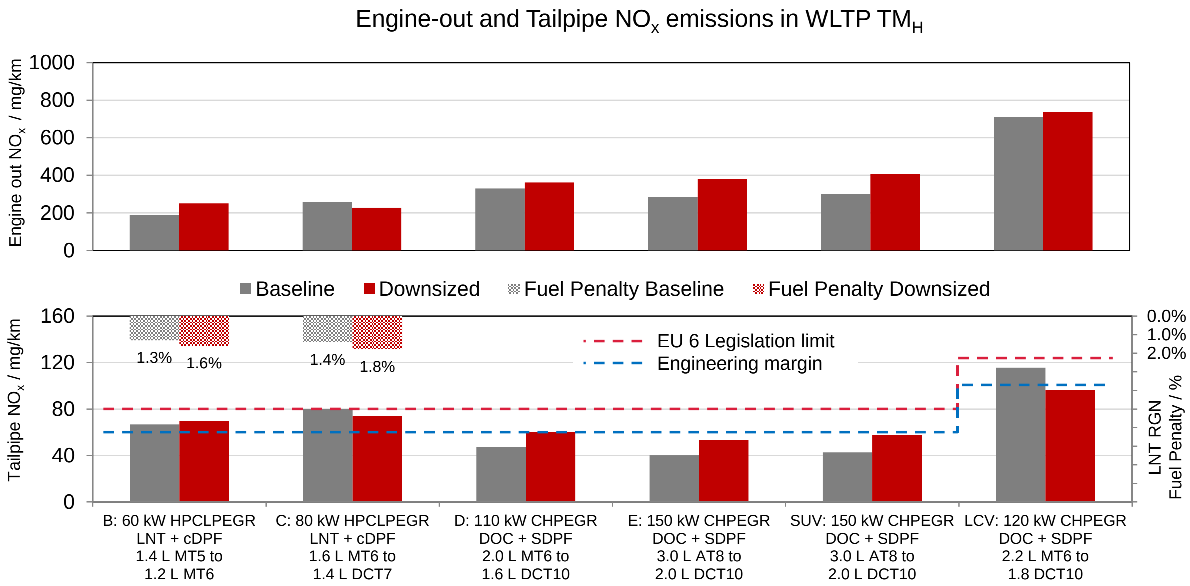


Figure 4.14 Engine and tailpipe NO_x emissions for the baseline and downsized variants of all Segments in the WLTP TM_H cycle (all equipped with Start & Stop). The fuel penalty coming from the LNT regeneration is also shown for the Segments B and C (which mount a LNT system). The EU6 legislation limit (80 mg/km) is shown in the graph, while a selected engineering margin is plotted for the segments (60 mg/km for all passenger cars and slightly more aggressive in comparison for LCV: 80 mg/km).

Through segments, the following conclusions can be extracted, when focusing on the WLTP TM_H cycle:

- The **Segment B** mounts a LNT and a MT5 transmission. The baseline version fulfills with an affordable emissions margin the limits; around 63 mg/km of NO_x tailpipe-based. The downsized version, equipped with a MT6 transmission, presents considerably higher engine-out emissions (~250 Vs 200 mg/km). Finally, the tailpipe targets are fulfilled but with a low engineering margin, around 71 mg/km. The fuel penalty in both cases ranges from 1.3 to 1.4 %, for the baseline and downsized version respectively.

- The **Segment C** baseline mounts a LNT and a MT6 transmission, while the downsized version upgrades the transmission to a DCT7. There are significant differences when using a DCT transmission which allows to move the operating points to engine areas where the NO_x emissions are lower; but normally penalizing the fuel economy potential. In such a way, the baseline version of this vehicle presents slightly higher engine-out NO_x emissions than the downsized, and hence the tailpipe values are lower in the downsized version than in the baseline. In any case and for both cases, there is barely any engineering margin for the tailpipe NO_x emissions: 79 mg/km for the baseline and 73 mg/km for the downsized version.
- To sum up, these vehicle will probably exceed the tailpipe limits on a real driving cycle compliant with the current RDE guidelines with a moderate conformity factor of 1.5. A combined system or even the exchange of the aftertreatment system by an active SCR solution might be an alternative for such segment. In addition, the downsized version of the segment C considers a transmission upgrading from a MT6 to a DCT7 transmission. This also allows to optimize the shifting strategy to slightly reduce the engine-out emissions. Therefore, it is even possible to decrease the engine-out emissions slightly. In addition, the fuel penalties coming from the LNT regeneration are increasing within the cycles, reaching a value of 1.8 % for the downsized version of the segment C. This fuel penalty would also increase when running a RDE cycle, and it should be considered in the trade-off.
- The **Segment D** with a close-coupled SDPF is able to fulfill with sufficient engineering margin for both the baseline and downsized versions. Nevertheless, the downsized version increases significantly the tailpipe emissions.
- The **Segments E and SUV** present a very similar behavior, since both mount the same engine and powertrain. The baselines fulfill the emission limits with a sufficient engineering margin but this is considerably reduced for the downsized versions, with tailpipe emissions in the range of 65 mg/km. An upgrade of the aftertreatment system might be necessary if running on RDE.
- The **Segment LCV** presents the highest engine-out and tailpipe emissions of all segments. However the EU6 legislation allows a higher limit for such vehicles, 125 mg/km instead of 80 mg/km. The downsized version of LCV vehicles is able to fulfill the emissions limits with a certain margin (thanks to the shifting strategy of the DCT10); however the base variant presents a low engineering margin tailpipe based (115 mg/km). In the case of the downsized version and due to the highest average temperatures, the tailpipe emissions are lower than for the baseline version. Nevertheless, a high increase of the engine-out emissions might be expected on RDE, and a dedicated analysis should be performed.

4.3.3 Technology potentials

The simulation results for the various segments are presented as a summary in this section. The simulations are carried out to evaluate the impact of various technology upgrades on the CO_2 emissions for a particular segment, such as:

- Impact of downsizing an engine (shifting of operational load points to better BSFC regions),
- transmission upgrade,
- impact of mass and coast reduction,
- friction optimization,
- engine and emission reduction technology (EGR, VCR and VVT), and
- Hybrid components.

In order to assess the impact of each of the above mentioned technological change on the fuel consumption, the segment C results are chosen as a case study while the relevant data related to the other segments are summarized and compared. In the most of the cases, the variations and potentials will be similar, with only small variations. The main differences within the other segments will be anyway underlined. In the Appendix, all average results for all segments can be found. The Table 4.3 shows the summary of variants considered for Segment C, while Figure 4.15 shows an overview of the different results of the variants considered in the Segment C.

Segment C		**EGR**	**T/C**	**VVT**	**CR**	**Engine friction**	**Hybridization**	**Trans-mission**	**EATS**	**Coast down set**	**REF / kg**	**TML / kg**	**TMH / kg**
T1	Baseline	HP-CLP	single	not variable	not variable	baseline	without	MT6	LNT+CDPF	CD1	1459	1530	1679
T2	Start/Stop	HP-CLP	single	not variable	not variable	baseline	**start/stop**	MT6	LNT+CDPF	CD1	1459	1530	1679
T3	Downsizing/ Transmission/ M red Scenario I	HP-CLP	single	not variable	not variable	baseline	**Adv. start/stop**	**DCT7**	LNT+CDPF	CD1	**1469** **1326** **1182**	**1540** **1397** **1254**	**1689** **1546** **1402**
T4	M reduction Scenario II	HP-CLP	single	not variable	not variable	baseline	Adv. start/stop	DCT7	LNT+CDPF	**CD1-CD2-CD3**	**1182**	1254	1402
T5	Friction, resistance	HP-CLP	single	not variable	not variable	**-20% friction, el. coolant pump**	Adv. start/stop	DCT7	LNT+CDPF	CD2	1182	1254	1402
T6	Engine technologies	**CHP-CLP**	single	not variable	**2-step VCR**	-20% friction, el. coolant pump	Adv. start/stop	DCT7	LNT+CDPF	CD2	1182	1254	1402
T7	Aftertreatment	CHP-CLP	single	**variable**	2-step VCR	-20% friction, el. coolant pump	Adv. start/stop	DCT7	**DOC+SDPF**	CD2	1197	1269	1417
T8	Without weight reduction	CHP-CLP	single	variable	2-step VCR	-20% friction, el. coolant pump	Adv. start/stop	DCT7	DOC+SDPF	CD2	**1484**	1555	1704
T9	Hybrid, P0 , 48V	CHP-CLP	single	variable	Not variable	-20% friction, el. coolant pump	Adv. start/stop	DCT7	DOC+SDPF	CD2	1205	1276	1425

Table 4.3 Summary of variants for Segment C.

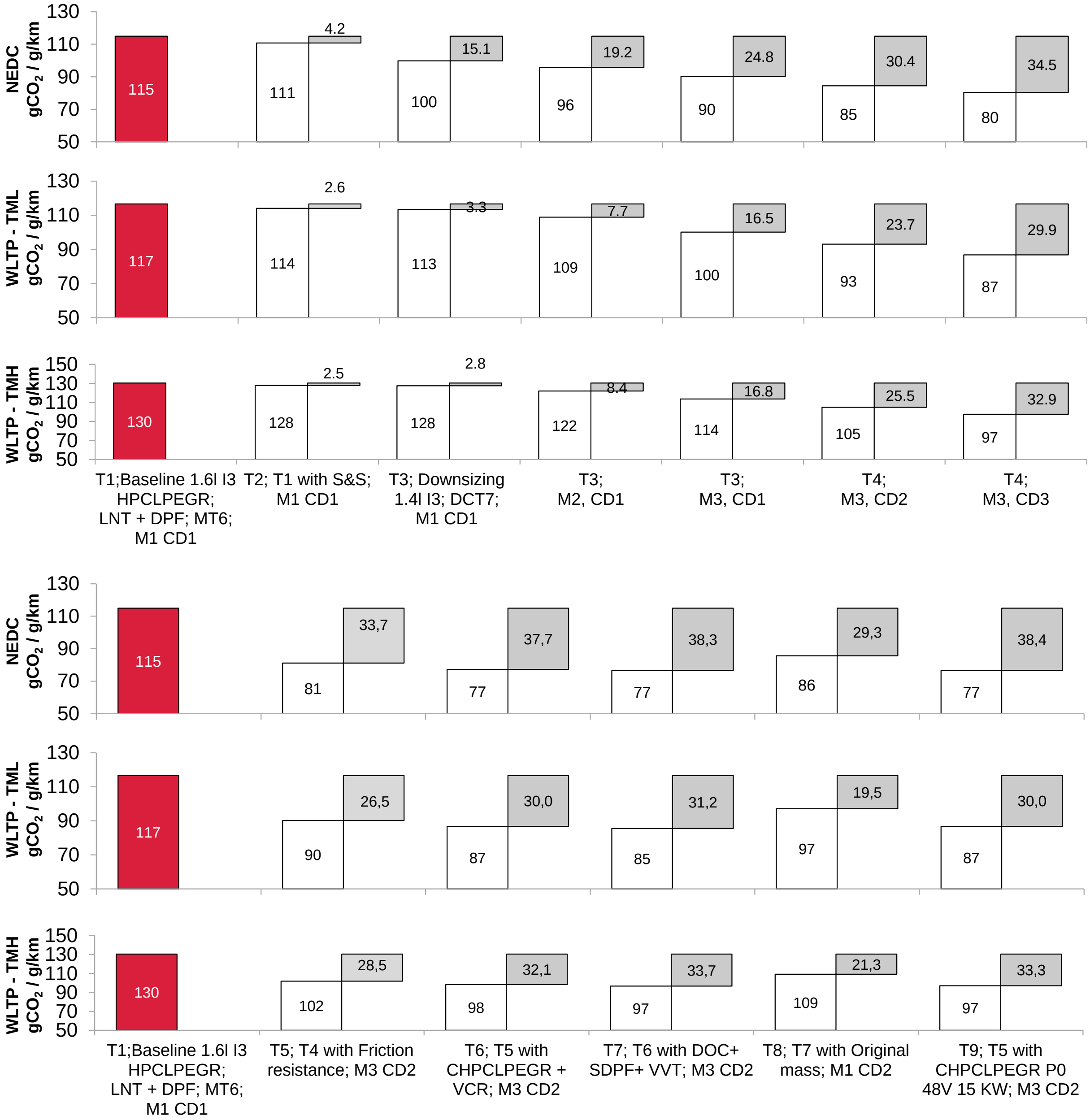


Figure 4.15 Overview of results for Segment C (EU cycles).

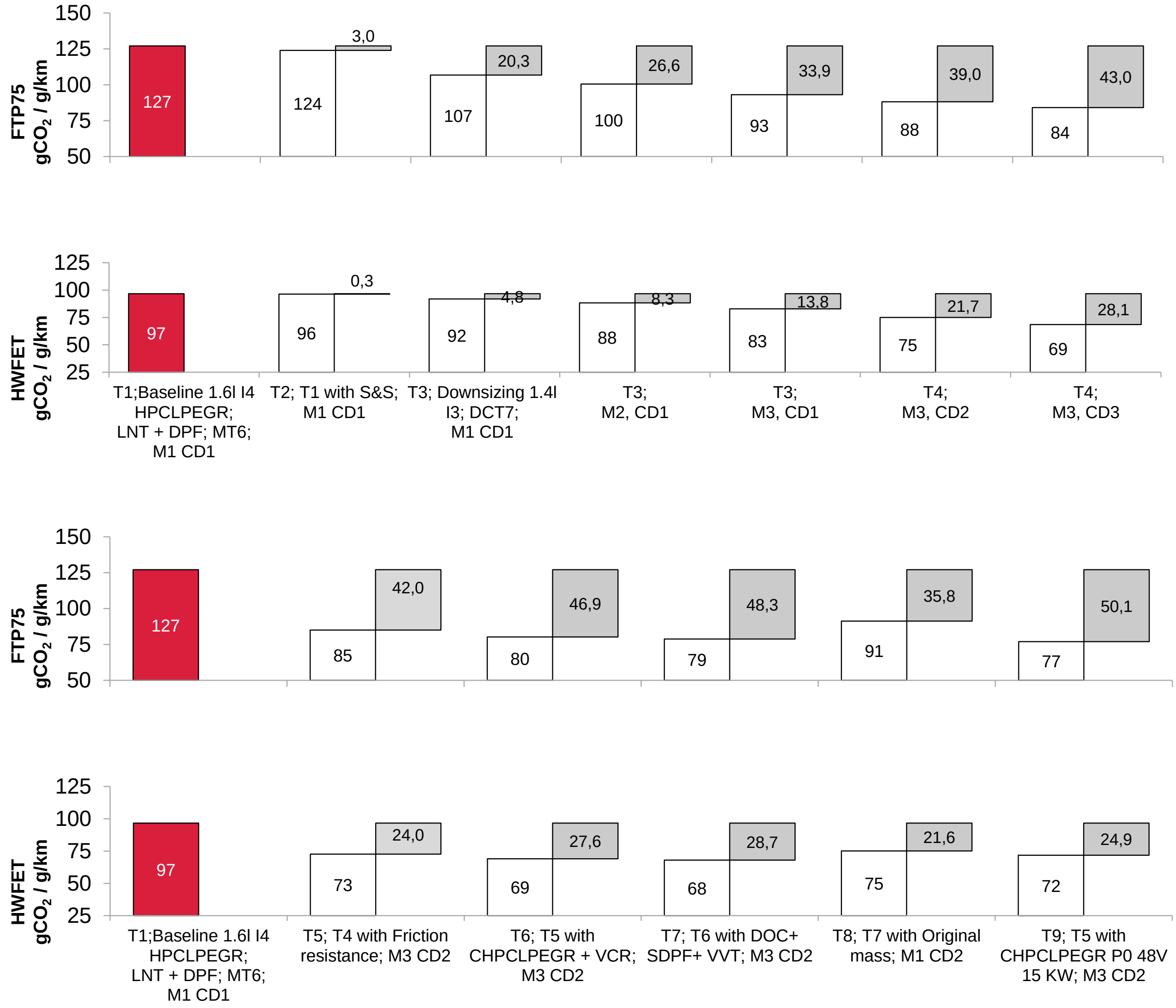


Figure 4.16 Overview of results for Segment C (US cycles).

4.3.3.1 Baseline

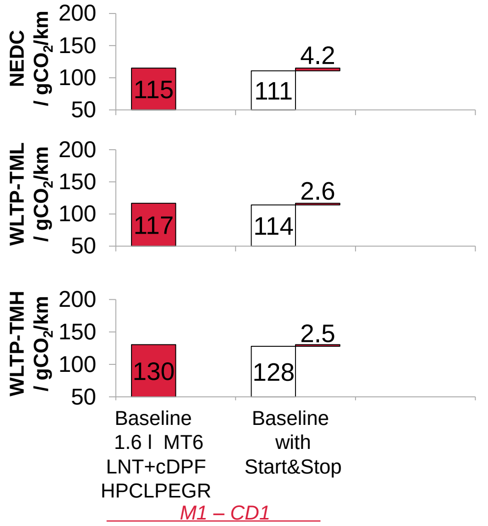
The base engine mounts an uncooled high pressure and cooled low pressure EGR system, which allows to maintain high EGR rates up to higher loads, while still keeping reasonable CO_2 emissions. This system is helpful for such a vehicle, which also mounts a LNT system as deNO$_x$ aftertreatment. Up to 250 °C, this system presents efficiencies similar or even higher than a SCR. However, at higher temperatures the efficiency decreases, forcing to reduce the Engine-out (EO) NO$_x$ levels in order to fulfil the legislation limits in the WLTP TM$_H$. Such systems equipped with a LNT, present the disadvantage in terms of CO_2, that they should run some regeneration or deNO$_x$ events in order to purge the LNT. However, the CO_2 values range from 115 g/km in NEDC up to 132 g/km in the WLTP TM$_H$. The increase is clear due to the higher energy requirements in the WLTP TM$_H$.

4.3.3.2 Impact of start-stop

The Figure 4.17 shows the impact of the start and stop technology on the NEDC and WLTP cycles. Engine stop at decelerating speeds lower than 3 km/h is implemented for this case. The start and stop (S&S) has a higher impact in the NEDC (~3.6 %) than in the WLTP TM$_H$

(~2 %). This benefit depends upon the idle fuel consumption calibration and the cycle itself. This engine presents a good fuel economy in idle which reduce the overall benefit coming from the S&S. At the same time, the NEDC presents longer stop phases during a shorter cycle than WLTP, which also brings up higher benefits in such cycles. While the impact of start stop is also observed in the FTP75 legislative cycle with a difference of around 3.5% over baseline version, the HWFET cycle presents no benefit, since the vehicle has barely any stop phase.

CO_2 emissions



Impact of Start&Stop

- Higher impact in the NEDC due to a higher Stop time and more stop phases
- 3.6 % CO_2 reduction in NEDC
- 2.2 % CO_2 reduction in WLTP - TM_L
- 1.9 % CO_2 reduction in WLTP - TM_H

	WLTC	NEDC
idle time / s	17	16
start stop active / s	190	206
cycle duration / s	1800	1180
start stop rate / %	10.5	17.5

Figure 4.17: Impact of start and stop technology.

The percentage benefits of the various segments for the NEDC and WLTP for the start and stop technology are also shown in the Figure 4.18 below. The segments E and SUV are equipped with an AT transmission which results in higher fuel consumption close to idle due to lower efficiencies in the torque convertor at low loads and low torques. The torque convertor stays open at such points. The other segments have manual transmission and have similar benefits. The segment LCV also shows lower benefits (percentage wise) since the main inefficiencies of such vehicle occur at high speed due to the unfavourable aerodynamics, reducing the relative benefit of the S&S, but not the absolute one.

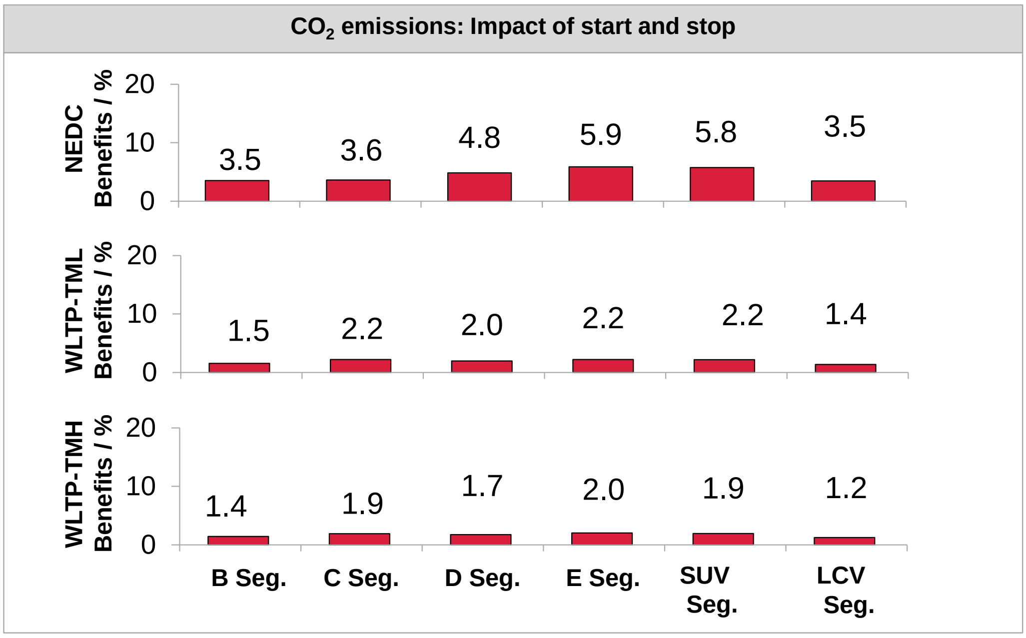


Figure 4.18 Segmentwise percentage benefits of Start and stop functionality.

4.3.3.3 Impact of engine downsizing and transmission upgrade

The downsizing of the engine is generally a favourable approach to enable the engine to operate at desired BSFC regions to reduce the fuel consumption. In addition, the transmission upgrade from manual to DCT allows to further optimize the shifting strategy at some cases. The downsizing is realised by means of increasing the specific power of the engine, but the reduction of the low-end torque might force a transmission redesign, e.g. adjustment of the final drive. In all cases, the final drive should be adjusted or even upgraded in order to keep a similar performance:

- The Segment B mounts a MT6 in the downsized version instead of a MT5 transmission in the baseline. One gear extra is advisable in order to keep the performance at the same level. At the same time, the final drive should be increased in order to keep the gradeability in the first gear, due to the lower low-end torque available.
- For Segment C, the MT6 is exchanged for a DCT7 transmission, which offers higher flexibility due to have one gear more, avoiding shorter gears.
- The Segments D and LCV mount now a DCT10 (wet clutch) instead of a MT6, allowing to have longer gears. Wet clutch is necessary for engine torque higher than 250 Nm and D-class vehicle hast approx.. 350 Nm.
- The Segments E and SUV mount a DCT10 (wet clutch) instead of an AT8, which offers similar performance but allows to have longer gears.

In case of segments E, SUV and LCV, the downsizing of the engine was realised through an additional turbocharger resulting in a two-stage boosting system, so as to ensure the downsized engine provided similar performance in power and torque requirement as provided by the baseline engines. The segmentwise changes considered for engine downsizing and transmission upgrade are summarized in the table below.

Segment	Baseline with start and stop	Downsized Version
B	1.4 L I3; 5 speed MT 1st-TC	1.2 l I3; 6 speed MT 1st-TC
C	1,6 L I4; 6 speed MT 1st-TC	1,4 L I3; 7 speed dry clutch DCT 1st-TC
D	2.0 L I4; 6 speed MT 1st-TC	1.6 L I4; 10 speed wet clutch DCT 1st-TC
E	3.0 L I6; 8 AT speed 1st-TC	2.0 L I4; 10 speed wet clutch DCT **2st-TC**
SUV	3.0 L I6; 8 AT speed 1st-TC	2.0 L I4; 10 speed wet clutch DCT **2st-TC**
LCV	2.2 L I4; 6 speed MT 1st-TC	1,8 L I4; 10 speed wet clutch DCT **2st-TC**

Table 4.4: Technology changes for engine and transmissions for defined technology step

For the engines with a torque demand greater than 350 Nm, a wet clutch based 10 speed DCT is advisable. Due to the presence of a wet clutch which requires further energy to move the oil through the system (“wet” system), a higher load demand from the engine is required especially at lower loads. This scenario may change in the long future by introducing variable pumps, which may result in higher efficiency at lower loads but this was not considered in the present simulation study. Alternatively, greater number of gears provides higher flexibility in implementing shifting strategies (either closer to best BSFC regions or finding best compromise between fuel consumption and NO_x emission in case of diesel engines),

In Diesel engines, unlike the gasoline ones, the impact of downsizing of engines is also linked to the increase of NO_x emissions, but first the operating points distribution depending on the engine size and transmission upgrade is discussed. Figure 4.19 shows the fuel share distribution for the Segment B vehicle in the NEDC and WLTP TM_H for both the base and downsized versions, when both mount a MT transmission (MT5 and MT6 respectively). On one hand, it can be clearly seen the non-optimized and relatively low load distribution of the points in the NEDC, while the much higher loads achieved during the WLTP cycle. Therefore, the required energy in the WLTP is higher but the average engine efficiency tends to be higher in this cycle than in the NEDC, especially for MT vehicles. In addition, even if not drastic (since the downsized step is only from 1.4 l to 1.2 l), the average specific engine loads increase for the downsized version.

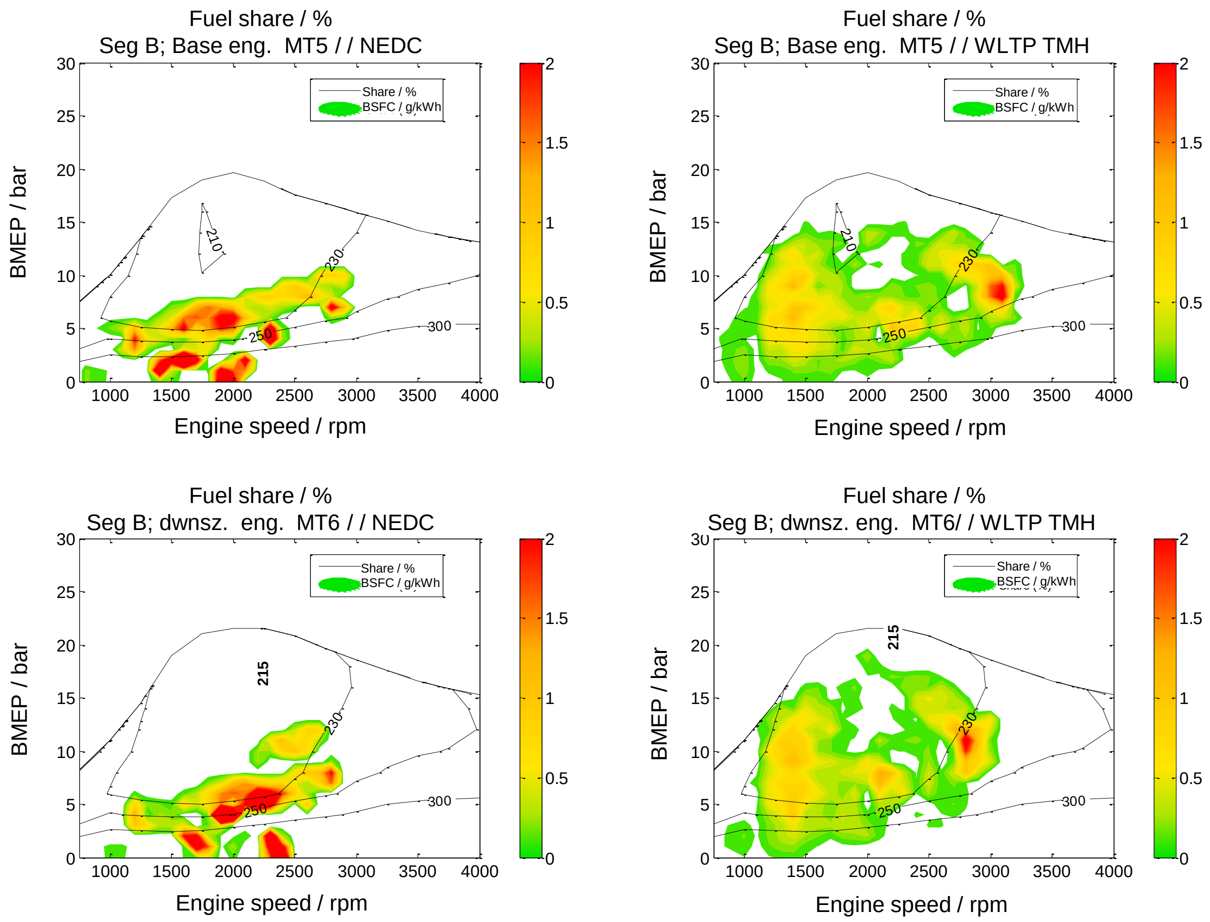


Figure 4.19 Fuel share distribution for NEDC and WLTP-TMH cycles for engine downsizing and transmission upgrade

Figure 4.20 show the same comparison but for the C class. In this case, the upgrade of the MT6 of the base vehicle for a DCT7 into the downsized version allows to cluster the operating points for both cycles around a more efficient area of the engine. However, the optimization of the shifting strategy must also consider the NO_x emissions. Since the original gear shift of the NEDC is fixed and the shifting is far from being optimized. There exist a much higher potential of CO_2 reduction in the NEDC cycle with respect to that of the WLTP, when installing DCT transmissions instead of MT ones. For the WLTP, the potential still exists but lower than in the case of the NEDC.

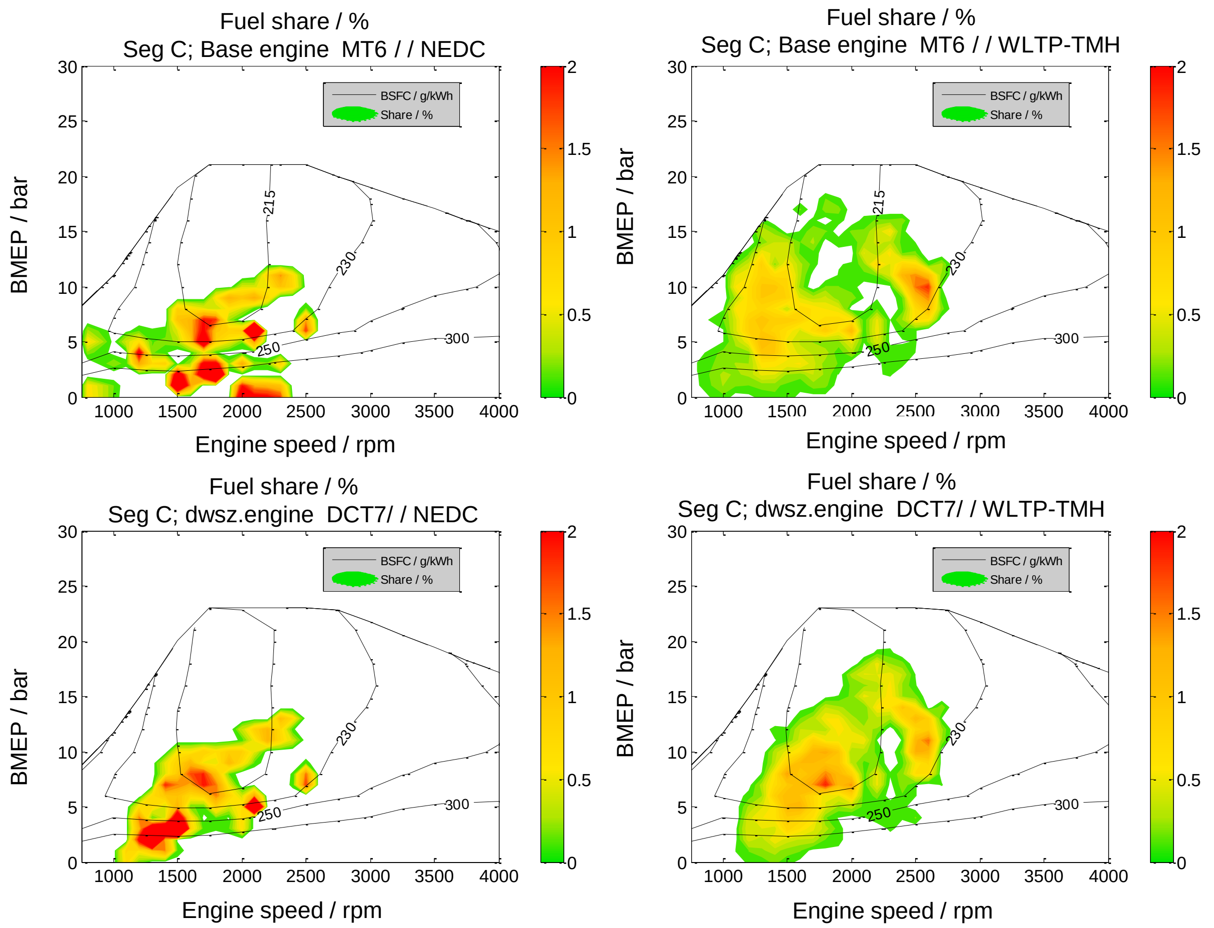


Figure 4.20 Fuel share distribution for NEDC and WLTP-TMH cycles for engine downsizing and transmission upgrade

Finally, Figure 4.21 compares now the NO_x share distribution of the baseline and downsized vehicles of the B and C segments in the WLTP TM_H. In the case of the B segment, the share contributions are shifted to regions of higher engine out NO_x emissions when comparing the downsized vehicle with respect to the base vehicle. This produces an increase of the final engine out NO_x emissions and finally also the tailpipe emissions. However, in the case of the downsized version of the C segment vehicle, the DCT7 transmission allows to also cluster the operating points to areas where the NO_x emissions are lower. This is clear when looking at the Figure 4.21.

Anyway, these plots also shows the higher power demand required for the WLTP cycle with respect to the NEDC cycle. Furthermore, the increasing cost of deNO_x operations (increased regeneration events in LNT and Adblue consumption for SCR catalyst based systems) should also be balanced against the benefits achieved from reduction in fuel consumption.

As explained and for the case example of the segment C, downsizing is a crucial step since this segment would represent a hypothetical limit for the implementation of an aftertreatment layout with one LNT as the only deNO_x system (focused in the EU market). The increase of the engine out NO_x requires the maximum possible performance from the LNT system to meet the future emission legislations for vehicles in this segment class.

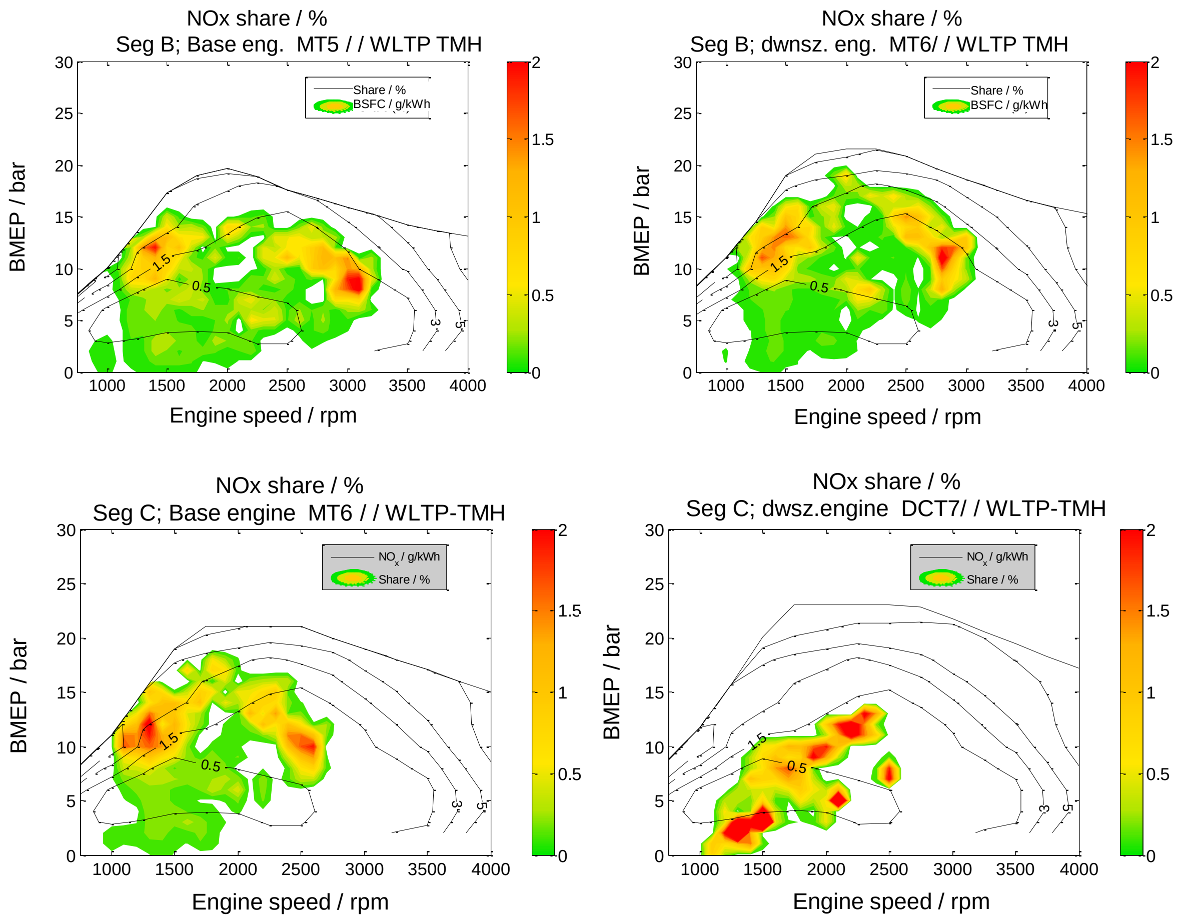
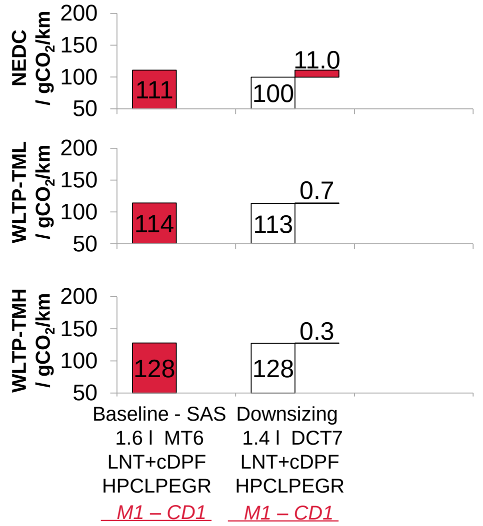


Figure 4.21 NO_x share distribution for WLTP-TMH for baseline and downsized engines. Top plot: Segment B. Bottom plot: Segment C.

The Figure 4.22 shows the reduction in CO_2 emissions in Segment C due to engine downsizing and transmission upgrade for NEDC and WLTP cycles. The benefit is about 10.5 g in the NEDC while in the WLTP the benefit is much lower, around 1.6 g/km for the WLTP TM_H. The NEDC results show the maximum benefits since the shifting strategy can be optimized with the DCT7 in contrast with the baseline where the transmission was manual and therefore the gear shift was fixed.

In the case of WLTP cycle, the guidelines of the UNECE (based on the Steven tool) are used in order to calculate the gear shifting in the case of the MT vehicle. Therefore, the baseline engine in this case is already operating in relatively efficient areas of the engine, getting an appropriate fuel consumption. However, the Figure 4.14 shows how both the baseline with S&S and the downsized variants of the Segment C gets no engineering margin into the WLTP in terms of emissions. The Baseline of the Segment C presents 78 mg/km in terms of TP NO_x emissions, while the downsized around 73 mg/km. Even being a poor margin for the downsized case, this reduction is achieved by optimizing the shifting strategy of the downsized engine, due to the DCT transmission installation, towards operating points where the NO_x is much lower. Hence the possible fuel economy is restricted due to that; moreover the higher fuel penalty coming from the LNT regeneration (see Figure 4.14) further reduces the possible fuel reduction with the new Segment version. In total, the possible benefit of the downsized version with an upgraded transmission is relatively low, only about 0.3 %-0.6% for the considered case and WLTP TM_H and TM_L respectively. But still in both cases the emissions engineering margin is insufficient and for future RDE legislation, i.e. the LNT system is not sufficient.

CO_2 emissions



Impact of Downsizing and Transmission change

- Higher impact in NEDC, due to improved flexibility in shifting strategy due to the 7-speed DCT.
- No change in vehicle coast down.
- Percentage benefits :
  - NEDC 9.9 %
  - WLTP-TML 0.6 %
  - WLTP-TMH 0.3 %

Figure 4.22 Impact of engine downsizing and transmission upgrade for Segment C

The Figure 4.23 shows the segment wise trend achieved by engine downsizing and transmission upgrades when comparing with the baseline version already mounting the start and stop. While segments C, E and SUV show a positive benefit, the segments B, D and LCV show an increase in CO_2 emissions at some points.

In case of segment B, the switch from 5 speed manual to 6 speed manual offers very little flexibility in moving the gear shifting patterns closer to the preferred shifting line in the NEDC. Due to this, the increased engine loads in the downsized engine result in a small CO_2 increase when comparing with baseline version with start and stop. In the WLTP, the downsized version however presents some benefits (2-3%) with respect to the baseline.

For the Segments D and LCV, the benefit is clear in the NEDC since the gear shifting can be optimized with the new transmission. In the WLTP cycles, the fuel consumption is slightly higher due to the required wet clutch. Nevertheless, such transmission is required for the installation of the P2 hybrid vehicle later and present clear benefits in terms of comfort and performance.

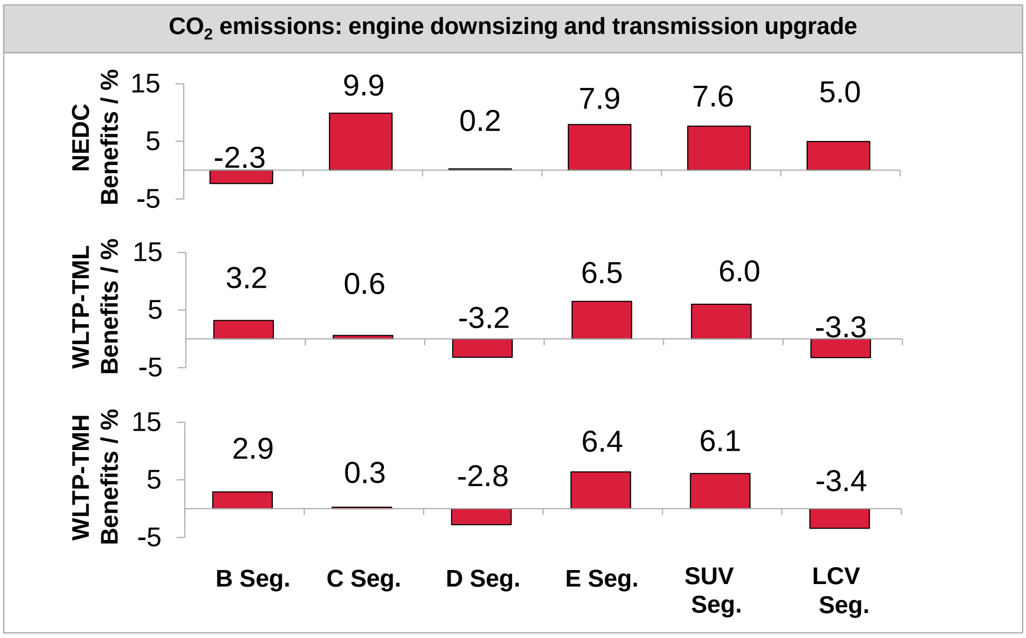
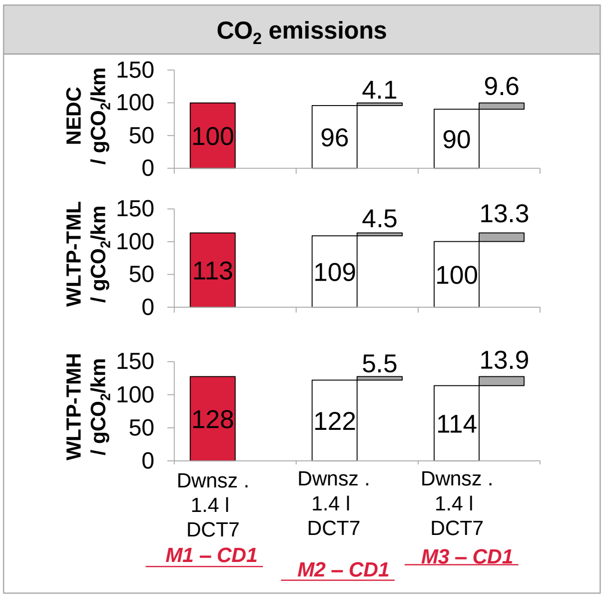


Figure 4.23 Segmentwise impact of engine downsizing and transmission upgrade.

4.3.3.4 Impact of mass reduction

The reduction of vehicle mass minimizes the load on engine resulting in benefits in fuel consumption. This impact was simulated in 2 stages, where in first stage 10 % reduction in vehicle mass was considered, by changing the vehicle mass (in running order) by 10 % as denoted by M2. In a further step, another 10 % of vehicle mass reduction (M3) was carried out to simulate conditions of a lighter vehicle mass with exactly the same downsized engine and upgraded transmissions. The engine and vehicle coast down (excluding mass) are maintained the same in order to compare the impact of the mass reduction. The final drive is adjusted and decreased in order to have longer gears, which gives some further benefit in terms of fuel consumption. For the manual transmission vehicles, the gear ratios are also modified in order to keep similar vehicle properties; while for the automatic transmission vehicles, the shifting strategy is also readjusted in order to improve the BSFC or even reduce the NO_x emissions, as it happens with the C segment downsized version.

As shown in Figure 4.24, the first mass reduction (M2) results in 3.7 % benefit for NEDC and 4.3 for WLTP-TM_H. The second mass reduction (M3) results in higher benefits for both cycles, due to the fact that the engine-out NO_x emissions are significantly being reduced. As shown in Figure 4.25, the operating points are moved slightly to regions of lower BSFC in the map for the 10 % reduction while in the case of 20% mass reduction, the shifting strategy is highly concentrated in this region of low BSFC resulting in higher benefits in WLTP-TM_H cycle. The Figure 4.26 also shows the engine-out NO_x share, showing how due to the lower energy requirements, the average specific NO_x can be increased but still having lower final EO NO_x: this makes possible this higher benefit step. This optimization gives an extra 2-3% fuel consumption benefit for the considered case.



Impact of Mass reduction

- M1:original vehicle mass with downsized engine and updated transmissions.
- M2→ 10% mass reduction
- M3→ 20 % mass reduction
- 10 % mass reduction
  - NEDC 4.1%
  - WLTP-TM_H 4.3 %
- 20 % mass reduction
  - NEDC 9.9 %
  - WLTP-TM_H 11.2 %

Figure 4.24 impact of mass reduction on CO_2 emissions for segment C.

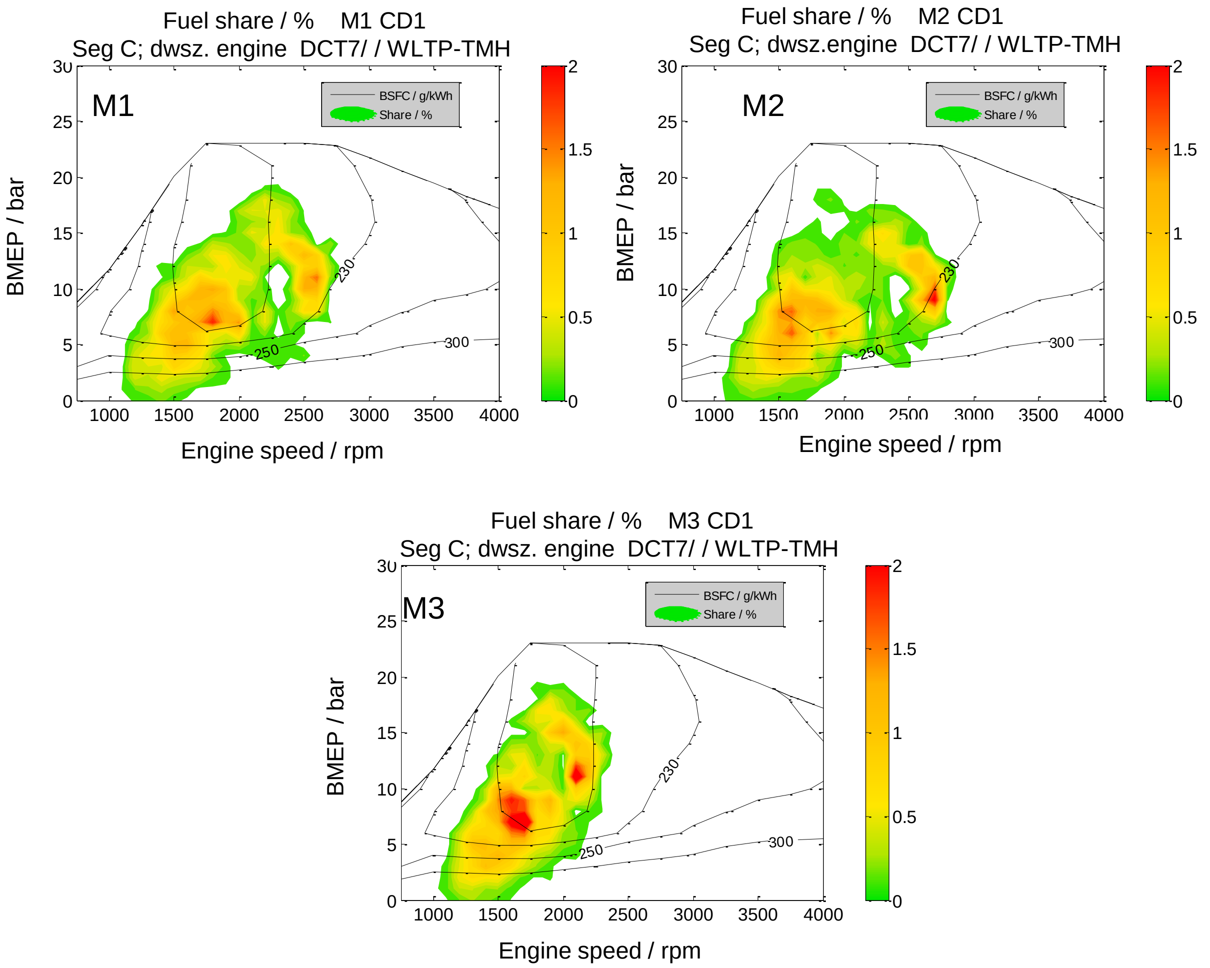


Figure 4.25 Variation in fuel share distribution due to mass reduction scenarios.

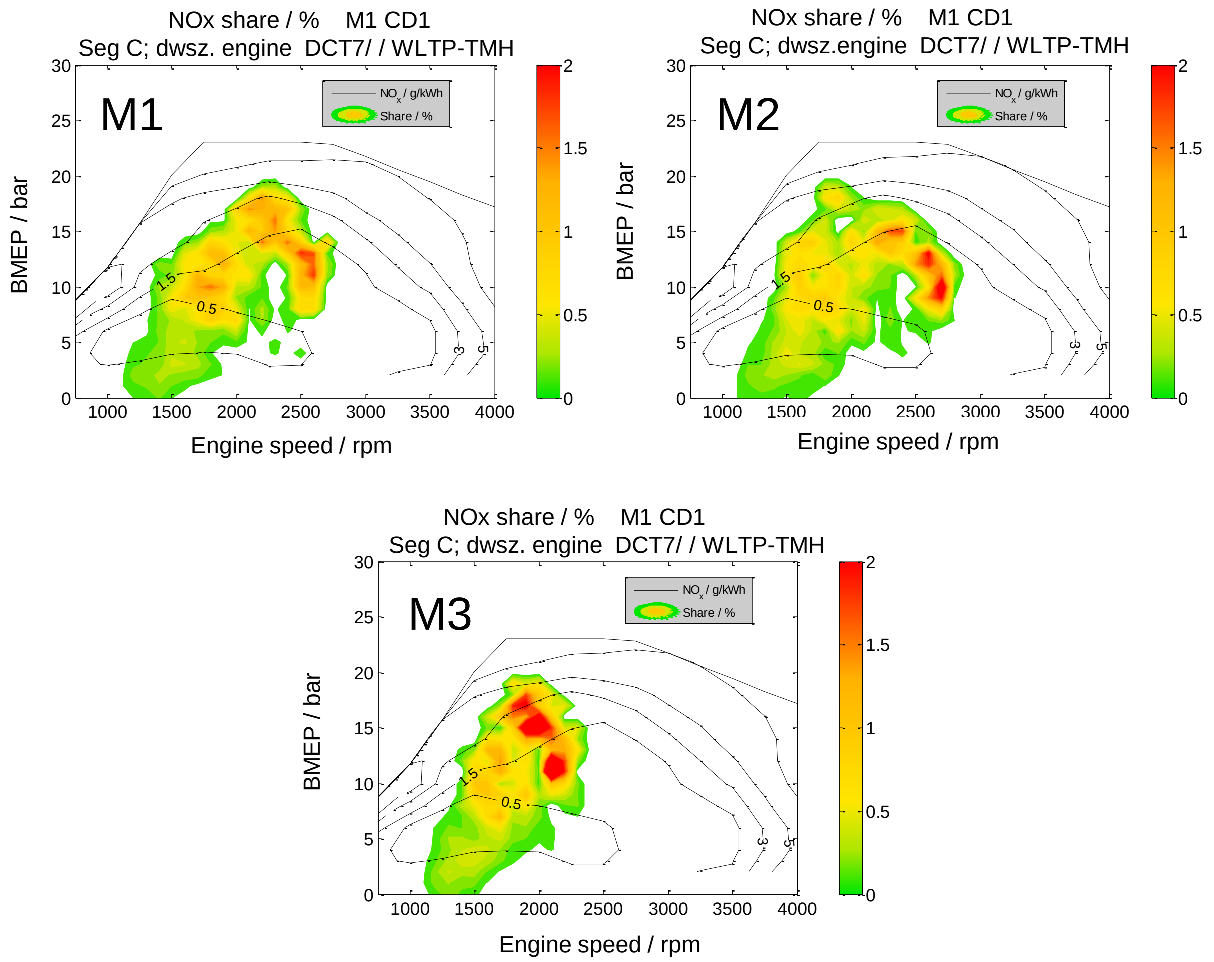


Figure 4.26 Variation in the EO NO_x share distribution due to mass reduction scenarios.

The Figure 4.27 shows the impact of the mass reduction within all the segments. Excepting the segment C, the first step of mass reduction achieves similar fuel consumption reduction than the second step. For the Segment B and NEDC, the benefits are much higher than in other segments, since the potential is higher for manual transmissions: the engine is operated at lower efficiency areas whereas the potential improvements are higher.

In all the other cases, all the systems mount already automatic transmissions. Therefore, the shifting strategy tends to be optimized closed to the BSFC line, allowing less improvement potential than in the segment B, which mounts a MT.

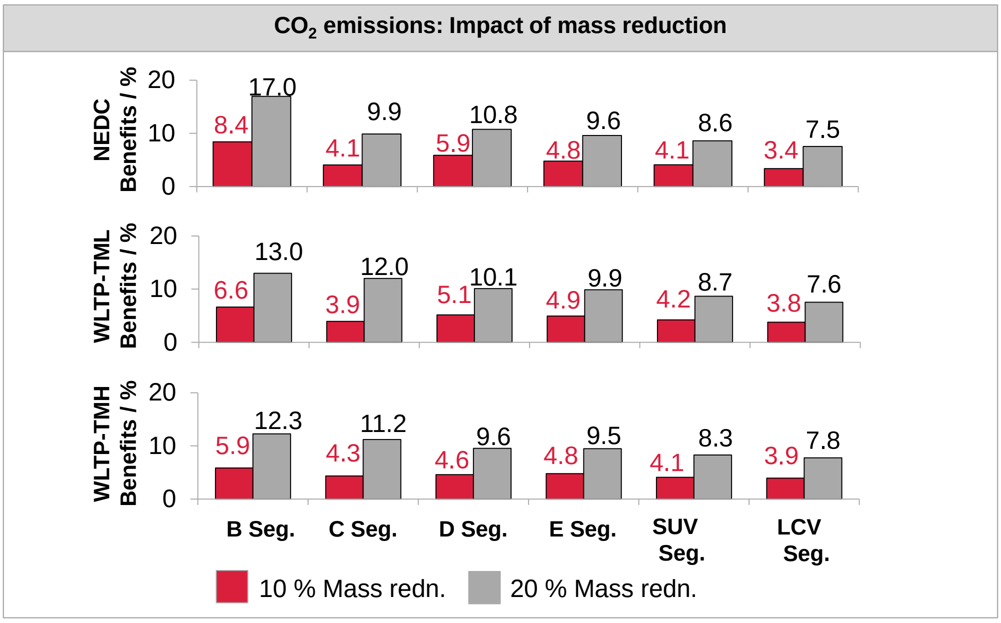
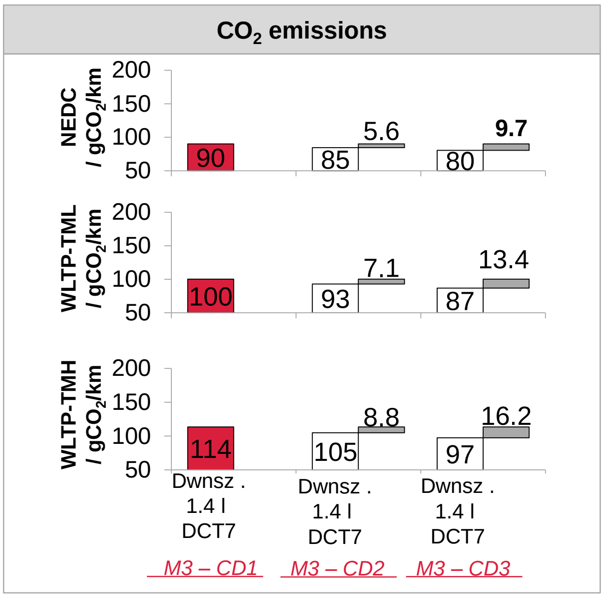


Figure 4.27 Segmentwise impact of the mass reduction.

4.3.3.5 Impact of coast down

Apart from the mass reduction scenarios, the impact of coast down reduction was also investigated through the simulations. The rolling resistance co-efficient of the tyre was reduced by 20% and 35% to derive CD2 and CD3 respectively. The downsized engine, upgraded transmissions and the 20 % reduction of vehicle mass were maintained for all cases in order to compare the impact of coast down reduction. The effects of the coast down reduction are similar to the ones of the mass, by overall reducing the energy requirements.



Impact of coast down

- All variants for this comparison include
  - 20 % weight reduction,
  - engine downsizing
  - 7 speed DCT.
- Resistances Red.: CD1→ CD2
  - NEDC 6.2 %
  - WLTP TM_H 7.7 %
- Resistances Red.: CD1→ CD3
  - NEDC 11.1 %
  - WLTP TM_H 14.8 %

Figure 4.28 Impact of coast down reduction scenarios for segment C.

As can be seen in Figure 4.28, the coast down reduction scenarios result in benefits of 6.2 % and 11.1 % for NEDC, and for CD2 and CD3 respectively. And 7.7% and 14.8% in the case of the WLTP –TM_H. Since the vehicle coast down plays a crucial role at higher vehicle speeds, the WLTP cycle shows higher potential for benefit from coast down reduction compared to the NEDC cycle.

As can be seen in the Figure 4.29, the fuel share distribution for the three coast down scenarios shows the reduction in engine load due to subsequent decrease in the vehicle coast down.

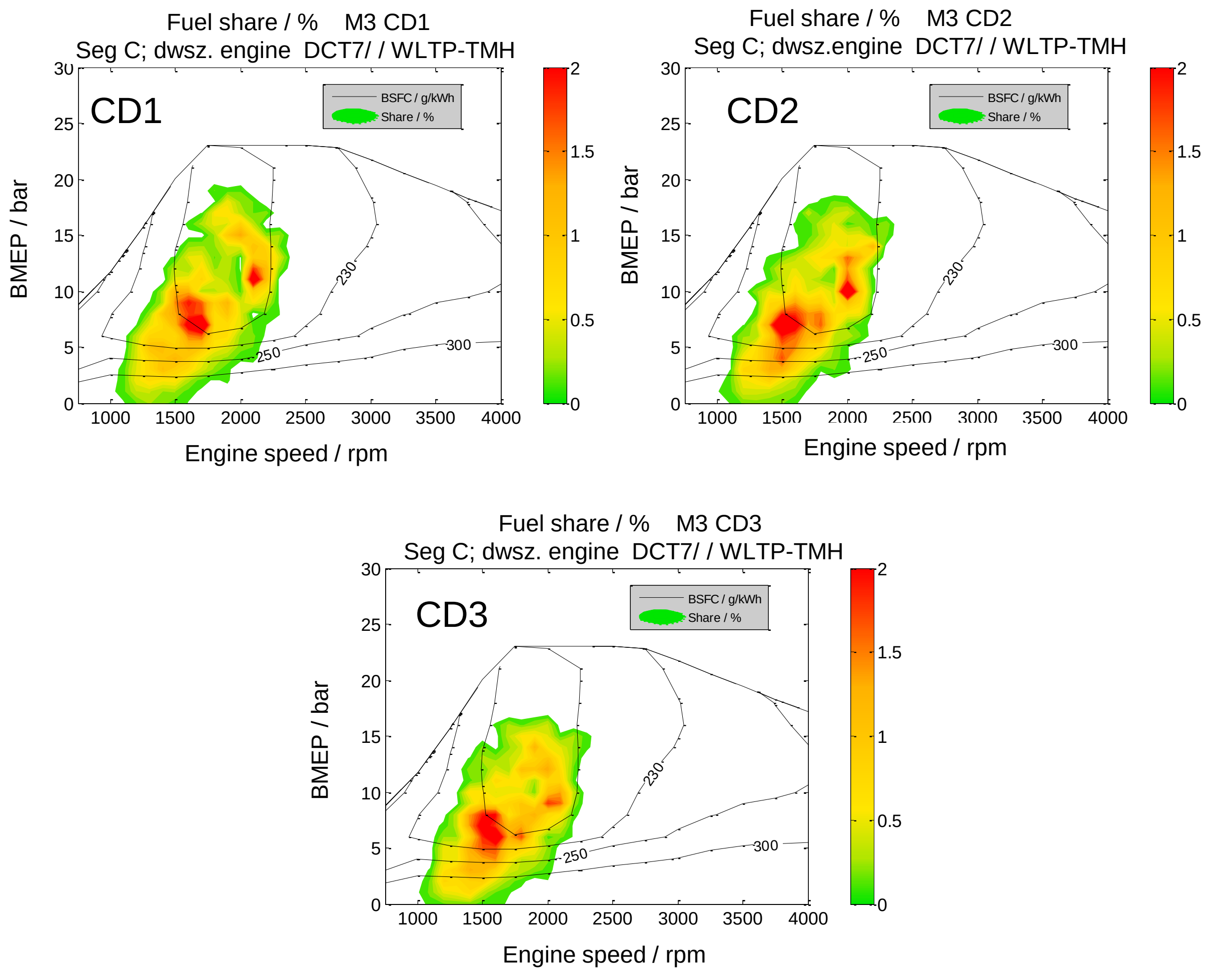


Figure 4.29 Fuel share distribution of the impact of the coast down for segment C in WLTP-TM_H.

The Figure 4.30 shows the relative trend for all the segments. In general, the first coast down reduction is giving slightly more relative benefits in terms of fuel consumption reduction than the second reduction, also supported by the fact that the first RR reduction is higher than the second one. Additionally, the average BSFC is lower when working at higher loads than at lower ones, but at the same time, the lower energy requirements reduce the final fuel consumption. Nevertheless, the main contributor to the final fuel consumption is driven by the aerodynamic coefficient, since its contribution to the final drag resistance is higher especially at higher vehicle speeds (see Figure 4.1), when the fuel consumption is also higher. Therefore, achieving significant aerodynamic reductions might lead to higher CO_2 reduction potential.

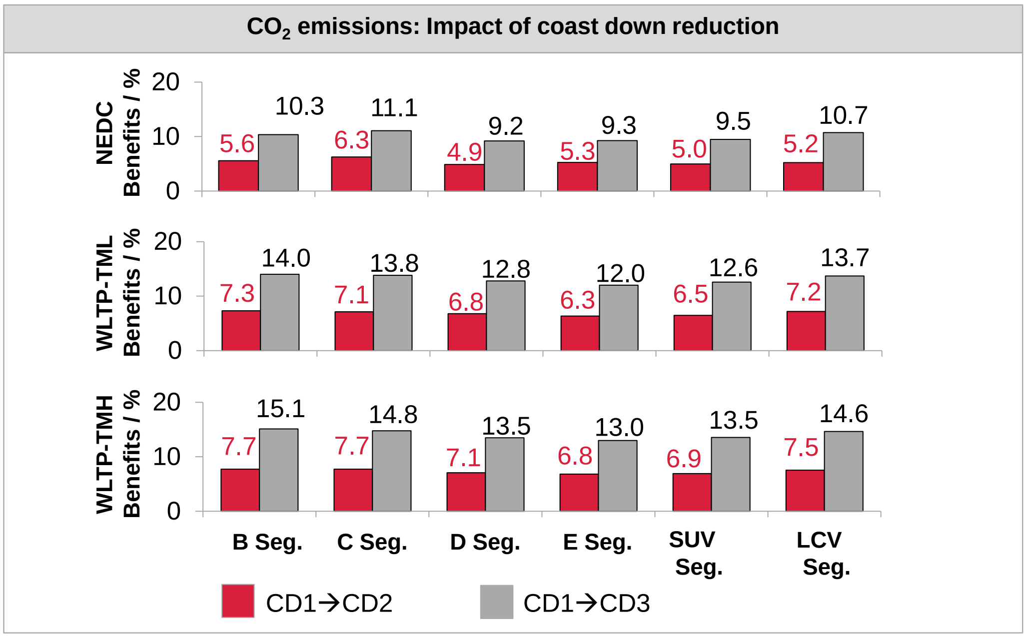


Figure 4.30 Segmentwise impact of the coast down reduction.

4.3.3.6 Impact of friction reduction

The base vehicle considered for comparison in this scenario was the vehicle with:

- Downsized engine
- Upgraded transmission
- 20 % mass reduction (M3)
- Coast down scenario 2 (CD2)

The final effective friction reduction was achieved through various measures which comprise general adjustments in the camshaft bearings, main bearing and piston off-set. In total, and based on the FEV benchmark experience, an average friction reduction of 20% might be achieved in the future with such measures.

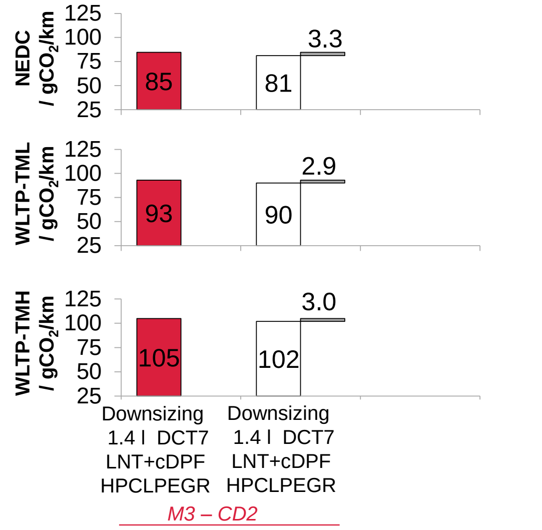
Moreover, an electrical water pump is installed into the system. The pumping requirements can therefore be adapted to the system needs: at lower loads and when the engine is still cold, the pumping requirements are lower, thus reducing the overall load coming from the auxiliaries. In combination with this, the split cooling allows to separate the coolant circuit into two: one for the cylinder heads and EGR coolers; and one for the engine block. The effective pumping requirements can be adjusted with the support of the electrical water pump: mainly no pumping needed for the engine block when the system is cold, and adapted pumping for the other circuit (with a lower effective coolant mass). In total, the following benefits are achieved:

- The engine block and thus the oil circuit is heated up faster, reducing the overall friction during the cold phase.
- The required energy coming from the electrical water pump can be optimized depending on the requirements. It can even be disconnected when the system is cold, and then switched on when needed. This produces also an overall reduction of the friction, since for a mechanical water pump, this cannot be disconnected.

- The split cooling and the smart pumping control accelerates in general the warm-up, reaching sooner an appropriate temperature range for the coolant and the oil circuits.

As can be seen in Figure 4.31, the final benefit is 3.9 % in NEDC and around 2.8% benefit in WLTP-TM$_H$ cycle. The benefits are remarkable during the warm-up phase of the engine, and therefore higher in the NEDC than in WLTP, which presents a shorter warm-up phase within a longer cycle.

CO_2 emissions



Impact of Friction reduction

- Friction reduction measures include camshaft bearings, main bearing , piston off-set adjustments.
- Electrical water pump and split cooling technologies are also included.
- Percentage Reductions
  - NEDC 3.9 %
  - WLTP-TML 3.1 %
  - WLTP-TMH 2.8 %

Figure 4.31 Impact of the optimized friction for segment C.

The Figure 4.32 compares the traces of the systems in Figure 4.31 for the WLTP TM$_H$ cycle. The top plot shows the average specific CO_2 calculated time by time (g/km). The differences between the two systems are higher at the beginning of the cycle when the cold phase is still running. The medium plot compares the coolant temperature and the average friction mean effective pressure (FMEP) for the two systems: for the optimized powertrain, the coolant temperature rises faster and the FMEP is lower during all cycle. The bottom plot finally shows the speed profile of the cycle.

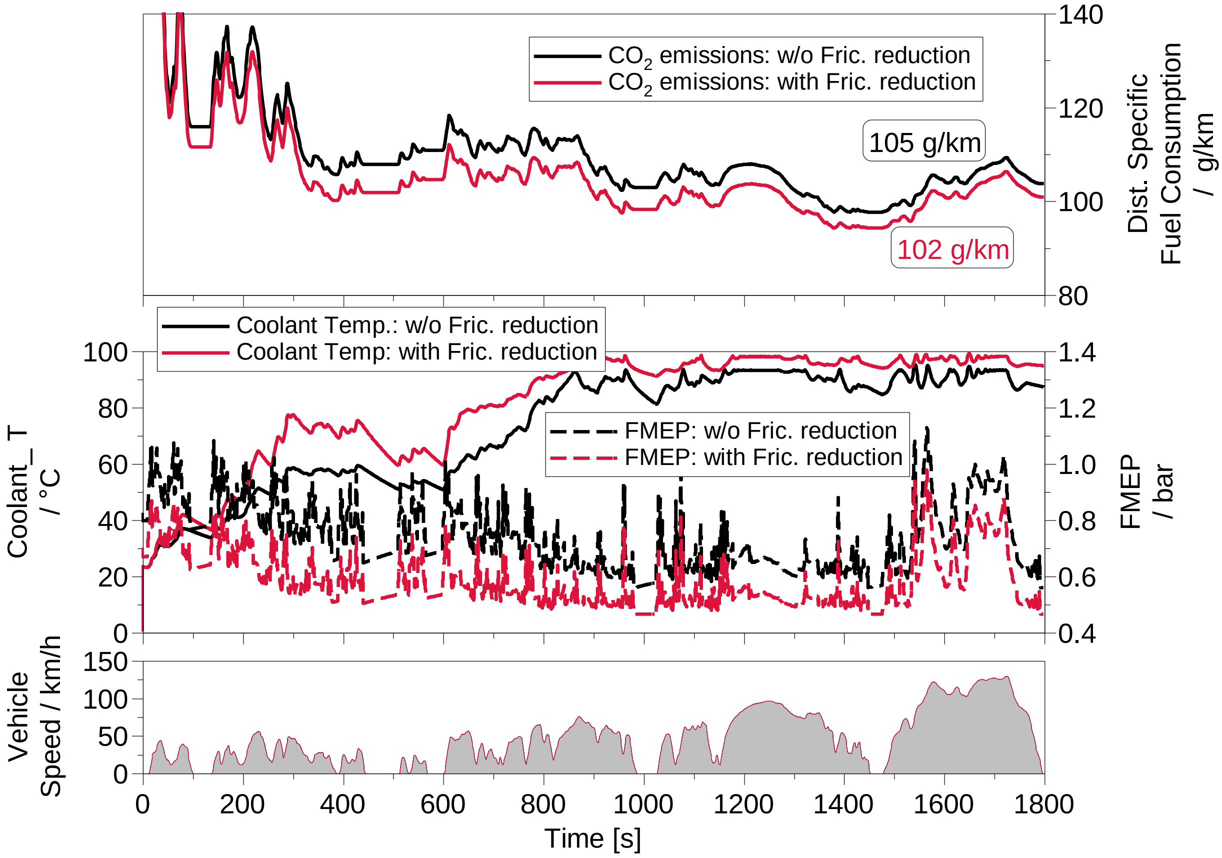


Figure 4.32 Fuel consumption, coolant temperature and FMEP with an engine with and without optimized friction.

The segmentwise trend is shown in the Figure 4.33. The potential is usually higher for the engines working with lower specific power and for the NEDC cycle when comparing with the WLTP. In the case of the Segment B, the potential for the NEDC is considerably higher since the gear shift is not optimized and thus, the system works at areas with lower efficiency, reaching higher improvement potential. All the other segments already mount an automatic transmission.

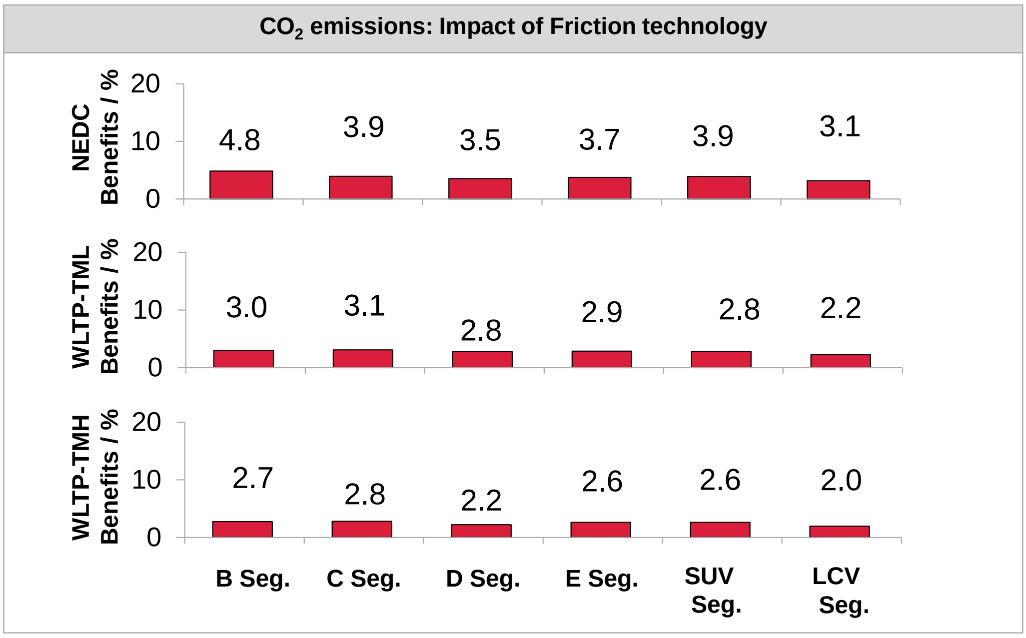


Figure 4.33 Segmentwise trend for the optimized friction.

4.3.3.7 Impact of engine technology upgrade

The base vehicle considered for comparison in this scenario was the vehicle with

- Downsized engine
- Upgraded transmission
- 20 % mass reduction (M3)
- Coast down scenario 2 (CD2)
- Optimized friction (base engine measures, split cooling and electrical water pump)

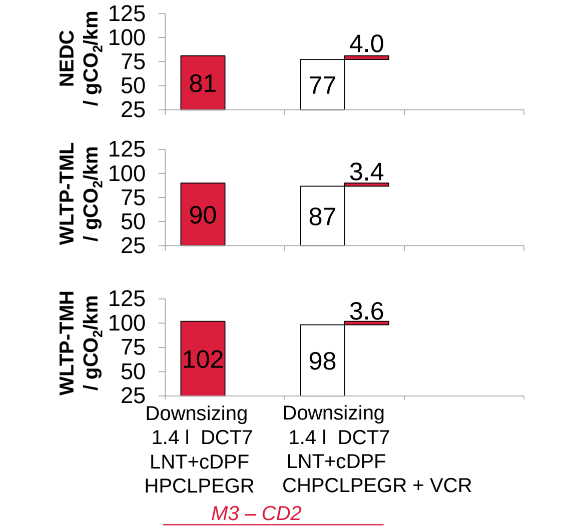
For segment C, the engine upgrade includes cooled high pressure EGR and a cooled low pressure EGR (CHPCLPEGR) from a base system which already mounts uncooled high pressure and cooled low pressure EGR (HPCLPEGR) systems. In addition, a 2-stage variable compression ratio (VCR) is installed.

The first measure by adding one coolant system to the high pressure EGR loop allows to increase the HPEGR rates at lower loads and/or still getting some fuel consumption benefit, since the temperature of the high pressure loop is lowered and the density increases. Therefore, the system can increase the HPEGR rates, especially at lower loads, with similar or even slightly higher turbocharger efficiency and volumetric efficiency of the system. The quantified fuel consumption benefit lies in the range of 1.5 % in the NEDC (maximum) and slightly lower for the WLTP (~0.5-1%).

The 2-stage VCR enables the engine to modify the compression ratio depending on the conditions. By having a higher compression ratio (in this case, 17.5) at lower loads, better fuel consumption benefits can be realised coming from a more efficient combustion. However, at higher loads, reducing the compression ratio (14.5) permits to reduce the overall friction and the NO_x emissions. In total, around 3-4% can be obtained by means of the VCR, also slightly higher in the NEDC.

The Figure 4.34 shows the total benefit, 5.4 % in the NEDC and 3.3-3.5 % in the WLTP. The benefit is therefore slightly higher in the NEDC.

CO_2 emissions



Impact of Engine Technology

- Cooled high pressure EGR and variable compression ratio technologies are evaluated
- Compression Ratio high→ 17.5
- Compression Ratio low → 14.5
- Percentage benefit
  - NEDC 4.9 %
  - WLTP- TML 3.8 %
  - WLTP-TMH 3.6 %

Figure 4.34 Impact of engine technology upgrade for segment C

Also, the impact of these engine technology upgrades can be clearly noticed in the comparison of the BSFC maps, shown in Figure 4.35. The top plots show the fuel share distribution in the NEDC (left) and WLTP TM_H (right) for the base powertrain including the optimized friction, while the bottom plots show the fuel share for the upgraded version.

The fuel share over the upgraded version lies into points with better BSFC (the map only considers the high compression ratio map of the VCR) due to the VCR and the CHPCLPEGR. The load points and thus the fuel share is similar for both powertrains (same gear ratio and similar shifting strategy), where the improvements are coming from the improved BSFC as reflected in the revised engine maps.

Segment	Add-on technologies
B	CHPCLPEGR (Base is HPCLPEGR)
C	CHPCLPEGR + 2-stage VCR
D	VVT
E	VVT
SUV	VVT
LCV	VVT

Table 4.5 Considered technologies for the engine upgrade.

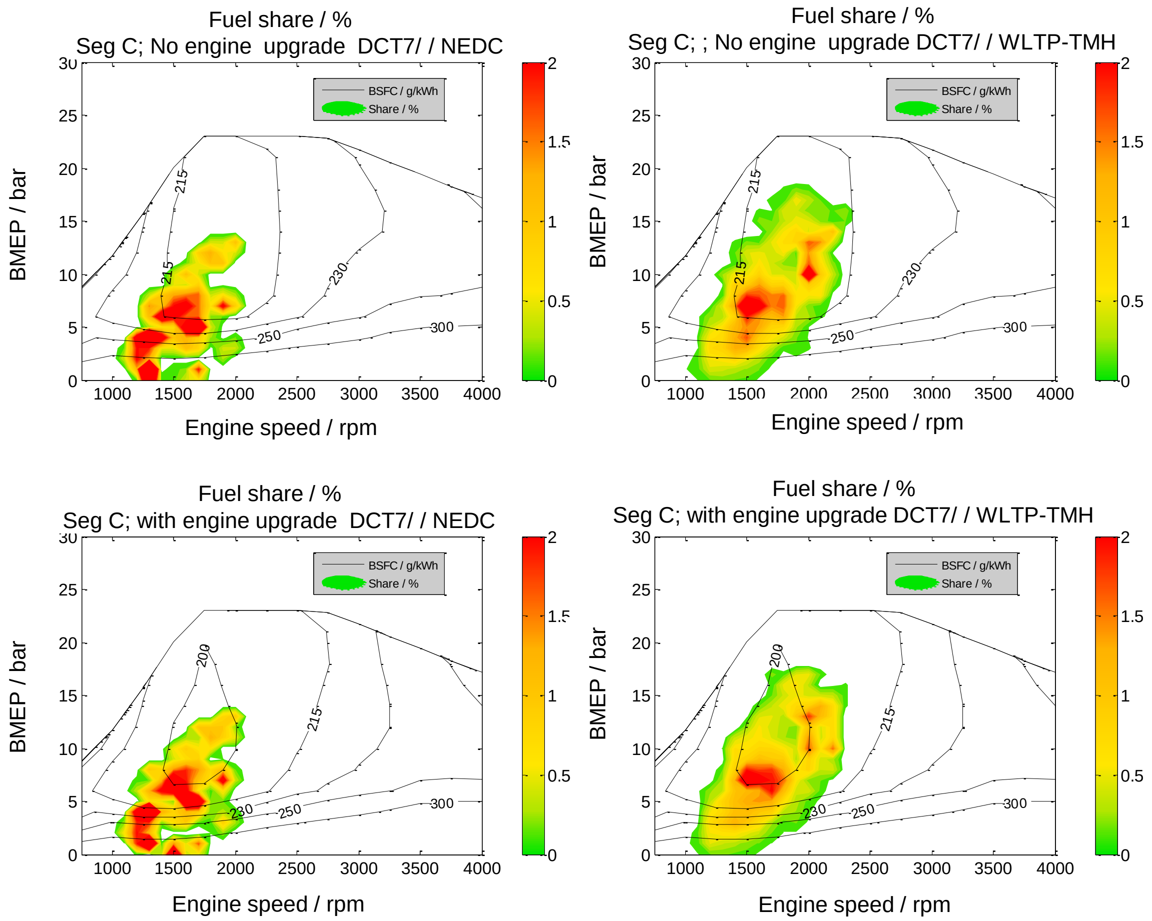


Figure 4.35 Fuel share plot of engines with and without engine technological upgrades

The comparison of this step for the different segments is not really meaningful, since the upgrade is different depending on the considered vehicle. The Table 4.5 shows the considered technologies, while Figure 4.36 shows the segmentwise trend for such upgrades.

The Segment B mounts a coolant system to the HPEGR loop, upgrading from uncooled HPCLPEGR to a CHPCLPEGR, getting a benefit in the range of 0.3-0.8 % of fuel consumption in the cycles. Some deviations are explained due to the regeneration events of the LNT on such segment. The C-Segment includes a higher level of system upgrading, while the Segments D, E, SUV and LCV include only a VVT system. The benefits for such system are around 0.2-0.7 % due to have a lower fuel penalty for heating, more noticeable in the NEDC than in WLTP.

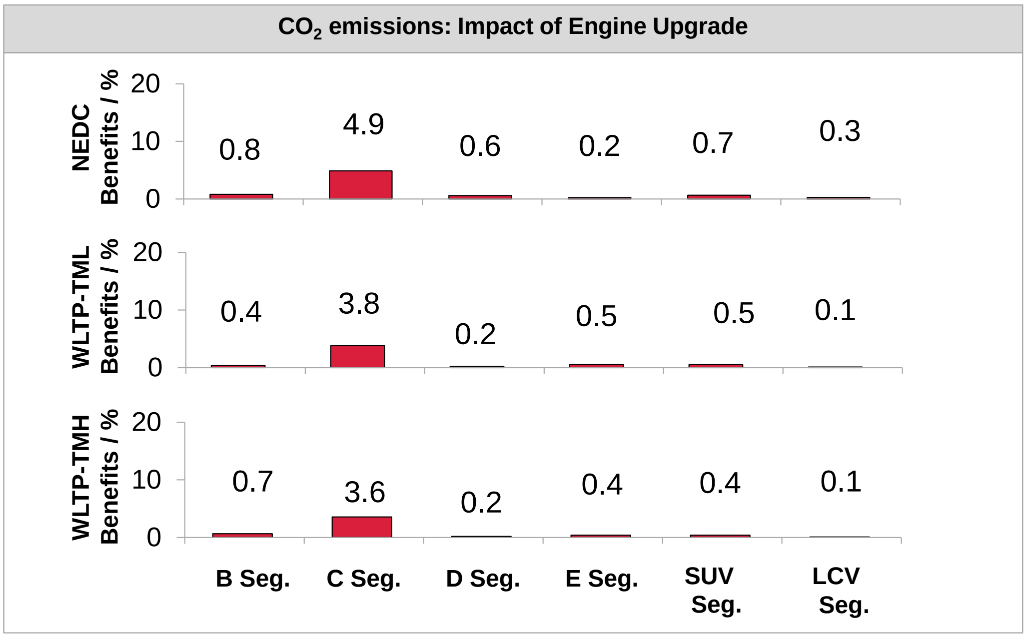


Figure 4.36 Segmentwise trend for the engine upgrade.

4.3.3.8 Impact of upgrading the Aftertreatment technology

For the segments B and C, the aftertreatment base technology considers one LNT+cDPF system, which is now upgraded into to a DOC + close-coupled SDPF system, by also including an AdBlue dosing system. In addition, a VVT system is installed for improving the heating strategy. The aftertreatment systems considered for this study are described in the Figure 4.37: left is the base engine, while right shows the upgraded aftertreatment.

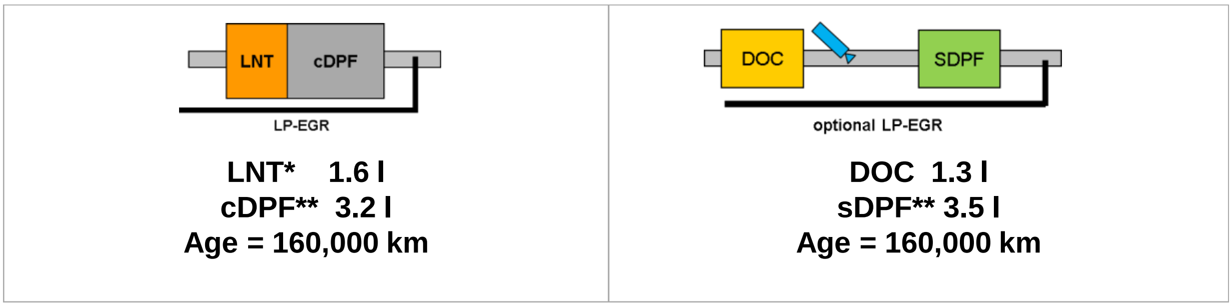
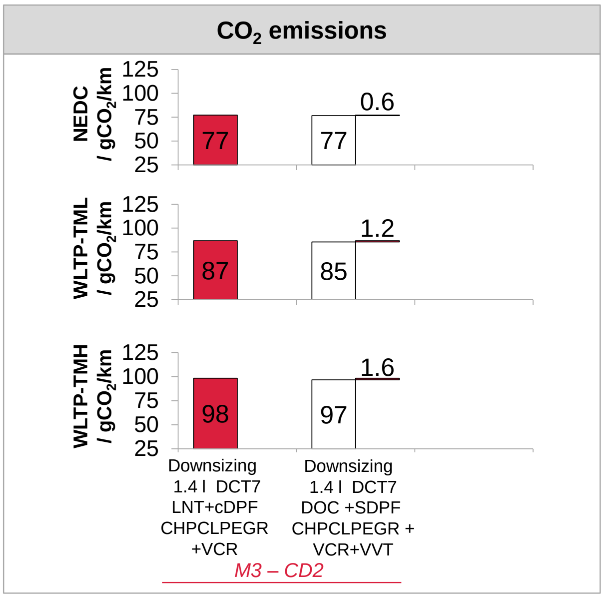


Figure 4.37 Aftertreatment systems for segment B and C.

Furthermore, the improved deNO$_x$ efficiency, especially in the WLTP cycles, enables to further optimize the shifting strategy for the DCT and the engine-out NO$_x$ to better BSFC regions. The fuel penalty coming from the LNT is now saved, while the VVT involves some fuel penalty in order to support the system heating. The total benefits in the range of 0.8-1.3 % as shown in Figure 4.38, slightly higher in WLTP TM$_H$, due to the LNT regenerations are no longer needed. Nevertheless, the NO$_x$ emissions can also be minimized, possibly being a more suitable solution for future RDE conditions. The AdBlue consumption might be addressed for the cycles, however, this is far from the expected demand into real driving conditions.



Impact of Emission Reduction

- VVT used for heating catalysts for better deNOx efficiency compared to LNT based system.
- Percentage reductions
  - NEDC 0.8 %
  - WLTP-TML 1.4 %
  - WLTP-TMH 1.6 %

Figure 4.38 Impact of change of aftertreatment system.

Finally, the Figure 4.39 shows the comparison of the potential benefit for the Segments B and C. The potential in the Segment B is higher since this system is mounting a MT. Therefore, the load points are usually running in lower efficiency areas of the map, leaving more space for optimization. In the case of the Segment C, the shifting strategy is already optimized from the base by means of the DCT transmission, leaving less space for improvement. Anyway, the factors range from 0.8 % up to 1.9 %.

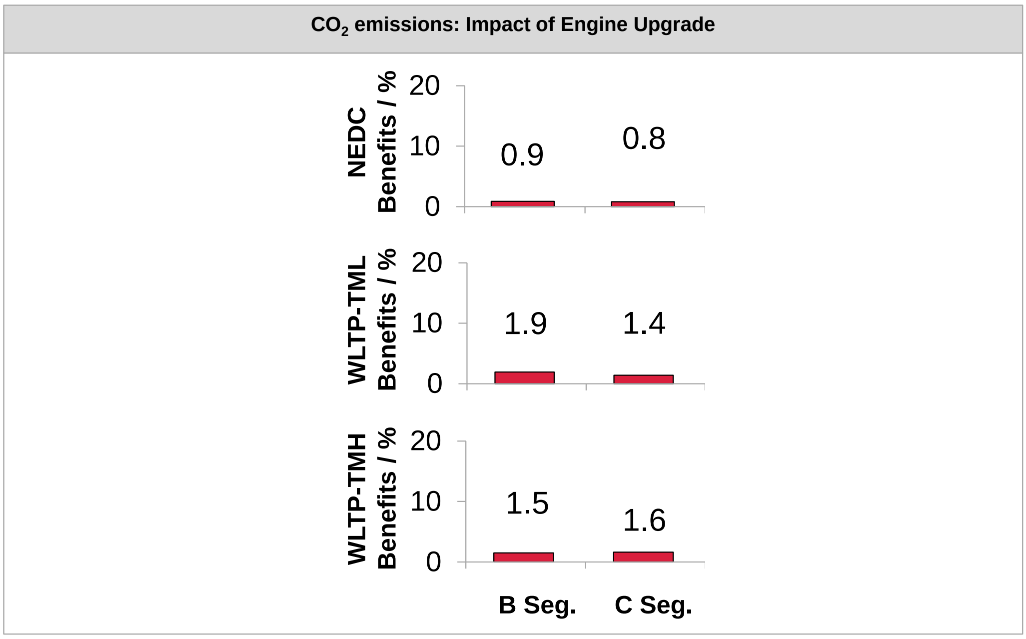


Figure 4.39 Segmentwise trend for the aftertreatment upgrade.

4.3.3.9 High spec engine

For the next step of the comparison, a high-spec engine is presented by combining the main engine technologies shown in the previous sections. Therefore, the Table 4.6 shows the base powertrain and the add-on technologies considered for this step.

Segment	Baseline with start and stop	Add-on technologies
B	1.4 L I3; MT5 M3 CD2 FR LNT+cDPF HPCLPEGR	VVT, DOC+SDPF (instead of LNT+cDPF), CHPCLPEGR
C	1.6 L I4; DCT7 M3 CD2 FR LNT+cDPF HPCLPEGR	VCR, VVT, DOC+SDPF (instead of LNT+cDPF), CHPCLPEGR
D	2.0 L I4; DCT10 1st-TC FR DOC+SDPF CHPEGR	VCR, VVT CHPCLPEGR
E	3.0 L I6; DCT10 2st-TC FR DOC+SDPF CHPEGR	VCR, VVT CHPCLPEGR
SUV	3.0 L I6; DCT10 2st-TC FR DOC+SDPF CHPEGR	VCR, VVT CHPCLPEGR
LCV	2.2 L I4; DCT10 2st-TC FR DOC+SDPF CHPEGR	VCR, VVT CHPCLPEGR

Table 4.6: Technology changes for engine and transmissions for defined technology step. FR = Friction reduction.

The Figure 4.40 shows the impact in the segment C vehicle, which considers the optimized friction reduction, the VCR, the VVT and the aftertreatment upgrade. In total, the benefit is coming mainly due to the VCR, giving a possible benefit of 6.2 % in the NEDC, and slightly lower in the WLTP cycles (~4.6%).

The

Figure 4.41 also compares the traces of the base engine for this case and the high-spec version in the WLTP TM_H cycle. The top plot shows the CO_2 comparison, the medium plot the engine-out and tailpipe NO_x while the bottoms plot the vehicle speed. The following can be observed:

- The VCR and CHPCLPEGR technologies offer a higher reduction potential of the fuel consumption at lower loads.
- The installation of an active SDPF system instead of a LNT allows to increase the engine-out NO_x emissions by getting some space for improvement of the fuel

consumption. At the same time, the higher deNO_x efficiency of the SDPF system in the highway part where the exhaust temperatures are higher is evident, reaching lower tailpipe NO_x emissions.

- By using a SDPF system, the LNT regeneration modes can be avoided.

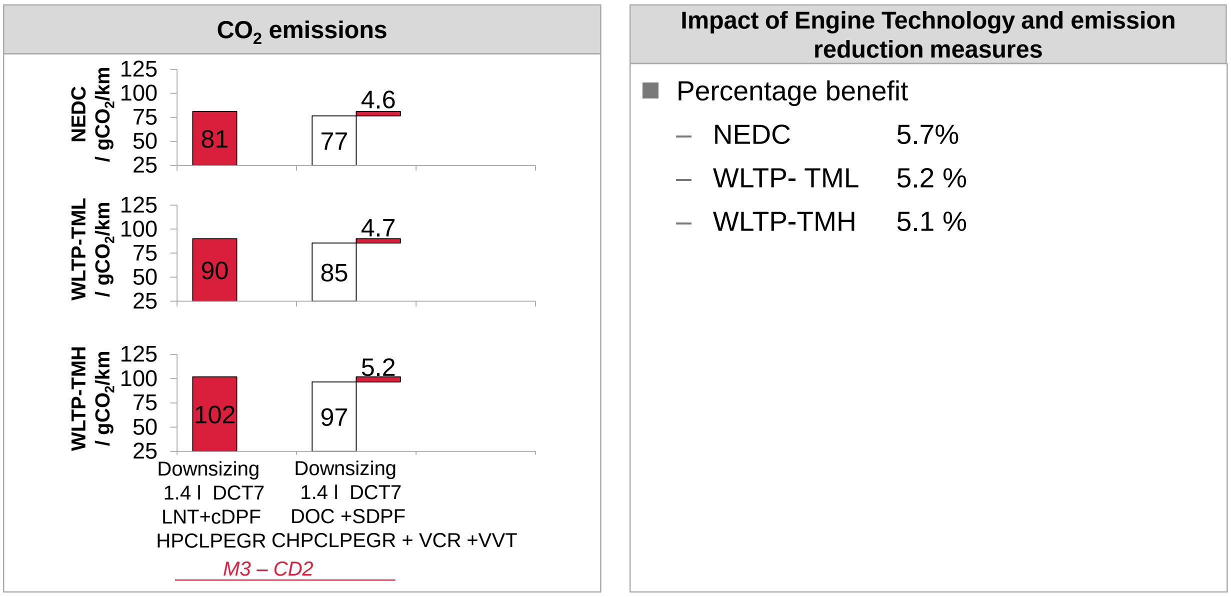


Figure 4.40 Impact of engine technology and emission reduction technology

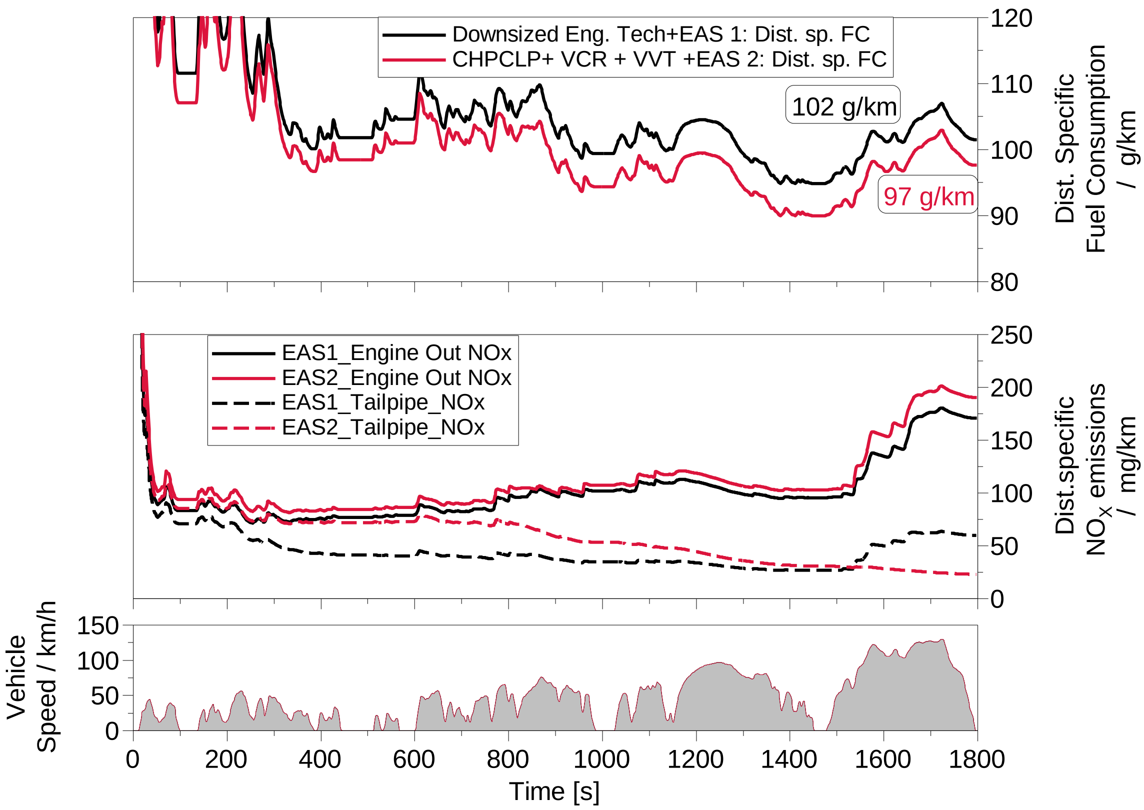


Figure 4.41 Impact of engine technology and DOC+SDPF aftertreatment system

The comparison of the high-spec CO_2 reduction potentials within all segments is shown in the Figure 4.42. The Segment B potential is lower since the VCR upgrade was not considered in such a small vehicle. At the same time, the considered technologies usually get lower potential when the engine has higher specific power. Therefore, the potential of the technologies over the segment D, E, SUV and LCV is becoming lower than in the case of the Segment C. In the following plot, the benefit is shown as higher for the heavier segments, since they are starting from a lower equipped base engine (only HPEGR) and they are upgrading to a cooled HP and cooled LP EGR engine.

Anyway, the benefits tend to be higher for the NEDC, in the range of 6-7 % and lower for the WLTP (~3-5%). The LCV vehicle shows considerable lower benefit potential in the WLTP cycles since the loads are increasing significantly due to the penalized aerodynamics. In such a case, the VCR (the system must work more time at the lowest compression ratio) and CHPCLPEGR gets much lower potential when working at higher loads.

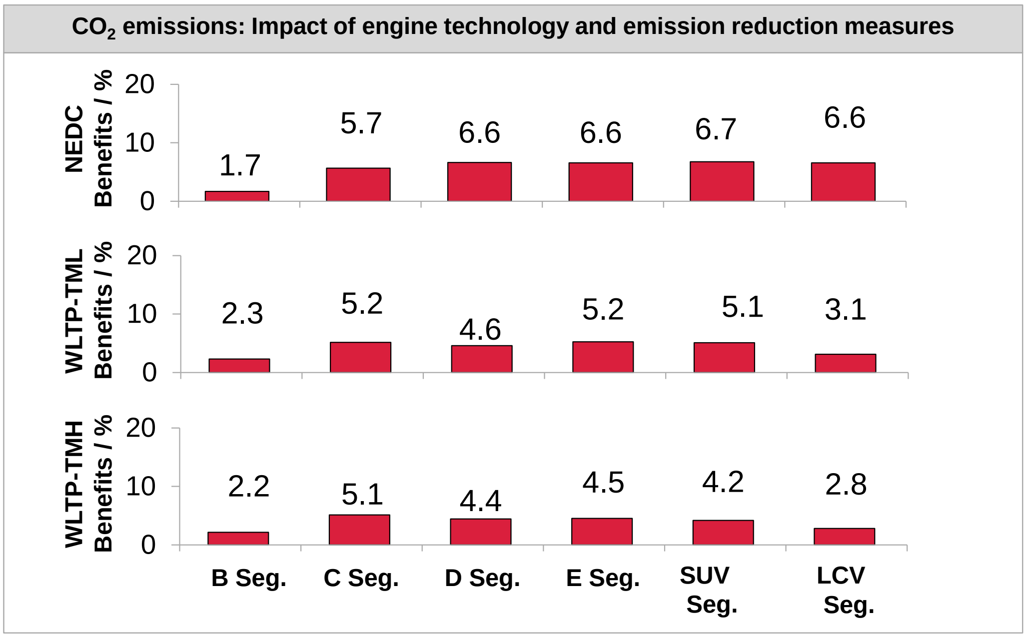


Figure 4.42 Segmentwise trend for engine technology and emission reduction benefits.

4.3.3.10 High spec engine with the original vehicle mass and coast down reduction

The next step of the comparison includes the high-spec engine analysed in the previous chapter, but with the original mass (M1) and only one step of coast down reduction (CD2). For this comparison, the comparison is set out with the following 3 variants:

- Downsized version with upgraded transmission and no step on mass and coast down reduction (no engine upgrade).
- High spec engine with the M3 CD2.
- High spec engine with the M1 CD2.

The objective is looking at different scenarios regarding the fuel consumption reduction potential. The Table 4.7summarizes the considered variants.

Version	Variant 1	Variant 2	Variant 3
Engine	Downsized engine with S&S		
Transmission	Upgraded transmissions		
Mass	Downsized vehicle mass (M1)	20 %reduced mass (M3)	Downsized vehicle mass (M1)
Coast down	Original coast down (CD1)	Reduced coast down (CD2)	Reduced coast down (CD2)
Add-on Technologies	None	Friction, engine, emission reduction	Friction, engine, emission reduction

Table 4.7 Comparison with original vehicle mass

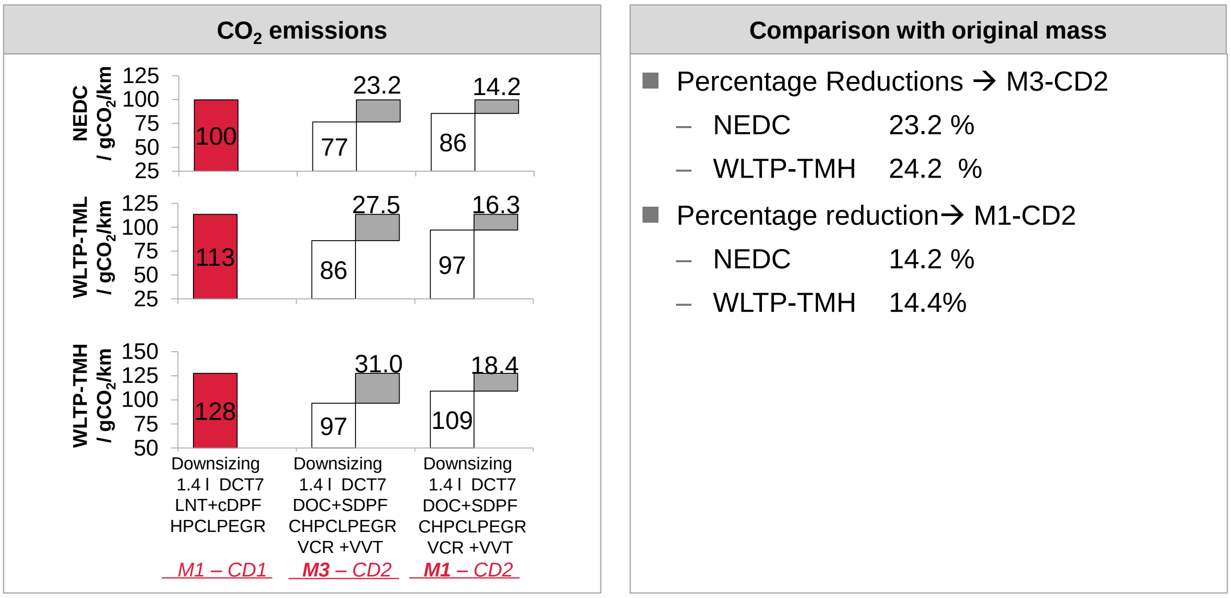


Figure 4.43 Comparison of original vehicle mass with all technologies (except hybrid)

The Figure 4.43 compares the results of the three variants for the Segment C. The total benefit of the variant 2 is around 23.2 % for the NEDC and 24.2 % for the WLTP TM_{H}. When considering the Variant 3, the benefits are 14.2 for the NEDC and 14.4 % for the WLTP TM_{H}.

The Figure 4.44 shows the segmentwise trend for the high spec engine with mass and coast down reduction – M3 CD2- (Variant 1→ Variant 2) and high spec engine with coast down reduction only –M1 CD2-(Variant 1→ Variant 3), where the Variant one is M1 CD1. The improvement potential is generally slightly lower for the heavier vehicles with higher specific power. In all cases, excepting the Segment C, the total improvement potential is higher in the NEDC with respect to the WLTP. This difference is due to the downsized version of the Segment C presented a shifting strategy (DCT7) based on getting lower engine out NO_x (this variant mounts a LNT as $deNO_x$).

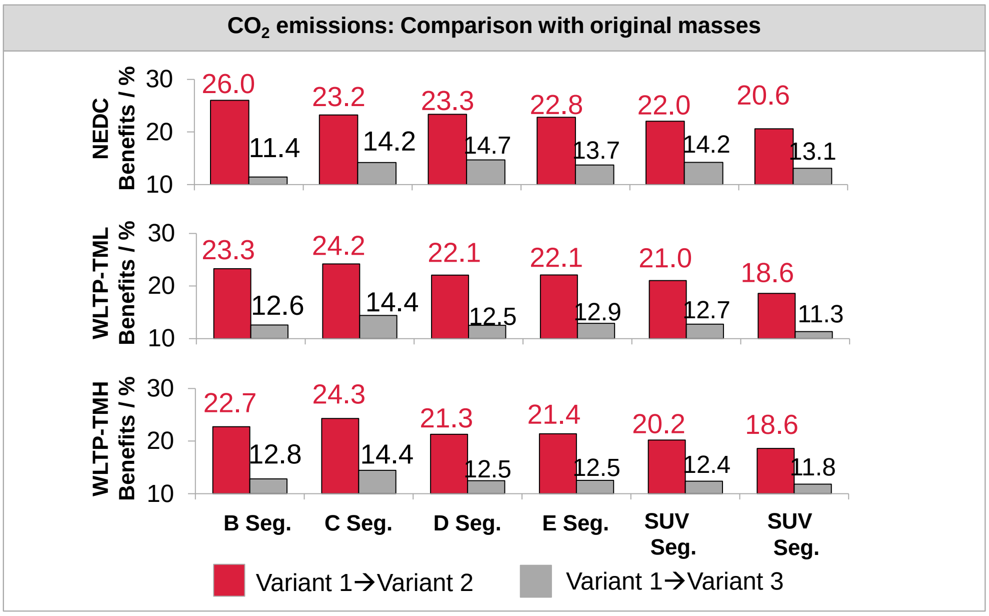


Figure 4.44 Segmentwise trend for the high spec engine with mass and coast down reduction – M3 CD2- (Variant 1→ Variant 2) and high spec engine with coast down reduction only –M1 CD2- (Variant 1→ Variant 3). Variant one is M1 CD1.

4.3.3.11 Impact of Hybrid technology

As a last step in the analysis, hybrid technologies are considered with a medium spec engine version. This upgraded package considers therefore a certain level of engine upgrading (not as complete as the one shown in the high spec engine) and the introduction of hybrid systems, which will differ depending on the segment. Therefore, the total benefits will be calculated in the bar plots as relative benefit with respect to the baseline which considered only the optimized friction reduction, i.e. the total benefit shown is not only due to the mild hybrid systems but also due to the some engine upgrade.

The Table 4.8 summarizes the considered technologies, which differ upon the segments. In addition, the hybrid components consider some extra mass which is conveniently added to the system (reducing the overall benefit of the add-on technologies). For a full description of the hybrid systems, see Chapters 2.8.3 and 4.3.1.6 (the last one also shows the description of the hybrid strategy).

Segment	Baseline with start and stop	Add-on technologies
B	1.2 L I3; MT6 M3 CD2 FR LNT+cDPF HPCLPEGR	BSG P0 48 V 15 kW (~+25 kg) CHPCLPEGR
C	1.4 L I3; DCT7 M3 CD2 FR LNT+cDPF HPCLPEGR	BSG P0 48 V 15 kW (~+25 kg) CHPCLPEGR
D	1.6 L I4; DCT10 1st-TC FR DOC+SDPF	P2 350 V 35 kW (~+80 kg) VVT

	CHPEGR	CHPCLPEGR
E	2.0 L I4; DCT10 2st-TC FR DOC+SDPF CHPEGR	P2 350 V 40 kW (~+80 kg) VVT CHPCLPEGR
SUV	2.0 L I4; DCT10 2st-TC FR DOC+SDPF CHPEGR	P2 350 V 40 kW (~+80 kg) VVT CHPCLPEGR
LCV	1.8 L I4; DCT10 2st-TC FR DOC+SDPF CHPEGR	P2 350 V 45 kW (~+80 kg) VVT CHPCLPEGR

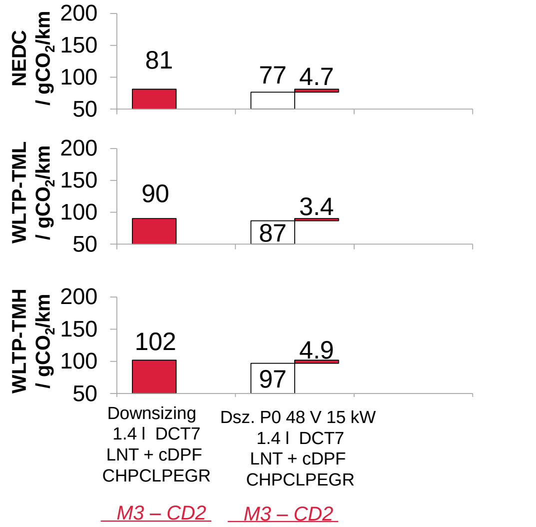
Table 4.8: Technology changes for engine and transmissions for defined technology step. FR = Friction reduction.

<u>P0 Hybrid system</u>

Both Segments B and C mount a mild hybrid P0 System by means of a BSG system. The results for the Segment C are shown in the Figure 4.45. The CO_2 benefits in the NEDC are 5.8 % while lower in the WLTP TM_H, around 4.1%. The following facts should be considered:

- The P0 hybrid system itself gets a benefit around 5 % when supporting the engine boosting at some phases.
- The coolant installed into the HP EGR system, jointly with the already existing cooled LPEGR loop, gives another 0.5-1 % benefit.
- The reduction of the fuel penalty coming out of the LNT regeneration events, due to the engine NO_x reduction gives another extra ~0.3% benefit in the NEDC, but not higher in the WLTP, since the system still needs to have a similar strategy. Anyway, some benefit can still be obtained for example in the WLTP TM_H and the C Segment vehicle.
- However, the extra mass considering the P0 system gets some penalty in the range of 1-1.5 % (~25kg).

CO$_2$ emissions



Impact of Hybrid systems

- Percentage Reductions.
  - NEDC 5.8 %
  - WLTP-TML 3.7 %
  - WLTP-TMH 4.8 %

Figure 4.45 CO$_2$ potential for the P0 48V 15 kW (BSG) variant with respect to the system with friction reduction. The add-on technologies comprise also the CHPCLPEGR system with respect to the one with HPCLPEGR.

In order to better understand the influence of the hybrid system itself without considering the effect of any other technology, the following Tables show the CO$_2$ potential benefits by simulating the hybrid variant with the P0 functionality enabled and disconnected. In such a way, the real benefit coming out of the hybrid system can be understood without the influence of other technologies or strategies.

Cycle	CO_2 [g/km]		
	Base	Hybrid	Benefit
NEDC	77.3	73.5	-5.1%
WLTP TML	82.9	76.8	-5.2%
WLTP TMH	93.1	87.7	-6.1%
FTP-75	83.63	74.9	-11.6%
HWFET	67.4	66.9	-0.8%

Table 4.9 CO_2 potential by installing only the P0 48V 15 kW (BSG) into the Segment B without considering the fuel penalty coming from the mass.

Cycle	CO_2 [g/km]		
	Base	Hybrid	Benefit
NEDC	80.8	76.5	-5.6%
WLTP TML	90.2	86.7	-4.1%
WLTP TMH	101.4	97.0	-4.5%
FTP-75	83.3	76.9	-8.3%
HWFET	72.6	71.8	-1.1%

Table 4.10 CO_2 potential by installing only the P0 48V 15 kW (BSG) into the Segment C without considering the fuel penalty coming from the mass.

Therefore, the following conclusions can be extracted:

- The system benefit into the cycles is similar and in the range of 4-5% due to mainly reduce the total power required from the engine.
- The average BSFC, thus the average efficiency, is slightly reduced in both cases.
- The influence over the LNT strategy might be individually optimized. However, in this project a coordinator mainly optimized for the WLTP TM_H cycle has been selected, getting a better performance into such cycle. Thus, the number of total regeneration events can be therefore reduced. This fact influences to have a slightly higher benefit into the WLTP TM_H than in the WLTP TM_L: 0.9 % higher CO_2 potential in the Segment B and 0.4 % higher in the Segment C for the WLTP TM_H.

As an example of the strategy, the Figure 4.46 shows the operation strategy for the hybrid variant of the Segment C in the WLTP TM_H cycle. The BSG recuperates the energy during braking of the vehicle, as shown in the bottom plot where the negative red peaks show recuperation phases. At the same time, the BSG boosting torque is applied during high engine torque request to keep away the system from the full load and the areas with higher engine-out NO_x. The state of charge (SOC) is balanced to get a similar state at the beginning and end of cycle. The most of the BSG support for such powertrain is applied at the end of the cycle at the highway area, where the main benefits regarding CO_2 and NO_x reduction might be achieved.

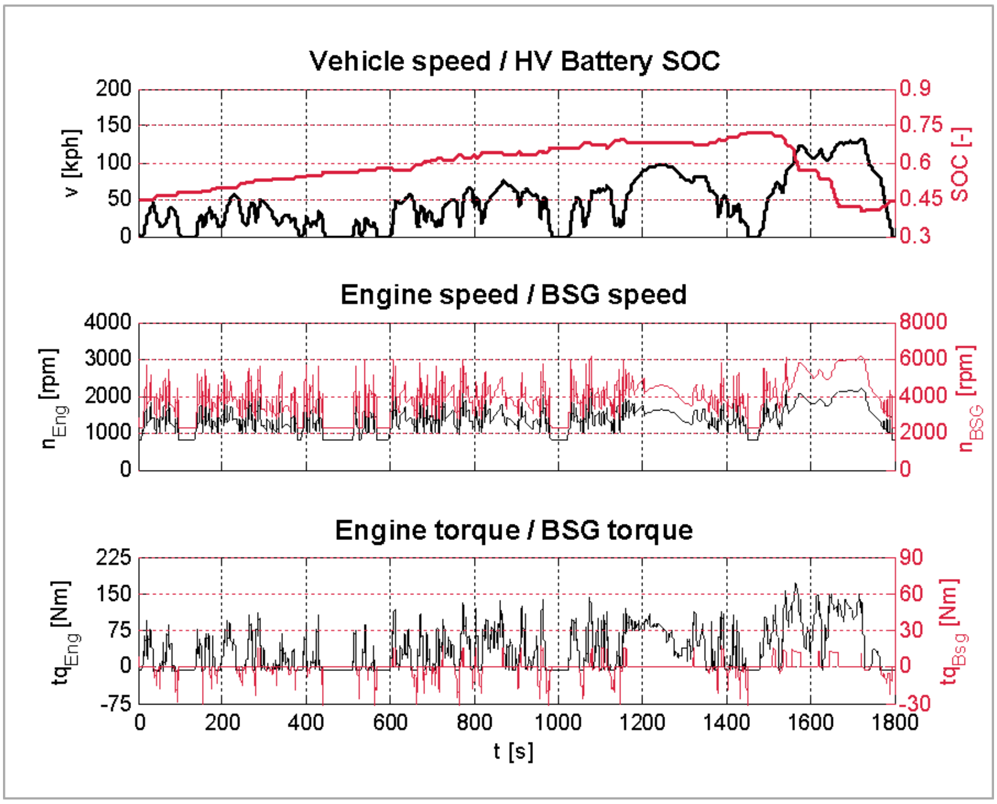


Figure 4.46 Operation strategy for the Segment C P0 hybrid vehicle in the WLTP TM_H.

The Figure 4.47 shows now the engine operating points for the hybrid configuration with the BSG functionality enabled and disabled. As shown before, the red points representing the hybrid strategy are cancelling the high load requests by the combustion engine, reducing the overall energy required by the powertrain and at the same time avoiding areas with high engine out NO_x. The strategy for the other cycles for both the Segment B and C is therefore similar.

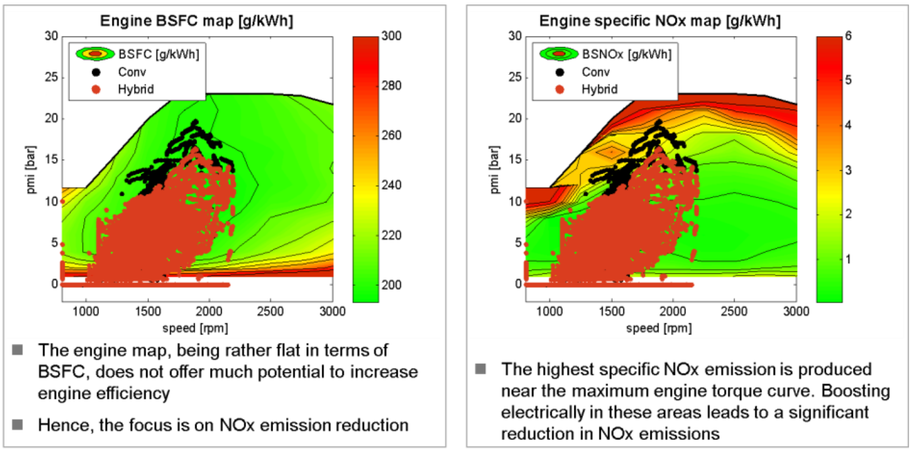


Figure 4.47 Comparison of the engine operating points for the Segment C hybrid vehicle in the WLTP TM_H. Left: BSFC map. Right: $BSNO_x$ map.

For completeness, the Figure 4.48 and Figure 4.49 show the operation strategy and engine operating points for the Segment C and FTP75 cycle. The same strategy and comments mentioned before may also be subscribed. Nevertheless, the FTP75 cycle presents a particularity which heavily influences the final tailpipe NO_x emissions. The cycle gets a sharp acceleration in the range of the 200 s when the engine is still cold. At that point, the engine-out NO_x emissions are high and at the same time, the aftertreatment systems do not reach an appropriate efficiency, especially in the case of the SDPF system (installed in the heavier segments). Therefore, the BSG might be used to boost at that phase by simply cancelling the high load peak. Nevertheless, the fulfilment of the US limits requires of a more refined strategy than in the case of the EU market, which is out of the scope of the present project.

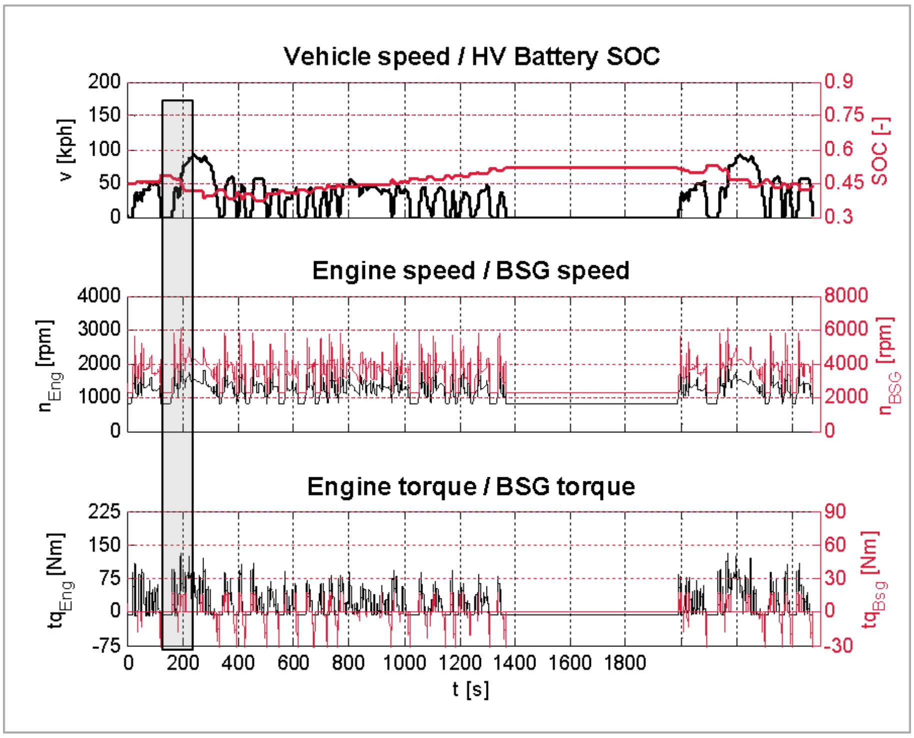


Figure 4.48 Operation strategy for the Segment C P0 hybrid vehicle in the FTP75.

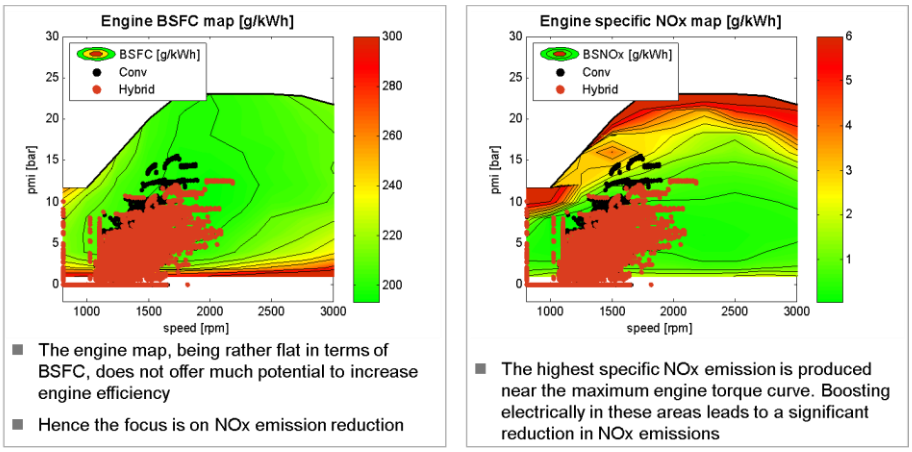


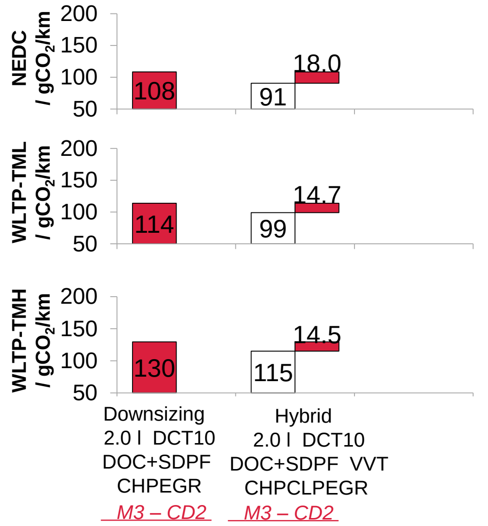
Figure 4.49 Comparison of the engine operating points for the Segment C hybrid vehicle in the FTP75. Left: BSFC map. Right: BSNO$_x$ map.

<u>P2 Hybrid system</u>

The segments D, E, SUV and LCV mount a full parallel P2 System. The results for the Segment E are shown in the Figure 4.50. The CO_2 benefits in the NEDC are 16.5 % while lower in the WLTP TM$_H$, around 11.2 %. The following facts should be considered:

- The P2 hybrid system itself gets a benefit around 10-13 % when supporting the engine boosting at different phases.
- The added cooled LPEGR loop, jointly with the already existing cooled HPEGR loop, gives another 2-4 % benefit, higher in the NEDC.
- The VVT gives a marginal CO_2 benefit (~0.5%), but also supports the faster engine warming.
- However, the extra mass considering the P2 system gets some penalty in the range of 2-3 % (~80kg).

CO_2 emissions



Impact of Hybrid systems

- Percentage Reductions.
  - NEDC 16.5 %
  - WLTP-TML 12.9 %
  - WLTP-TMH 11.2 %

Figure 4.50 CO_2 potential for the Segment E P2 350V 40 kW (Full Parallel) variant with respect to the system with friction reduction. The add-on technologies comprise also the CHPCLPEGR and VVT systems with respect to the one with HPEGR.

As in the previous chapter, the following tables show the CO_2 potential benefits by simulating the hybrid variant with the P2 functionality enabled and disconnected. In such a way, the real benefit coming out of the hybrid system can be understood without the influence of other technologies or strategies. In general, the maximum benefits are achieved in the FTP75 (15-18 %) and NEDC (13-15 %). The WLTP cycles get lower benefit ranging from 8 to 12.5 %. Finally, the benefits in the HWFET are marginal or even penalizing the final CO_2. The reason is that such cycle is quite specific to measure the fuel economy at a medium-high speed without barely any boosting phase.

Cycle	CO_2 [g/km]		
	Base	Hybrid	Benefit
NEDC	93.6	80.8	-13.7%
WLTP TML	103.1	91.9	-10.9%
WLTP TMH	117.3	106.1	-9.5%
FTP-75	100.9	82.7	-18.0%
HWFET	80.8	82.3	+1.9%

Table 4.11 CO_2 potential by installing only the P2 350V 35 kW (BSG) into the Segment D without considering the fuel penalty coming from the mass.

Cycle	CO_2 [g/km]		
	Base	Hybrid	Benefit
NEDC	106.5	90.5	-15.0%
WLTP TML	112.6	99.1	-12.0%
WLTP TMH	128.7	115.2	-10.5%
FTP-75	112.4	93.1	-17.2%
HWFET	87.3	88.0	+0.8%

Table 4.12 CO_2 potential by installing only the P2 350V 40 kW (BSG) into the Segment E without considering the fuel penalty coming from the mass.

Cycle	CO_2 [g/km]		
	Base	Hybrid	Benefit
NEDC	109.7	95.6	-12.9%
WLTP TML	118.8	108.7	-8.5%
WLTP TMH	136.2	125.3	-8.0%
FTP-75	112.8	95.5	-15.3%
HWFET	95.6	97.6	+2.1%

Table 4.13 CO_2 potential by installing only the P2 350V 40 kW (BSG) into the Segment SUV without considering the fuel penalty coming from the mass.

Cycle	CO_2 [g/km]		
	Base	Hybrid	Benefit
NEDC	129.6	116.9	-9.8%
WLTP TML	153.7	144.8	-5.8%
WLTP TMH	170.3	161.5	-5.2%
FTP-75	132.0	115.0	-12.9%
HWFET	124.2	128.6	+3.5%

Table 4.14 CO_2 potential by installing only the P2 350V 45 kW (BSG) into the Segment LCV without considering the fuel penalty coming from the mass.

As an example of the strategy, the Figure 4.51 and Figure 4.52 show the operation strategy for the P2 hybrid variant of the Segment E in the WLTP TM_H and FTP75 cycles. As happened for the P0, the system supports into the engine boosting by reducing the overall energy required from the combustion engine. Since the P2 systems present higher power and battery capacity than the P0 system, the overall potential is higher: the electrical engine is operating the most of the time. The state of charge (SOC) is balanced to get a similar state at the beginning and end of cycle. In addition, an important NO_x reduction might also be achieved.

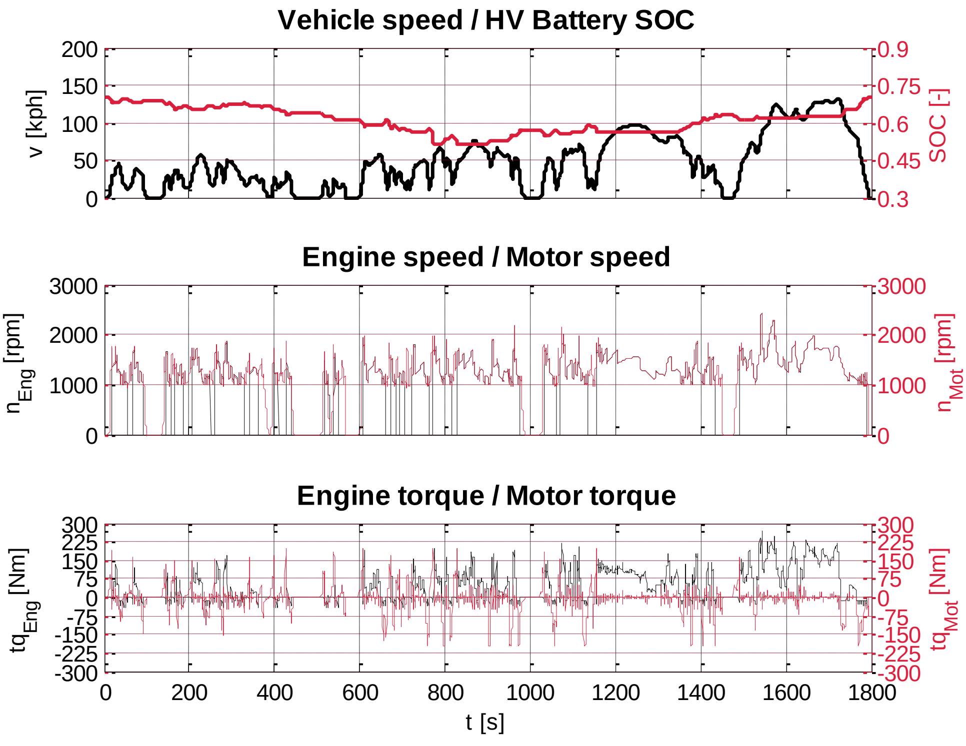


Figure 4.51 Operation strategy for the Segment E P2 hybrid vehicle in the WLTP TM_H.

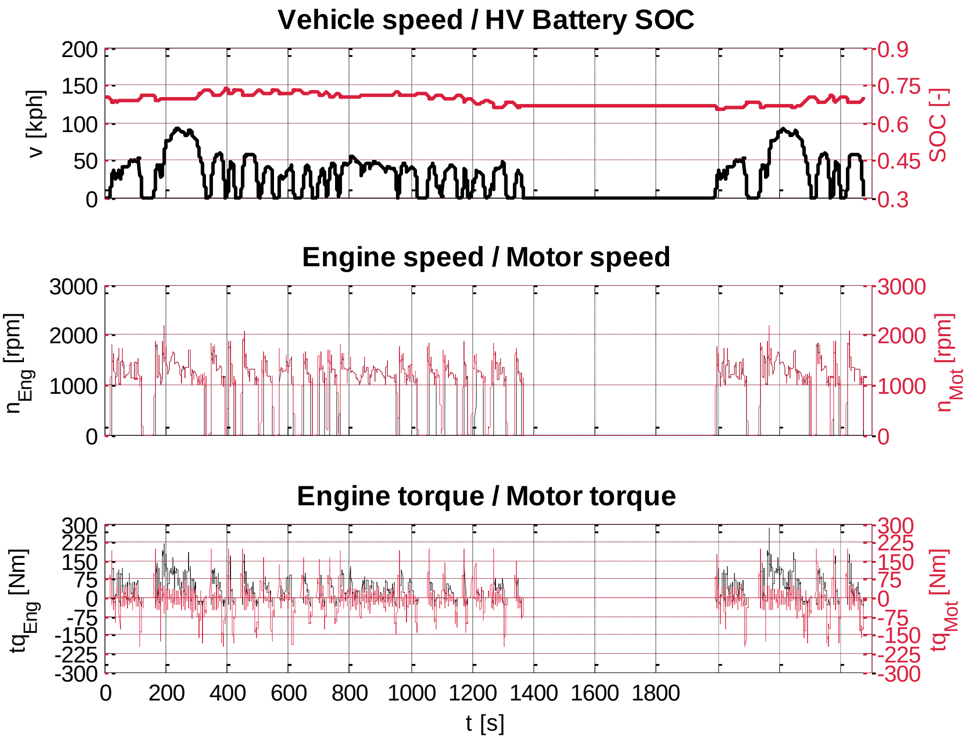


Figure 4.52 Operation strategy for the Segment E P2 hybrid vehicle in the FTP75.

Segmentwise CO_2 reduction potentials

The Figure 4.53 summarizes the segmentwise potentials for the hybrid systems with respect to the base variant with friction reduction. The following conclusions are extracted:

- For the Segment B, the P0 potential by also considering the cooled high pressure cooled low pressure EGR (base is uncooled HP and cooled LP EGR), is in the range of 6 %. The differences between the NEDC and WLTP are mainly driven here due to the LNT regeneration optimization. The final benefits are higher than in the case of the segment C due to the non-optimized shifting strategy (Segment B mounts a MT transmission), giving more space for optimization when installing the hybrid system. Furthermore, the power design is the same for both segments, but the general power requirements for the Segment B are lower.
- For the Segment C, the upgrading technologies are the same as for the Segment B. The benefit for the NEDC is similar to the one of the Segment B, but slightly lower for the WLTP.
- The heavier Segments (D, E, SUV and LCV) mount a P2 full parallel system. The benefits are clearly higher in the NEDC with respect to the WLTP. In addition, the SUV and LCV also present lower reduction potential due to the combination of a penalized aerodynamics and higher mass, especially in the case of the LCV. Therefore, for the LCV, such technology seems less attractive than for the Segment E.

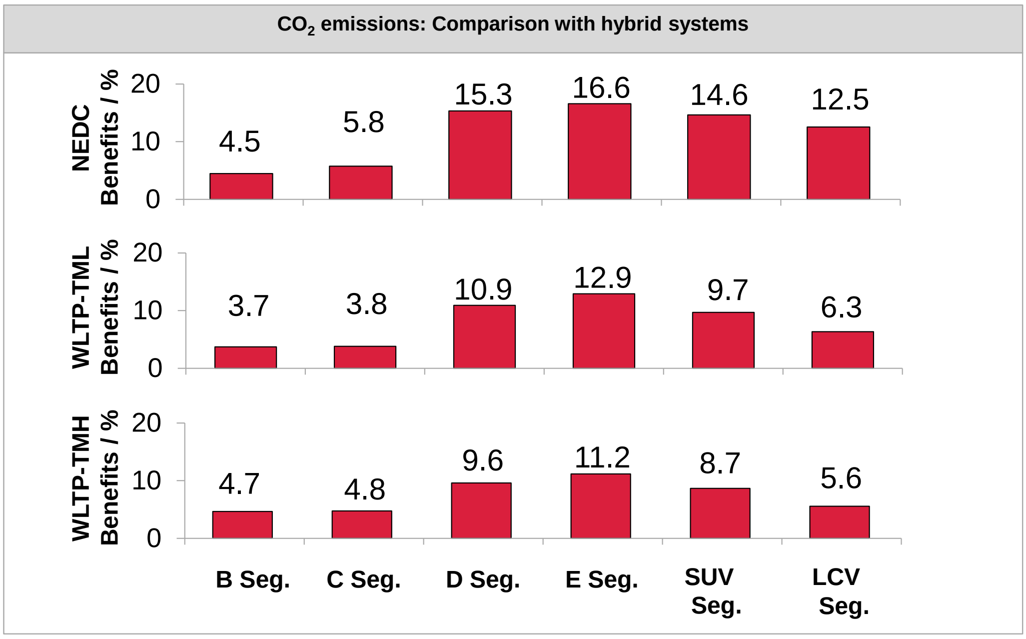


Figure 4.53 Segmentwise trend for the hybrid version. The base variant already considers the friction reduction, but no engine technology.

4.4 Gasoline Simulation

4.4.1 Model set-up and boundary conditions

4.4.1.1 Modelling Approach

For the vehicle simulation with gasoline engines a similar toolchain like in the Diesel engines part based on GT-Suite model was used. GT-Suite is commercial simulation software for engine and vehicle simulation. The FEV model shown in Figure 4.54 consists of all relevant sub models for simulation of vehicle, engine, vehicle control, ICE thermal model, driver, transmission, auxiliaries, shift strategy, raw and tailpipe emissions. In the detailed model all relevant ECU functions are included e.g. catalyst heating or fuel cut off during coasting. The model has various detailed sub model levels, a base level and one or two advanced levels. The degree of detail increases in the advanced levels to consider more realistic effects. For example the auxiliary power consumption of the board net is in the base level only a simple constant power consumption. In this project a more advanced level for simulation of intelligent alternator management including a 12V battery model and alternator model was used.

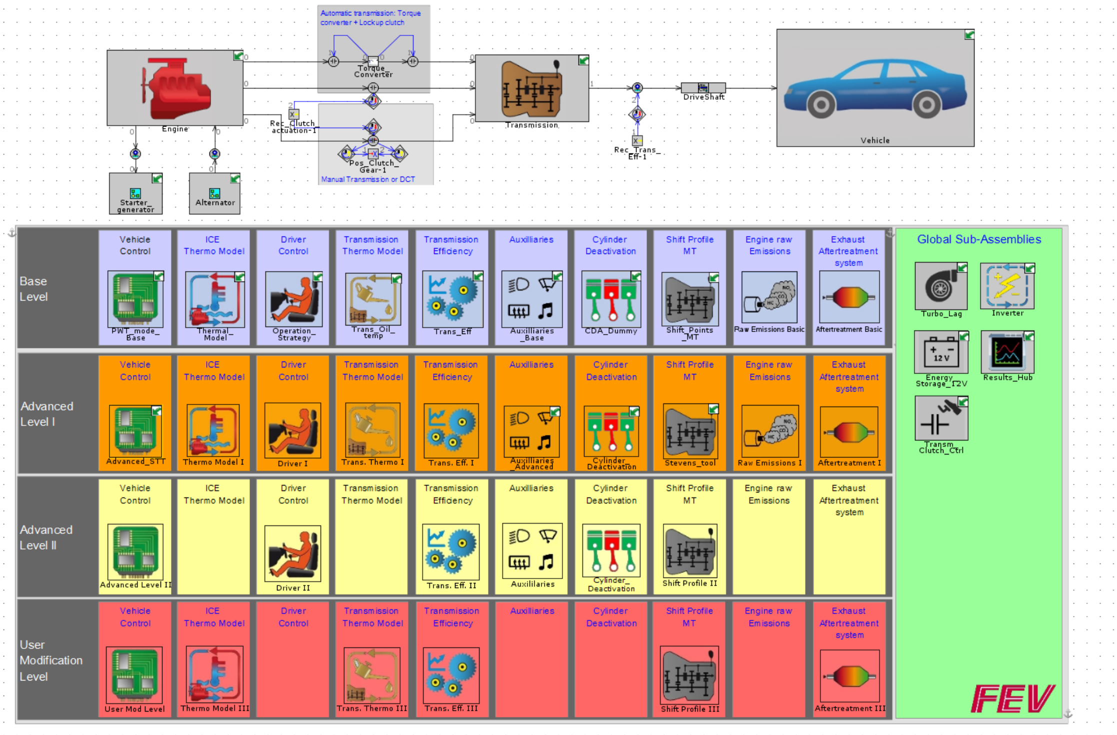


Figure 4.54: FEV GT-Suite Simulation Model

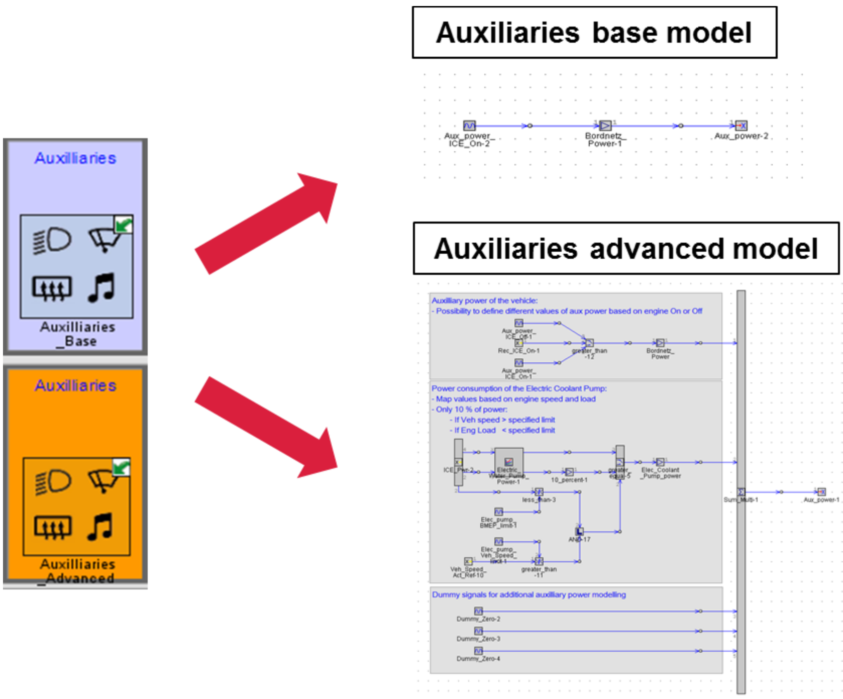


Figure 4.55: Base and advanced level of auxiliaries

The fuel consumption was simulated with the help of stationary engine maps derived from measurement data and engine simulations. The stationary fuel consumption was corrected in the dynamic driving cycle with correction factors for dynamic spark timing, actual engine temperature and dynamic air/fuel ratio based on typical vehicle calibration.

4.4.1.2 Engine warm-up model

The engine warm-up is considered by using a thermal model of the engine and measurement data from various engines. The thermal model consists of 4 thermal resistances, the cylinder mass, the coolant mass, the engine oil mass and the engine block mass. By using the FEV database the heat transfer between the thermal resistances are modelled. With this model the engine coolant and oil temperature during various driving cycles are simulated. The influence of the engine temperature on the fuel consumption is considered by the friction influence of the engine. With increasing engine temperature the engine friction decreases. The decreasing behaviour of engine friction with warm-up is modelled by using a friction correction depending on the actual engine oil and coolant temperature in the driving cycle. This correction was derived from measurement data of various engines at 30°C, 60°C and 90°C.

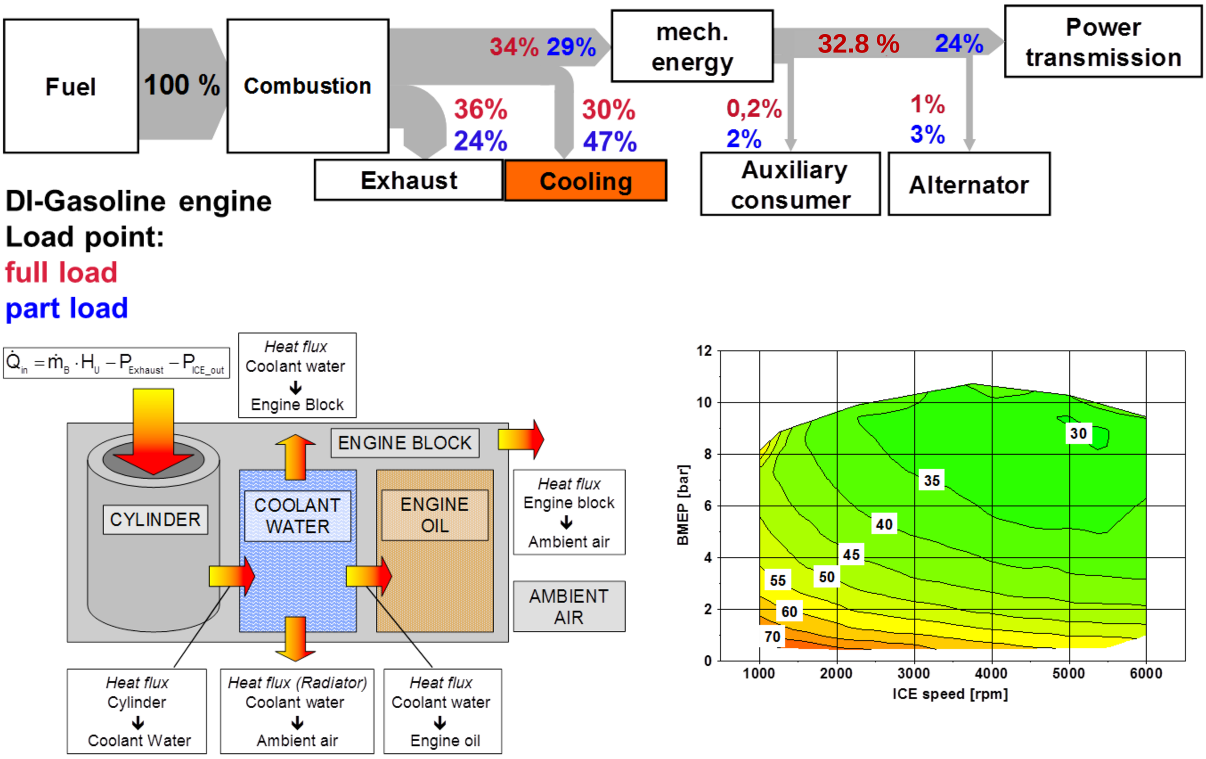


Figure 4.56: Overview engine warm-up model

4.4.1.3 Transmission model

The transmission simulation is based on efficiency maps measured on transmission test bench for each gear, various transmission temperatures and transmission speeds. To consider the influence of the transmission oil temperature on the transmission efficiency a thermal model of the transmission was used in a similar way like the engine thermal model. Losses for auxiliaries like transmission oil pump and actuators are considered in the efficiency calculation. In this project MT, DCT, and AT are simulated.

The transmission shift strategy was calculated based on the engine fuel consumption map and the gear ratios. For each output power of the engine, the operation point with the minimal fuel BSFC was calculated. The intersection of this operation points as the optimum BSFC operation line. The target of an optimal shift strategy is to operate the engine as close as possible to this optimal operation line. For that the shift maps are calculated to shift up or down if the engine operation point in the higher or lower gear is more optimal. This shift strategy is implemented in the model as shift maps for up- and downshifting for each gear depending on engine speed and accelerator pedal value.

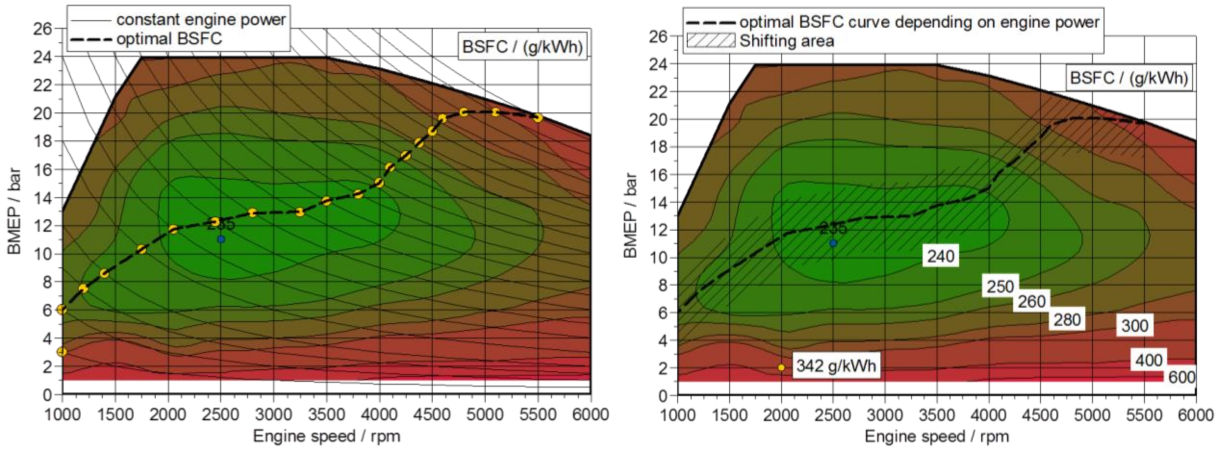


Figure 4.57: Optimal operation line (left) and operation area of shift strategy (right)

4.4.1.4 Engine map calibration for various technologies

FEV uses for the calculation of gasoline engine maps an in-house tool that can consider various gasoline engine technologies and the effects on fuel consumption.

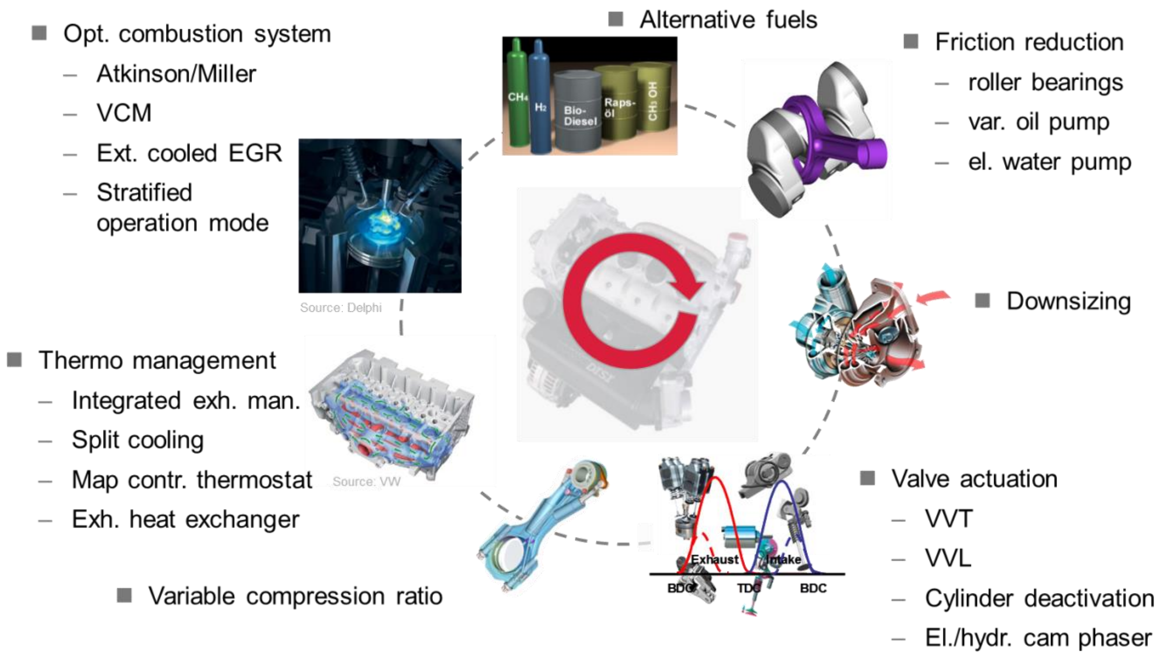


Figure 4.58: Gasoline engine technologies examples for fuel consumption map simulation

The effect of each technology on 5 efficiencies was evaluated by measurements or detailed engine process simulations. This effects from the engine technology database can be added to base engine measurements.

For the calculation of engine maps as input for the vehicle simulation a separate calculation process is used to modify base engine maps with additional technologies. An example for the engine map calculation is shows in Figure 4.60.

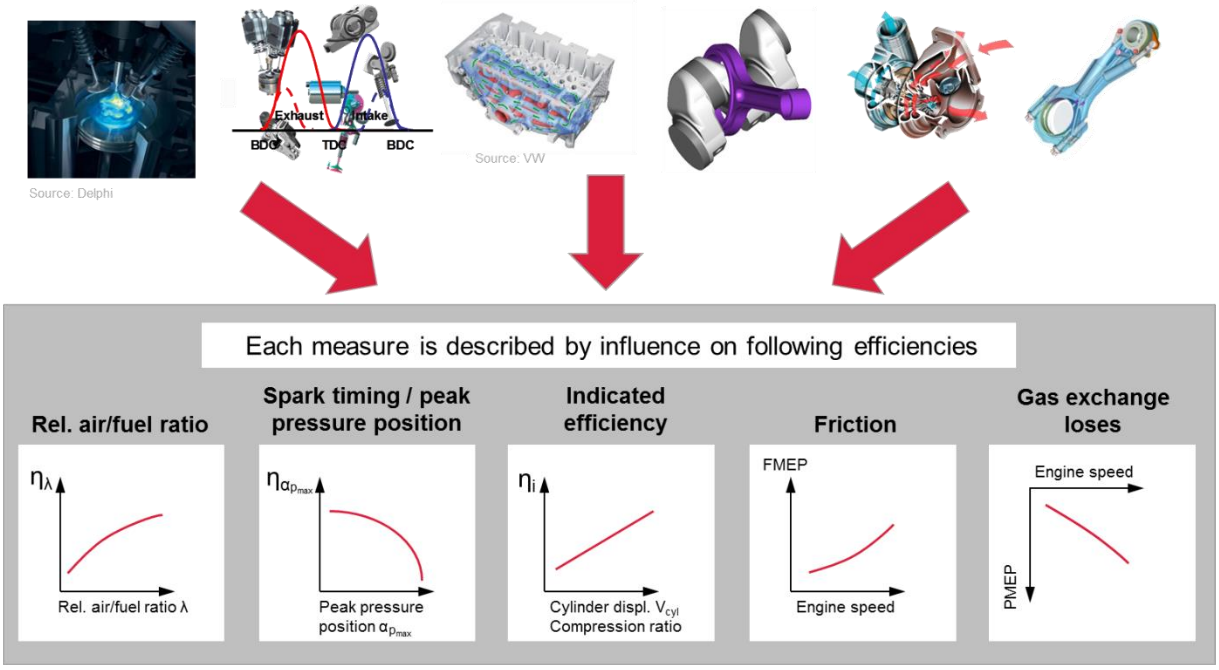


Figure 4.59: Effects of engine technology on efficiency

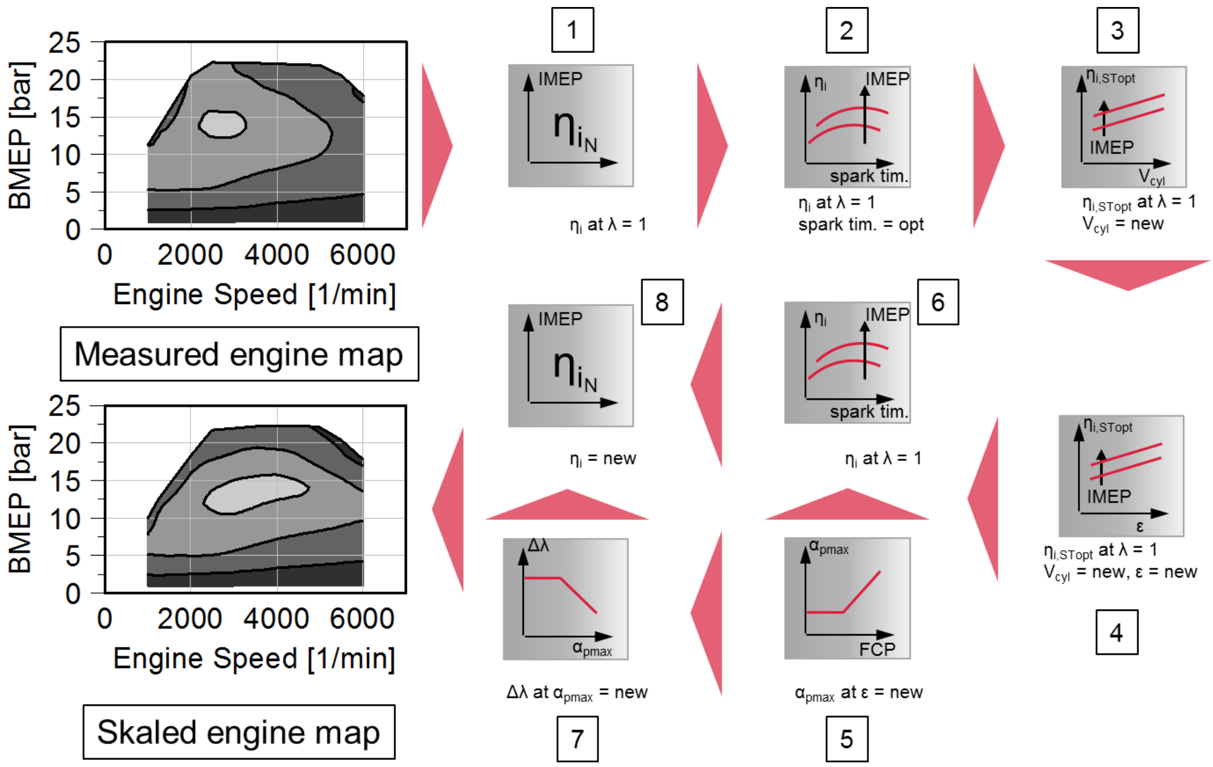


Figure 4.60: Engine map scaling process for increased CR

In a 1. Step an engine map without fuel enrichment and stoichiometric air fuel ratio is calculated. In a second step the knock limitation is eliminated from the map and a map with optimal peak pressure position in the whole map is calculated. This engine map is the base for technology modifications like change of compression ratio or displacement. The fuel consumption changes if the cylinder displacement is modified due to higher or lower wall heat losses and various quench effects. This is considered in step 3. In the 4th step the compression ratio is modified and the effect of changed efficiency implemented. A changed compression ratio and cylinder displacement also affects the knocking behaviour of the engine. This is considered by using a correlation between final compression pressure in the end of the compression stroke and the achievable peak pressure position without knocking combustion (step 5). The effect of the retarded combustion timing on the efficiency is then considered in step 6. A shift of combustion centre to earlier or later position leads to a change in exhaust gas temperature and this requires a higher or lower fuel enrichment for component protection. In step 7 the required enrichment to limit the exhaust gas temperature to the component limit is calculated. In step 8 the influence of changed air-fuel ratio on the fuel consumption map is considered and a new BSFC map is calculated. This BSFC map is used as input for the drive cycle simulation in GT-Suite.

4.4.1.5 Hybrid powertrain modelling

FEV uses a optimization approach based on Design of Experiments methodology in combination with longitudinal dynamics simulation and numerical optimization to find the optimal hardware layout and operation strategy for various HEV types. This methodology is implemented in the FEV drivetrain optimization tool (FEV DOT). This process is shown in Figure 4.61.

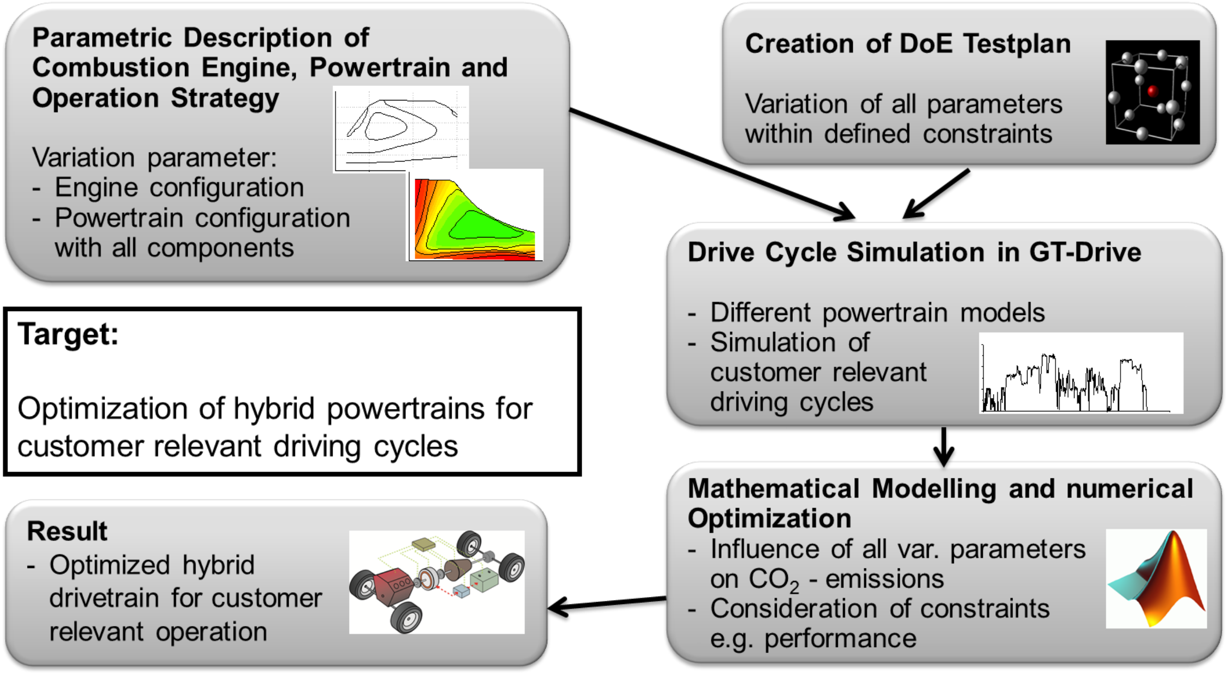


Figure 4.61: Optimization process FEV DOT

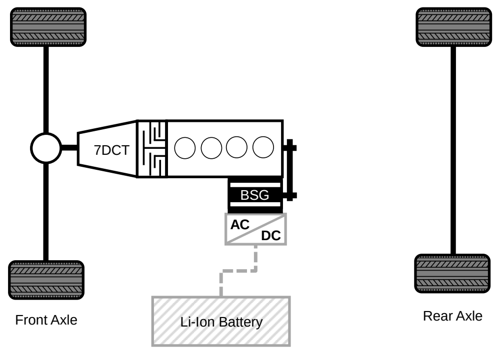
First step is the description of the powertrain and the operation strategy with a few number of variable parameters. This parameters were variated in DoE test plan and all combinations of the variable parameters from the test plan were simulated in GT-Suite in the target drive cycles. The results from the simulation were then modelled in a mathematical model in dependency of the input parameters. Inside this model a numerical optimization of the variable parameters to minimize fuel consumption under respect of the exhaust emissions and targets for performance were done. The result is an optimized powertrain that fulfil the driving requirements and emission targets with minimal fuel consumption.

In the following part the boundary conditions of the optimization of the used hybrid powertrains in this project (P0 hybrid, P2 parallel hybrid) will be described.

P0 hybrid

In this configuration the engine is coupled, by means of a belt, with a 48V generator. The limited power of the electric machine (and the limited torque that can be transmitted with the belt) does not allow to drive pure electrically the vehicle. However it is possible to perform the regenerative braking, and to use the energy recuperated in order to boost the engine.

Hybridization P0 Architecture



P0 hybrid features

- Electric machine coupled to the ICE by means of a belt
- Possibility to perform electric boosting during low power phases, and regenerative braking

- Voltage: 48 V
- Battery type: Li-Ion
- Battery capacity: 0.53 kWh
- Power e-motor: 15 kW
- Torque e-motor: 50 Nm
- Additional weight: 33 kg

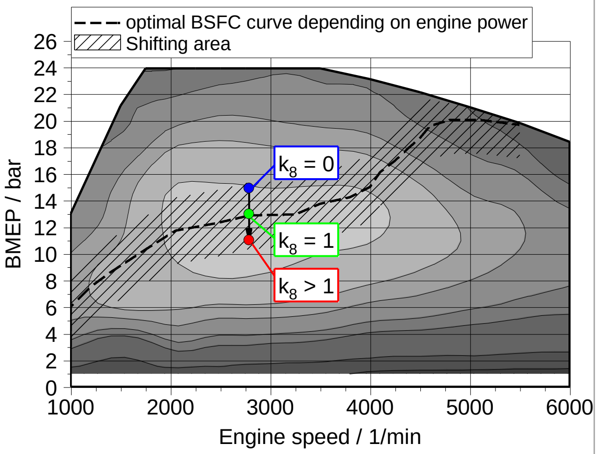
Figure 4.62 Architecture and specification of a P0 hybrid vehicle used in B and C segment

In the optimization process, the hybrid strategy works in a close cooperation with the shift-strategy, in order to reach the optimal trade-off between CO_2 emissions and SOC balance.

First of all an optimal BSFC line is defined basing on the engine specific consumption map. The shift strategy works in order to get the operation points of the engine inside a range of this line. The range becomes closer to the optimum with higher number of gears of the transmission.

During the cycle the BMEP difference between the operating points and the optimal line is calculated. The K8 parameter defines the fraction of this difference that has to be covered by the generator boosting. Then the hybrid strategy works in order to achieve a more close distribution, of the engine operating points, to the optimal BSFC line. So brake energy is used to support the engine with power of the electrical machine and reduce the engine power. This is done at operation points were the engine load is higher than the optimum to reduce engine power and improve engine efficiency at the same time.

Hybridization P0 strategy

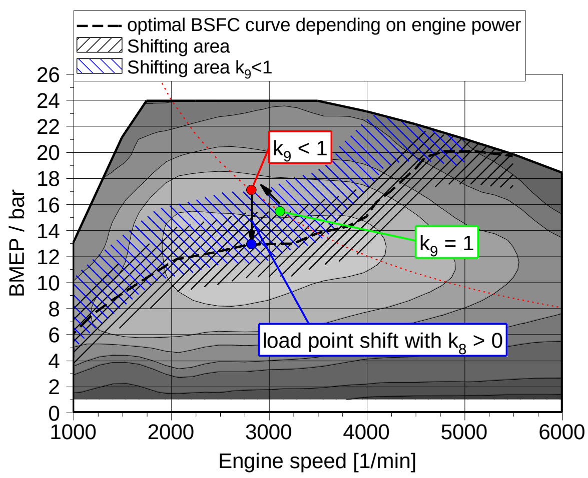


K8 Parameter

- Operating strategy for electric boosting in mild hybrid vehicle
- If BMEP of actual operation point → BMEPopt than load point shift towards BMEPopt
- K8=0 → no load point shift
- K8=1 → load point shift to BMEPopt
- K8>1 → load point shift below BMEPopt

Figure 4.63 K8 Parameter used for electric boosting within P0 operating strategy

Hybridization P0 strategy



K9 Parameter

- Operating strategy parameter k9 influences shift strategy of automatic transmission:
- K9<1 → upshift at higher pedal value, load point shift to higher load and lower speed
- K9=1 → no load point shift
- K9>1 → upshift at lower pedal value, load point shift to lower load and higher speed

Figure 4.64 K9 Parameter used to optimize the shift points within P0 operating strategy

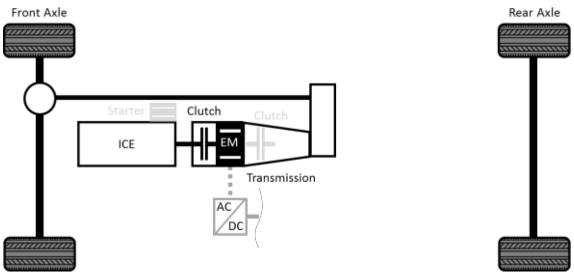
The second key parameter of the P0 strategy, is the K9 parameter. This parameter causes an up-shift to a higher load and lower speed region, of the operating points. In this way the load difference between the actual operating points and the optimal BSFC line increases, and so also the potential for boosting without increasing the BSFC.

To conclude, with this hybrid configuration, it is possible to recuperate a limited amount of energy during the braking phases. This is due to the limited torque and power of the electric machine. Always for the same reason is not possible to drive the vehicle pure electrically. Then the recovered energy can be used for boosting the engine. The hybrid strategy (K8 and K9), together with the shift-strategy defines the criterion for boosting: make the engine work as close as possible to the BSFC optimal line. This is a kind of downspeeding approach for the combustion engine. It follows from this the final benefits in terms of CO_2 emissions.

P2 hybrid In this hybrid configuration the electric machine is embedded between the engine and the transmission, and is separated from both by two clutches. During braking phases the engine is decoupled form the electric machine that enables regenerative breaking.

The key factor of the P2 hybridization is the possibility to drive the vehicle pure electrically. The power of the electric machine allows to recuperate almost completely the energy that come from the braking phases. Task of the strategy is to define the optimal method to use this recuperated energy.

Hybridization P2 Architecture



Full hybrid P2 features

- Electric machine embedded between the engine and transmission by means of two clutches
- Possibility to perform regenerative braking
- Possibility to perform pure electric driving during low vehicle speed phases

D-Segment

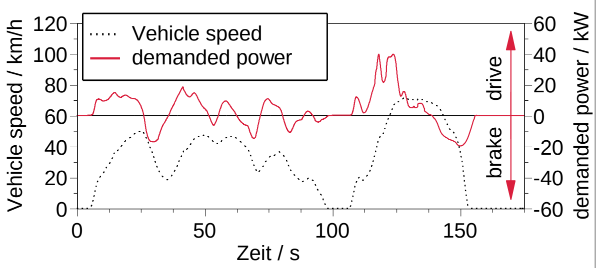
- Voltage: 350 V
- Battery type: Li-Ion
- Battery capacity: 1.1 kWh
- Power e-motor: 35 kW
- Torque e-motor: 175 Nm
- Additional weight: 52 kg

Figure 4.65 Architecture and specification of a P2 hybrid vehicle used in D and E segment

As first, the power demanded for driving is calculated as function of the vehicle resistances and driving profile. The P2 hybrid strategy uses two parameters in order to find the best configuration possible between CO_2 emissions and SOC balance.

The first parameter K1 defines the power demanded threshold below which the vehicle performs full electric driving.

Hybridization P2 strategy



$$P_{demand} = \left[\underbrace{m_{veh} \cdot g \cdot f_R \cdot \cos\alpha}_{\text{rolling resistance}} + \underbrace{m_{veh} \cdot g \cdot \sin\alpha}_{\text{climbing resistance}} + \underbrace{e \cdot m_{veh} \cdot a}_{\text{acceleration resistance}} + \underbrace{\frac{\rho_a}{2} \cdot c_W \cdot A \cdot v_{veh}^2}_{\text{air resistance}} \right] \cdot v_{veh}$$

inertia coefficient $e = 1 + \frac{J}{m_{veg} \cdot r_{dyn}^2}$

Demanded power

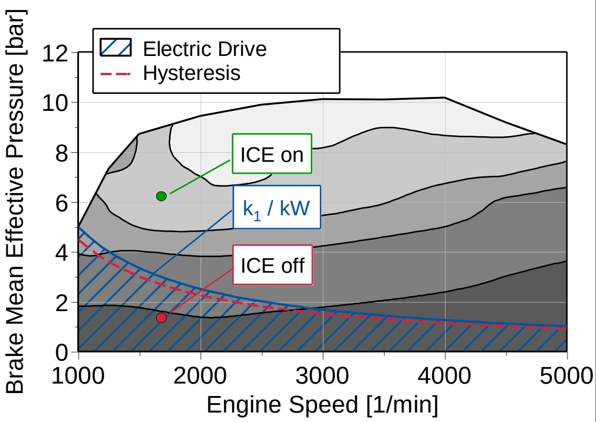
Demanded power calculated with road resistance equation:

Dependency of:

- vehicle mass
- air resistance
- rolling resistance
- driving profile

Figure 4.66 Calculation of the demanded power with P2 operating strategy

Hybridization P2 strategy



K1 Parameter

Operating strategy parameter k_1 defines a power threshold, below which the vehicle performs full electric drive

ICE on:

1. $P_{demand} > k_1$ or
2. $P_{demand} + P_{charge} > k_1$

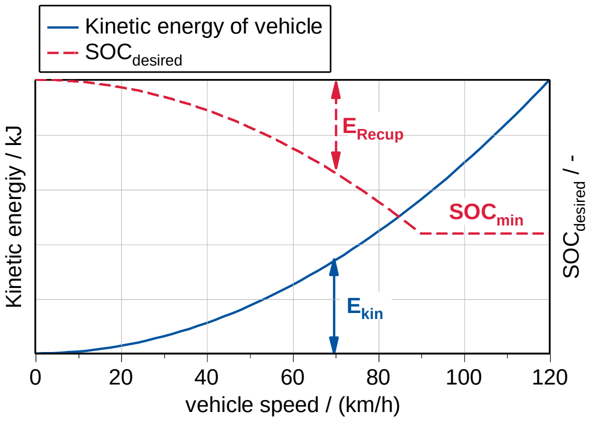
Electric drive, ICE off:

1. $P_{demand} \leq k_1$ and
2. $P_{demand} + P_{charge} \leq k_1$

Figure 4.67 K1 Parameter used to define the power threshold for full electrical drive within P2 operating strategy

The second parameter has the task to balance the SOC by the means of electric machine or engine boosting, depending on the SOC level. More properly, a SOC desired value is defined step by step depending on the potential recoverable energy that is function of the instantaneous kinetic energy of the vehicle.

Hybridization P2 strategy	K2 Parameter



Operating strategy parameter k_2 influences battery charging

$$P_{charge} = k_2 \cdot (SOC_{desired} - SOC)$$

$$SOC_{desired} = \left(SOC_{max,\,legal} - \frac{E_{Recup}}{E_{Bat}} \right)$$

$$E_{Recup} = \eta_{EM} \cdot \eta_{LE} \cdot \eta_{Bat} \cdot E_{kin}$$

$$E_{kin} = \frac{1}{2} \cdot e \cdot (m_{veh} + m_{payload}) \cdot v_{veh}^2$$

$$e = 1.07$$

$$m_{payload} = 100kg$$

SOC: state of charge

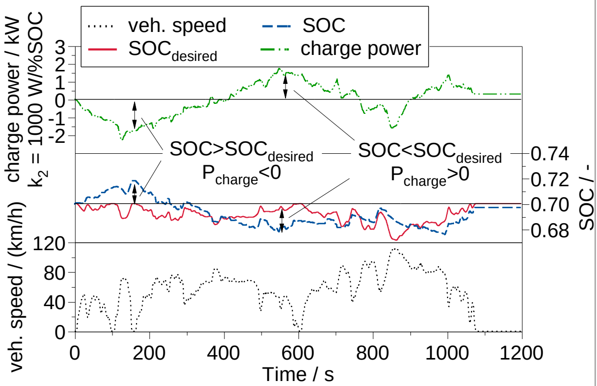
Figure 4.68 K2 Parameter used to balance the battery charging within P2 operating strategy (1/2)

The difference between the desired and actual SOC is calculated for each step.

If the difference is negative it means that the SOC is higher than the desired one and so the electric machine operates boosting on the engine. While, if the difference is positive the engine applies more torque, than the one needed to drive the vehicle, in order to charge the battery (engine boosting).

The second parameter K2 influences directly the power that has to be provided for boosting.

Hybridization P2 strategy	K2 Parameter



Operating strategy parameter k_2 influences battery charging

$$P_{charge} = k_2 \cdot (SOC_{desired} - SOC)$$

$$SOC_{desired} = \left(SOC_{max,\,legal} - \frac{E_{Recup}}{E_{Bat}} \right)$$

$$E_{Recup} = \eta_{EM} \cdot \eta_{LE} \cdot \eta_{Bat} \cdot E_{kin}$$

$$E_{kin} = \frac{1}{2} \cdot e \cdot (m_{veh} + m_{payload}) \cdot v_{veh}^2$$

$$e = 1.07$$

$$m_{payload} = 100kg$$

SOC: state of charge

Figure 4.69 K2 Parameter used to balance the battery charging within P2 operating strategy (2/2)

To conclude, thanks to the bigger size of the electric machine, and to the P2 hybrid architecture, almost all the energy needed to brake the vehicle can be recuperated. The most of this energy is used to drive the vehicle during the drive-away phases (until the threshold power K1 is reached). In this way, it is avoided the use of the engine during these remarkable low efficiency phases. In order to balance the SOC and to use an eventual excess of energy the K2 parameter defines the boosting power that has to be delivered respectively by the electric machine or by the engine.

4.4.2 Technology potentials

The impact of various technology upgrades on the CO_2 emissions is discussed in this section. Therefore, the segment C results are chosen as a case study with detailed explanations, while the percentage benefits in other segments are summarized in an additional plot for each technology step. The investigated measures in this study are:

- Start&Stop function
- Downsizing (from NA basis to TC engine)
- Further downsizing combined with transmission upgrade and weight reduction
- Reduction of vehicle weight and vehicle resistances
- Transmission upgrade from manual to automated (in segment C only)
- Friction reducing measures
- Variable valve lift (VVL)
- Miller cycle
- Low pressure exhaust gas recirculation (LP-EGR)
- Variable compression ratio (VCR)
- Dedicated EGR (in segment D and E only)
- Extreme downsizing
- P0/P2 hybridization

Segment C	Engine size / l	Cyl.	Spec.power / kW/l	EGR	T/C	Valvetrain	Compression Ratio	Engine friction	Hybridization	Transm.	Mass set	Coast down set
Baseline	1.8	4	53	w/o EGR	NA	DVVT	not variable	baseline	without	5-speed MT	M1	CD1
Start&Stop	1.8	4	53	w/o EGR	NA	DVVT	not variable	baseline	**start/stop**	5-speed MT	M1	CD1
Downsizing	**1.4**	**4**	**68**	w/o EGR	**single**	DVVT	not variable	baseline	without	5-speed MT	M1	CD1
Start&Stop	1.4	4	68	w/o EGR	single	DVVT	not variable	baseline	**start/stop**	5-speed MT	M1	CD1
Downsizing +Transmission upgrade +Weight reduction	**1.0**	**3**	**95**	w/o EGR	single	DVVT	not variable	baseline	start/stop	**6-speed MT**	**M2**	**CD1**
Mass red. Scenario II	1.0	3	95	w/o EGR	single	DVVT	not variable	baseline	start/stop	6-speed MT	**M3**	**CD1/CD2/CD3**
Transmission	1.0	3	95	w/o EGR	single	DVVT	not variable	baseline	advanced start/stop	**7-speed DCT**	M3	CD2
Friction package	1.0	3	95	w/o EGR	single	DVVT	not variable	**optimized**	advanced start/stop	7-speed DCT	M3	CD2
Engine technologies	1.0	3	95	w/o EGR	single	**DVVT +VVL**	not variable	optimized	advanced start/stop	7-speed DCT	M3	CD2
Engine technologies	1.0	3	95	w/o EGR	**2-stage**	**DVVT +VVL +Miller**	not variable	optimized	advanced start/stop	7-speed DCT	M3	CD2
Engine technologies	1.0	3	95	**cooled LP**	2-stage	**DVVT +VVL**	**two-step VCR**	optimized	advanced start/stop	7-speed DCT	M3	CD2
Engine technologies	1.0	3	95	**cooled LP**	2-stage	**DVVT +VVL +Miller**	not variable	optimized	advanced start/stop	7-speed DCT	M3	CD2
w/o weight reduction	1.0	3	95	cooled LP	2-stage	DVVT +VVL +Miller	not variable	optimized	advanced start/stop	7-speed DCT	**M1**	**CD2**
Extreme downsizing	**0.8**	**3**	**119**	**cooled LP**	**2-stage**	**DVVT +VVL**	not variable	optimized	advanced start/stop	7-speed DCT	M3	CD2
Top spec engine + Hybrid	**1.0**	**3**	**95**	**cooled LP**	**2-stage**	**DVVT +VVL +Miller**	not variable	optimized	**P0 with 48V**	7-speed DCT	M3	CD2
Extreme downsizing + Hybrid	**0.8**	**3**	**119**	**cooled LP**	**2-stage**	**DVVT +VVL**	not variable	optimized	**P0 with 48V**	7-speed DCT	M3	CD2

Table 4.15 Summary of variants of segment C

In the following figures (Figure 4.70 to Figure 4.73) the Overview of the CO_2-emission results for the C- segment is shown. With all configurations the following driving cycles were simulated: NEDC, WLTP TML, WLTP TML, FTP75, US HWY.

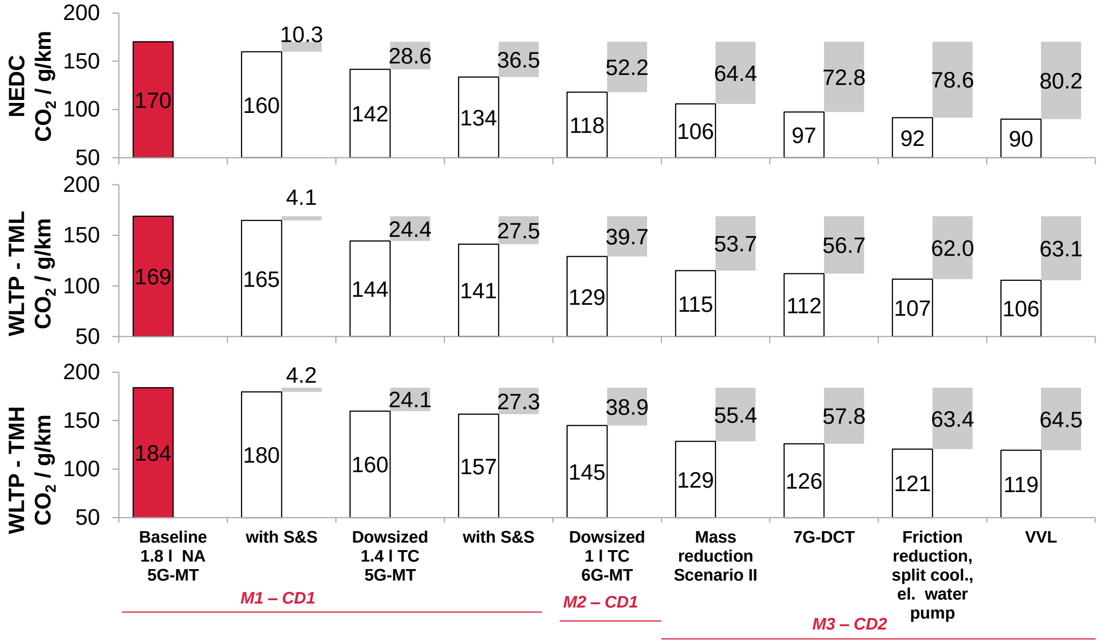


Figure 4.70 Overview of results for segment C in the EU cycles (part 1/2)

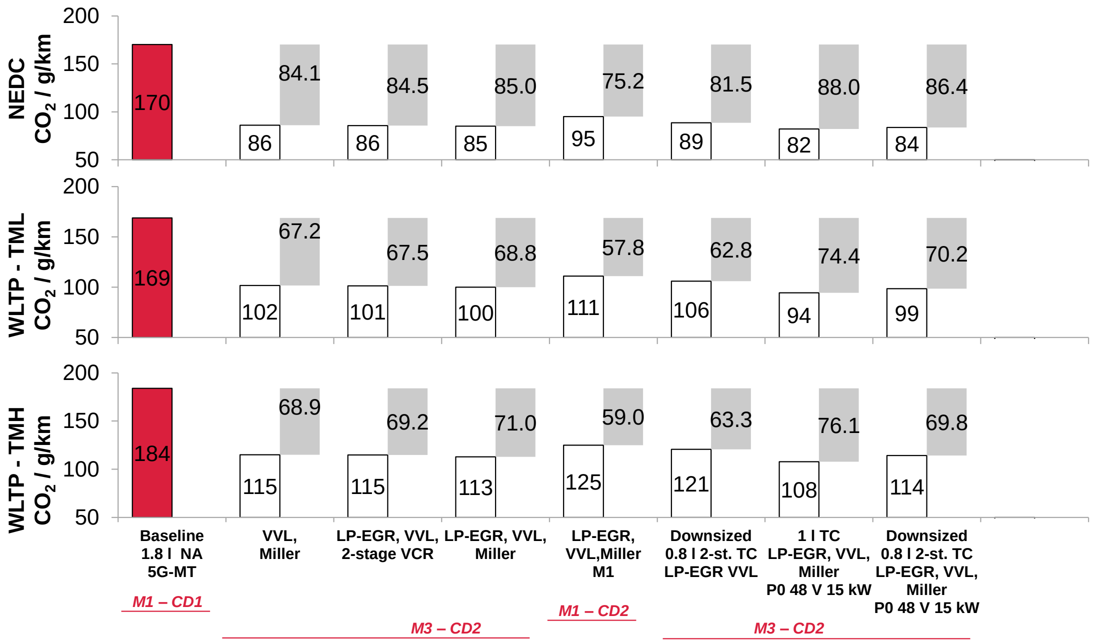


Figure 4.71 Overview of results for segment C in the EU cycles (part 2/2)

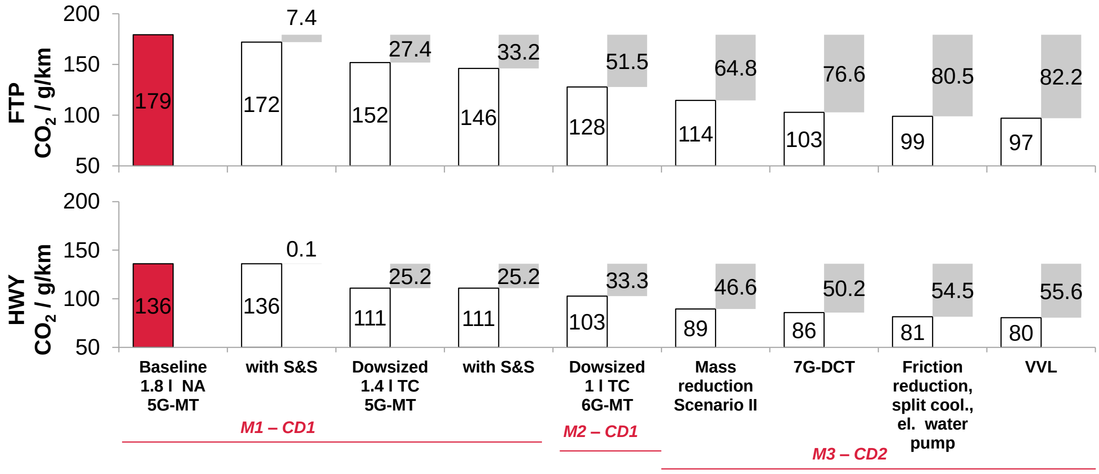


Figure 4.72 Overview of results for segment C in the US cycles (part 1/2)

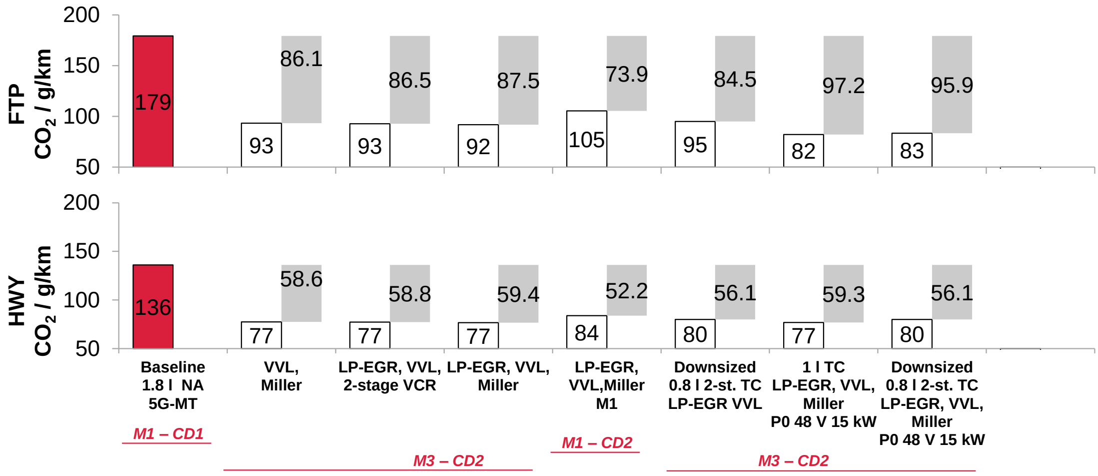


Figure 4.73 Overview of results for segment C in the US cycles (part 2/2)

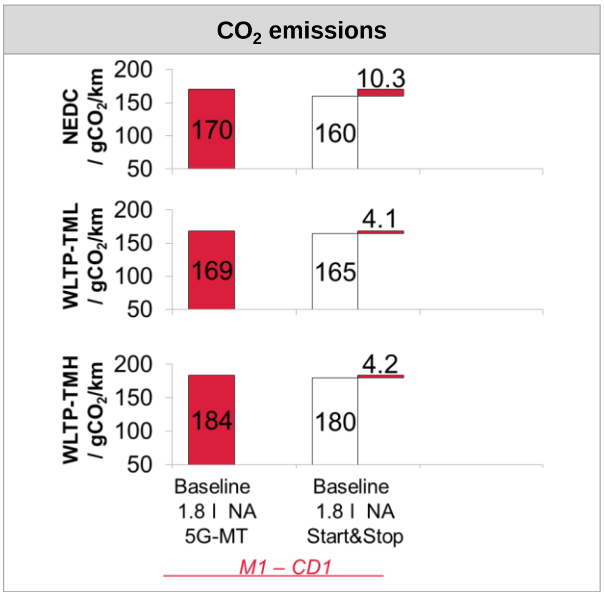
4.4.2.1 Baseline

The baseline variant in each segment is a naturally aspirated (NA) engine equipped with variable valve timing (VVT) on the intake and the exhaust side. In segment B and C a five speed manual transmission was selected, whereas in the segment D and E an eight speed automated transmission was used as base configuration. The specific power through all the segments rages is between 50 kw/l (segment B) and 60 kw/l segment E.

Despite the higher energy requirements of the WLTP cycles, lower CO_2 emissions might appear for NA engines within the WLTP TM_L than in the NEDC (see Figure 4.74). In case manual transmissions this is due to the more beneficial shift points available within the WLTP, which avoid the typically very low load operation points given by NEDC shift points. When using an automated transmission, as it was done in segment D and E, lower CO_2 emissions within the WLTP TM_L appear due to torque converter losses in the stopping phases of the driving cycles, which have a higher proportion within the NEDC.

4.4.2.2 Impact of start&stop

The Start&Stop technology allows to shut off the combustion engine in vehicle stop phases and start it again right before the next drive away. The idle fuel mass flow is completely avoided in this phases. Due to a higher absolute stopping time and even higher relative stopping time compared to the cycle duration the highest impact of this technology can be seen within the NEDC. For the investigated 1.8 l NA engine a benefit of 6 % can be achieved during the NEDC, whereas the fuel saving potential within the WLTP is around 2.5 %. In the CO_2 benefits through all simulated cycles can be seen. There is a clear trend of an increasing potential towards bigger vehicle segments and to bigger engine displacement due to the higher idle fuel mass flow. In D and E segment the idle fuel mass flow is additionally increased by the torque converter within the 8 speed automated transmission.



Impact of Start&Stop

Engine shut off in stopping phases during the cycle

- 6.0 % CO_2 reduction in NEDC
- 2.5 % CO_2 reduction in WLTP - TML
- 2.3 % CO_2 reduction in WLTP - TMH

	WLTP	NEDC
start&stop active time / s	190	206
cycle duration / s	1800	1180
start&stop rate / %	10.5	17.5

Figure 4.74 Impact of start&stop technology on the base line variant in segment C

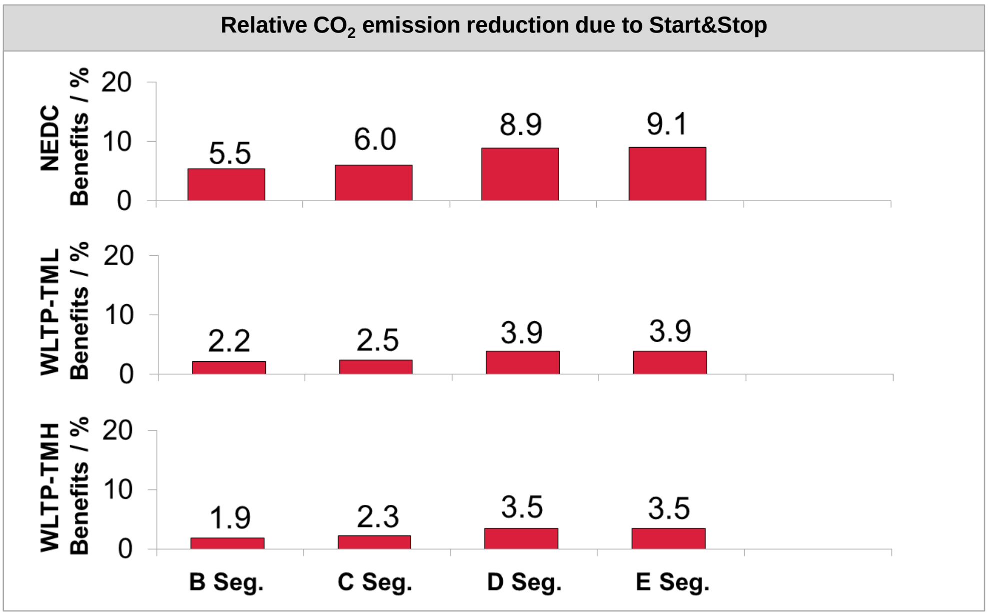
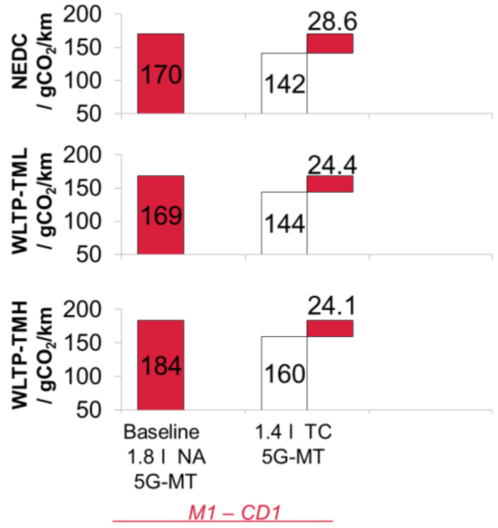


Figure 4.75 Segmentwise trend for percentage benefits of start&stop

4.4.2.3 Impact of Downsizing

In the first downsizing step the 1.8 l NA engine has been replaced by a 1.4 l TC engine. The downsizing degree of 22 % leads to moderate engine operation point shift to higher load. Additionally the increased torque curve of the turbo charged engine allows a downspeeding by a reduced final drive gear ratio while still keeping the same elasticity performance. In this case the final gear ratio can be decreased from 3.8 to 2.8. Both the downsizing and downspeeding result in a shift of the engine operation points towards higher loads and lower engine speeds and so to an area with higher efficiency shown in Figure 4.77. Due to the lower load and the fixed shift strategy this measure is more effective within the NEDC. In WLTP the shift strategy is not fixed for all engines like in NEDC but is calculated depending for the engine full load curve, the vehicle mass and the vehicle residence forces. It tends to operate each engine vehicle configuration at the lowest possible engine speed and so at relatively high loads. This circumstance reduces the potential of downsizing as well as the downspeeding effect compared to the NEDC. Figure 4.78 shows the CO_2 reduction for the other segments, where a similar behaviour for the B segment (downsizing grade of 23 %) can be seen. In D and E segment, where the downsizing grade is even slightly higher (25 % in D seg.; 33 % in E seg.), the CO_2 potential is lower because of the use of automated transmissions for these variants. The optimized shift strategy in the NEDC and the WLTC with automatic transmission allows in each case an engine operation near the BSFC optimum line and almost independent from the final gear ratio. The downspeeding effect as can be seen for the manual transmissions in B and C segment doesn't occur in the same way for the automated transmissions.

CO_2 emissions



Impact of downsizing

Downsizing: 1.8 l → 1.4 l

Downspeeding:
overall ratio in 5th gear from 3.8 → 2.8

- 16.8 % CO_2 reduction in NEDC
- 14.4 % CO_2 reduction in WLTP - TML
- 13.1 % CO_2 reduction in WLTP - TMH

Figure 4.76 Impact of downsizing from a NA to a TC engine in segment C

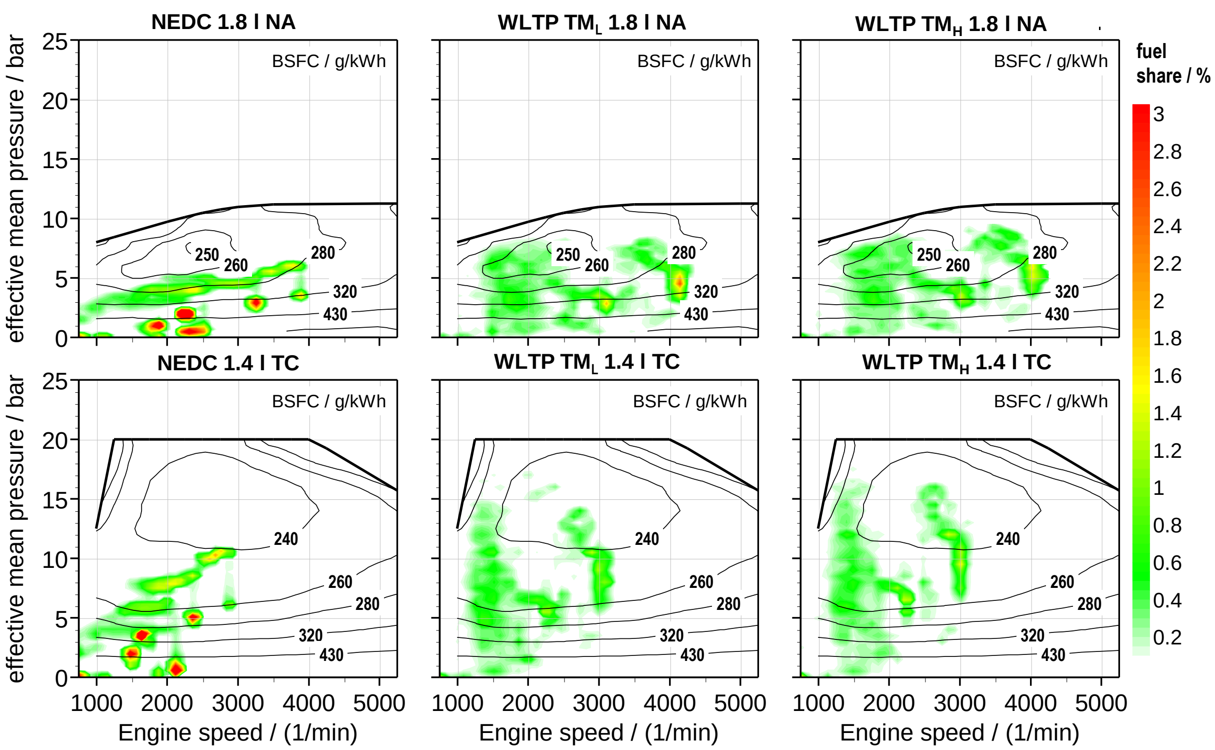


Figure 4.77 Fuel share distribution in NEDC and $WLTP_{TMH}$ for engine downsizing in segment C

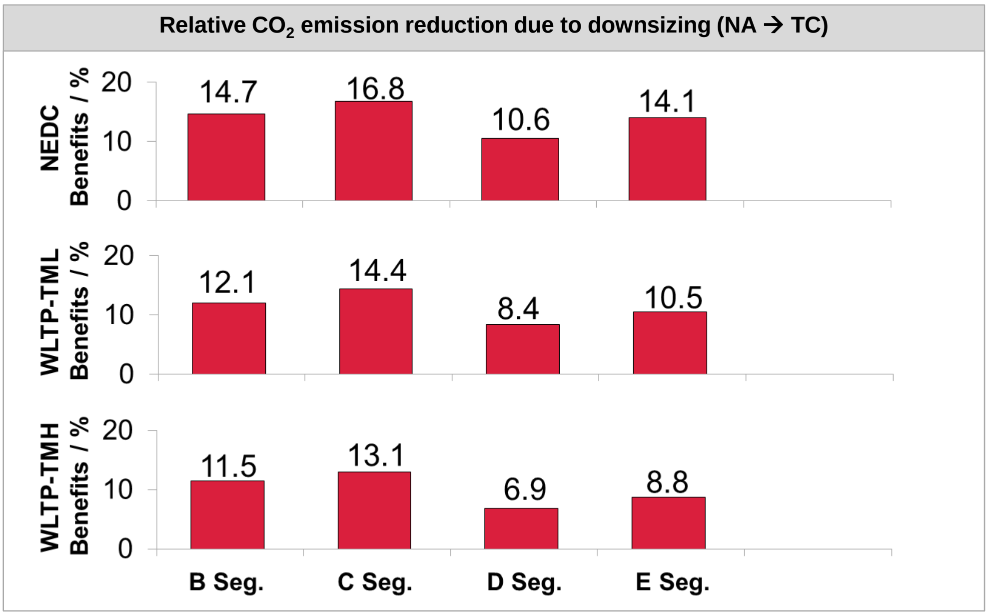
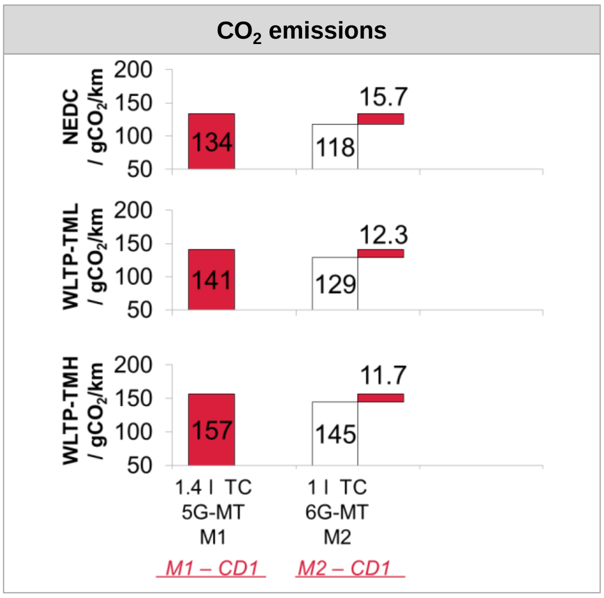


Figure 4.78 Segmentwise percentage benefits of downsizing from a NA to a TC engine

4.4.2.4 Impact of a downsizing, transmission and weight package

The second downsizing step to a 1 l TC engine goes along with an upgrade from a 5 speed to 6 speed manual transmission and a vehicle weight reduction of 10 %. However, and even though the 1 l TC engine reaches the demanded power of 95 kW, the torque at low engine speed is below the one of the 1.4 l TC variant due to the achievable torque with a single stage turbo charger (reduced low-end torque). To keep the same climbing performance in the highest gear on the motorway the final gear ratio must be increased from 2.8 to 3.1. Due to the demand for the gradeability in the first gear and the maximal gear ratio step from gear to gear, the total gear spread of the transmission must be increased and this leads to a demand of higher gear number with downsized engines. As a result of this measure the engine speeds in the NEDC and the WLTC are slightly increased (see Figure 4.80), which has a negative impact on the CO_2 emissions. Additionally, the potential of a further downsizing is reduced compared to first downsizing step from a 1.8l NA engine to a 1.4l TC engine due to the stronger impact of the knock limitation with the smaller displacement and the lower BSFC benefit for the load point shift at higher loads. Therefore, despite the mass reduction of 10 % the overall CO_2 benefit in this step is lower compared to the downsizing step before.

The results for the B segment in Figure 4.81 can be explained in the same way as for the C segment example. In D and E segment the transmission was upgraded from an 8 speed automatic transmission to a 10 speed DCT, where additionally the torque converter losses could be avoided and therefore the transmission efficiency is strongly improved at driveaway and low vehicle speed.



Impact of downsizing + transmission + weight

Downsizing: 1.4 l → 1 l

Transmission: 5G-MT → 6G-MT

Overall ratio in last gear: 2.8 → 3.1

Weight reduction: M1 → M2 (-10 %)

- 11.7 % CO_2 reduction in NEDC
- 8.7 % CO_2 reduction in WLTP - TML
- 7.5 % CO_2 reduction in WLTP - TMH

Figure 4.79 Impact of downsizing, transmission upgrade and weight reduction in segment C

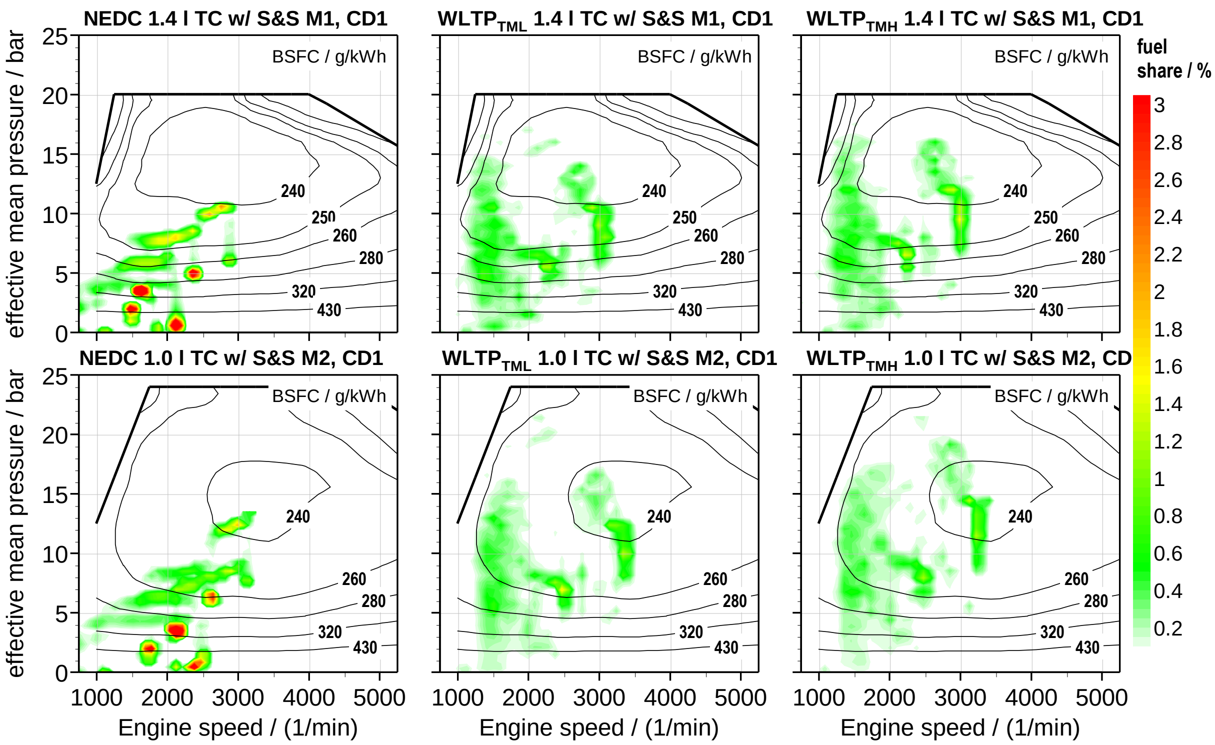


Figure 4.80 Fuel share distribution in NEDC and $WLTP_{TMH}$ for engine downsizing, transmission upgrade and weight reduction in segment C

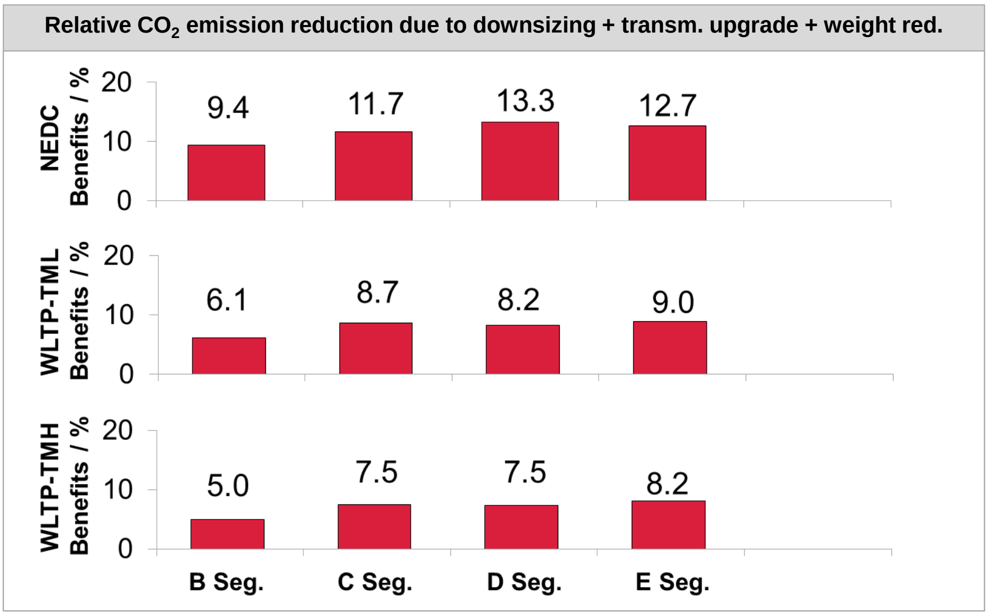


Figure 4.81 Segmentwise percentage benefits of downsizing, transmission upgrade and weight reduction

4.4.2.5 Impact of mass reduction

Figure 4.82 presents a mass variation of the C segment 1 l TC variant with a 6 speed manual transmission. The results show a nearly linear relationship. A mass reduction of 10 % in the first step leads to a CO_2 benefit of around 5 % through all investigated cycles. The 20% weight reduction in a second step also doubles the CO_2 emission reduction.

It is to be mentioned that the slightly higher percentage benefit within the NEDC is caused by final drive adjustment due to lower torque demand for gradeability, since in the D and E segment, where the automated transmission was used, the opposite effect occurs.

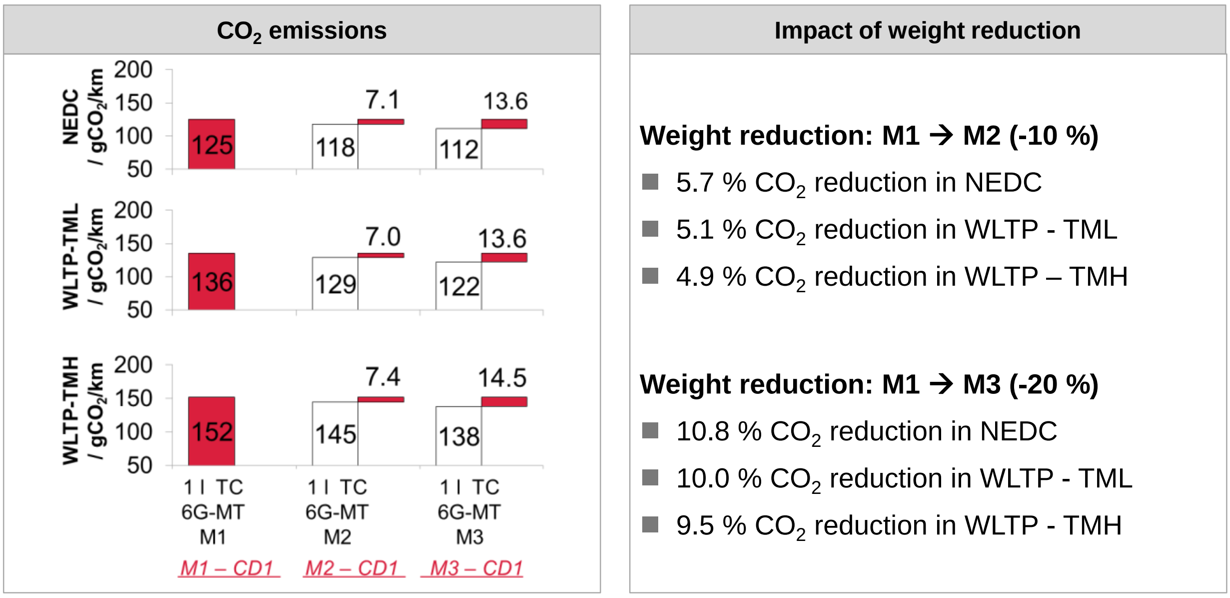


Figure 4.82 Vehicle weight variation in segment C

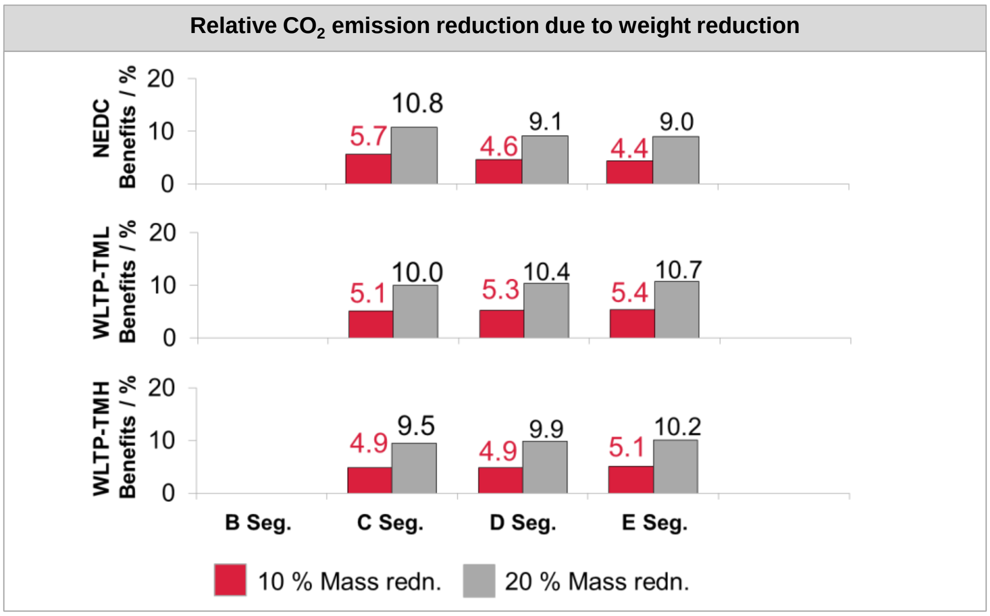


Figure 4.83: Segmentwise percentage benefits of weight reduction

4.4.2.6 Impact of rolling resistance and drag resistance

The variation of the rolling and drag resistance was performed with the C segment 1 l TC variant with a 6 speed manual transmission and the second mass reduction step (M3). The results depicted in Figure 4.84 show a clear tendency of a higher impact towards the cycles with higher vehicle speeds and higher vehicle test masses.

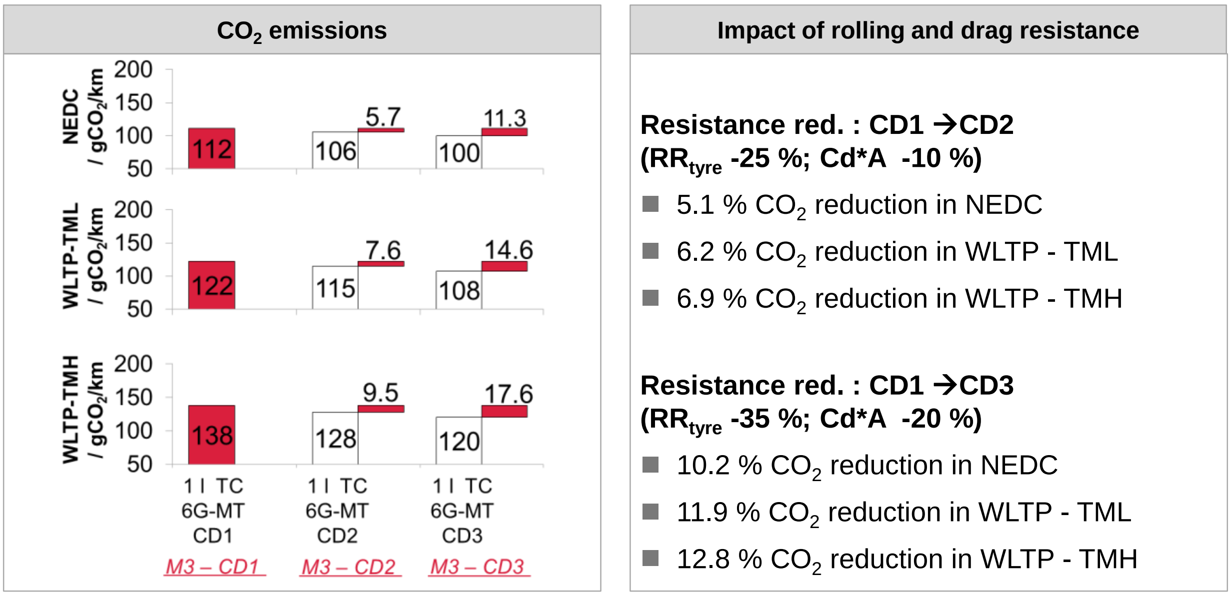


Figure 4.84 Vehicle resistance variation in segment C

For all segments it can be seen in Figure 4.85, the first reduction is giving slightly more relative benefits than the second reduction.

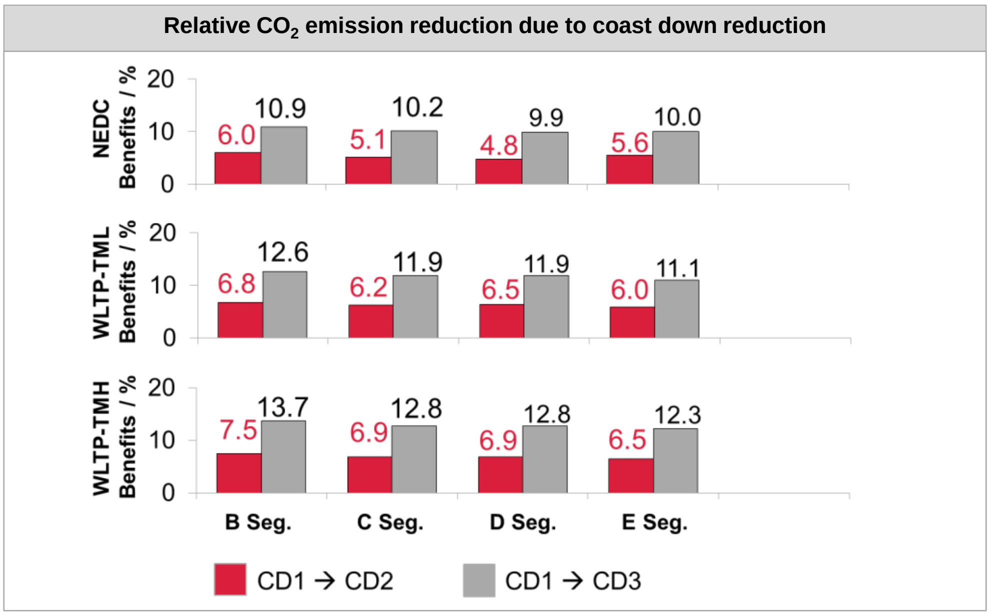


Figure 4.85: Segmentwise percentage benefits of coast down reduction

4.4.2.7 Impact of transmission upgrade

In the C segment an additional transmission upgrade was simulated, where the 6 speed manual transmission (MT) was replaced by a 7 speed automated dual clutch transmission (DCT). The results in Figure 4.86 show a much higher impact in the NEDC than in the WLTP cycles. The optimized shift strategy, which is given for all simulated cycles when using an automated transmission, allows in each case an engine operation near the BSFC optimum line. As can be seen in Figure 4.87 the much more disadvantageous shift points with MT which have to be used within the NEDC, results in a strong improvement regarding the engine operation points when using a DCT. Since the MT shift strategy for the WLTP considers the given full load curve and vehicle properties with the tendency to operate already at lower speeds and higher loads, the same transmission upgrade results in a much lower benefit compared to the NEDC.

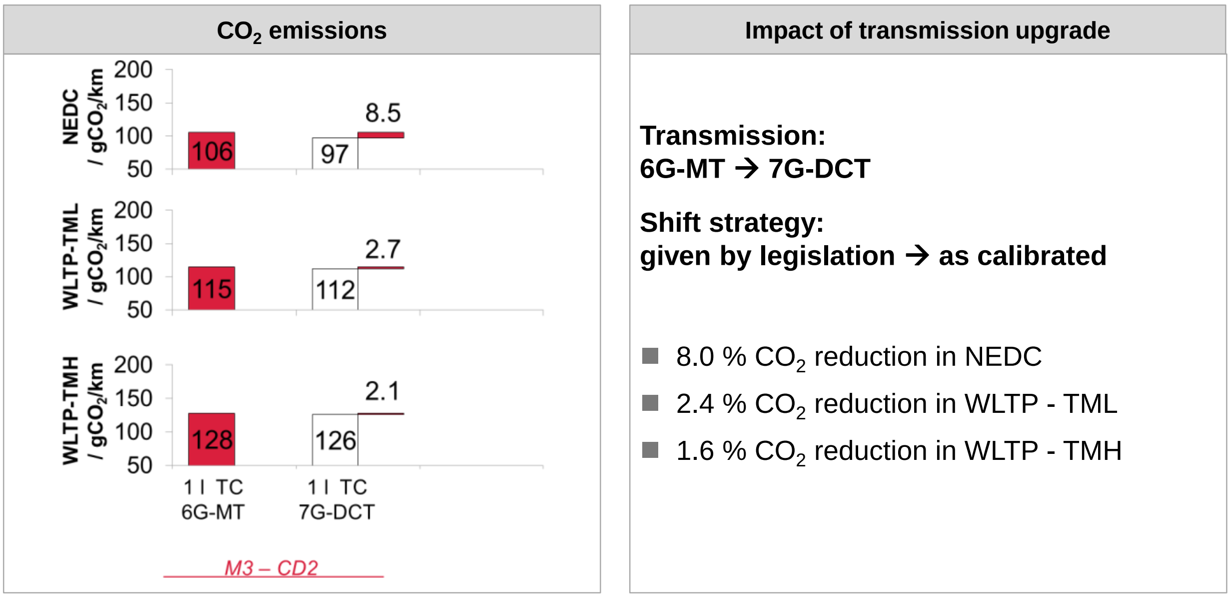


Figure 4.86 Impact of transmission upgrade from 6 speed MT to 7 speed DCT in segment C

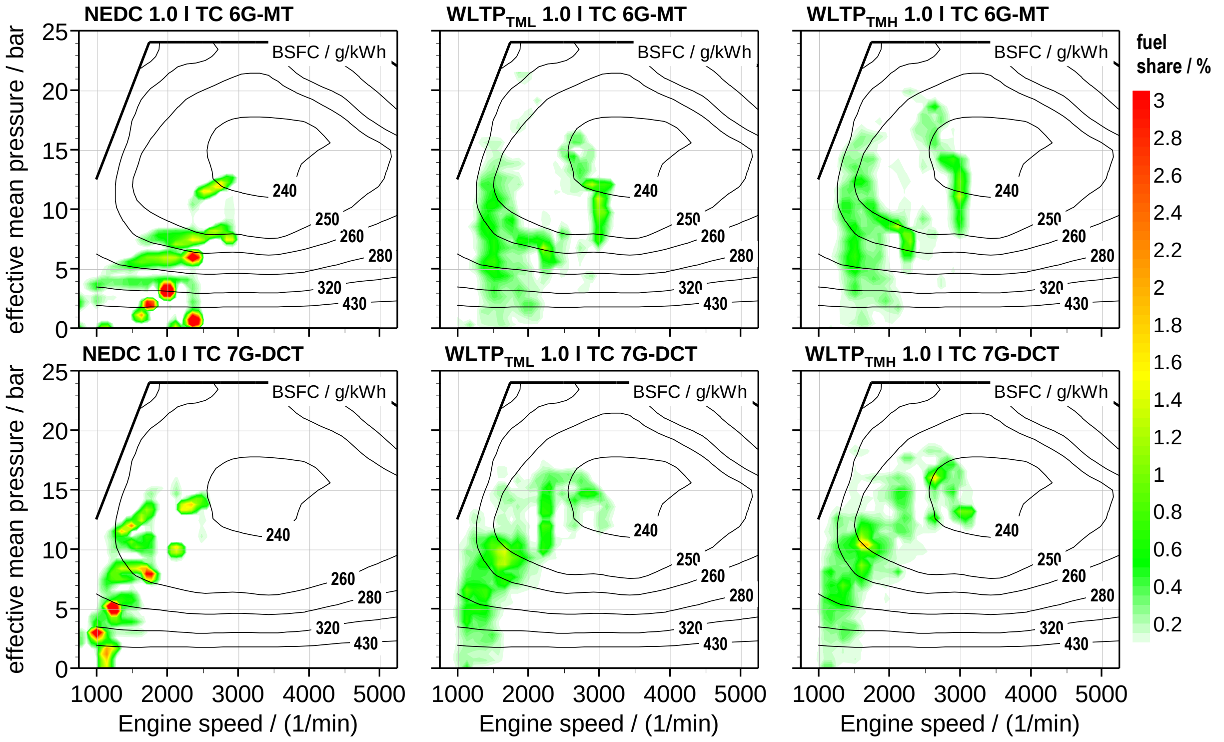


Figure 4.87 Fuel share distribution in NEDC and WLTP$_{TMH}$ for transmission upgrade (6 MT → 7 DCT) in segment C

4.4.2.8 Impact of friction reduction

In terms of friction reduction three measures were applied. The first one is an engine friction reduction by 20%, which simply was realized by a factor of 0.8 multiplied with the base FMEP map. This factor covers the potential achievable by base engine measures (e.g. camshaft roller bearing, optimized main bearing layout, crankshaft off-set, etc.). The influence on the BSFC map is depicted in Figure 4.90, where at almost the same engine operation points the break efficiency increases. As a second modification a split cooling system was applied. The effect of this system is a faster engine warm-up and so a reduced engine friction over the entire driving cycle. The third measure was the implementation of an electrical water pump. Even though the required energy to supply the electrical pump must be generated at the cost of lower conversion efficiency, this can also be actuated on demand. Therefore, the strategy may be optimized to optimize the process reaching an average friction reduction.

In Figure 4.89, in the upper diagram the coolant temperature curve, the oil temperature curve and the friction trace are shown for the NEDC with and without a split cooling system and an electrical water pump. The same plot is given for the WLTP-high cycle in the bottom diagram. It can be seen that in the NEDC the split cooling affects almost the entire cycle duration, whereas during the WLTP-high cycle only about 60 % of the cycle duration are influenced by this measure, due to the smaller share of engine war-up time. This fact explains the slightly higher benefit within the NEDC (Figure 4.88). Similar results occur for the other simulated segments represented in Figure 4.91.

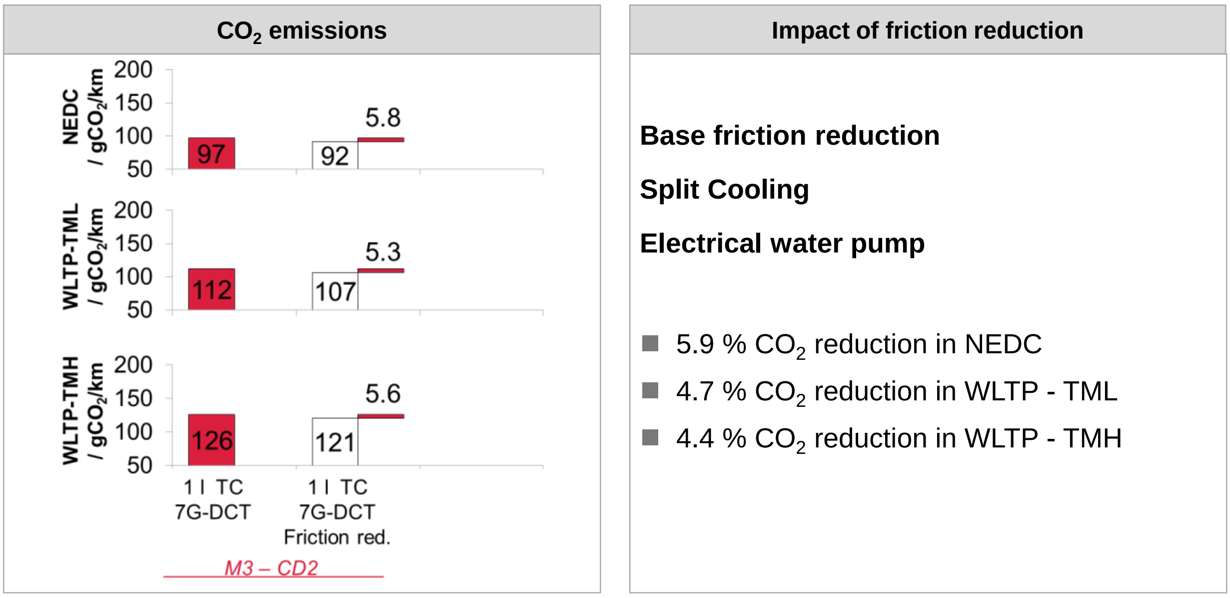


Figure 4.88 Impact of friction reduction in segment C

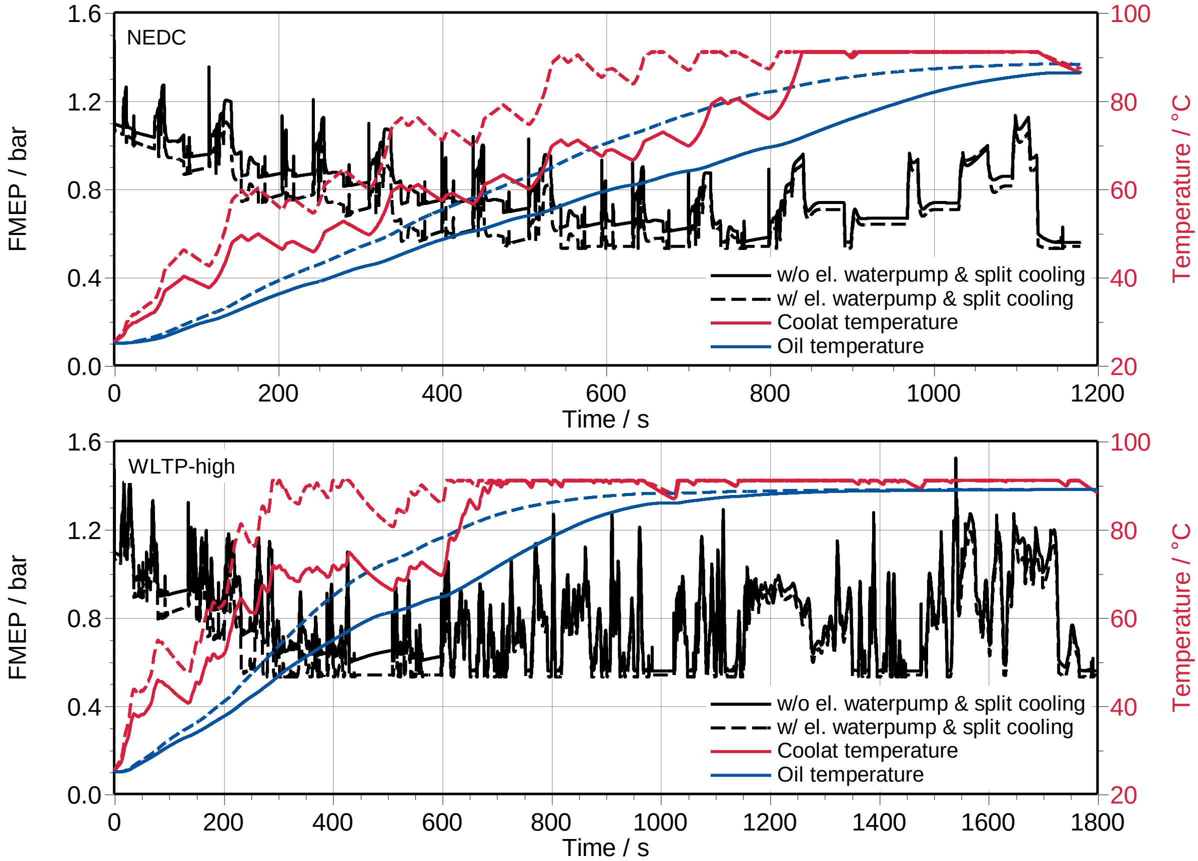


Figure 4.89 Temperature and FMEP traces in NEDC and $WLTP_{TMH}$ w/ and w/o el. water pump and split cooling in segment C

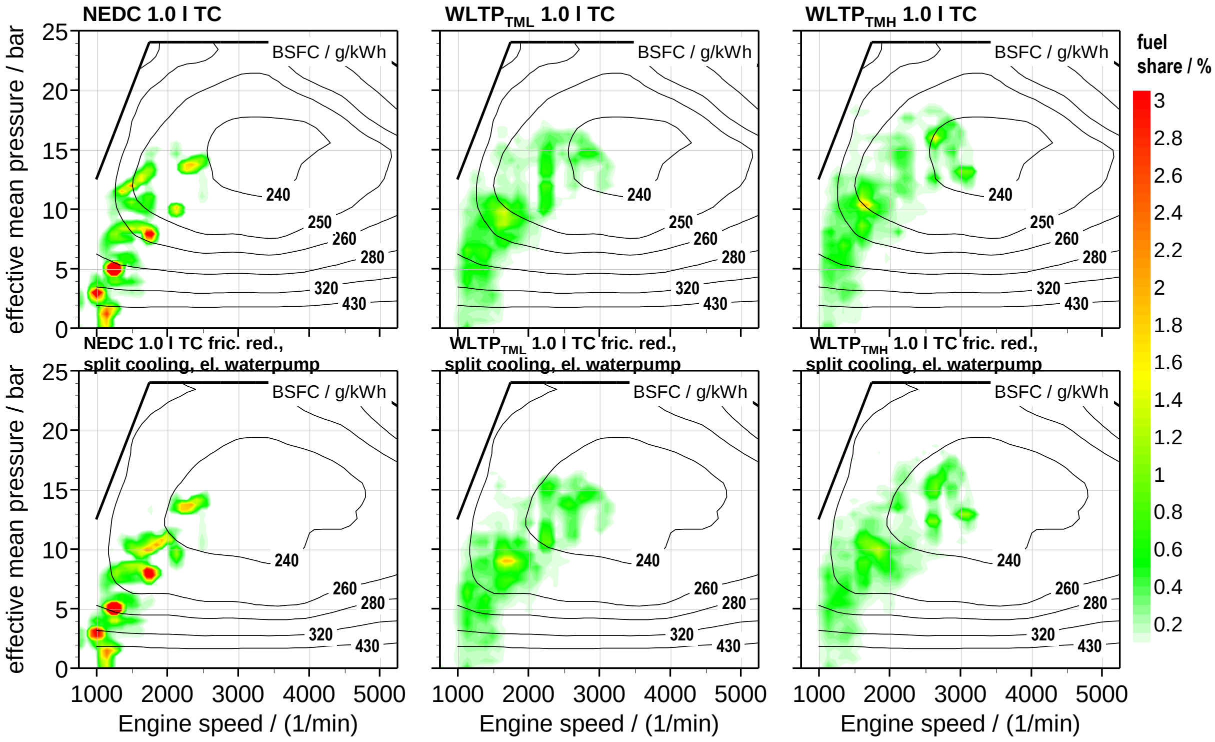


Figure 4.90 Fuel share distribution in NEDC and WLTP$_{TMH}$ for friction reduction in segment C

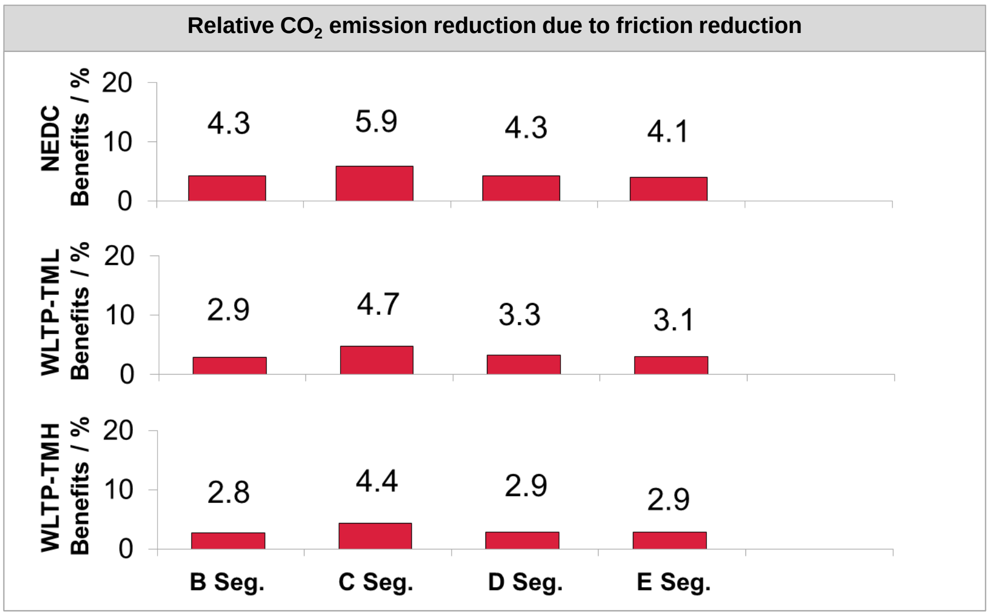


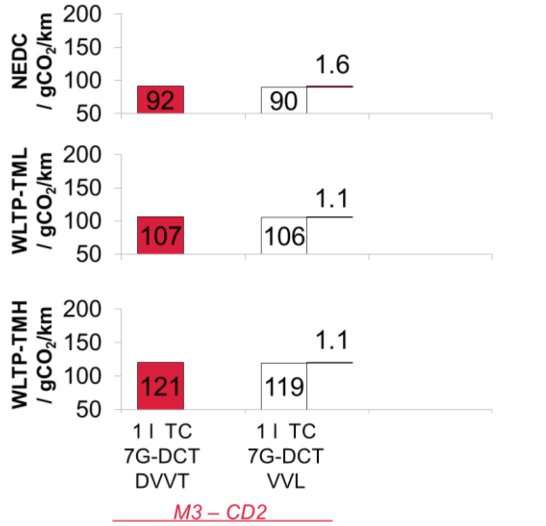
Figure 4.91 Segmentwise percentage benefits of friction reduction measures

4.4.2.9 Impact of variable valve lift

The variable valve lift on the intake side is an engine technology to dethrottle the engine in lower partload. This leads to lower gas exchange losses due to early intake valve closing and therefore reduced fuel consumption in low part load operation. For this variant the simulation model stay the same but the engine map was modified to represent the effect of intake VVL.

In Figure 4.93, the maps and the operation areas in the driving cycles of the two compared variants are shown. The impact of the VVL technology can be seen on the improved BSFC contours in the bottom left corner of the VVL maps. In Figure 4.94 the effect of VVL for the various segments is shown. With increasing engine displacement to vehicle weight ratio, the effect of fuel consumption measures that affects the lower part load increases. Therefore the fuel consumption benefit increases slightly to bigger vehicle segments.

CO_2 emissions



Impact of variable valve lift

Effective in low load areas by reducing throttling losses

- 1.8 % CO_2 reduction in NEDC
- 1.0 % CO_2 reduction in WLTP - TML
- 0.9 % CO_2 reduction in WLTP - TMH

Figure 4.92 Impact of VVL technology in segment C

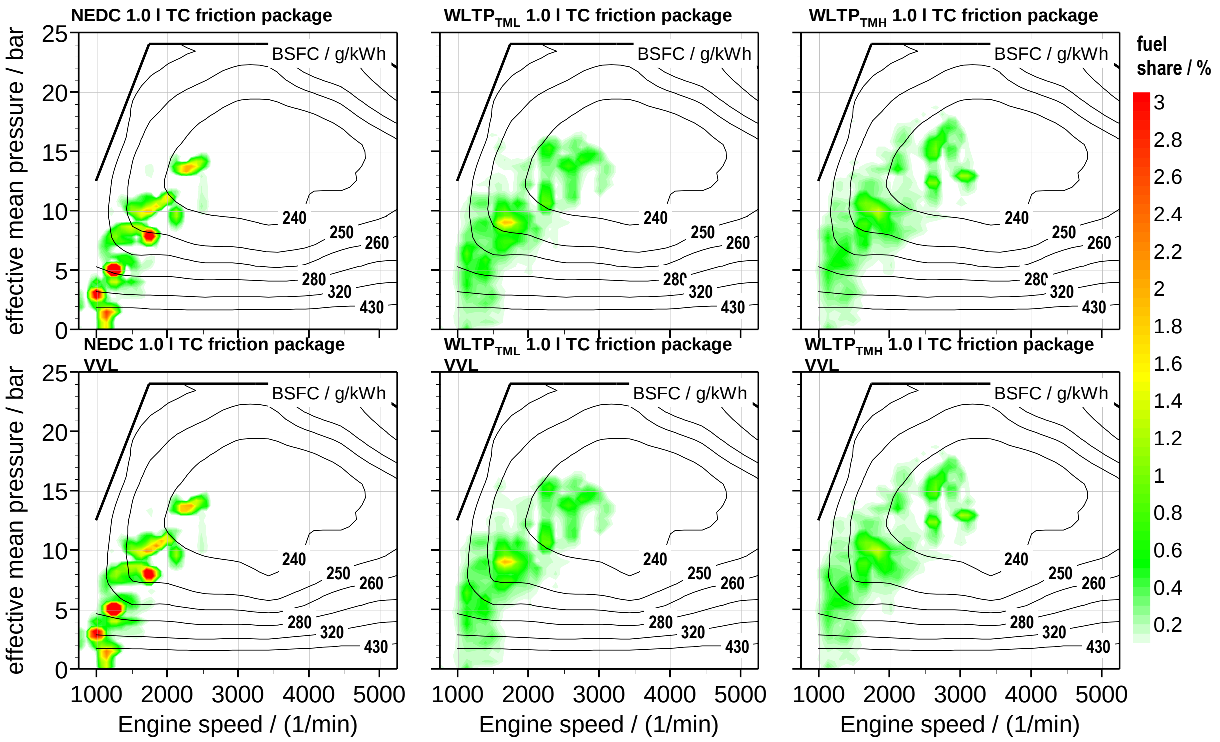


Figure 4.93 Fuel share distribution in NEDC and $WLTP_{TMH}$ for VVL technology in segment C

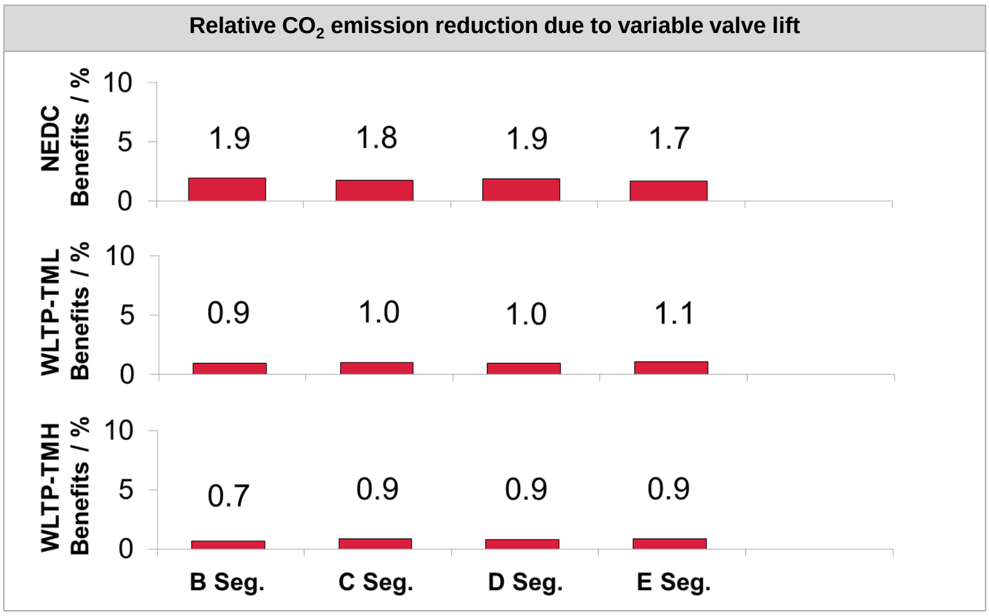


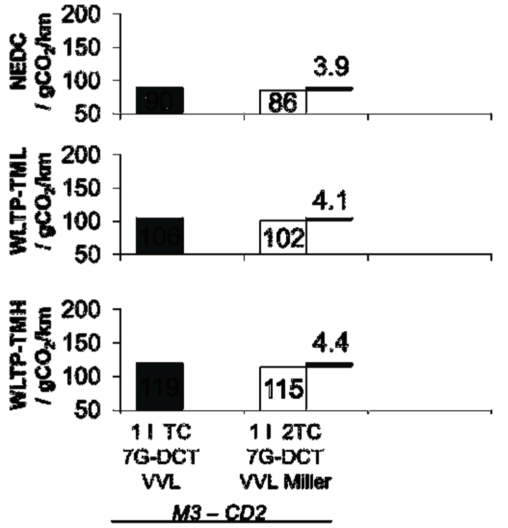
Figure 4.94 Segmentwise percentage benefits of VVL technology

4.4.2.10 Impact of Miller cycle

The Miller Cycle affects the entire engine map, as can be seen by the BSFC contours in Figure 4.96. Realized by variable valve timing with early or late intake valve closing, the effective compressing ratio can be reduced. This leads to a lower pressure and temperature in the end of the compression stroke and reduces the knock tendency of the engine. Due to that the geometric compression ratio can be increased to have equal pressure and temperature like the base engine. This leads to an increased expansion ratio and improved fuel consumption. Here the compression ratio was increased from 10 to 12, which mainly affects the low load area of the engine map. In the medium load area an additional efficiency increase occurs by outsourcing a part of the compression to the external charger to have the same air mass in the cylinder like the base engine. Due to the fact that the air coming out of the compressor passes the intercooler, for the same load point the same amount of air has a lower peak compression pressure and so a lower peak compression temperature. Therefore, the knock tendency is reduced and so a more efficient ignition timing becomes possible.

As can be seen in Figure 4.96, the efficiency increase in the part load area has a slightly stronger impact than the knock tendency reduction effect in the higher loads. For all investigated segments the implementation of the Miller cycle shows slightly higher benefits in the more part load dominated NEDC.

CO_2 emissions



Impact of Miller cycle

Low load and medium non charged area:
Efficiency increase due to a longer expansion stroke than compression stroke and compression ratio increase (10 → 12, to keep a similar effective compression ratio)

Near NA full load and medium charged area:
Efficiency increase from reduced knocking tendency by moving part of the compression stroke work to the compressor after which the charged air is cooled by passing the intercooler

High charged area:
Fast efficiency decrease due to higher charging effort

- 4.4 % CO_2 reduction in NEDC
- 3.8 % CO_2 reduction in WLTP - TML
- 3.7 % CO_2 reduction in WLTP - TMH

Figure 4.95 Impact of Miller cycle in segment C

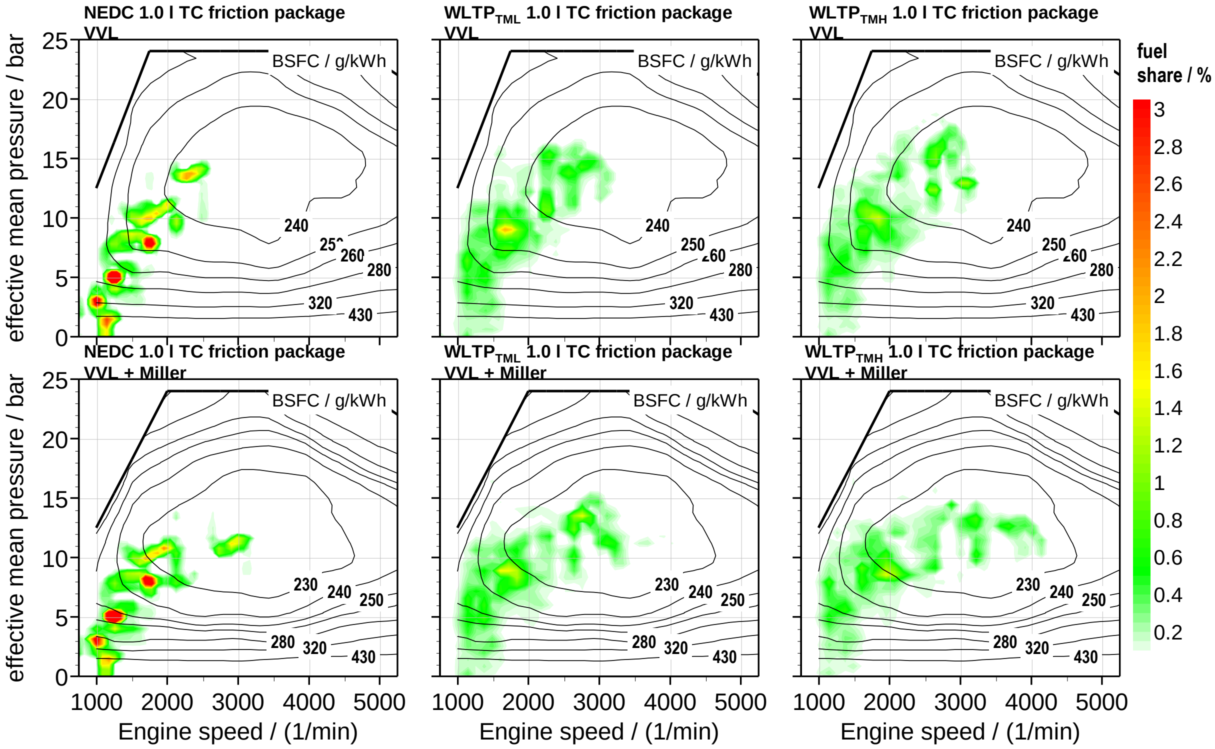


Figure 4.96 Fuel share distribution in NEDC and WLTP$_{TMH}$ for VVL technology and additionally Miller cycle in segment C

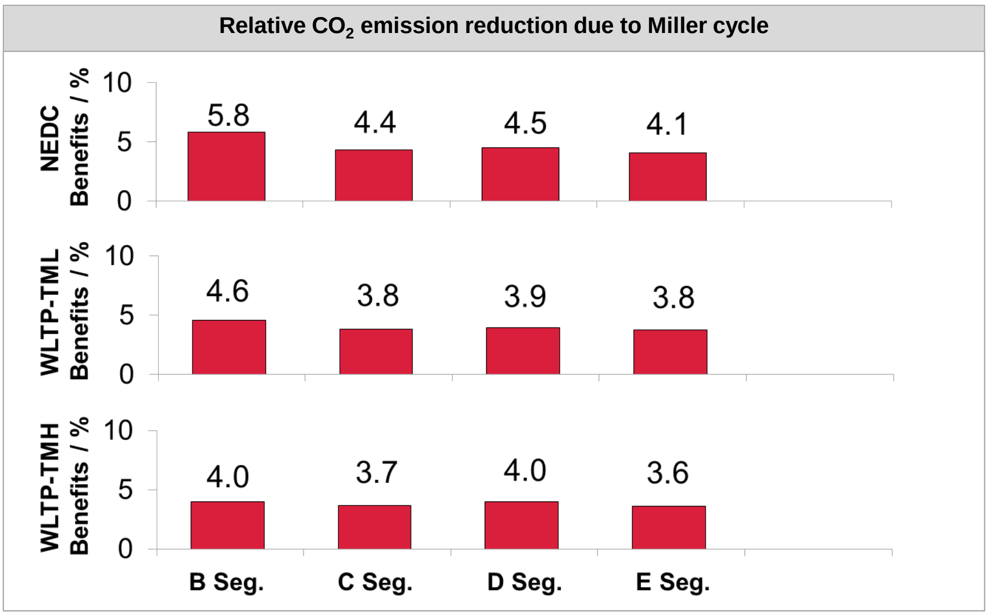


Figure 4.97 Segmentwise percentage benefits of Miller cycle added to the VVL variant

4.4.2.11 Impact of adding EGR and 2-step VCR to a VVL concept

As the Miller cycle, the combination of 2 step variable compression ratio (VCR) and low pressure exhaust gas recirculation (LP-EGR) also affects the entire engine map. The influence on the fuel consumption is shown by the BSFC contours in Figure 4.99.

In low load areas the increased compression ratio has a high impact on the combustion efficiency. Whereas, at higher loads, a lower compression ratio allows earlier ignition timing. In knock limited areas the combustion retarding effect of LP-EGR leads to a further ignition advance and therefore a further increase of efficiency

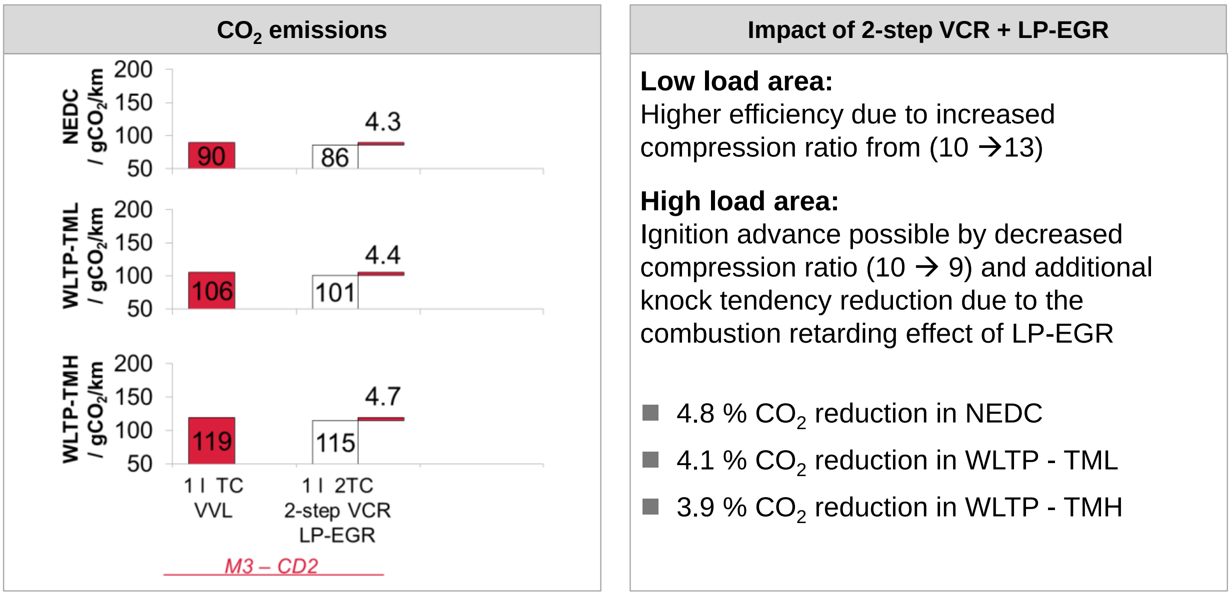


Figure 4.98 Impact of two step VCR and low pressure EGR added to the VVL variant in segment C

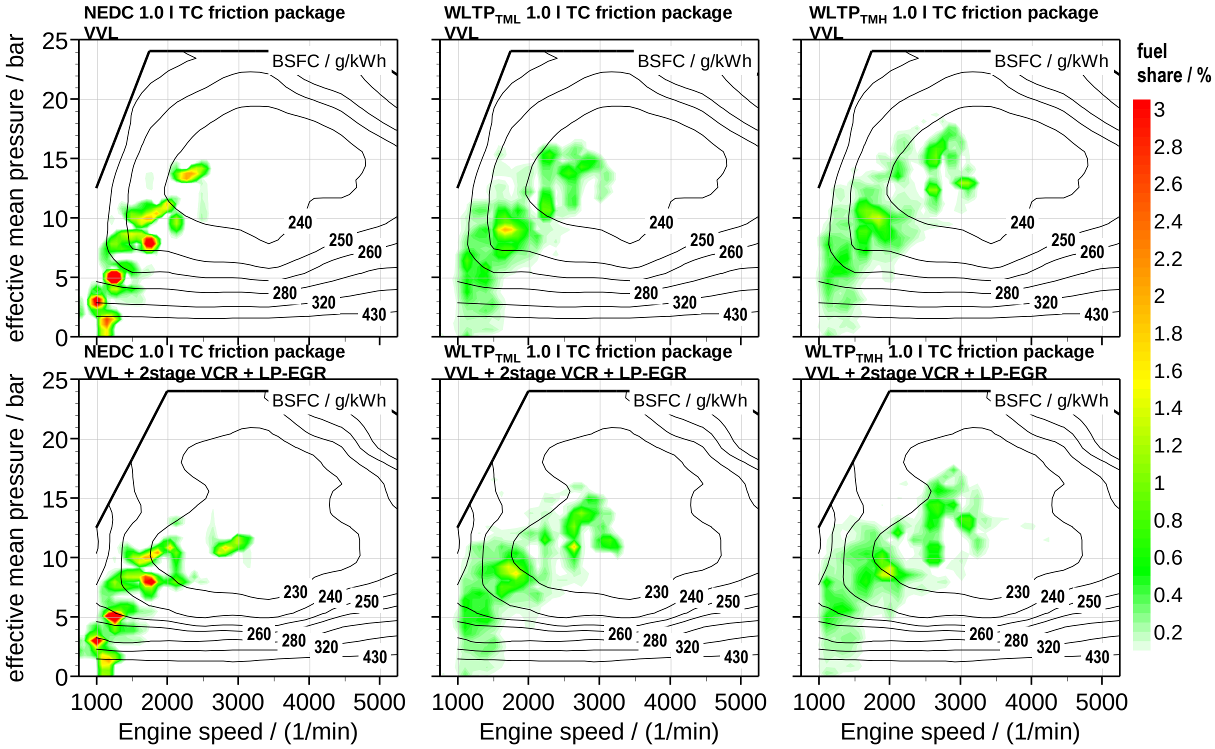


Figure 4.99 Fuel share distribution in NEDC and $WLTP_{TMH}$ for VVL technology and additionally two step VCR and low pressure EGR in segment C

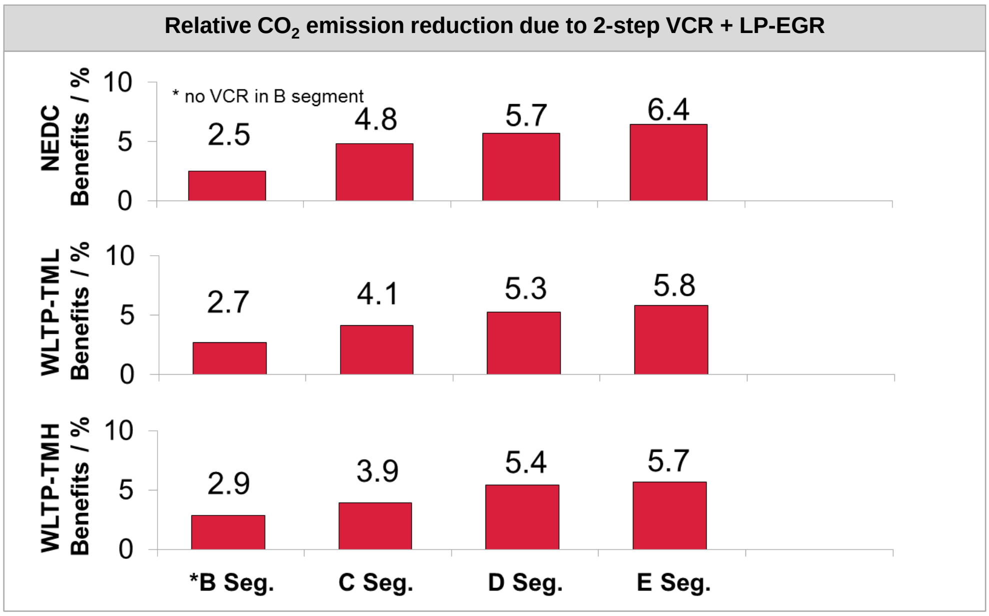
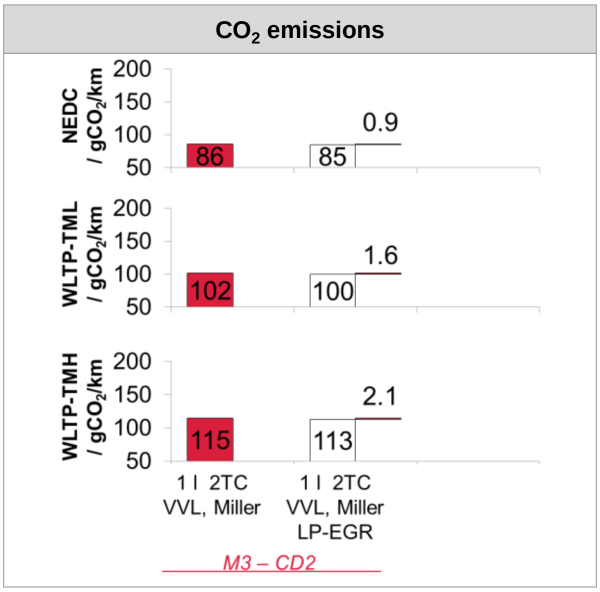


Figure 4.100 Segmentwise percentage benefits of two step VCR and low pressure EGR added to the VVL variant

4.4.2.12 Impact of adding low pressure EGR to a Miller concept

Adding low pressure EGR to the Miller cycle concept leads to further knock tendency reductions and so efficiency improvement in high engine load areas (depicted in Figure 4.102). Due to this fact, also a further increase of the compression ration from 12 to 12.5 could be realized affecting additionally the low load operation area slightly. Anyway, a clear trend of higher benefits with in the WLTC can be seen in for segment C and also for the other segments in .



Impact of LP-EGR on Miller cycle

High load area:
Additional knock tendency reduction due to the combustion retarding effect of LP-EGR.

- 1.0 % CO_2 reduction in NEDC
- 1.6 % CO_2 reduction in WLTP - TML
- 1.9 % CO_2 reduction in WLTP - TMH

Figure 4.101 Impact of low pressure EGR added to the Miller variant in segment C

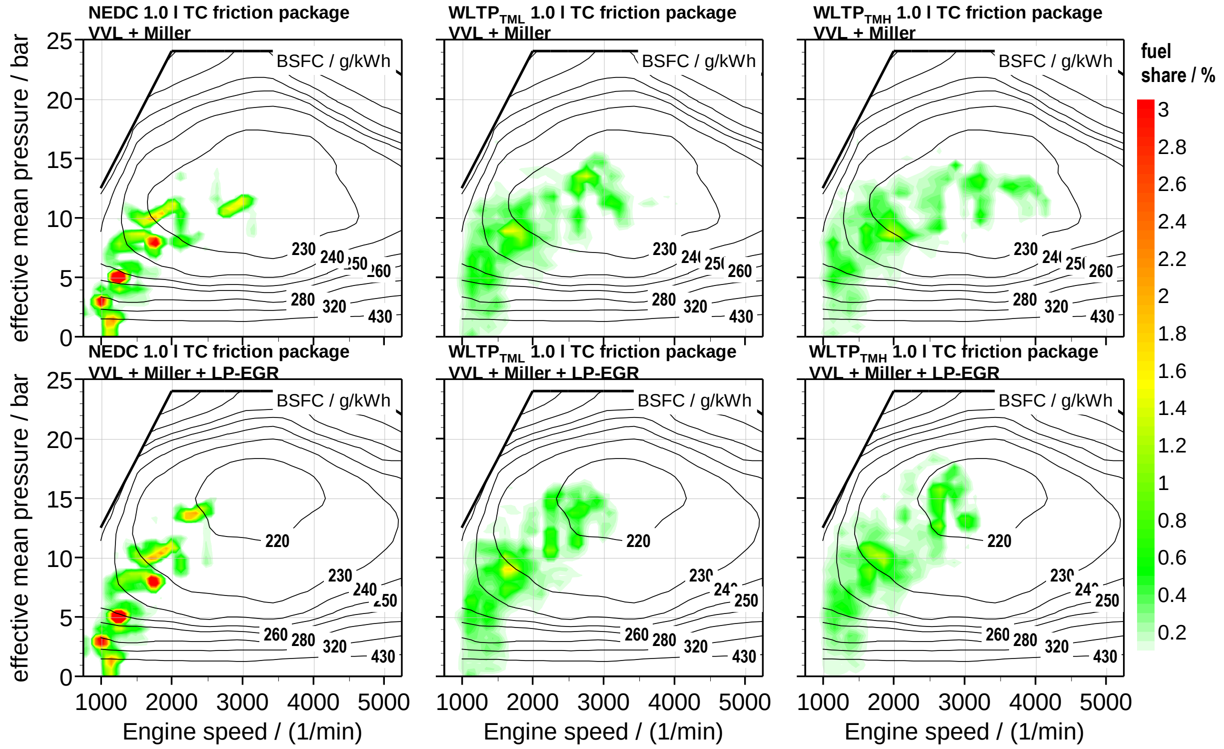


Figure 4.102 Fuel share distribution in NEDC and WLTP$_{TMH}$ for VVL+Miller technology and additionally low pressure EGR in segment C

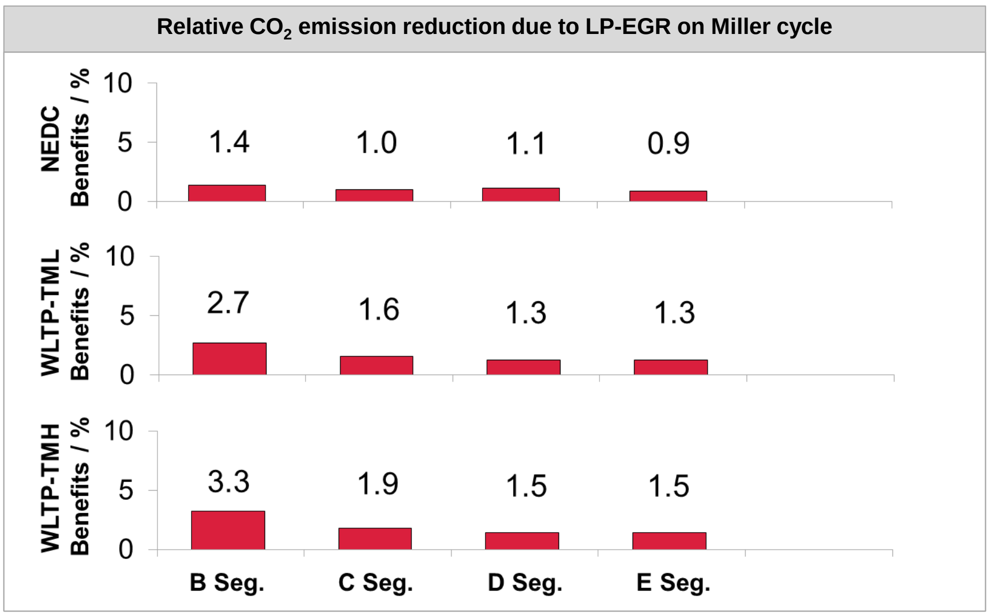


Figure 4.103 Segmentwise percentage benefits of low pressure EGR added to the Miller variant

4.4.2.13 Impact of extreme downsizing

The extreme downsizing was realized by increasing the mean effective pressure up to 27 bar BMEP in segment C and even 35 bar BMEP in segment D and E. To sustain the higher cylinder pressure level, a stronger crank train has to be applied, which directly goes along with an increase of friction losses. Also the thermodynamic efficiency decreases, due to the lower volume to surface ratio of the combustion chamber causing higher wall heat losses. With respect to the high boosting level, even without Miller cycle, a two stage boosting system became necessary as well as a compression ratio reduction from 12.5 to 11. Together, all this affects lead to worse efficiencies within the entire engine map, which can be seen in Figure 4.105.

The shift of the engine operation points towards higher load doesn't compensate the described disadvantages, which actually leads to penalties regarding the CO_2 emissions through all the investigated segments and driving cycles as shown in Figure 4.104.

There is a trend of lower disadvantages from WLTP TM_H to NEDC, due to the lower average engine load level within the NEDC and so a higher impact of operation point shifting. The same trend can be seen from segment C to segment E. Here, the higher power to weight ratio, which usually increases towards higher vehicle classes, leads to lower average engine load levels and so a higher impact of operation point shifting, too.

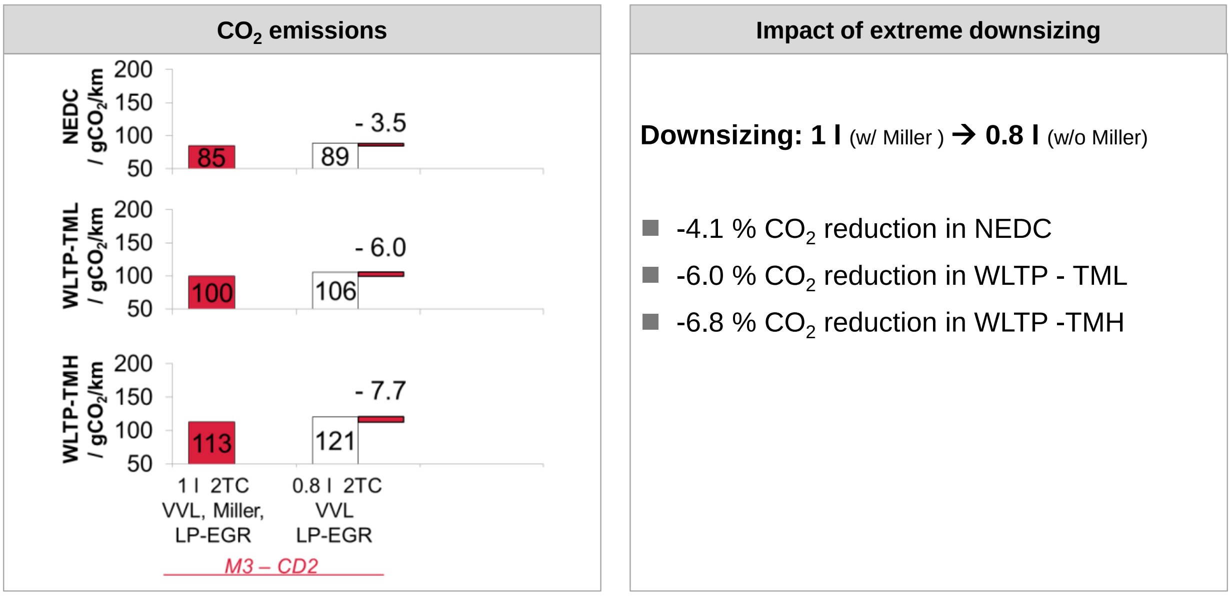


Figure 4.104 Impact of extreme downsizing in segment C

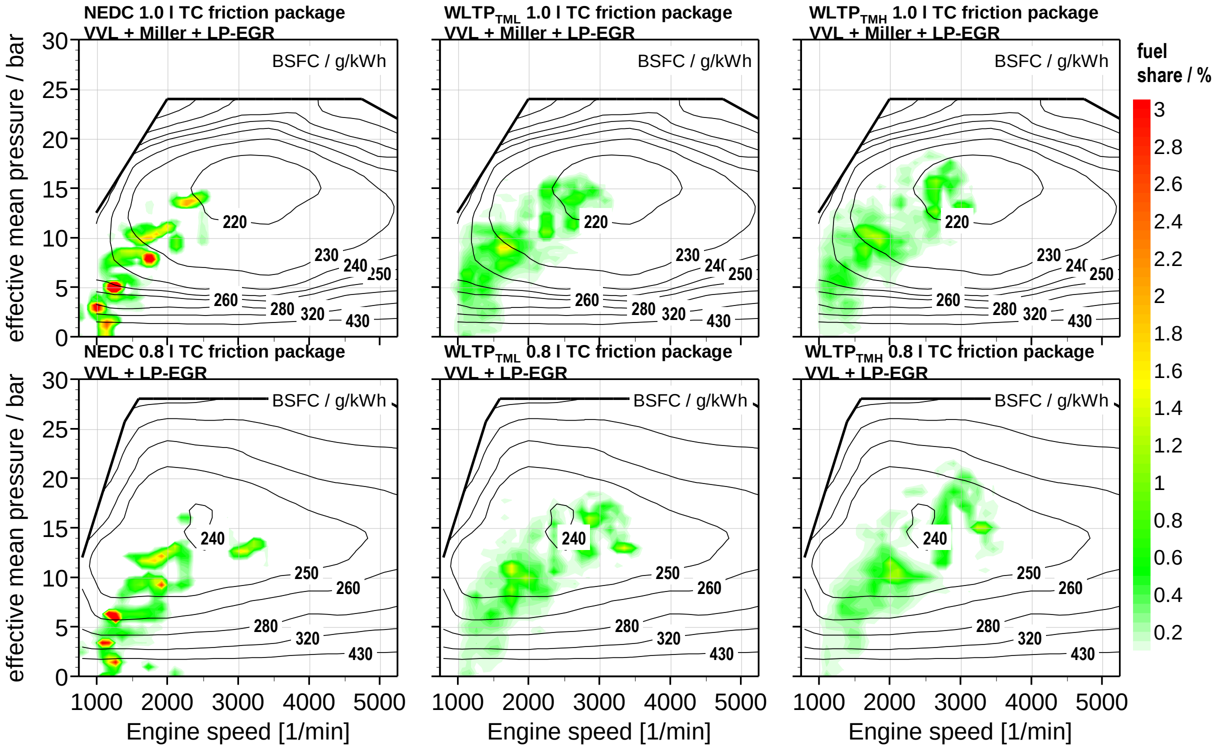


Figure 4.105 Fuel share distribution in NEDC and WLTP$_{TMH}$ for VVL+Miller+LP-EGR technology and extreme downsizing with VVL+LP-EGR in segment C

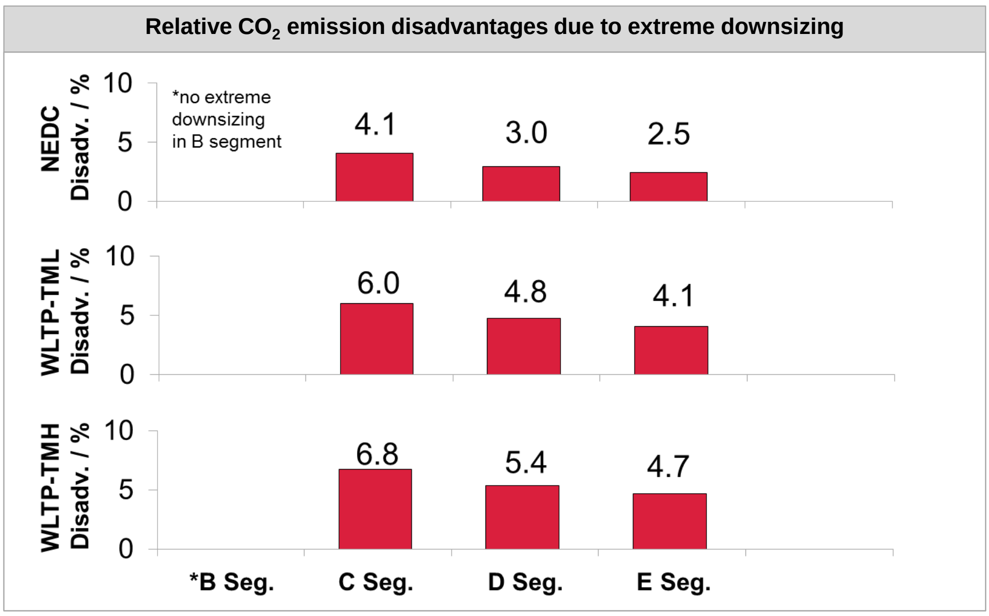


Figure 4.106 Segmentwise percentage disadvantage of extreme downsizing

4.4.2.14 *Impact of P0 hybridization*

As shown in the results below, the benefits that are possible to achieve with the P0 hybridization is appreciably higher in the WLTP-Low cycle (for the 1l Miller variant also in the WLTP-High cycle).

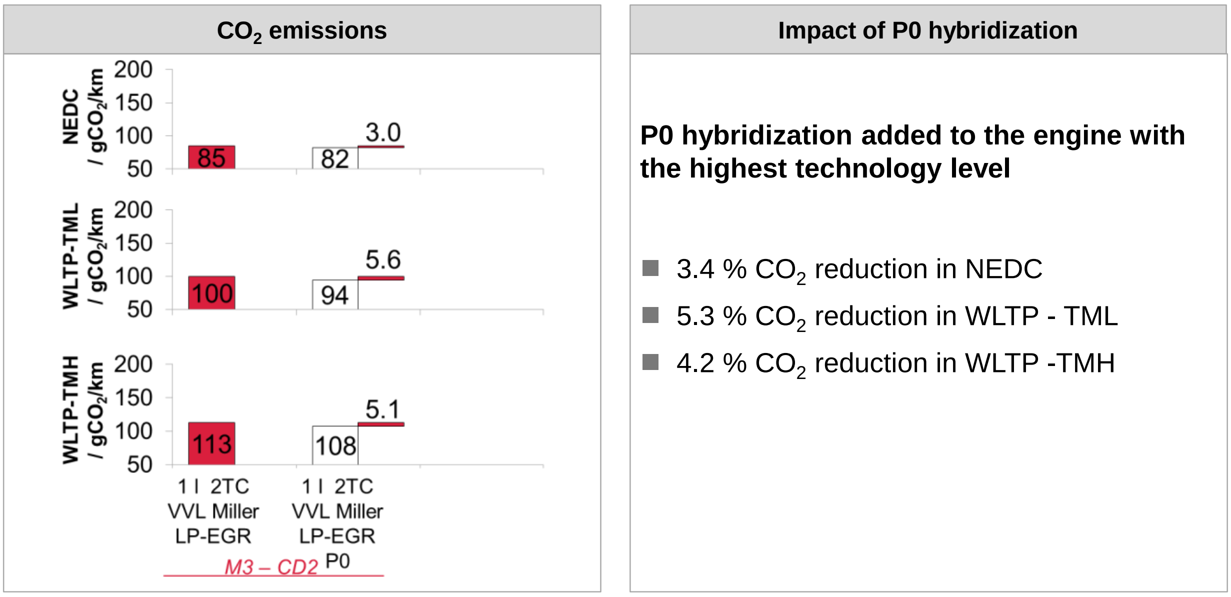


Figure 4.107 Impact of P0 hybridization added to the 1 l 2stg.-TC VVL+Miller+LP-EGR variant in segment C

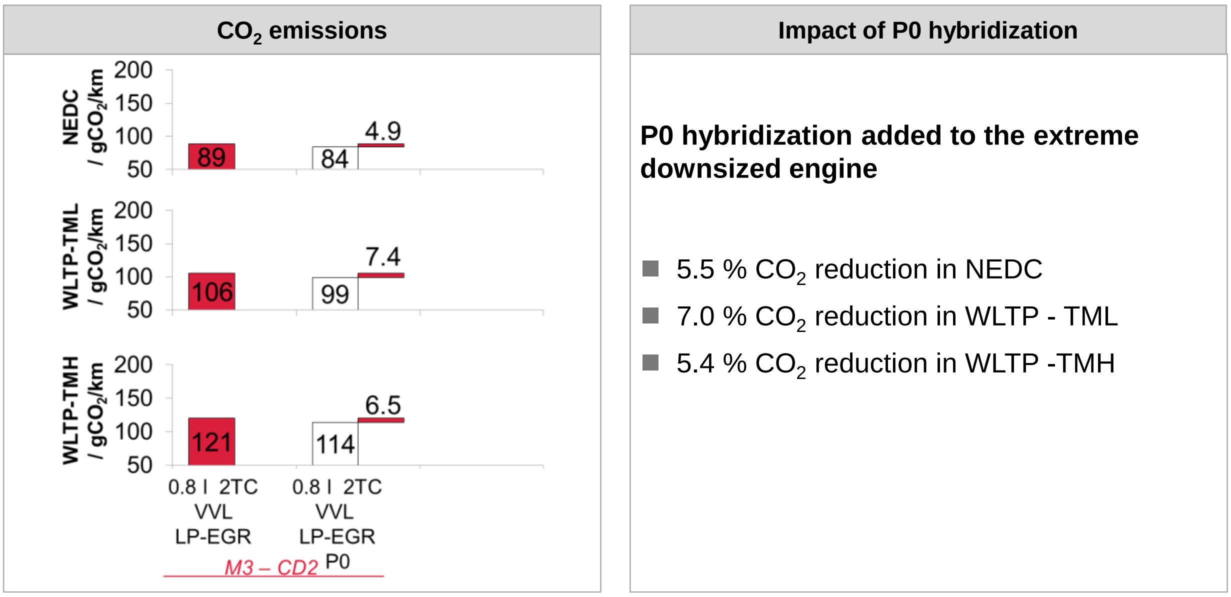


Figure 4.108 Impact of P0 hybridization added to the 0.8 l 2stg.-TC VVL+LP-EGR variant in segment C

This difference is mainly due to the given speed profile of the cycle. A dynamic cycle like the WLTP presents more and stronger breaking phases and so more recoverable energy is available. The P0 hybrid strategy then operates a load point's shift towards higher load and lower speed, so that the generator is able to apply the available energy to boost and to get the operation points closer to the BSFC optimum line. This can be clearly seen from the Figure 4.109.

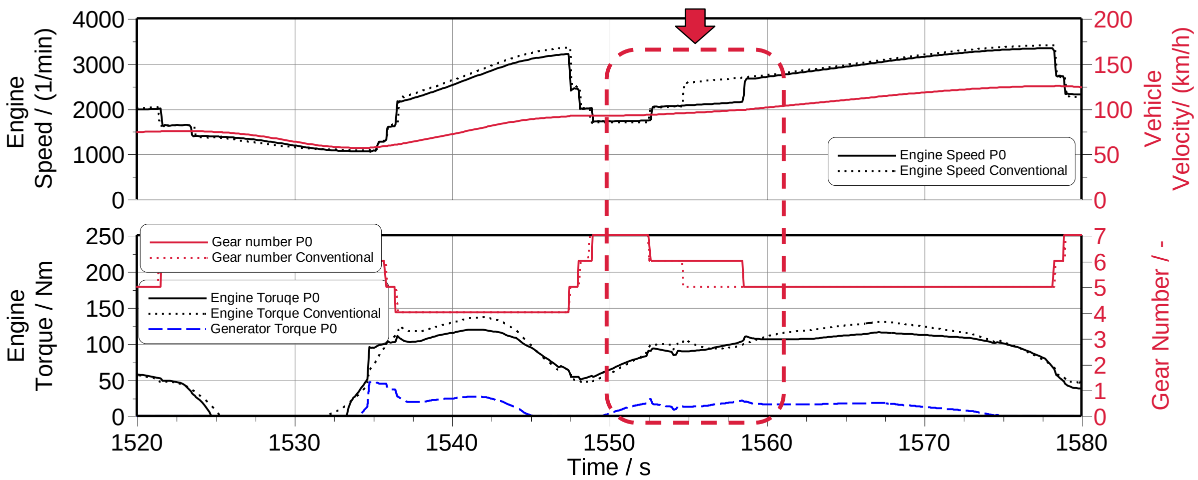


Figure 4.109 Example for combination of shift point optimization and boosting by the P0 operation strategy within $WLTP_{TMH}$ in segment C

From this operation follows a reduction of the overall energy demanded to the engine and also a benefit in terms of fuel consumption and CO_2 emissions.

From the picture below it is possible to see the effect of the generator boosting, particular during the high load phases.

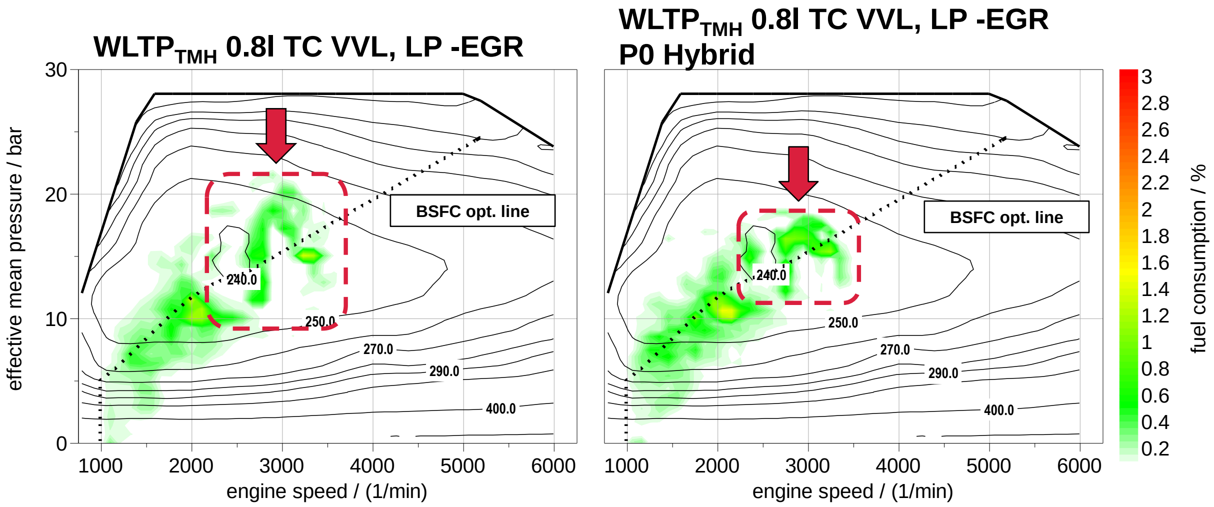
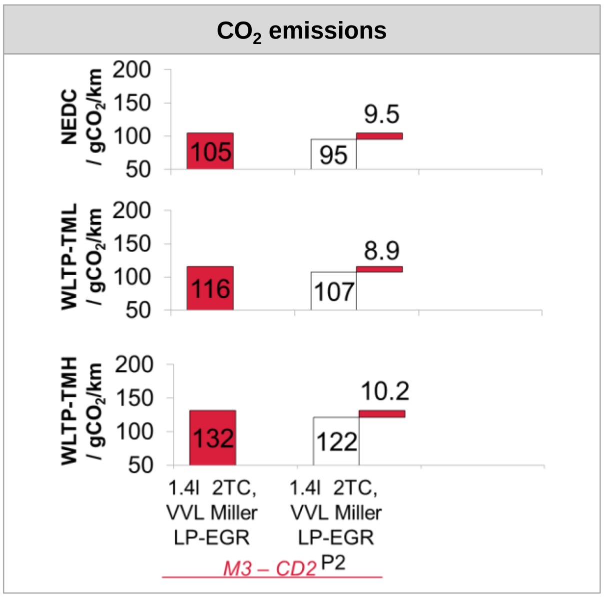


Figure 4.110 Fuel share distribution in $WLTP_{TMH}$ for 0.8 l 2stg.-TC, VVL+LP-EGR w/ and w/o P0 hybridization in segment C

4.4.2.15 Impact of P2 hybridization (Segment D and E)

The trend of fuel consumption benefits by P2 hybridization presents clearly a higher fuel saving on the NEDC cycle in comparison to the WLTP cycle. The reason of this deviation is caused by the P2 hybrid strategy.

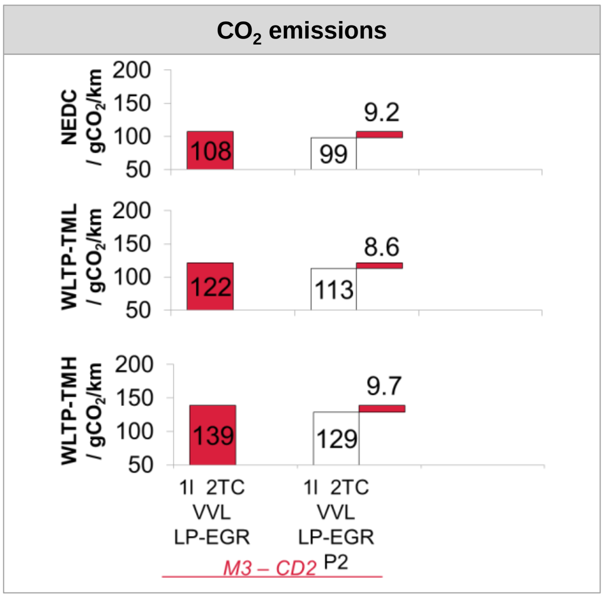


Impact of P2 hybridization

P2 hybridization added to the engine with the highest technology level

- 9.1% CO_2 reduction in NEDC
- 7.7 % CO_2 reduction in WLTP - TML
- 7.7 % CO_2 reduction in WLTP - TMH

Figure 4.111 Impact of P2 hybridization added to the 1.4 l 2stg.-TC VVL+Miller+LP-EGR variant in segment D



Impact of P2 hybridization

P2 hybridization added to the extreme downsized engine

- 8.5 % CO_2 reduction in NEDC
- 7.1 % CO_2 reduction in WLTP - TML
- 7.0 % CO_2 reduction in WLTP - TMH

Figure 4.112 Impact of P2 hybridization added to the 1 l 2stg.-TC VVL+LP-EGR variant in segment D

As mentioned in chapter 2.8.3, only the lower power phases are performed in pure electric driving. Focusing on a specific sector of the NEDC cycle, as in the diagram below, it can be shown that during the beginning of the acceleration phase, and during the constant speed and breaking phases the engine is turned off and disengaged from the transmission.

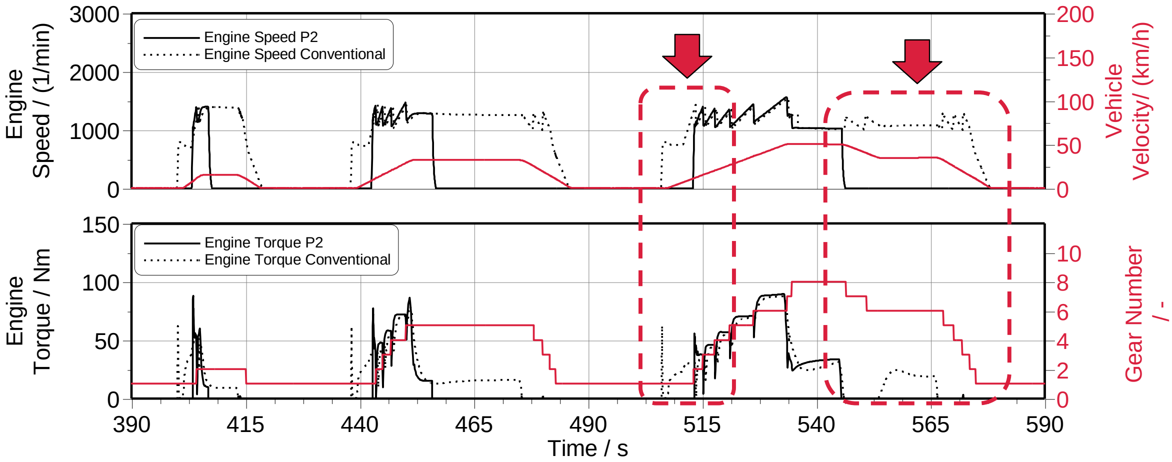


Figure 4.113 Example for the P2 operation strategy within NEDC in segment D

The NEDC cycle presents a considerable part of the consumption in a very low load and speed area of the BSFC map (see). Thanks to the hybridization the low efficiency operation area is completely avoided, and principally from this comes the CO_2 emission benefit.

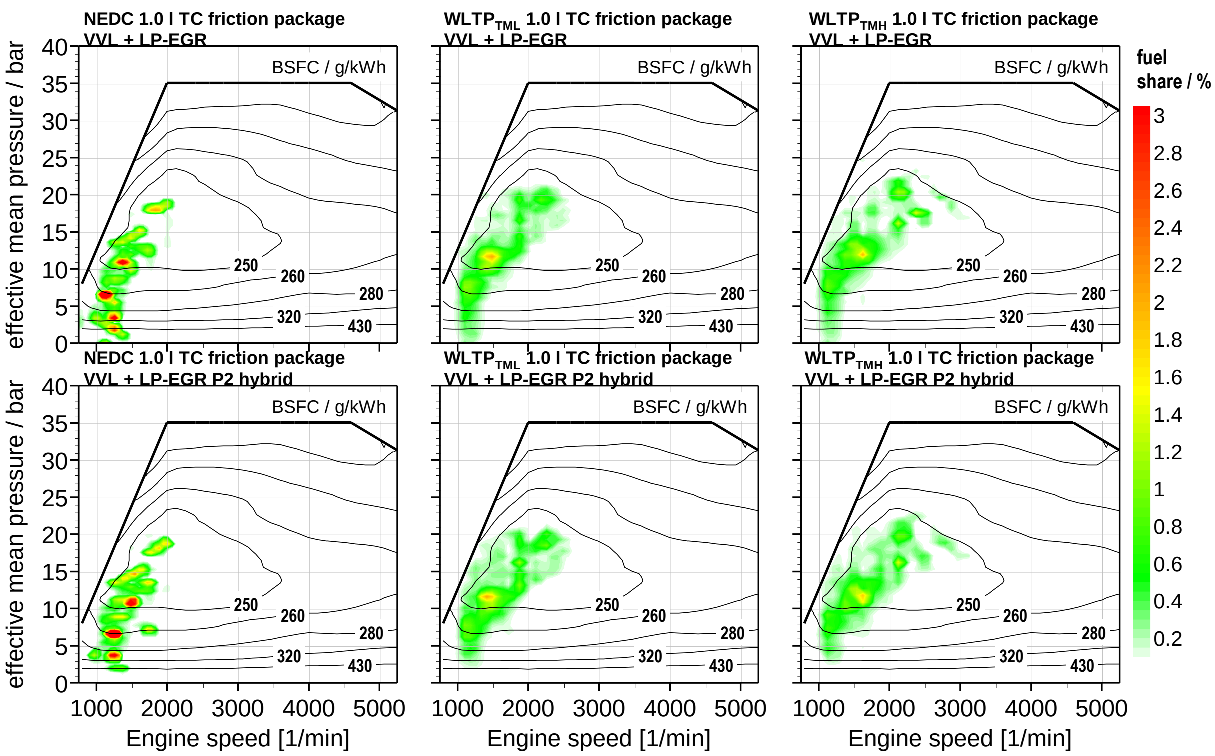


Figure 4.114 Fuel share distribution in NEDC and $WLTP_{TMH}$ for 1 l 2stg.-TC, VVL+LP-EGR w/ and w/o P2 hybridization in segment D

For the WLTP cycle the hybrid strategy operates in the same way, but the operation points that this cycle presents are generally on higher load. Also with this cycle we are able to avoid a part of the low load operation and reduce fuel consumption, but on the potential is lower than in the NEDC cycle.

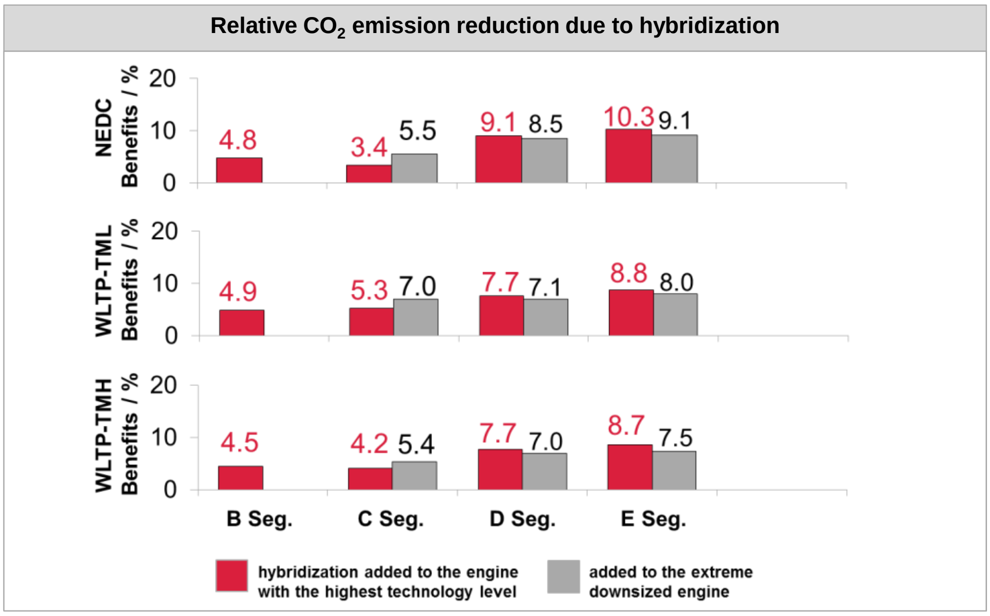
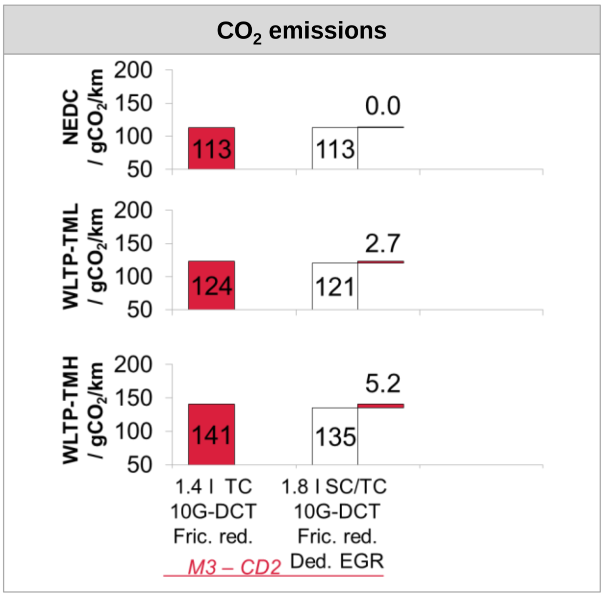


Figure 4.115: Segmentwise percentage benefits of hybridization (P0 in segment B and C; P2 in segment D and E)

4.4.2.16 Impact of dedicated EGR



Impact of dedicated EGR

Dedicated EGR compared with the also friction reduced version of the second downsizing step

- 0.0 % CO_2 reduction in NEDC
- 2.2 % CO_2 reduction in WLTP - TML
- 3.7 % CO_2 reduction in WLTP - TMH

Figure 4.116: Impact of dedicated EGR compared with the second downsizing step in segment D

The “dedicated EGR”e technology investigation was done according the following SAE paper: *“A Demonstration of Dedicated EGR on a 2.0 L GDI Engine”, Chadwell, C., Alger, T., Zuehl, J., and Gukelberger, R SAE Int. J. Engines 7(1):434-447, 2014* . Keeping the boundary condition of 75 kW/l specific power, an engine with 1.8 l displacement and 135 kW rated power was chosen for segment D and an 2.4 l 180 kW engine for segment E. As it is described in the named paper, also an mechanical compressor (SC) was applied for transient behaviour reasons.

To keep a fair comparison for the “dedicated EGR” version, a variant with less differences as possible was selected as basis (an also friction reduced downsized engine with the same transmission, weight and CD scenario) within this section.

A better efficiency especially in the high load area, but also in the part load area can be seen for the dedicated EGR version in Figure 4.117 (segment D) and (segment E). In the segment D the dedicated EGR version achieves more or less the same fuel consumption as the selected basis within the NEDC, whereas it becomes better within the WLTP cycles. This trend is simply explained by the higher impact of the EGR technology on the high load area of the engine map due to the reduced knock limitation. In the E segment principally the same effects appear. But due to the higher displacement compared to the base engine an increase of the CO_2 emissions by 5.9 % can be seen within the NEDC and only a slightly benefit of 0.6 % with the WLTP TM_H cycle. Main disadvantage of the presented dedicated EGR system is the relative low specific power and therefore the low downsizing degree of the engines compared to the Miller system with external cooled low pressure EGR. With a better and optimized boosting system and improved specific power maybe a higher fuel consumption potential can be achieved. Key features for that are improved charge motion, optimized injection systems and combustion chamber design.

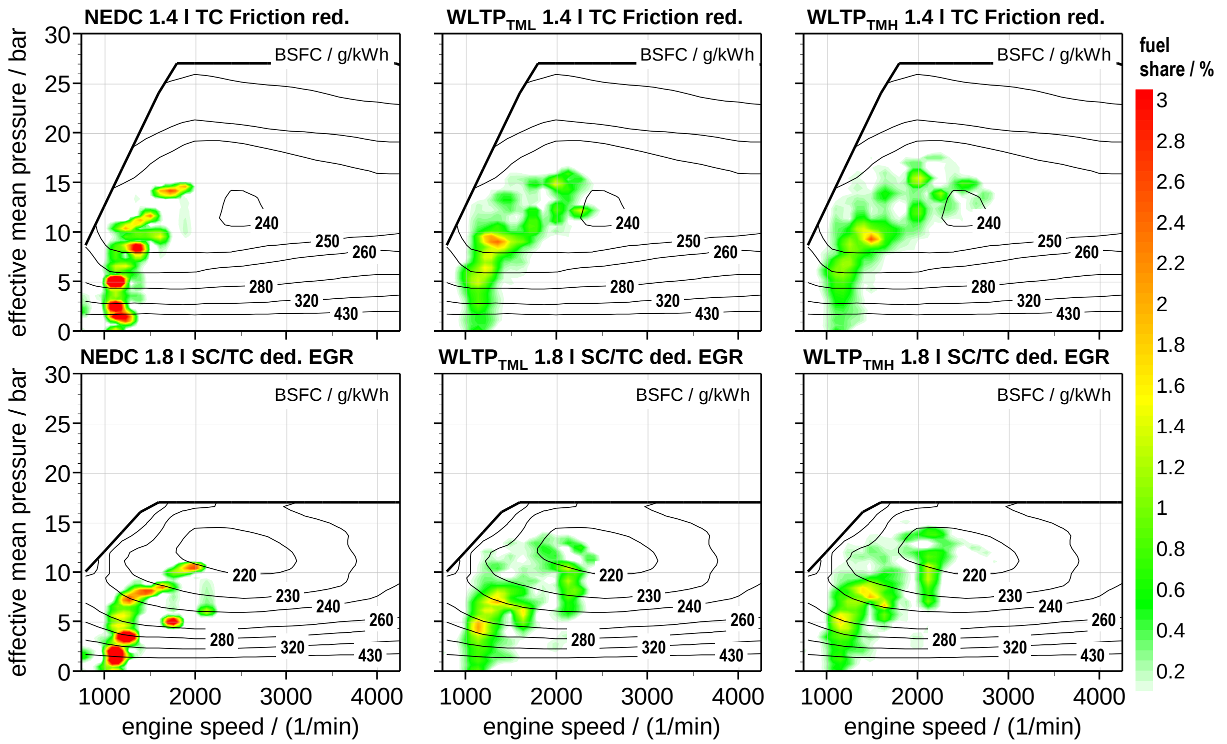


Figure 4.117: Fuel share distribution in NEDC and WLTP for dedicated EGR in segment D

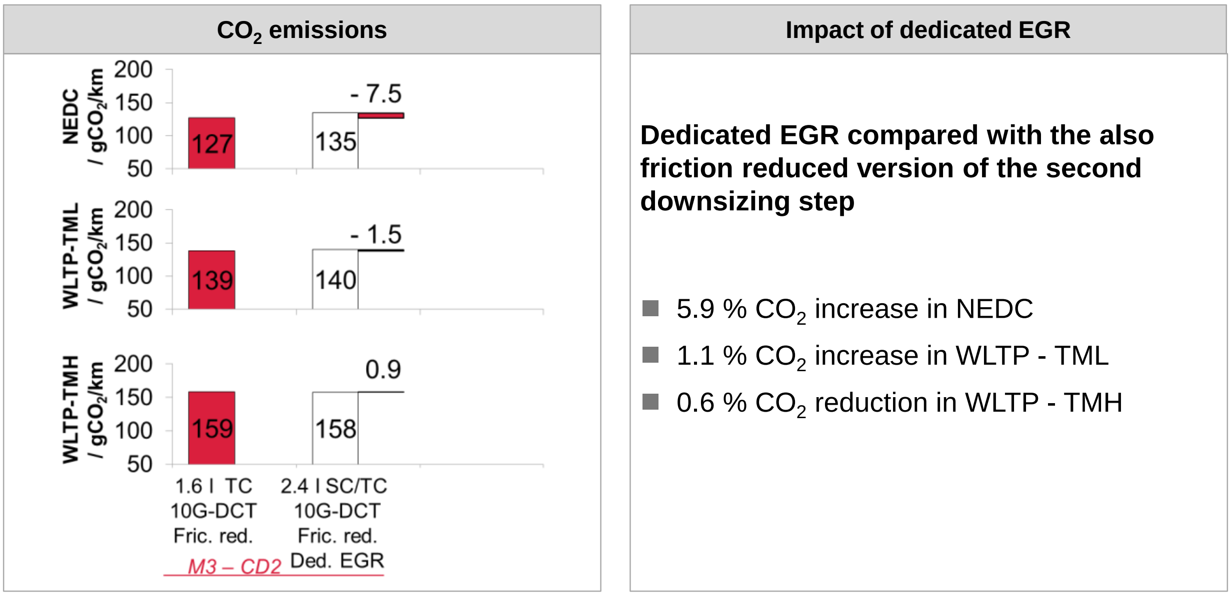


Figure 4.118: Impact of dedicated EGR compared with the second downsizing step in segment E

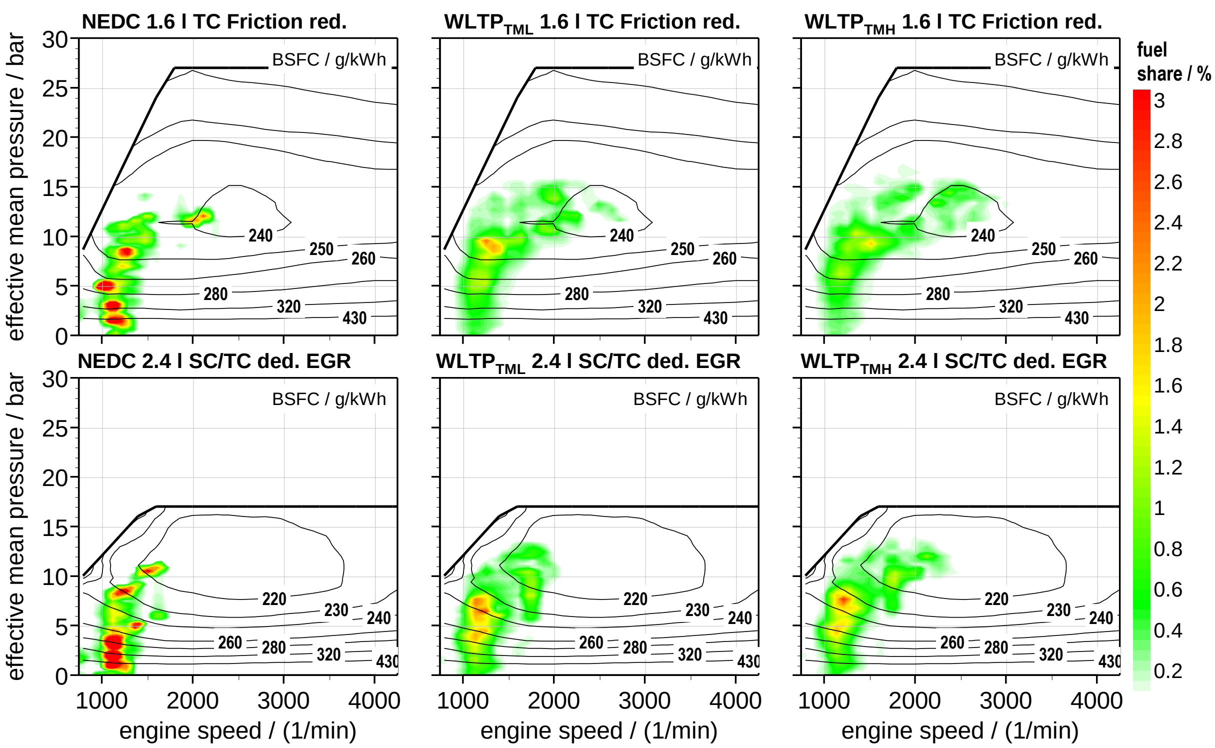


Figure 4.119: Fuel share distribution in NEDC and WLTP for dedicated EGR in segment E

5 CO2 potential with respect to final cost

This Chapter compares the CO_2 potentials achieved by the different technologies applied to the variants, with respect to their costs. For such a comparison, the steps of mass reduction and coast down reduction should be avoided, since the cost calculation of these steps was not taken into account within the project.

- The relevant technologies and packages for all segments are analyzed in detail.
- The cost for every technology is underlined as function of the CO2 reduction potential.
- The line of the 95 €/ g CO_2 /km is also shown (Penalty that it should be paid in case of CO_2 emissions exceedance based on NEDC cycle).
- The best configurations for the Segments are remarked.

Important remarks:

- The mass and coast down reductions cost are not calculated.
- The plots show absolute values, but not relative benefits. It is normally more difficult to get CO_2 benefits when the base for comparison is more efficient, i.e. The CO_2 absolute potential is usually lower when the system is more efficient.

5.1 Diesel

Therefore, and for the diesel case, there will be two sets of comparison per segment:

1. Start and Stop, and Downsizing influence with respect to the baseline. For example, and for the case of the Segment C, this is depicted in the Figure 5.1.

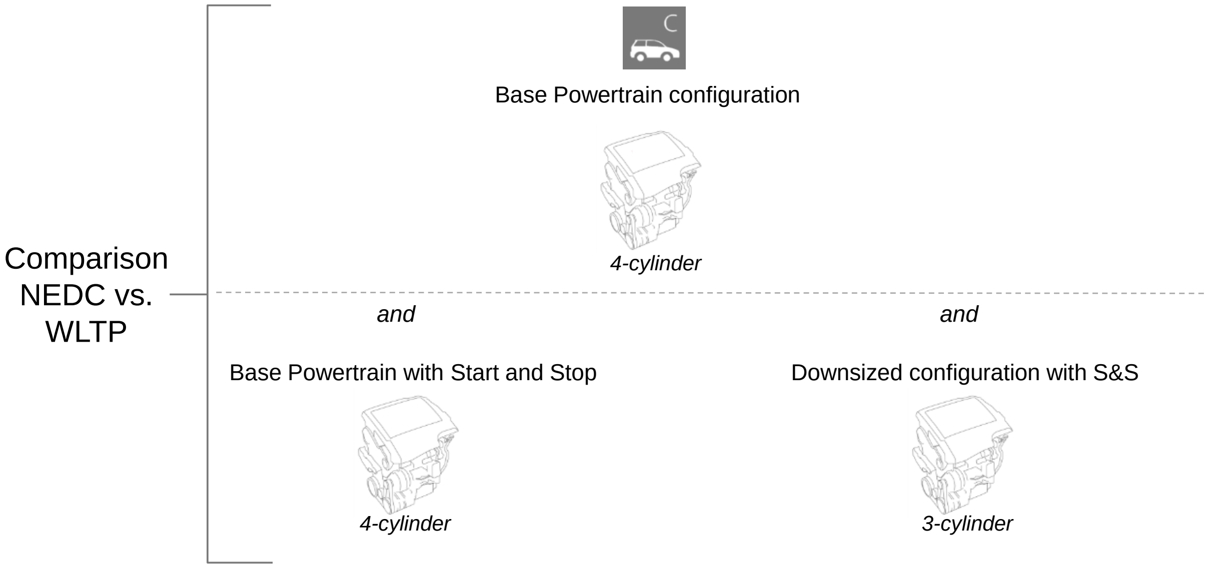


Figure 5.1 Analysis of Downsizing and Start and Stop with respect to the baseline, for the case of the Segment C.

2. Different Engine technologies and Hybrid versions with respect to the first variant with the lowest mass and first coast down reduction. For the Segment C, this is shown in the Figure 5.2.

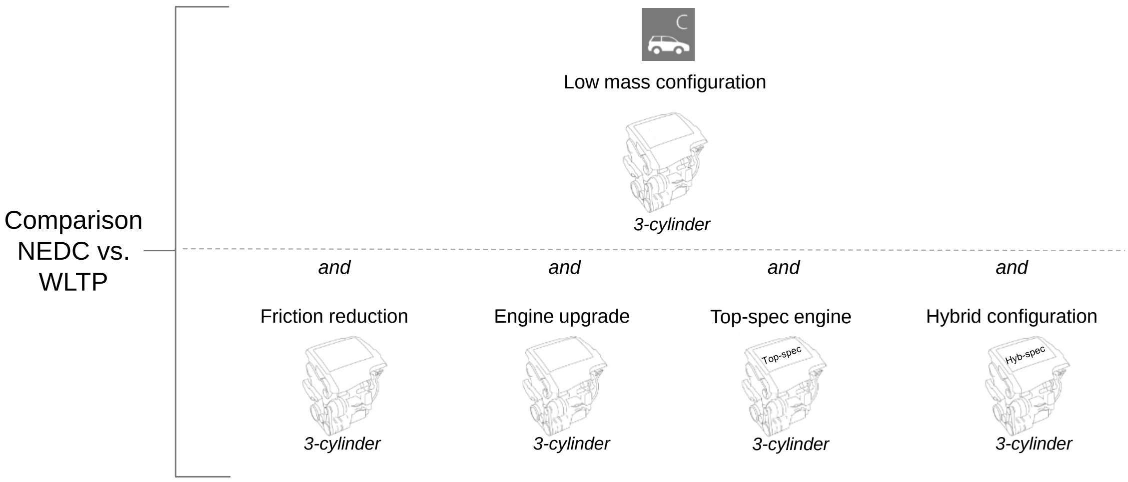


Figure 5.2 Analysis of different powertrain technologies, for the case of the Segment C.

This study allows understanding which technologies may be more attractive to the OEMs in order to reduce the CO_2 emissions. Nevertheless, the values shown in the figures are absolute values, and not relative, showing the total cost of every package with respect to the total CO_2 reduction potential. On the other hand, the heavier vehicles with higher specific powered engines will usually have less potential of improvement due to the base values are already high, and the average engine load would also be. And the most of the technologies will have more potential benefits in the low load regions. At the same time, the potential benefits are also depending on the cycles, i.e. the NEDC usually presents higher potential of improvement than the WLTC.

The Segments B and C present similar results, and thus only the results of the Segment C will be presented in detail. In the case of the heavier segments (D, E, SUV and LCV), even with small differences, the conclusions are also similar; hence the Segment SUV will be discussed in detail. For all cases, all figures can be found in the Appendix.

5.1.1 Segment C

The Segment C analysis will be partially applicable to the Segment B, since the conclusions are similar, with the only exception that the VCR system is not installed in the Segment B vehicle.

The base available powertrain of the Segment C already presents a good idle calibration, and the transmission is manual; therefore, the potential benefit for the Start and Stop technology is lower than for other segments which mount for example AT transmissions with much worse idle efficiency due to the torque convertor. Anyway, the Start and Stop is an interesting technology, especially for the NEDC. The ratio of cost against total CO2 potential is in the range of 20 € / gCO_2/km. For the WLTP, the ratio is lower but still in the range of 30-50 € / gCO_2/km.

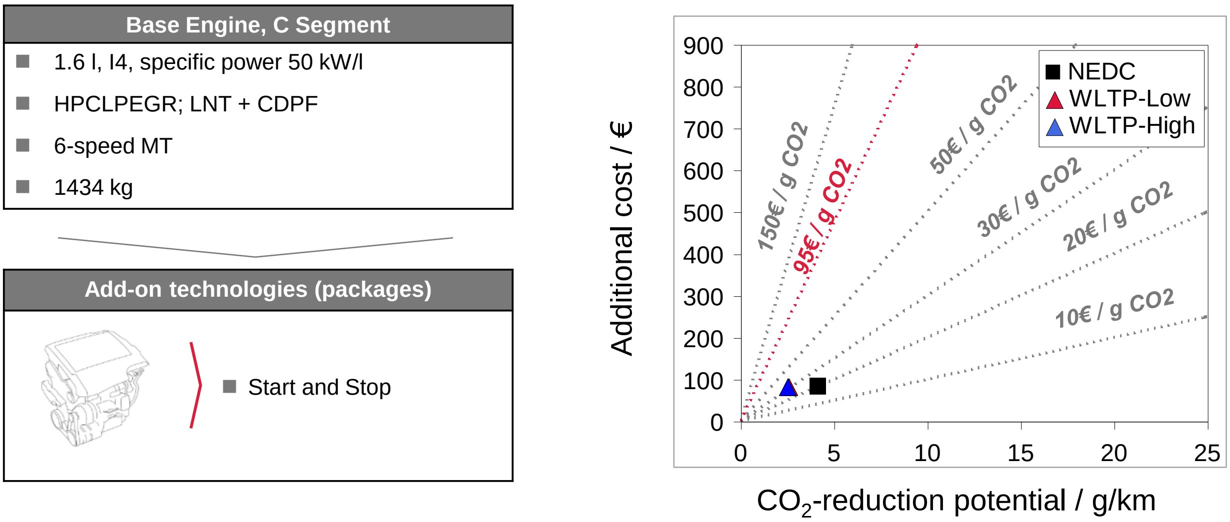


Figure 5.3 Segment C cost analysis for 4 technological start and stop

The second step includes one downsizing step from 1.6 to 1.4 L, but also the upgrade of the transmission from a MT6 to a DCT7 with a wet-clutch transmission. The benefit in the NEDC is really attractive, with a total potential around 15 g/km for a delta cost about 400 €, driven mainly due to the DCT7 cost. This gives a good ratio around 25 € / gCO_2/km. The main reason is the fixed shifting strategy for the NEDC, with respect to the optimized shifting strategy achieved by a DCT7.

In the WLTP, the total benefit is around 3 g/km, with a ratio higher than 95 € / gCO_2/km. Therefore, the potential for such cycle is much lower than for the NEDC, and heavily affected by a shifting strategy which was optimized in order to reduce the NO_x emissions. Furthermore, it should be mentioned that the installation of the DCT transmissions have other benefits, such as better performance or comfort. In addition, the increase of the deNO_x efficiency may permit to further optimize the shifting strategy.

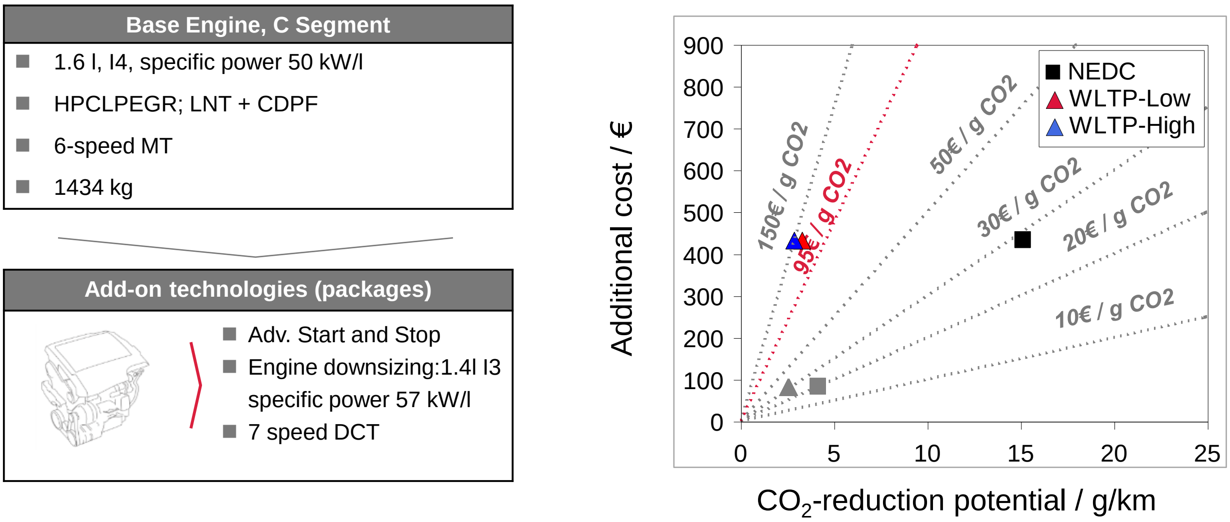


Figure 5.4 Segment C cost analysis for engine downsizing and transmissions upgrade

In the next figures, further powertrain strategies are compared with respect to the segment variant which present already the lowest mass (M3: 20 % mass reduction) and the first coast down reduction (CD2: 25 % RRC reduction and 10% CdA reduction). The baseline is already a downsized version with a DCT7, Advanced S&S, and a high pressure and cooled low pressure EGR systems.

The first add-on technology is then the application of friction reduction measures, which cover base engine measures, electrical water pump, split cooling and variable oil pumping. A benefit around 3.5 gCO_2/km of CO_2 is achieved with a ratio at all cases around 20 € / gCO_2/km.

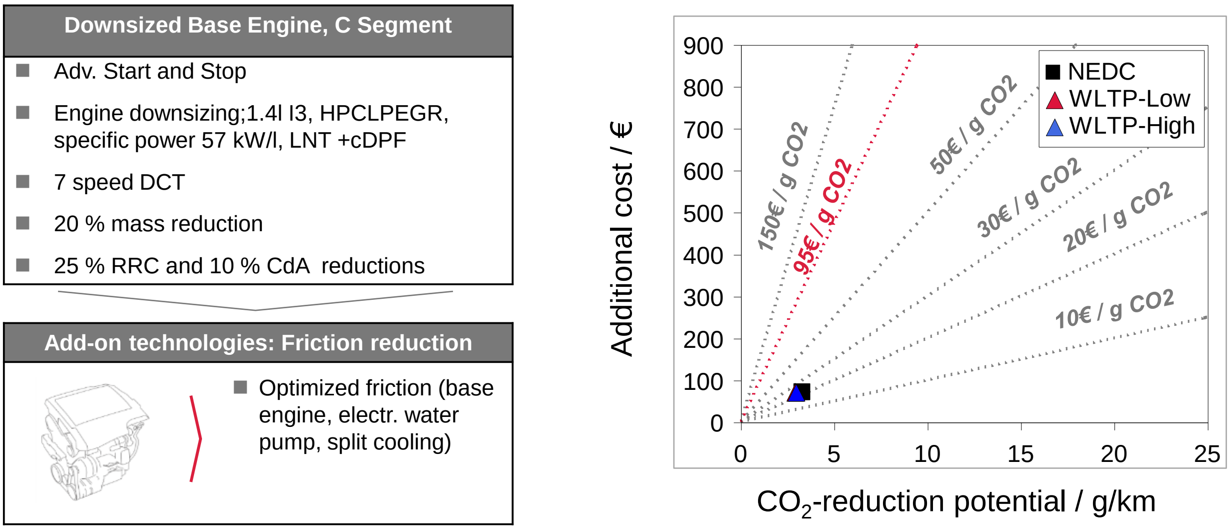


Figure 5.5 Segment C cost analysis for frictional upgrade.

The next upgrade includes two engine technologies, such as a double stage Variable Compression Ratio (VCR) and the installation of a coolant into the HPEGR loop. Due to those changes, up to 7 g/km can be achieved in total, getting an extra benefit around 4 g/km. The ratio is around 30 € / gCO_2/km for all the cycles. Anyway, the percentage of the benefit is higher for the NEDC, since the base is lower.

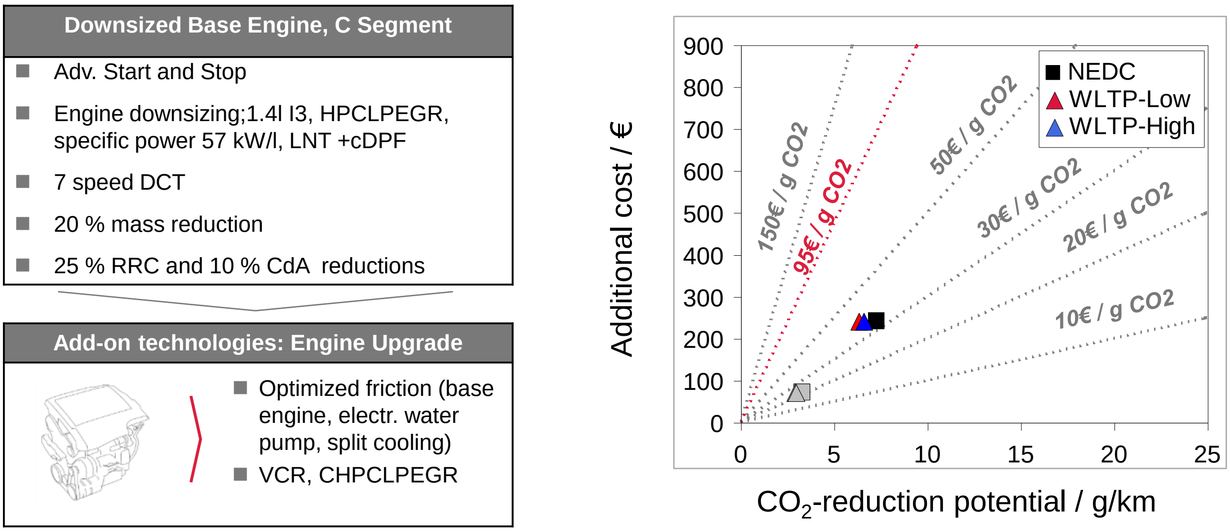


Figure 5.6 Segment C cost analysis for engine upgrade.

In the next step and as shown in Figure 5.7, the aftertreatment system is changed, and the LNT + cDPF system is substituted by a DOC+SDPF system. Furthermore, a variable valve timing (VVT) is installed in order to support the system heating. In total, the incremental benefit in terms of CO_2 is small and in the range of 1 g / km. In total and considering the optimized friction, the VCR, the cooled HPEGR and cooled LPEGR as well as the aftertreatment upgrade and VVT installation, the total cost ratio is in the range of 70 € / gCO_2/km. However, the SCR system presents clear benefits in terms of NOx reduction.

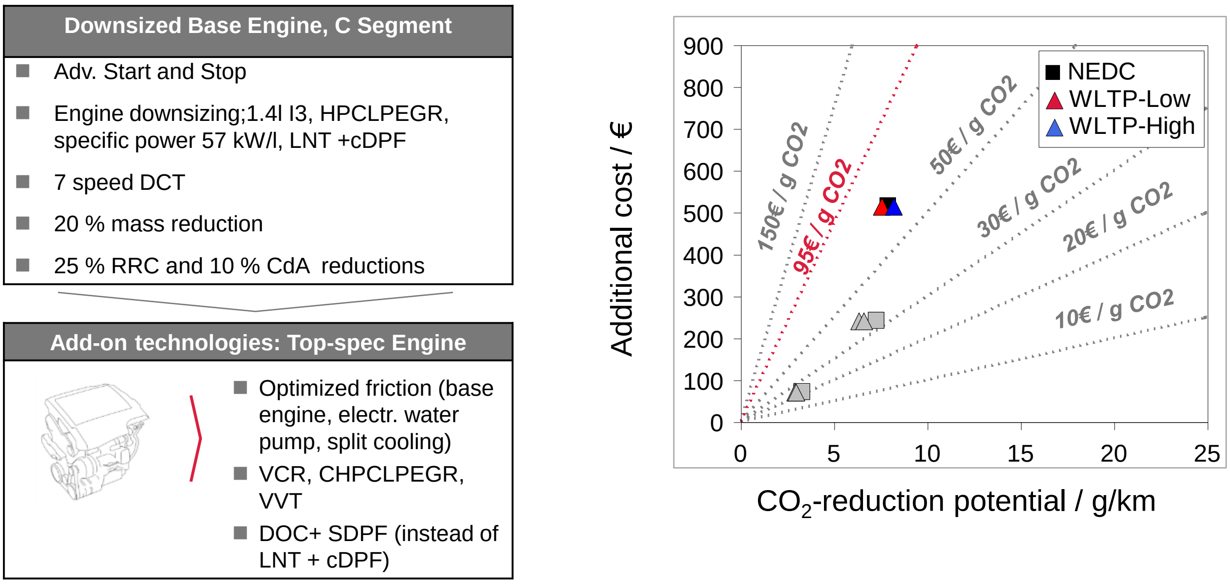


Figure 5.7 Segment C cost analysis for top-spec

The last step shows the effect by installing a hybrid system, comprising a P0 with a Belt Starter Generator (BSG) with 15 kW and 48 V. The engine technologies include now the optimized friction measures and the CHPCLPEGR system. The results are summarized in Figure 5.8. The total benefit is in the range of 8 g/km, slightly higher for the NEDC. The cost ratio is 95 € / gCO_2/km for the NEDC and slightly higher for the WLTP cycles (~120 € / gCO_2/km). The total extra cost by considering these technologies is around 800 € with respect to the baseline, considerably higher than for the cost of upgrading the engine technologies. In addition, it should be mentioned that the P0 allows also to reduce the engine-out NOx emissions, as shown in the Chapter 4.

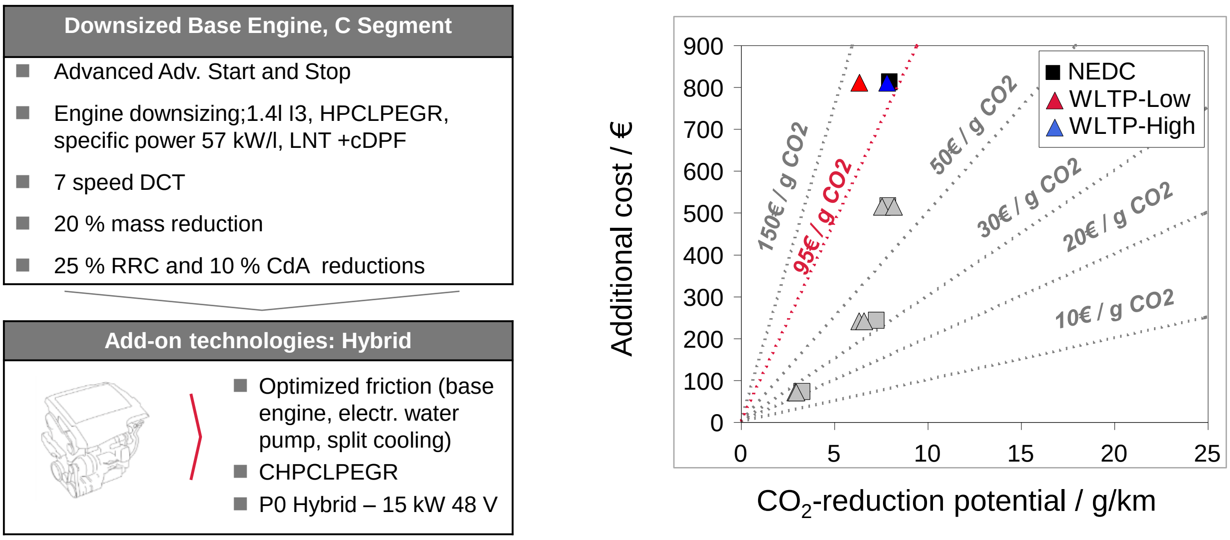


Figure 5.8 Segment C cost analysis for hybrid

5.1.2 Segment SUV

Due to be a much heavier vehicle with a different baseline concept, the SUV is also presented and commented here in this section. The results and conclusions are applicable directly to the Segment E, since both present a similar powertrain.

First of all, the potential benefit of the advanced Start and Stop for the SUV segment is higher than for other segments, since the base powertrain mounts an AT transmission, which presents a low efficiency at low loads and idle. The reason is due to the torque convertor overall stays open, the efficiency is low. Therefore, the ratio for the NEDC is 10 € / gCO_2/km, while for the WLTP is 20 € / gCO_2/km.

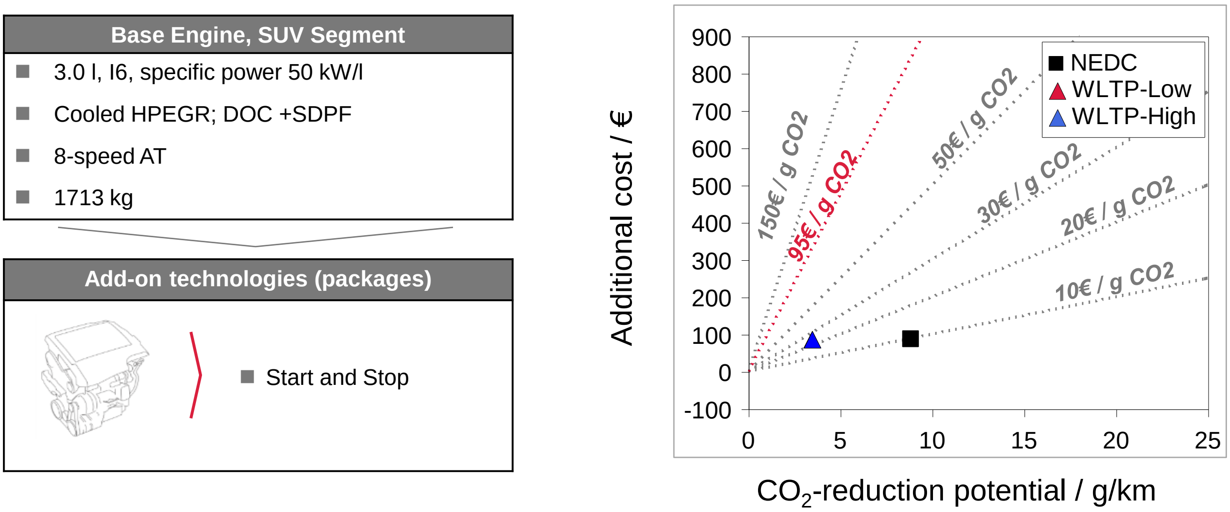


Figure 5.9 Segment SUV cost analysis for technological start and stop

The downsizing step involves moving from 3.0 L to 2.0 L and the change of the AT8 transmission for a DCT10 with a wet clutch. The NEDC total benefits are in the range of 20 g/km in total by also considering the advanced S&S. In the case of WLTP, the benefit is still high and in the range 14 g/km. The downsizing seems a very attractive step for such segments. However, the emissions increase due to the downsizing should be assessed.

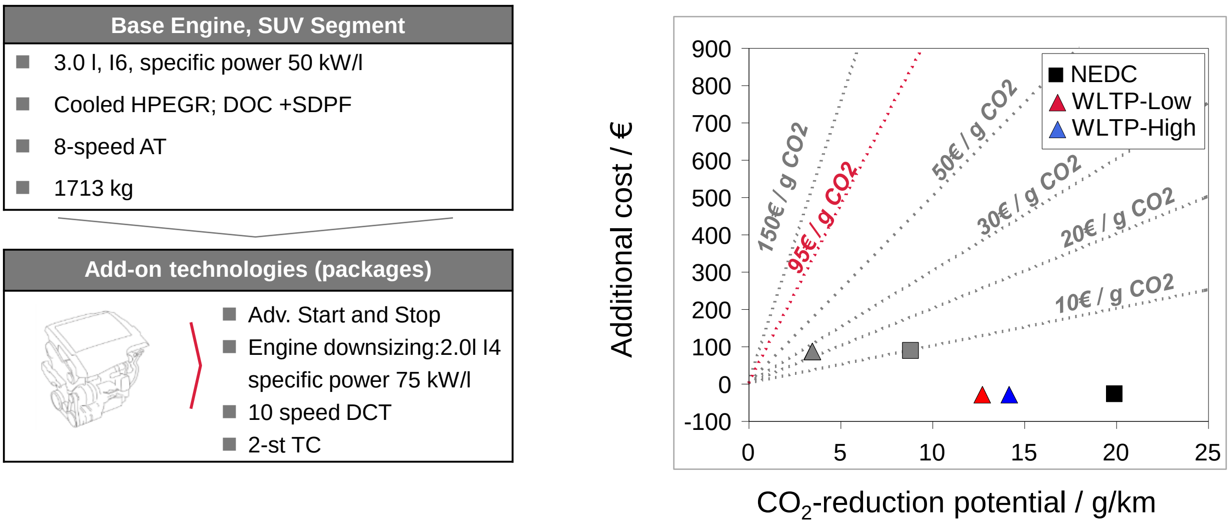


Figure 5.10 Segment SUV cost analysis for engine downsizing and transmissions upgrade.

As for the Segment C, now the next figures compare the different technology packages with respect to the segment variant which present already the lowest mass (M3: 20 % mass reduction) and the first coast down reduction (CD2: 25 % RRC reduction and 10% CdA reduction).

The absolute friction reduction benefit is higher into the SUV class since the base CO2 emissions are also higher. Therefore, the cost ratio is in the range of 10-20 € / gCO_2/km, and as usual, slightly lower for the NEDC than the WLTP.

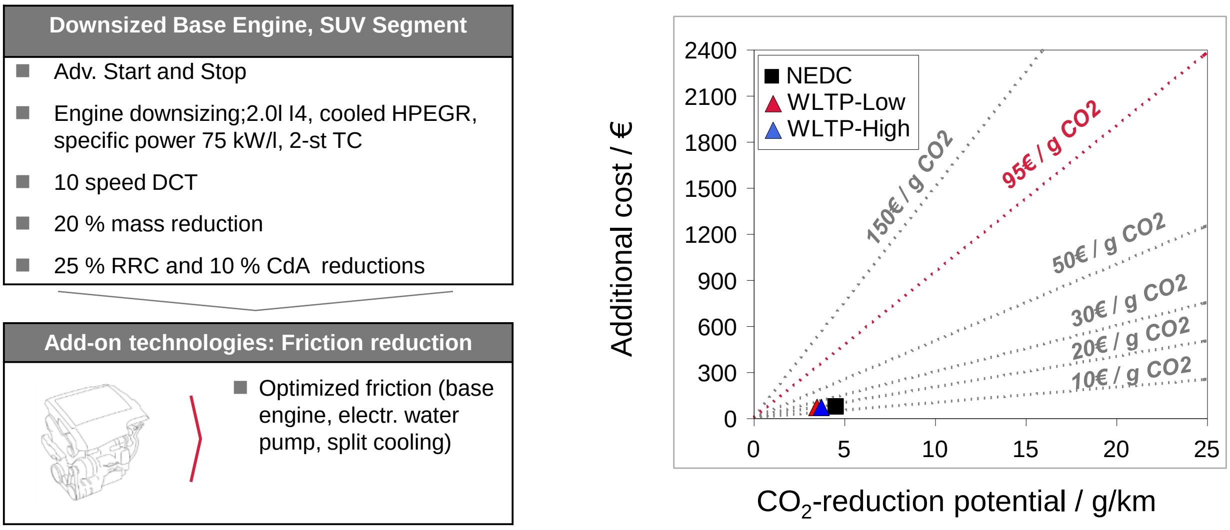


Figure 5.11 Segment SUV cost analysis for frictional upgrade

The base engine presents a heating strategy based on post-injections in order to accelerate the light-off of the aftertreatment systems. Thus, by installing a VVT system, the faster warming can be achieved by the VVT, and therefore there is no need to keep post-injections. The benefit is in the range of 1g for the NEDC and 0.5 g for the WLTP. The value of the complete package is now in the range of 20 € / gCO_2/km for the NEDC and 30 € / gCO_2/km for the WLTP cycles.

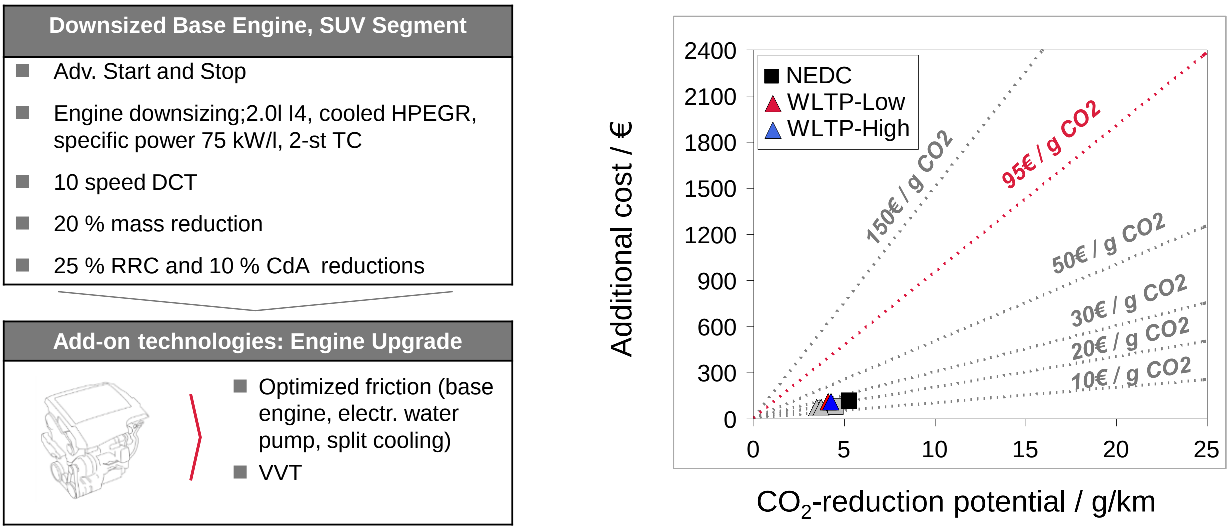


Figure 5.12 Segment SUV cost analysis for engine upgrade

As a next step, a double stage VCR system and CHPCLPEGR system are installed. The benefits of both systems together are around 7-8 g/km for the NEDC, and ~5 g/km for the

WLTP. In total, all the technologies achieved a ratio around 30 €/ g /km for the NEDC and ~40 € / gCO_2/km for the WLTP, even though lower for the NEDC.

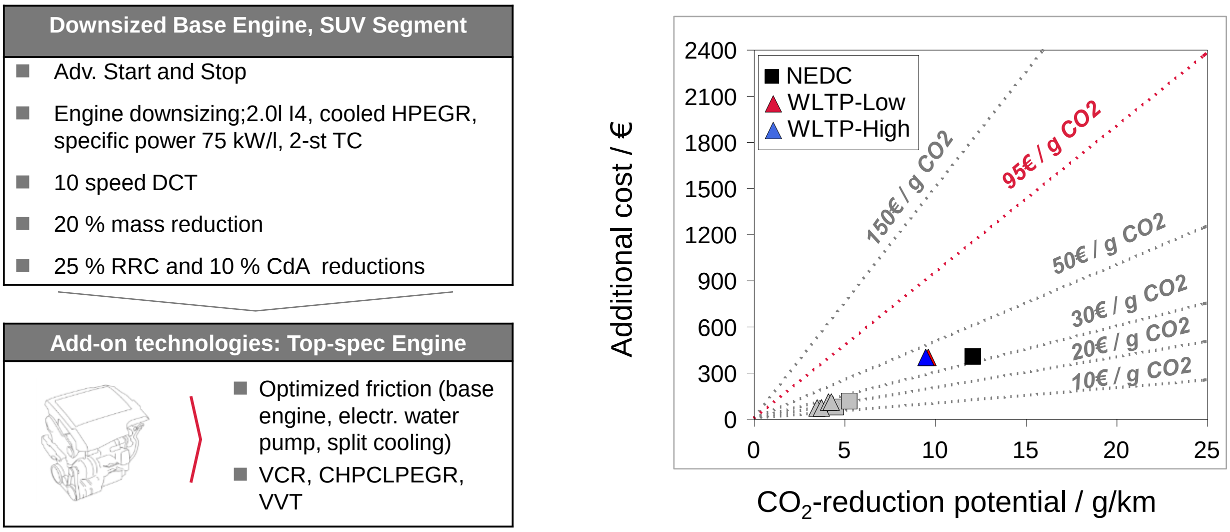


Figure 5.13 Segment SUV cost analysis for top-spec

In the Figure 5.14, it is shown now the potential by using a full parallel hybrid system (P2 configuration). The total benefit is clearly higher for the NEDC, in the range of 22 g/km, against 15 g/km in the WLTP. The cost ratio for the NEDC is in the range of 100 € / gCO_2/km and in the range of 150 € / gCO_2/km for the WLTP cycles. With such technology, there is a big potential to get high CO_2 reduction, however the cost should be assessed before installing them into the vehicles. If the order of the technologies would be different, the application of a hybrid system to a base engine would look like more attractive.

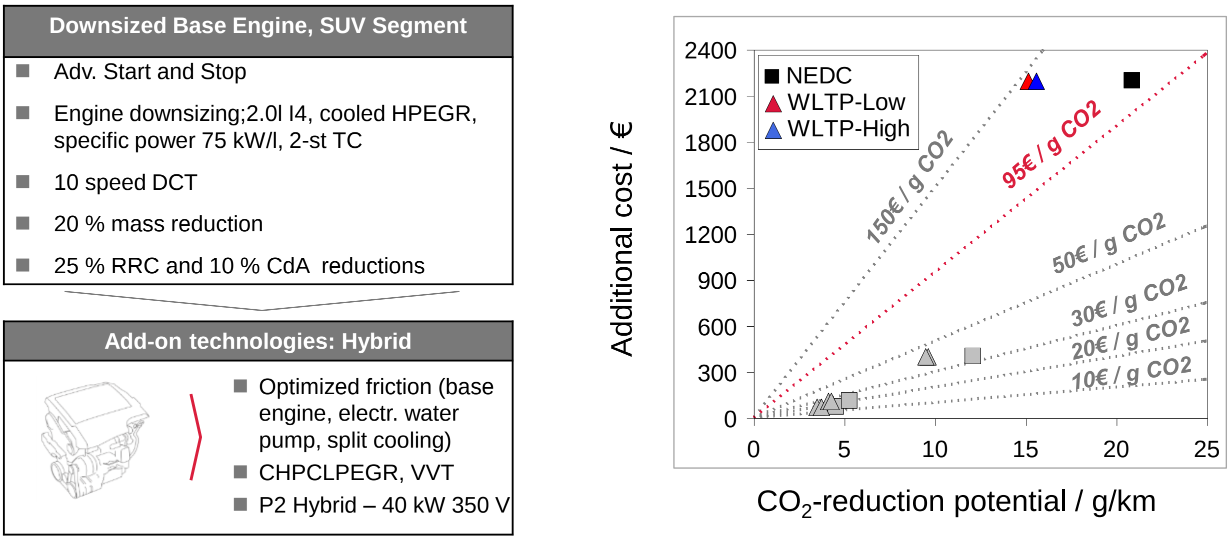


Figure 5.14 Segment SUV cost analysis for hybrid.

Nevertheless, the sequence of application of technologies affects this analysis, and it is much cheaper get higher reductions when the base value is higher. Once the system has been optimized, for example by the mass and coast down reduction and downsizing, reaching higher reductions is becoming more expensive and difficult.

5.2 Gasoline

Also for the gasoline case, there will be two sets of comparison per segment:

1. Start and Stop, and Downsizing influence with respect to the baseline.

2. Different Engine technologies and Hybrid versions with respect to the first variant with the lowest mass and first coast down reduction.

The segments B and C present similar results, and thus only the results of the Segment C will be presented in detail. In the case of segment D and E the E segment will be discussed in detail. For all cases, all figures can be found in the Appendix.

5.2.1 Segment C

The Start&Stop functionality is a low cost technology and so worthwhile to be applied also in lower segment. The ratio of cost against total CO_2 potential for the shown segment C variant in Figure 5.15 is less than 10 € / gCO_2/km for the NEDC. For the WLTP, the CO_2 potential is about the half as of the NEDC, which increases the cost CO_2 potential ratio to around 20 € / gCO_2/km

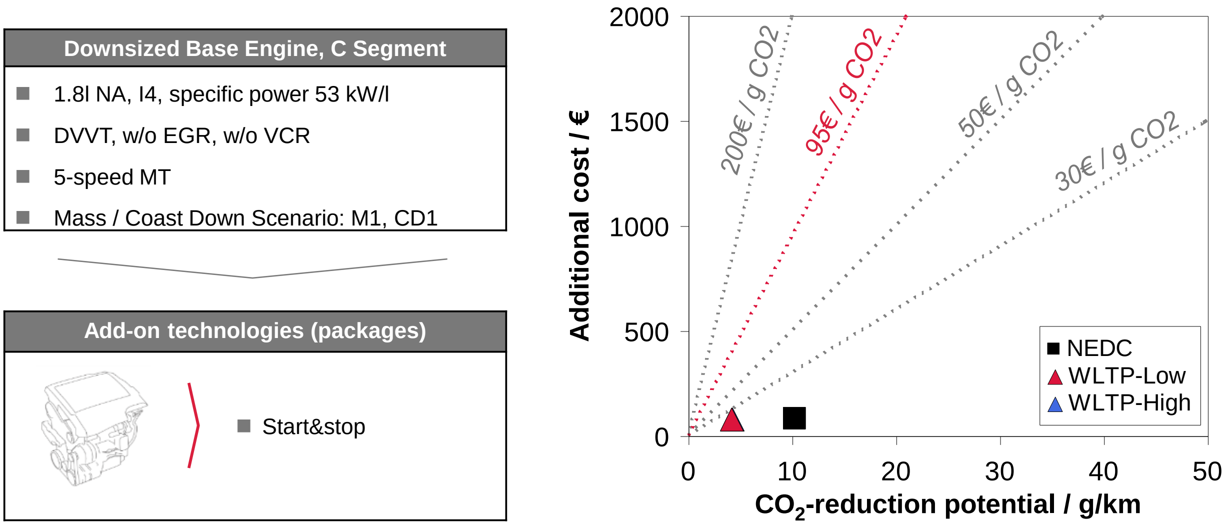


Figure 5.15 Segment C cost analysis for start&stop technology

Figure 5.16 shows the first downsizing step from the NA 1.8l baseline engine to a 1.4 l TC one in segment C. Since, the replacing of a NA engine by a TC engine normally gives big CO_2-reduction potential with manageable efforts, by now, the European market is dominated by moderate turbo charged engines in almost every vehicle class. For the shown segment C example, the reason is given by the very attractive cost versus CO_2 potential ratio below 15 € / gCO_2/km for the NEDC and slightly above 15 € / gCO_2/km for the WLTP.

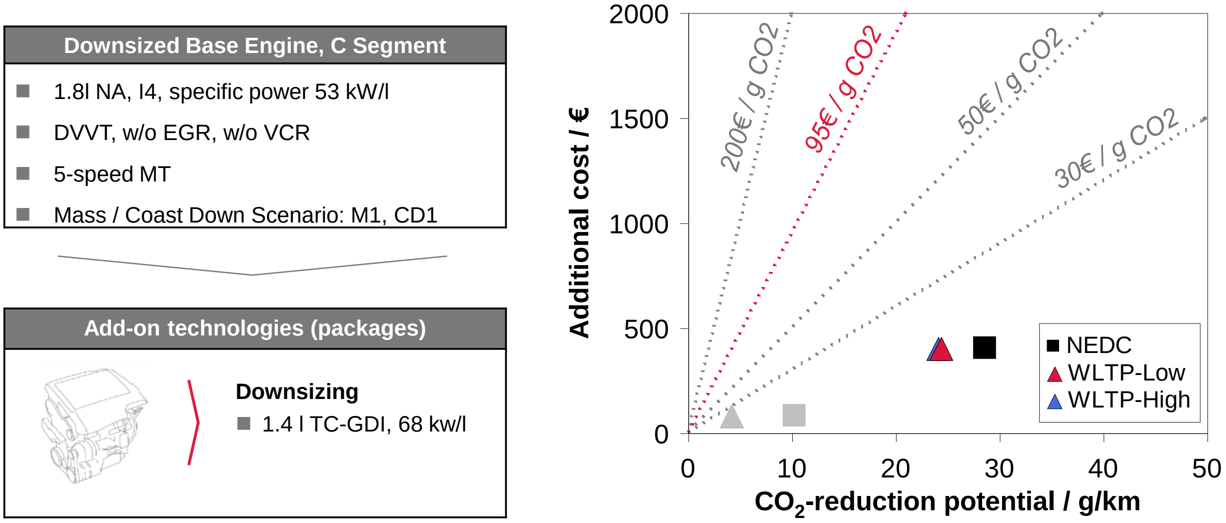


Figure 5.16 Segment C cost analysis for engine downsizing

In the following figures, further powertrain technologies will be compared with respect to the segment variant which present already the lowest mass scenario (M3: 20 % mass reduction) and the first coast down reduction step (CD2: 25 % RRC reduction and 10% CdA reduction). The baseline is always a further downsized 1 l TC I3 variant with a 6-speed MT.

The Figure 5.17 shows the transmission upgrade in segment C from a 6 speed MT to a 7-speed DCT, equipped with a dry clutch and maximum transmission input torque of 250 Nm. Due to the fixed NEDC shift points for an manual transmission, this upgrade leads to significant CO_2 benefits within this driving cycle, whereas for the WLTP only low CO_2 potentials are achievable. Compared to the relatively high cost for a DCT upgrade, a ratio of 60 € / g/CO_2km results for the NEDC. For WLTP the ratio exceeds the line of the 95 €/ gCO_2/km to values up to 240 € / gCO_2/km.

As already mentioned in the diesel section (see Chapter 4.3.3.3), the installation of the DCT transmission has also other benefits, such as better performance and comfort. At the same time, an automatic transmission is mandatory for energy recuperation strategies based on sailing or coasting for mild hybrid systems. Normally the customer pay an extra price for the better comfort of an automatic transmission and therefore the extra cost for CO_2-emission reduction can be reduced. In smaller segments, the willingness of the customer to pay more for automatic transmission is limited and therefore a DCT transmission as only CO_2-emission reduction measure does not seem cost effective.

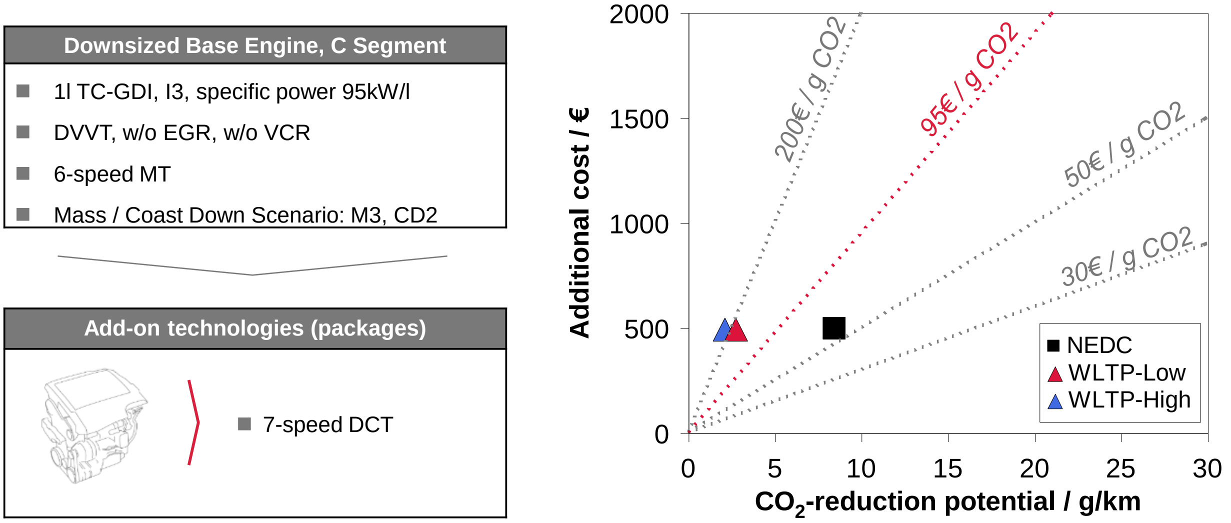


Figure 5.17 Segment C cost analysis for transmissions upgrade

In the next step, a friction package was applied covering base engine measures, variable oil pump, electric water pump and split cooling. With additional costs of around 80 €, in this case, a further CO_2-potential of around 5.5 gCO_2/km is achievable. As a result, the cost versus absolute CO_2-potential ratio improves to 40 € / gCO2/km for the NEDC and to 75 € / gCO_2/km for the WLTP respectively. The friction package only with the 1.0l TC DI engine with DCT transmission as base, achieves for NEDC and WLTP a cost to CO_2 emission benefit ratio of 15€ / gCO_2/km.

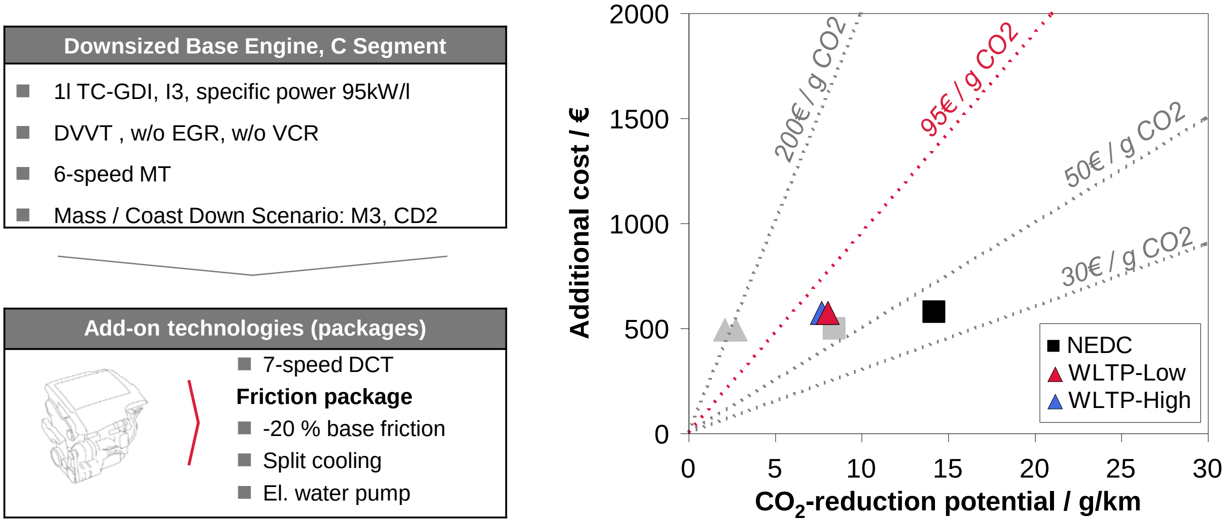


Figure 5.18 Segment C cost analysis for transmissions upgrade and for frictional upgrade

Figure 5.19 shows the analyses for the additionally installed variable valve lift technology. Also in this case the extra costs are relatively low (85 €) however, the CO_2 benefits do not exceed 2 gCO_2/km. In total, the cost versus absolute CO_2-potential ratio for the entire package stays in the same range of 40 € / gCO_2/km to 75 € / gCO_2/km and for the single VVL system at about 55€ / gCO_2/km in NEDC and 80€ / gCO_2/km in WLTP.

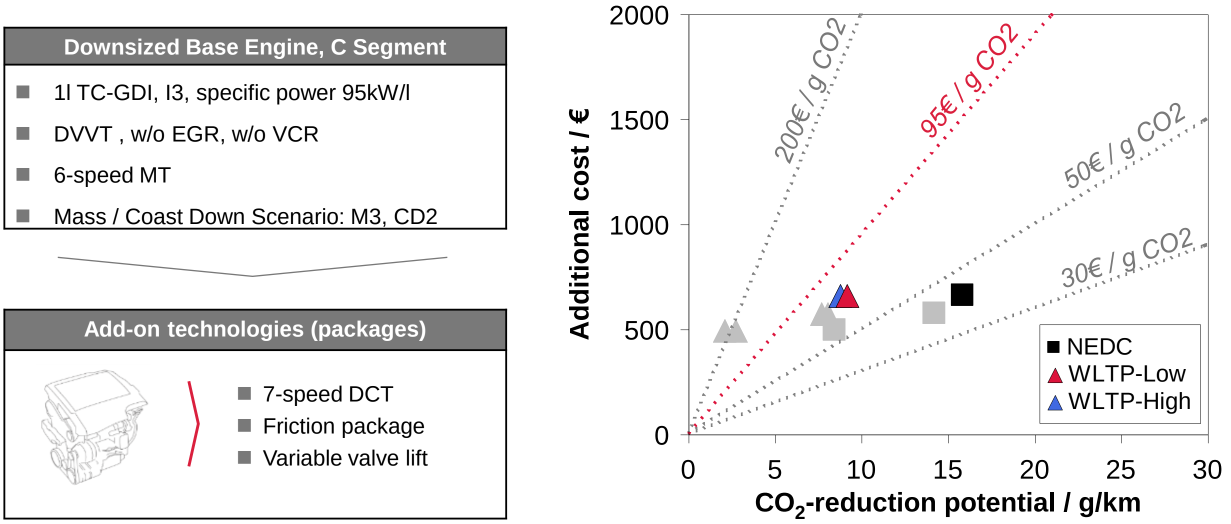


Figure 5.19 Segment C cost analysis for transmissions upgrade, for frictional upgrade and for variable valve lift

The installation of the Miller cycle gives a further benefit of around 4 gCO_2/km (Figure 5.20). With the additional costs of 178 €, the cost ratio for NEDC stays at almost the same value, whereas for the WLTP the ratio improves to around 65 € / gCO_2/km.

For the single Miller system with 2-stage boosting the cost ratio is about 45€ / gCO_2/km in NEDC and 40€ / gCO_2/km in WLTP.

The delta costs, in this step, arise due to the need of a two stage charging to cover the higher boosting requirements of the Miller cycle (see Chapters 2.4.1 and 4.4.2.10). In the segment B, where a one stage charging is still sufficient, actually, no extra costs arise in this step.

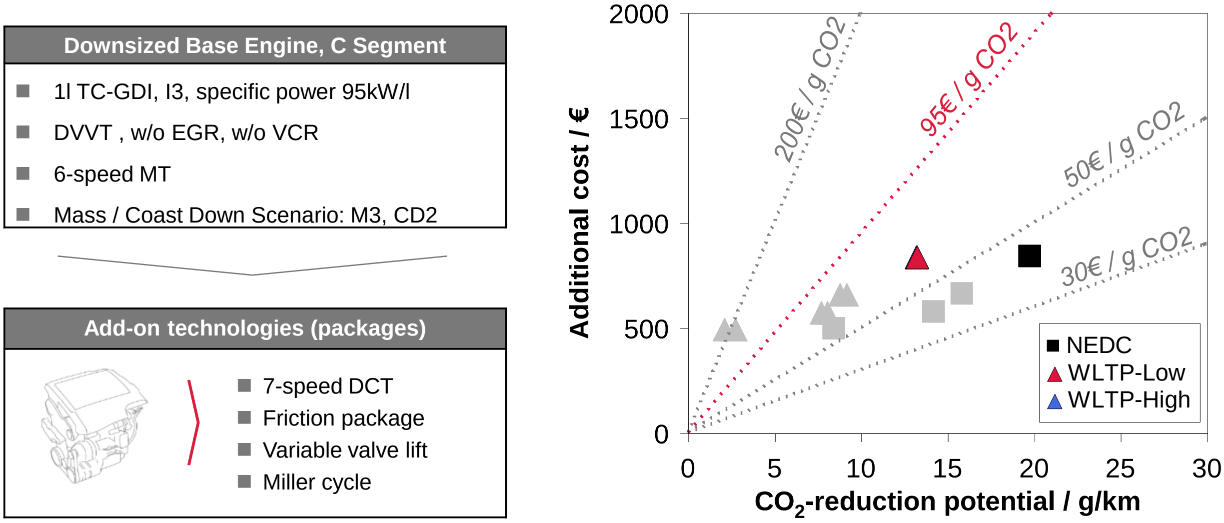


Figure 5.20 Segment C cost analysis for transmissions upgrade, for frictional upgrade, for variable valve lift and for Miller cycle

In the next step, instead of the Miller cycle, a two-step VCR system as well as low pressure EGR is installed (see Figure 5.21). Due to the use of LP-EGR, in this case also a two stage charging system is required. Therefore, together with the costs for the LP-EGR and for the VCR system, the total extra costs of 388 € occur. The further CO_2-benefit is around 4.54 gCO_2/km, with respect to the variant shown in Figure 5.19.

As a result, the cost versus total CO_2-potential ratio for the entire add-on package increases to 50 € / gCO_2/km for the NEDC and near to 80 € / gCO_2/km for the WLTP.

For the 2-stage VCR plus cooled LP EGR system with 2-stage boosting the cost ratio is about 90€ / gCO_2/km in NEDC and 85 € / gCO_2/km in WLTP.

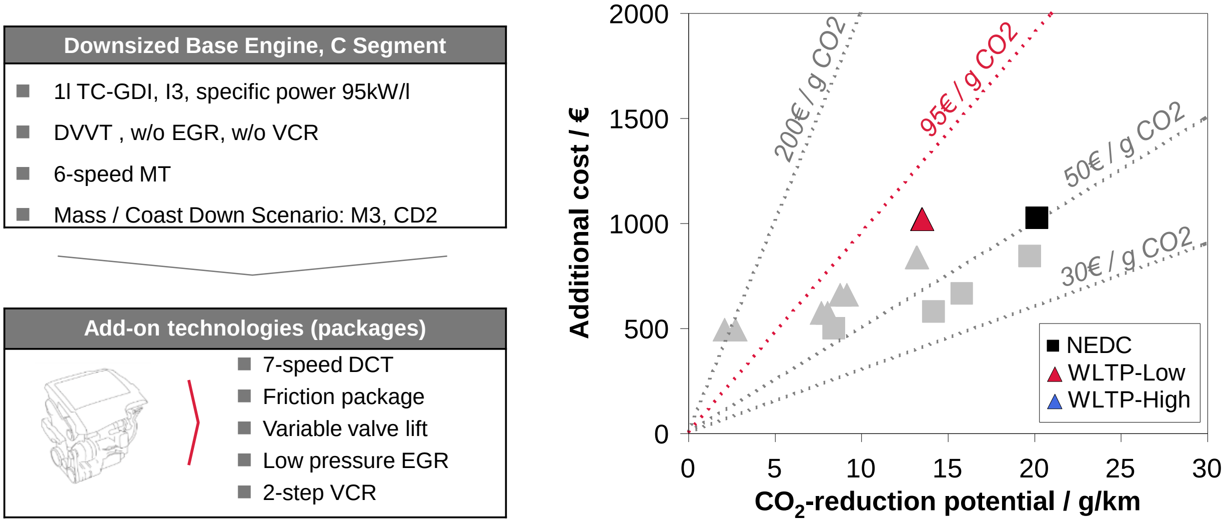


Figure 5.21 Segment C cost analysis for transmissions upgrade, for frictional upgrade, for variable valve lift, for low pressure EGR and for 2-step VCR

Going back to the Miller variant, Figure 5.22 shows the effect of an additionally installed low pressure EGR system. Since this measure improves the knock limitation at higher engine load, in WLTP the CO_2 emission reduction is higher. The total system cost ratios are ~45 € / gCO_2/km for NEDC and below 65 € / gCO_2/km for WLTP. The additional costs are 103 € and the CO_2-benefit is 0.9 gCO_2/km for the NEDC and up to 2.1 g CO_2/km for the WLTP.

For the cooled LP EGR system in addition to the Miller cycle engine the cost ratio is about 115€ / gCO_2/km in NEDC and 50 € / gCO_2/km in WLTP.

In conclusion, the Miller variant combined with LP-EGR shows a higher total CO_2-reduction potential and is at the same time more cost-efficient than the variant with two step VCR and LP-EGR.

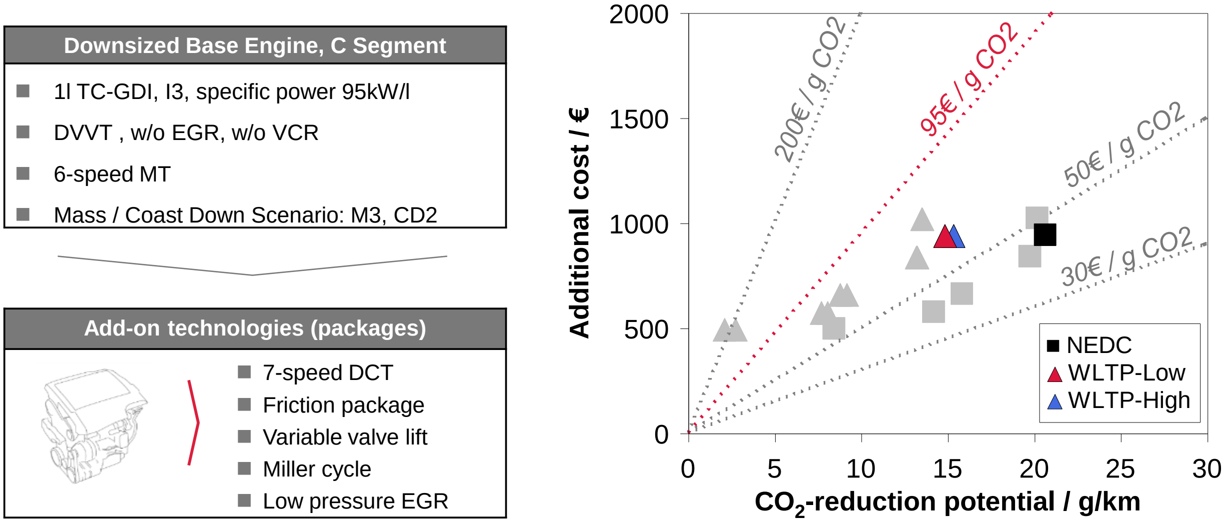


Figure 5.22 Segment C cost analysis for transmissions upgrade, for frictional upgrade, for variable valve lift, for Miller cycle and for low pressure EGR

A further downsized variant is shown in Figure 5.23. As already shown in Chapter 4.4.2.13, the extreme downsized version is less beneficial in terms of CO_2-emissions than the former three variants within this section. However, the total add-on package costs are in the same range. Therefore, the cost ratios end up at worse areas, compared to best 1 l TC version.

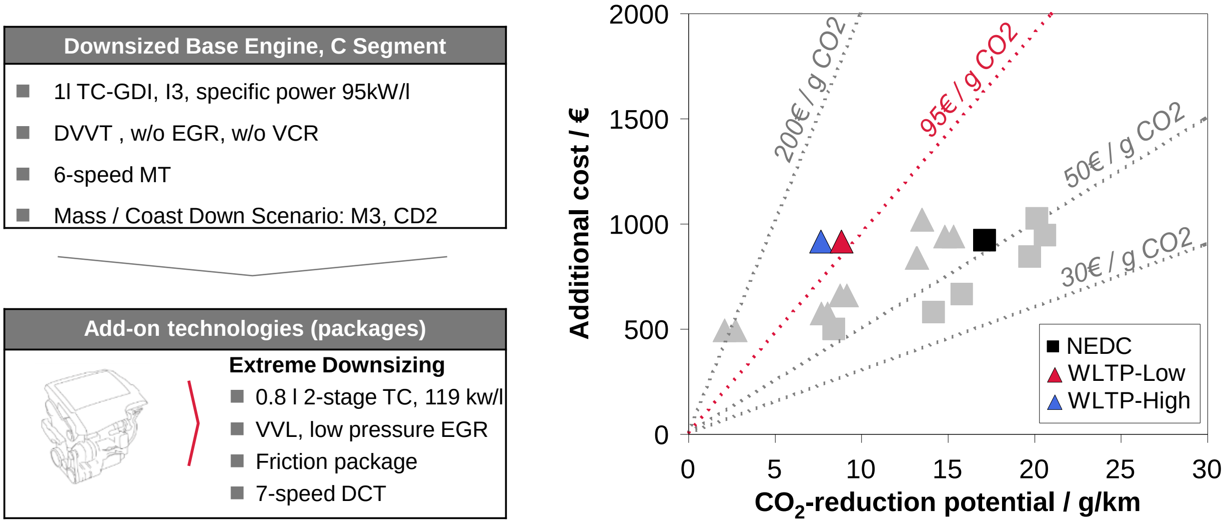


Figure 5.23 Segment C cost analysis for transmissions upgrade, for frictional upgrade, for variable valve lift, for low pressure EGR and for further engine downsizing

Figure 5.24 shows the analysis of P0 hybridization, installed to the best conventional version in terms of CO_2-emissions. The hybridisation leads to further improvement of 3 gCO_2/km in the NEDC and 5.1 gCO_2/km in the WLTP. The extra costs just for the 48V P0 hybridization are 694 €. Anyway, the overall cost ratios for the complete package are still below the critical line of 95 € / gCO_2/km. They are 69 € / gCO_2/km for the NEDC and about 80 € / gCO_2/km for

WLTP. In total, this package represents a cost-effective variant for a considerably high CO_2-reduction potential.

For the 48V P0 mild hybrid system in addition to the Miller cycle with cooled LP-EGR engine the cost ratio is about 230€ / gCO_2/km in NEDC and 135 € / gCO_2/km in WLTP.

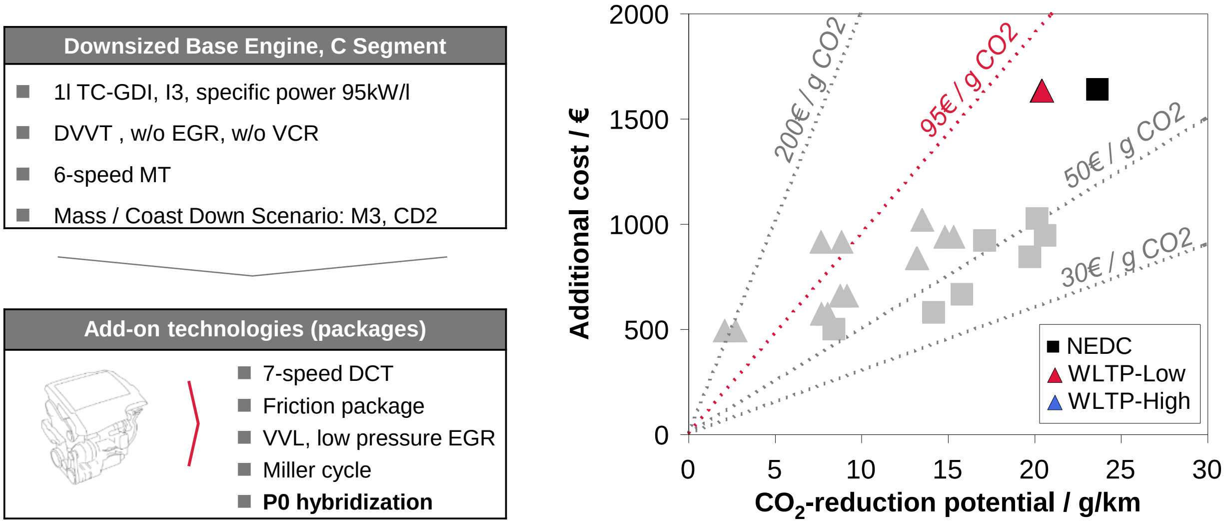


Figure 5.24 Segment C cost analysis for transmissions upgrade, for frictional upgrade, for variable valve lift, for Miller cycle, for low pressure EGR and for P0 hybridization

In a last step, also the extreme downsized version was equipped with a P0 hybridization. For the NEDC, this is a slightly less cost-effective technology package, compared to the P0 version in Figure 5.24. But with a cost ratio of 73 € / gCO_2/km, still a worthwhile one (Figure 5.25).

In case of the WLTP, the hybridized extreme downsizing version hardly exceeds the CO_2-potential of other variants without hybridization. At the same time, the total package costs are almost twice as high, which leads to cost ratios of over 100 € / gCO_2/km. As the conventional extreme downsized version, also the hybrid variant seems less attractive, with regard to the WLTP.

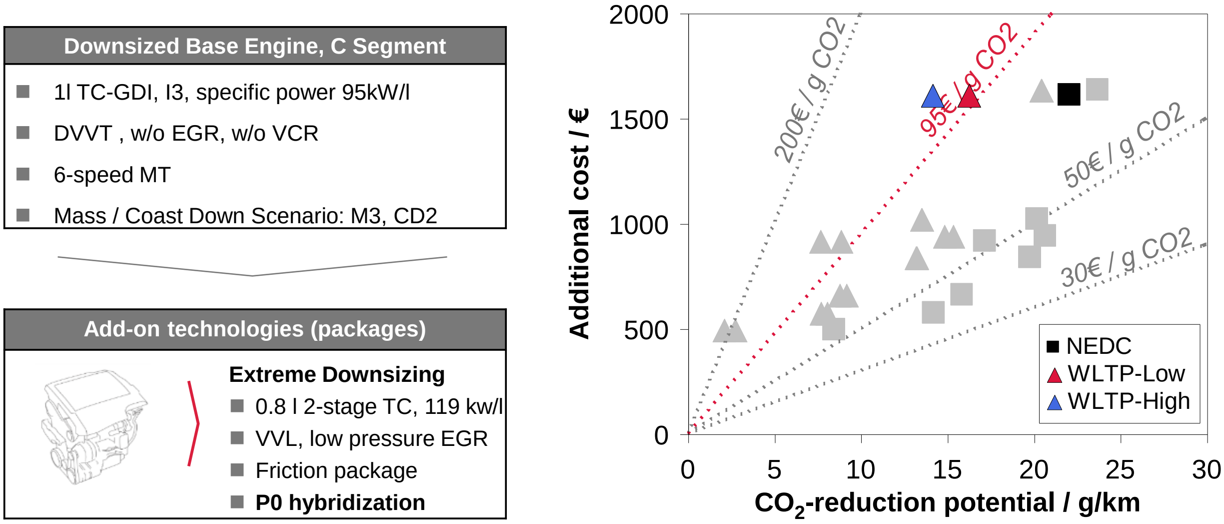


Figure 5.25 Segment C cost analysis for transmissions upgrade, for frictional upgrade, for variable valve lift, for further engine downsizing, for low pressure EGR and for P0 hybridization

5.2.2 Segment E

A much higher CO_2-potencial for the start&stop technology can be seen in segment D and E, as shown in Figure 5.26. One reason is, of course, the higher engine displacement, which is directly proportional to the idle fuel mass flow. However, the main impact, in this case, is the avoided torque converter losses of the automated transmission in the stopping phases of the driving cycles.

The high CO_2-reduction leads to very cost-effective ratios of 4 € / gCO_2/km for the NEDC and around 10 € / gCO_2/km for the WLTP.

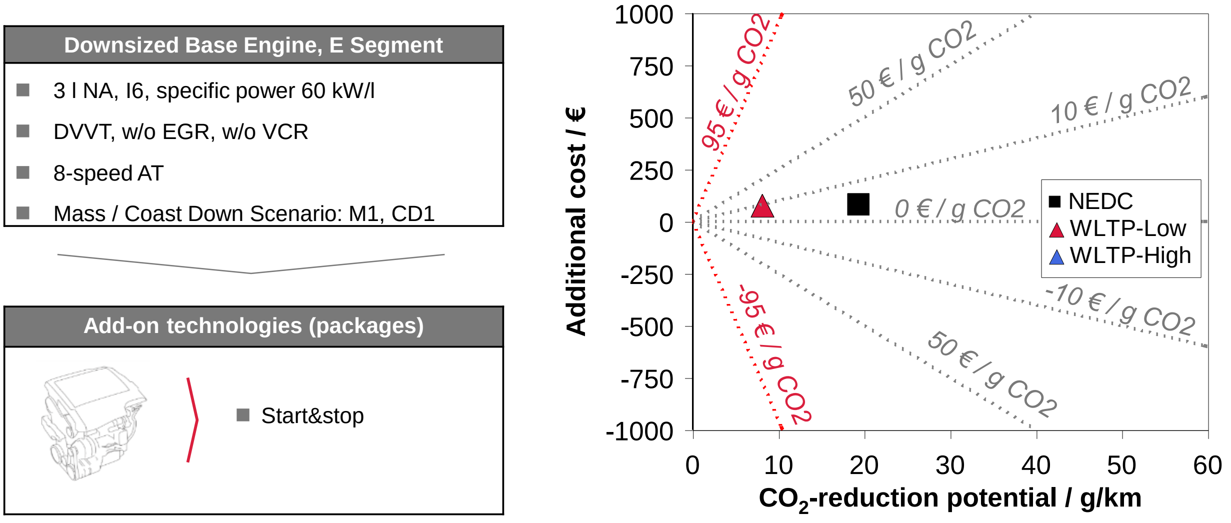


Figure 5.26 Segment E cost analysis for start&stop technology

An interesting case appears in the first downsizing step of segment D and E (see Figure 5.27). Along with the strong CO_2-reduction, also the package costs decrease. This effect occurs due to the step from a V6 to a I4 engine, which means highly reduced costs for crankcase, cylinder head, crank train, valve train and timing drive(see Chapter 3). Because of this fact, downsizing ,in these segments, already became a state of the art technology within the European market.

As for the E segment, further powertrain technologies will be compared with respect to the segment variant which present already the lowest mass scenario (M3: 20 % mass reduction) and the first coast down reduction step (CD2: 25 % RRC reduction and 10% CdA reduction). The baseline is now always a further downsized 1.6 l TC DI I4 variant with a 10-speed DCT. Unlike segment C, no additional transmission upgrade was performed within segment D and E. Without the cost intensive step from a manual to an automated transmission, the cost against CO_2-potencial ratios become much more attractive for each engine technology package, shown in the following Figures.

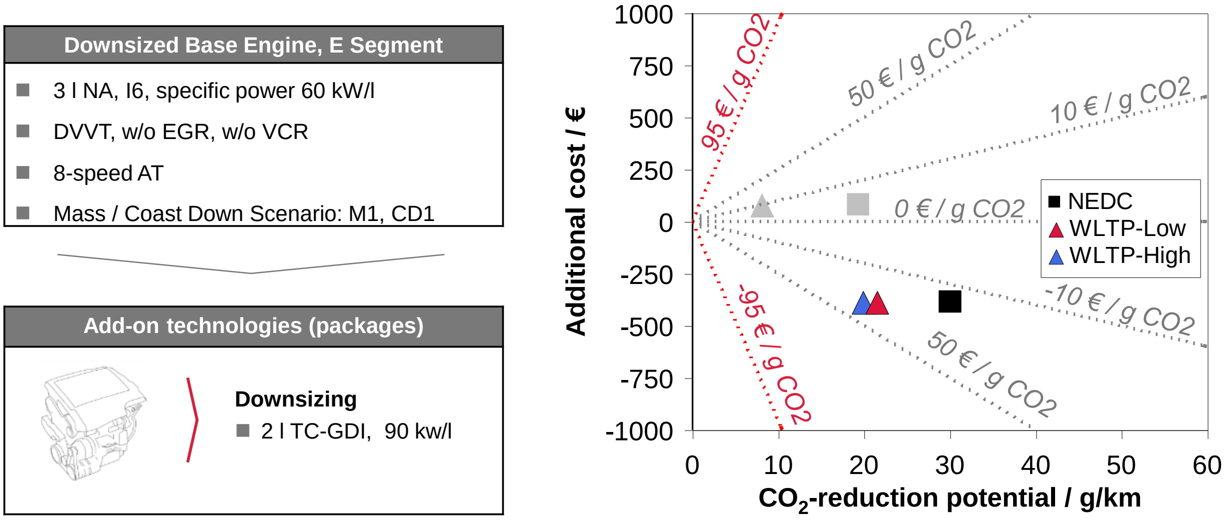


Figure 5.27 Segment E cost analysis for engine downsizing

The CO_2-reduction potential of around 5 gCO_2/km for the friction package (see Figure 5.28) is similar to the one in segment C. The additional cost of 80 € leads to a very good cost effectiveness of friction reduction also in segment E. But without the strongly cost driving transmission upgrade step, cost ratios for the friction optimized package in the heavier segments improve to values around 15 € / gCO_2/km for the NEDC and still below 20 € / gCO_2/km for the WLTP.

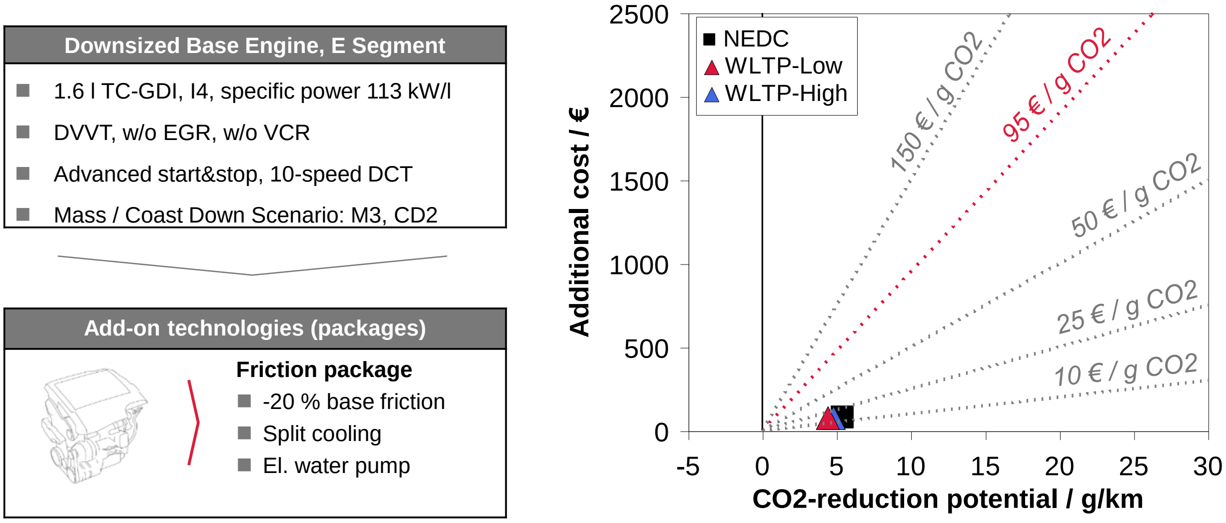


Figure 5.28 Segment E cost analysis for frictional upgrade

The variable valve lift technology, shown in Figure 5.29, leads to further CO_2-potentials of 2 gCO_2/km (NEDC) respectively around 1.5 gCO_2/km (WLTP). With extra costs of 110 €, the cost ratios for the entire technology package increase to 25 – 32 € / gCO_2/km.

For the VVL system in addition to the friction reduced downsized 1.6l TC DI engine the cost ratio is about 50€ / gCO_2/km in NEDC and 75 € / gCO_2/km in WLTP.

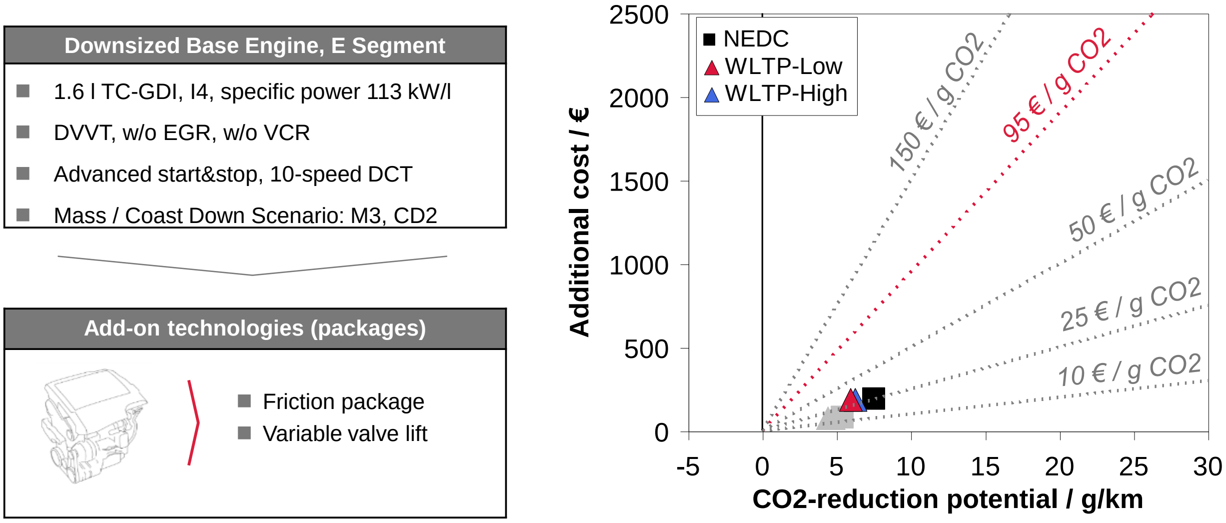


Figure 5.29 Segment E cost analysis for frictional upgrade and for variable valve lift

The use of the Miller cycle gives an additional CO_2-reduction of around 5 gCO_2/km with respect to the former variant. The total package costs increase by 104 € to 293 €. As a result, cost ratios improve slightly to 23 – 27 € / gCO_2/km.

For the Miller combustion with 2-step TC shows in this configuration the cost ratio of about 20€ / gCO_2/km in NEDC and 18 € / gCO_2/km in WLTP.

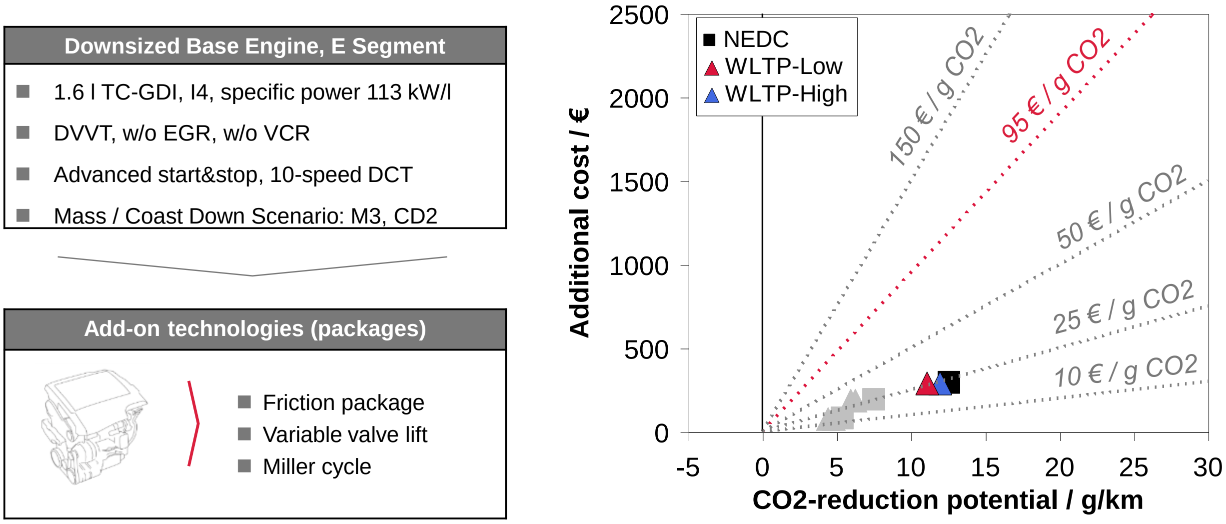


Figure 5.30 Segment E cost analysis for frictional upgrade, for variable valve lift and for Miller cycle

In Figure 5.31 a 2-step VCR system and LP-EGR was installed to the variant with friction optimization and VVL. CO_2-emission can be reduced significantly by 8 - 9 gCO_2/km within this step. However, extra cost of 385 € increase the overall package ratios to 36 – 42 € / gCO_2/km.

The package of LP-EGR and 2-step VCR shows in this configuration the cost ratio of about 48€ / gCO_2/km in NEDC and 42 € / gCO_2/km in WLTP.

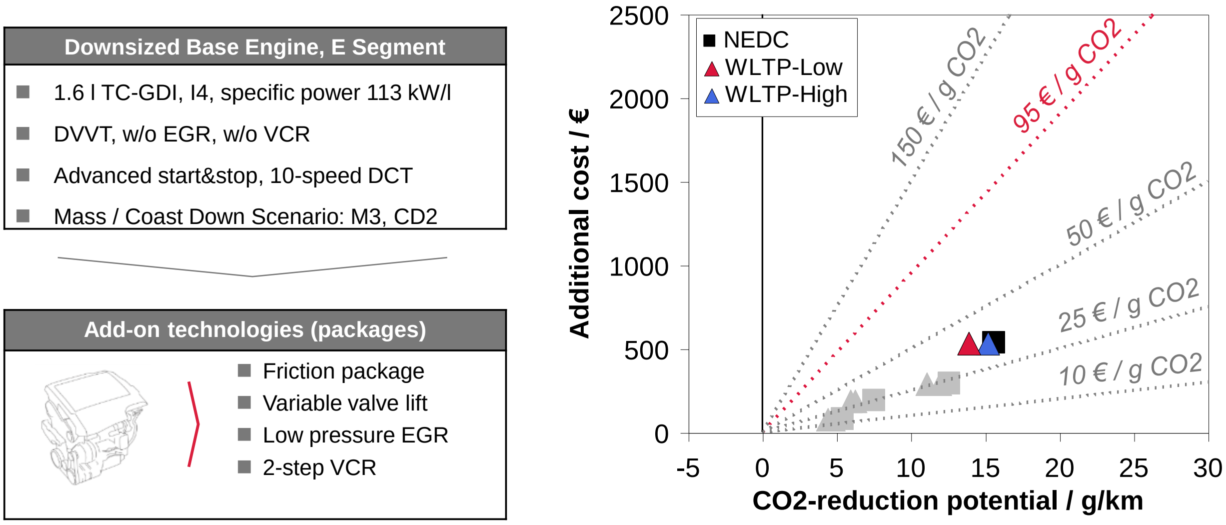


Figure 5.31 Segment E cost analysis for frictional upgrade, for variable valve lift, for low pressure EGR and for 2-step VCR

In Figure 5.32 LP-EGR was added to the Miller variant as shown in Figure 5.30. With further CO_2-potentials of 1 - 2 gCO_2/km and additional costs of 132 € the total ratio increases slightly, to value around 30 – 32 € / gCO_2/km.

Unlike segment C, the Miller version with LP-EGR doesn't reach the CO_2-reduction potential of variant with 2-step VCR. Nevertheless, it is again the more cost-efficient one.

The package of LP-EGR added to the Miller combustion variant in this configuration shows the cost ratio of about 120€ / gCO_2/km in NEDC and 60 € / gCO_2/km in WLTP.

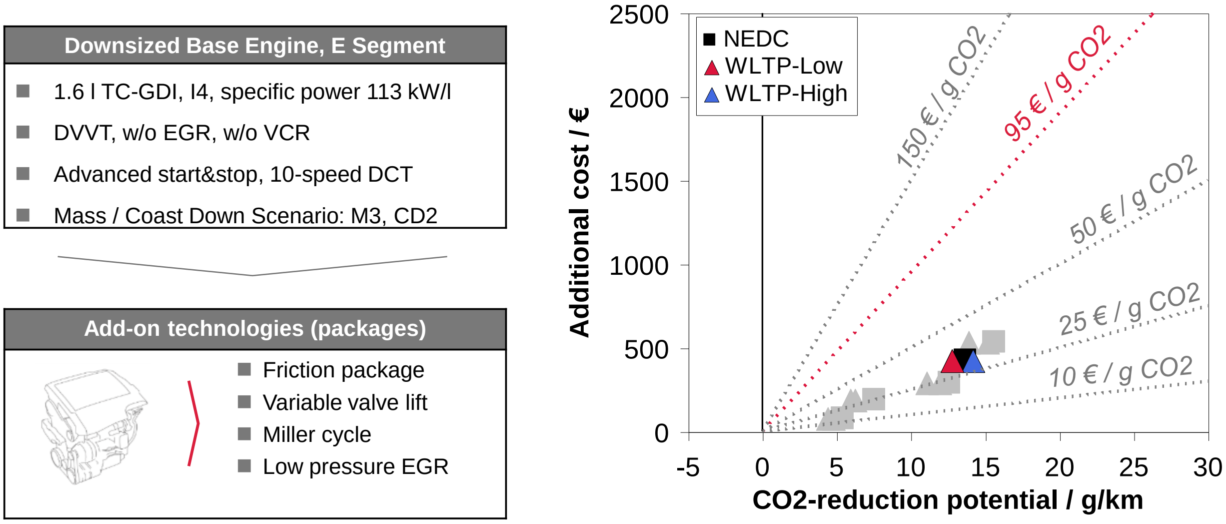


Figure 5.32 Segment E cost analysis for frictional upgrade, for variable valve lift, for Miller cycle and for low pressure EGR

As described in Chapter 4, the dedicated EGR variant was realised with a 2.4 l SC/TC engine, according to the SAE paper: *"A Demonstration of Dedicated EGR on a 2.0 L GDI Engine", Chadwell, C., Alger, T., Zuehl, J., and Gukelberger, R SAE Int. J. Engines 7(1):434-447, 2014.*

Compared to the same basis as the other technology packages of segment E within this section, the dedicated EGR variant shows CO_2-benefis of up to 5.6 gCO_2/km for the WLTP cycles. Whereas, in the NEDC the CO_2-emissions are even 2 gCO_2/km higher than for the base variant. Also the extra costs of 498 €, compared to the used basis, are relatively high, which is manly caused by the use of mechanical charger, as explained in Chapter 3.

As a result, the shown CO_2-benefit within the WLTP cycles can only be achieved with high cost ratios of 88 – 175 € / gCO_2/km.

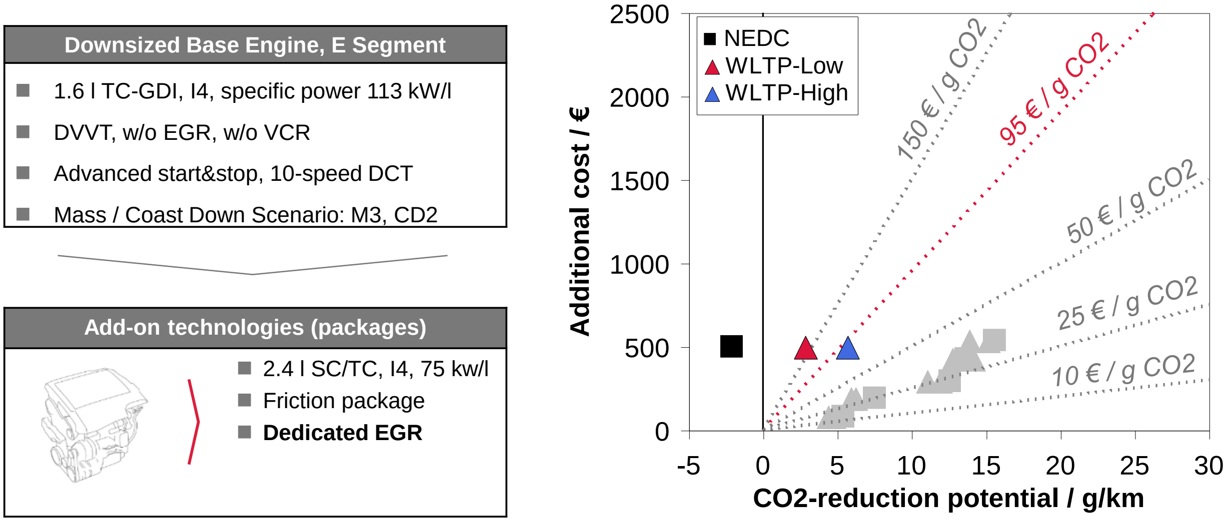


Figure 5.33 Segment E cost analysis for frictional upgrade, for engine variation and for dedicated EGR

As seen for segment C, the extreme downsized version in segment D and E is less beneficial in terms of CO_2-emissions than other less downsized variants within this section. However, the extra costs (266 €) for the extreme downsized engine now are lower, with respect to these variants. So, in total comparable cost ratios of 25 – 37 € / gCO_2/km emerge.

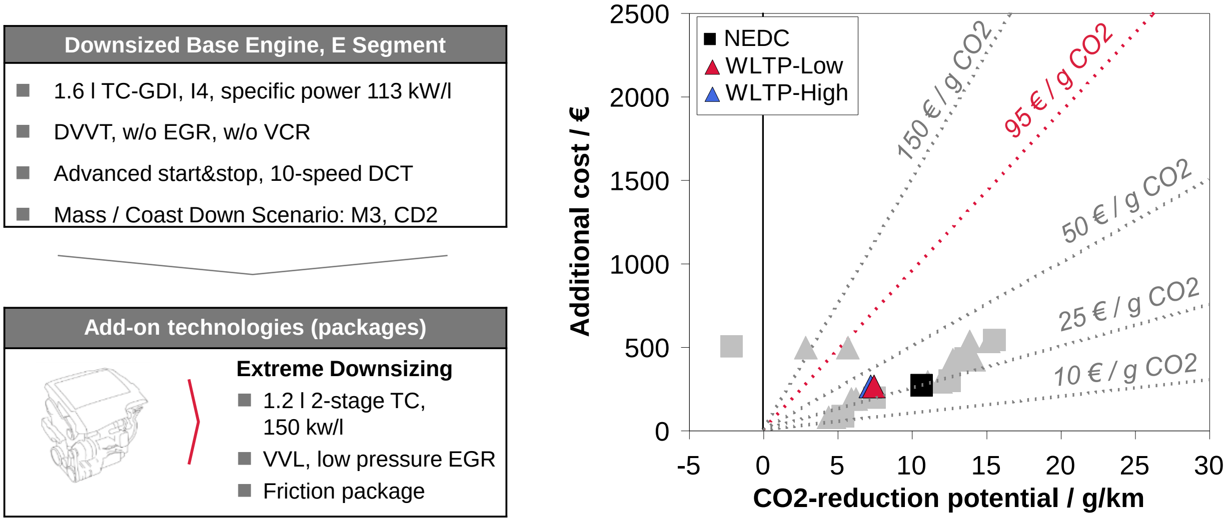


Figure 5.34 Segment E cost analysis for frictional upgrade, for variable valve lift, for low pressure EGR and for further engine downsizing

Figure 5.35 shows the analysis of P2 hybridization, installed to the best conventional version in terms of CO_2-emissions. Although, the hybridisation leads to further improvement of around 11.5 -13 gCO_2/km, the extra costs amount to 1978 € at the same time. The cost ratios increase to values around the critical line of 95 € / gCO_2/km. Anyway, technology package is still interesting, in terms of the achievable CO_2-reduction potential.

The P2 hybridization alone has a much worse cost to benefit ratio of 172€ / gCO_2/km in NEDC and 152€ / gCO_2/km in WLTP.

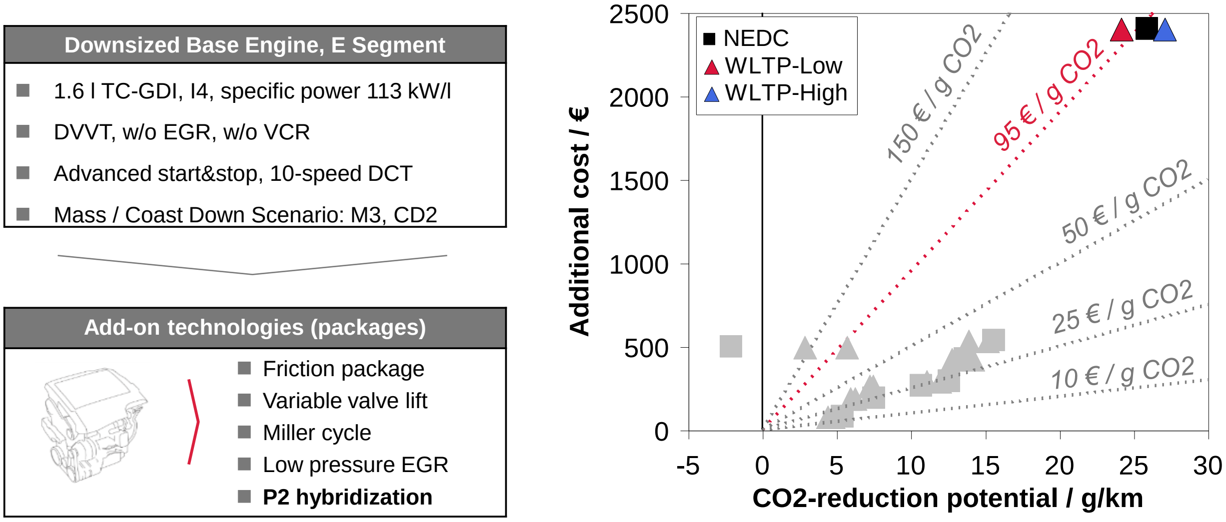


Figure 5.35 Segment E cost analysis for frictional upgrade, for variable valve lift, for Miller cycle, for low pressure EGR and for P2 hybridization

The installation of a P2 hybrid to the extreme downsized version, leads to higher CO_2-benefits than any variant without hybridisation, but less than the other hybrid package. With cost ratios of 102 – 122 € / gCO_2/km, it is less cost-efficient too.

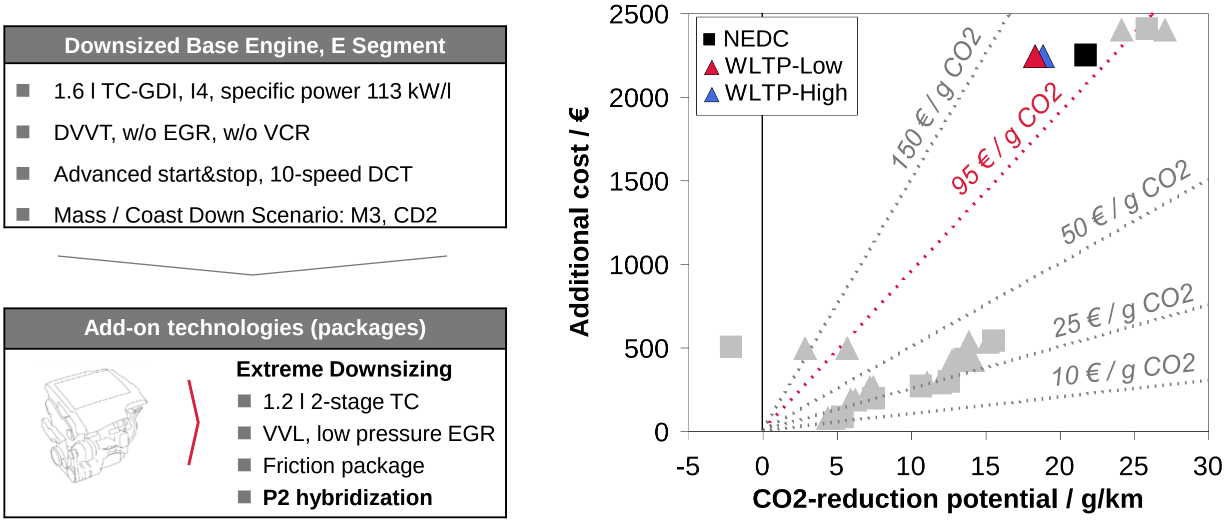


Figure 5.36 Segment E cost analysis for frictional upgrade, for variable valve lift, for low pressure EGR, for further engine downsizing and for P2 hybridization

6 Conclusions

This Chapter shows the main conclusions of the results, divided into:

- Cycle influence, focused in the EU market,
- mass and coast down,
- powertrain technologies,
- aftertreatment systems, and
- hybrid systems.

6.1 Cycle influence

The CO_2 emissions are considerably higher in the WLTP TM_H with respect to the NEDC ranging from 10 to 30% higher depending on the configuration. The influence of the new boundary conditions for the certification in WLTP is the cause for the most of the fuel consumption difference between NEDC and WLTC. The cycle itself leads to operation area of the engine at higher efficiency and can compensate partly the higher energy demand.

The most of the technologies gets more CO_2 reduction potential in the NEDC than in the WLTP, due to the engine is normally working at lower loads in the former and subsequently less efficient areas of the engine

- This is especially important in Manual Transmission vehicles: the gear shifting strategy in the NEDC is fixed and it cannot be optimized (apart from optimizing the gear ratios)
- In Hybrids, it is possible to get more energy to recuperate in the WLTP, which can lead to higher potential in such cycle with respect to the NEDC

6.2 Mass and coast down

The mass and coast down reduction steps influence heavily the final result:

- As an example, in Segment C both mass reduction Scenario II and coast down Scenario I are getting a total benefit of 23 g CO_2/km in the WLTP TMH cycle, over a base of 127 g /km (downsized version without mass reduction), which means ~18 % of the reduction is due to that
- The maximum final benefit for such case with the best analyzed powertrain configuration is ~30 %

6.3 Powertrain Technologies

The **Start and Stop (S&S)** from 3 km/h and the advanced version from ~15 km/h are interesting technologies when focusing in the NEDC and FTP75 cycles, since the stop phases share is high within the cycles, giving benefits which range from 4-5 % in MT or DCT vehicles and up to 10 % for AT vehicles (NEDC based). In the case of AT vehicles, these higher benefits come due to the low efficiency of the torque convertor at idle and low speed conditions that are eliminated with the engine stop.

However, in cycles such as the WLTP, where the share of the stop phases is much lower, the benefits of the S&S are lower, and in the range of 2% for MT and DCT vehicles and slightly higher for AT vehicles equipped with a torque converter (~2-3%). Nevertheless, it is an interesting technology which is already widespread into the market.

The **downsizing** in combination with other technologies such as turbocharging and DCT transmissions is proved to be an interesting technology, especially for gasoline:

- The costs can even be lower than for the original engine, especially when the steps are higher and involve reduction of the number of cylinders.
- The engine usually works at more efficient areas in the downsized version.
- The combination of the **downsized version with downspeeding strategies** (with automatic transmissions), especially for the case of the **dry-clutch DCT transmissions** and NEDC (the shifting strategy can be optimized) is beneficial.
- In diesel, the downsizing is limited due to the increase of the engine-out NOx emissions, i.e. the engine works higher loads with higher specific NOx emissions. When installing LNT as deNOx systems, the higher fuel penalty coming from the LNT regenerations can compensate the CO_2 reduction.
- The automatic transmissions get more potential in the NEDC, but this is becoming lower in the WLTP. However, comfort reasons should also be underlined especially for **wet clutch DCTs and ATs.**
- Regarding wet clutch **DCTs and ATs**, the relatively low efficiency of such systems at the lower loads is due to the need of oil pumping to the different transmission systems and cooling the clutches with oil flow. In the future, it might be expected to find variable oil pump with a smart control which allows increasing the overall efficiency of these systems, and therefore reducing the fuel consumption heavily.
- It is necessary to say that for some of the functionalities coming from the hybrid systems, such as coasting or sailing, the use of automatic transmissions (DCT or AT) is mandatory.

Measures for optimizing the **friction**, by considering base engine measures, electrical water pump, split cooling and variable oil pump, are interesting technologies, giving a good CO_2benefit at a low cost (~4% benefit at 10-20 €/g CO2). This is usually the first measure that is applied to the engines before considering any other technologies:

- An **average friction reduction** can be achieved by crankshaft offset to reduce the piston side forces, bearings adjustment, and in general improved materials and manufacturing processes. This leads to general 10-15 % reductions with virtually no cost.
- The **split cooling** allows separating the cooling circuits around the cylinder heads and engine block by only installing one additional thermostat. Therefore, the temperature of the circuit through the engine block can increase faster, reducing the warm-up phase, and reducing the effective friction. Nowadays, it can also be found the separation into three circuits by also considering one extra circuit for the EGR coolers (by also including two electrical water pumps for two of the circuits).
- The installation of an **electrical water pump** instead of a mechanical one also allows further reducing the friction by modulating the pumping energy, which can be lowered especially in the cold phases, supporting also a faster engine warm-up.

Other technologies are still interesting CO_2 potential reductions:

- **The combination of cooled High Pressure and cooled Low Pressure EGR** technologies make possible to have high EGR rates without compromising the turbocharger efficiency. At low loads, the cooled HPEGR allows high EGR rates but still recirculating gas with a higher density (lower temperature) due to the installation of the coolant; therefore the volumetric and turbocharger efficiencies keep high. At higher loads, the LPEGR circuit permits to keep high EGR rates up to areas close to the full load, without a low fuel penalty. Overall, this is an interesting technology for NOx reduction, but also to keep an appropriate fuel efficiency, with ratios ranging ~20-30 € / g CO_2/ km.
- **The Low Pressure EGR in Gasoline** is mainly a measure to reduce the knock limitation of the engine at higher load. The de-throttling at low load is favorable with internal EGR due to the hotter gas and therefore better combustion stability. At mid and high engine load the reduced knock tendency directly improve the efficiency. With low pressure EGR used up to the full load with advanced boosting technologies the compression ratio can be increased. That leads directly to an efficiency increase at part load. Cooled LP-EGR can be combined with other technologies like Miller combustion and 2-step VCR to use synergies.
- **2 stage VCR** system is in the range of 30-50 € / g CO_2/ km, still interesting for reaching further CO_2 reductions in the short future for gasoline, but also for diesel. The cost of a possible fully variable VCR might possibly not justify the possible extra benefit, since the system would become much more complex (this was not analyzed within the current project).
- **The Variable Valve Timing (VVT)** for diesel is mainly a heating strategy, used for emissions management especially when for example SCR systems are installed. In this project, the late exhaust valve opening has been explored:
    - The exhaust temperature raises faster, therefore the SCR and DOC systems get higher efficiency soon.
    - It can be used for managing internal EGR, by late exhaust valve opening.
    - The HC and CO emissions become lower, as the gases stay more time into the cylinder.
    - The combustion efficiency decreases, but the fuel penalty tends to be lower than the one achieved by means of other heating strategies, such as post-injections.
    - The exhaust mass flow tends to decrease, which reduce the space velocity of the exhaust gasses into the aftertreatment systems (slightly increasing the efficiency of these).
    - For LNT (and EU market) systems, the use of VVT is not mandatory; nevertheless it might be beneficial for allowing engine rich operation at lower loads. For this, the VVT allows a rich stable combustion from lower loads. Anyway, this measure is more emissions related, and might have a negligible influence on the CO_2 potential reduction.
- **Miller cycle** can be realized by early or late intake valve closing in combination with an increase of compression ratio. This leads to lower knock limitation at mid and high loads of gasoline engines and improve the FC by 3.5% to 4.5%. Due to the reduction of the volumetric efficiency an increased boosting demand occurs in comparison with conventional valve timings. This lead for power demands higher than 70kW/l displacement to the need of advanced boosting systems like VTG, e-booster or 2-stage TC. This increase the cost a lot while at lower specific power demand the Miller combustion is nearly cost neutral. The Miller combustion system in gasoline engines is

very beneficial at moderate extra cost and can be combined with other technologies like cylinder deactivation, lean combustion and cooled LP-EGR.

- **Dedicated EGR** is an efficient combustion system with high cooled EGR rates for gasoline engines. The knock limitation in higher loads is improved as well the engine dethrottled at lower loads. The fixed EGR rate from one of the 4 cylinders lead to a higher boosting demand that is compensated with an additional supercharger to the conventional turbocharger. The engine can achieve low BSFC values and has the drawback of low specific power even with 2-stage boosting combining supercharger and turbocharger. This leads to a lower potential of downsizing and lower FC saving potential compared to Miller combustion with cooled LP-EGR. Due to this no FC benefit in NEDC can be realized compared to a TC DI downsizing concept with optimized friction. In WLTP the FC reduction is about 2.2 to 3.7%.
- **Variable Valve lift in Gasoline (VVL)** is used to reduce the throttle losses in part load by realizing an early intake valve closing and set the air mass on demand without using the conventional throttle. This leads to strongly reduced gas exchange losses and improve the FC in the low part load. These systems are well known from BMW as Valvetronic or Toyota as Valvematic. The FC potential is in NEDC higher compared to WLTC due to the lower engine load used in the cycle. In NEDC the FC benefit is about 2% in WLTP about 1%.

6.4 Aftertreatment systems

The LNT systems are cost effective solutions for small segments, since the packaging is less restrictive and there is no need of installing AdBlue systems for dosing. However, the engine should be calibrated in order to get low engine out NOx emissions, since the LNT efficiency in the highway operation (with exhaust temperatures higher than 250 °C) is limited with respect to the SCR systems. This fact makes difficult to keep LNT systems-only for deNOx when considering real driving cycles, since the engine out NOx emissions will increase heavily and at the same time, the LNT efficiency will also decrease. Furthermore, a fuel penalty due to the system regeneration should be considered, and this penalty will also depend on the engine out NOx emissions. For example, and thinking of a Segment C vehicle and future RDE legislation, the installation of combined systems, e.g. LNT with a passive SCR, or the use of active SCR might be mandatory. Alternatively, mild hybrid configurations (such as the P0 BSG) which reduce the engine out NOx emissions in combination with low emissions shifting strategies might be assessed.

The close-coupled SDPF (SCR on filter) aftertreatment systems allows reaching higher deNOx efficiencies (alone or in combination with others) especially when the exhaust temperature increases, which maximize the efficiency (>85-90%) when the engine out NOx emissions are also the highest. The SCR however presents null or low efficiency when the exhaust temperature is lower than 200 °C. In order to reduce the NOx, it is necessary to dose NH3 (coming from urea) into the system, which normally is achieved by dosing AdBlue (aqueous solution which contains 32.5 % of urea). Such systems are usually called active SCR systems, and allow calibrating the engine at lower EGR rates than when using LNT systems, giving the possibility of getting some CO_2 benefit. Furthermore, there is no need to run rich modes to clean the system as it happens with the LNT.

As also mentioned before, and even if not considering within this project, there are other alternatives for deNOx aftertreatment, which are attractive to the OEMs and can already be found today into the market:

- LNT + passive SCR system, where the NH3 required for the SCR is produced when purging the LNT. This can give an extra 5-10 % of deNOx efficiency depending on the conditions.
- LNT + active SCR solutions, where the maximum efficiencies are reached in the cold phase thanks to the LNT, and at higher temperatures thanks to the SCR. This requires anyway the installation of the AdBlue related systems.
- SDPF (close-coupled) + underfloor SCR or ammonia slip catalyst, in order to also maximize the efficiency by taking profit of the AdBlue exceedance in the upstream SCR system.
- It might also be mentioned the use of cooled low pressure and high pressure EGR systems in order to reduce the engine out NOx.
- Other alternatives and combined solutions can also be found into the market by combining the main systems depicted before.
- Finally, it should be mentioned that the requirements for the US market in terms of emissions are much more aggressive than for the EU one, making that LNT only systems are not a valid alternative anymore. For the US market, the use of combined systems with active SCR systems and low pressure EGR is possible the best option.

For gasoline, the **TWC** is the common used system to reduce the tailpipe emissions. This system is very effective and can reduce the emissions by over 95% if the engine is operating at stoichiometric conditions. In areas were the engine is operated lean or rich the NOx or HC/CO emissions cannot be converted. Therefore lean or rich combustion has to be avoided in the driving cycle. In the future with respect to RDE maybe lean scavenging and fuel enrichment for component protection is not more possible and requires advanced boosting devices and active cooling of the exhaust gas with e.g. cooled exhaust manifolds and turbine housing.

The main challenge in terms of emission reduction for direct injection gasoline engines is the reduction of particle emissions. There are some possibilities to reduce particulate matter and particulate number emissions like advanced injection system combining PFI and DI, increased fuel rail pressure or the gasoline particulate filter (GPF). The GPF can be realized as stand-alone solution after a conventional TWC or as a coated GPF that also delivers the function of the TWC. This second solution is called 4WC. The 4WC has the main advantage that the additional cost is relative small due to the integrated solution with only one canning.

6.5 Hybrid systems

The **P0 configuration (Belt Starter Generator)** mounted in the Segments B and C, for both gasoline and diesel:

- It is able to reach a similar performance as the combination of other engine measures, but with a higher cost ratio, which ends around 70-95 €/ g CO_2.
- The potential might be slightly higher within the WLTP, due to have more energy recuperation potential in such cycle.

The **P2 configuration (Full parallel hybrid)** mounted in the Segments D, E, SUV and LCV for diesel and D, E for Gasoline:

- It is able to reach a similar performance as the combination of other engine measures, but with a higher cost ratio, which ends around 95€/ g CO_2.

- The potential by analyzing all the technologies within the hybrid package is higher in the NEDC, with respect to the WLTP, also influenced by all the other technologies

Regarding the hybrids and apart from the CO_2 potential, it can be mentioned:

- It has other benefits such as a better performance (by e-boosting, higher low-end torque), comfort and marketing.
- In case of the hybrid application, it might be an attractive solution by saving the cost of the engine technologies. A good example is Toyota, which usually installs hybrid components with a lower level technology at the engine source; in contrast with Europe manufacturers which normally invest more money into engine related technologies.
- Furthermore, for diesel it is possible to reduce the engine-out emissions by avoiding high load areas, and thus enabling cheaper aftertreatment configurations.
- There also exist other hybrid configurations, which have not been explored within this project and those which could lead to higher CO2 potentials. For example, the installation of plug-in hybrid vehicles (PHEV) in the gasoline segment.

7 Appendix: Diesel Engines

7.1 Summary of average CO_2 emissions for all segments

7.1.1.1 Segment B

	Segment B	EGR	T/C	VVT	CR	Engine friction	Hybridization	Transmission	EATS	Coast down set	REF / kg	TML / kg	TMH / kg
T1	Baseline	HP-CLP	single	not variable	not variable	baseline	without	MT5	LNT+CDPF	CD1	1249	1310	1438
T2	Start/Stop	HP-CLP	single	not variable	not variable	baseline	**start/stop**	MT5	LNT+CDPF	CD1	1249	1310	1438
T3	Downsizing/ Transmission/ M red. Scenario I	HP-CLP	single	not variable	not variable	baseline	start/stop	**MT6**	LNT+CDPF	CD1	**1229** **1107** **984**	**1290** **1168** **1046**	**1418** **1295** **1173**
T4	M reduction Scenario II	HP-CLP	single	not variable	not variable	baseline	start/stop	MT6	LNT+CDPF	**CD1 - CD2 - CD3**	**984**	1046	1173
T5	Friction, resistance	HP-CLP	single	not variable	not variable	**-20% friction, el. coolant pump**	start/stop	MT6	LNT+CDPF	CD2	984	1046	1173
T6	Engine technologies	**CHP-CLP**	single	not variable	not variable	-20% friction, el. coolant pump	start/stop	MT6	LNT+CDPF	CD2	984	1046	1173
T7	Aftertreatment	CHP-CLP	single	**variable**	not variable	-20% friction, el. coolant pump	start/stop	MT6	**DOC+SDPF**	CD2	999	1061	1188
T8	Without weight reduction	CHP-CLP	single	variable	not variable	-20% friction, el. coolant pump	start/stop	MT6	DOC+SDPF	CD2	**1244**	1305	1433
T9	Hybrid, P0 , 48V	CHP-CLP	single	variable	Not variable	-20% friction, el. coolant pump	Adv. start/stop	DCT7	DOC+SDPF	CD2	1007	1068	1196

Table 7.1 Summary of variants for Segment B

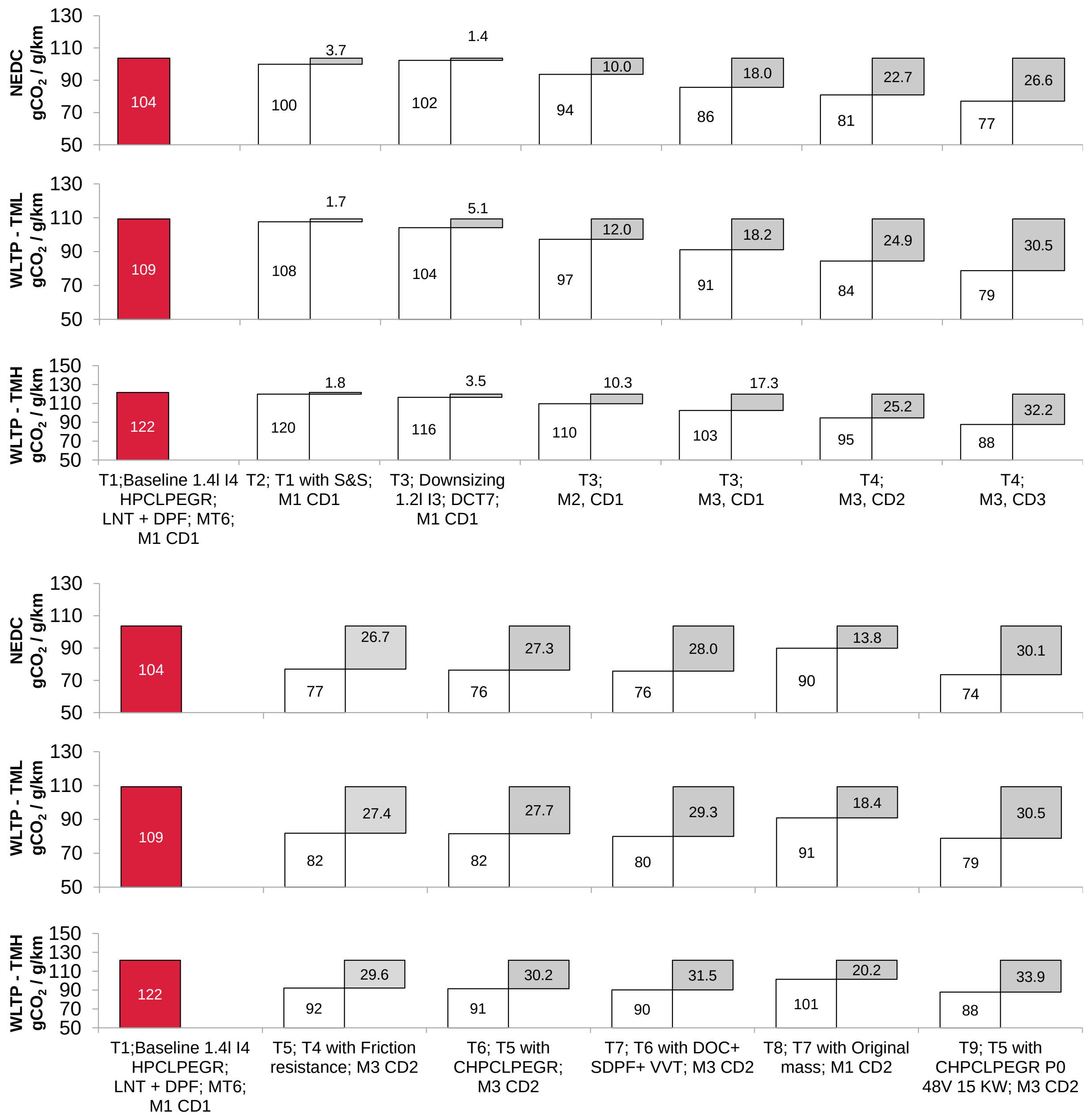


Figure 7.1 Overview of results for Segment B (EU cycles).

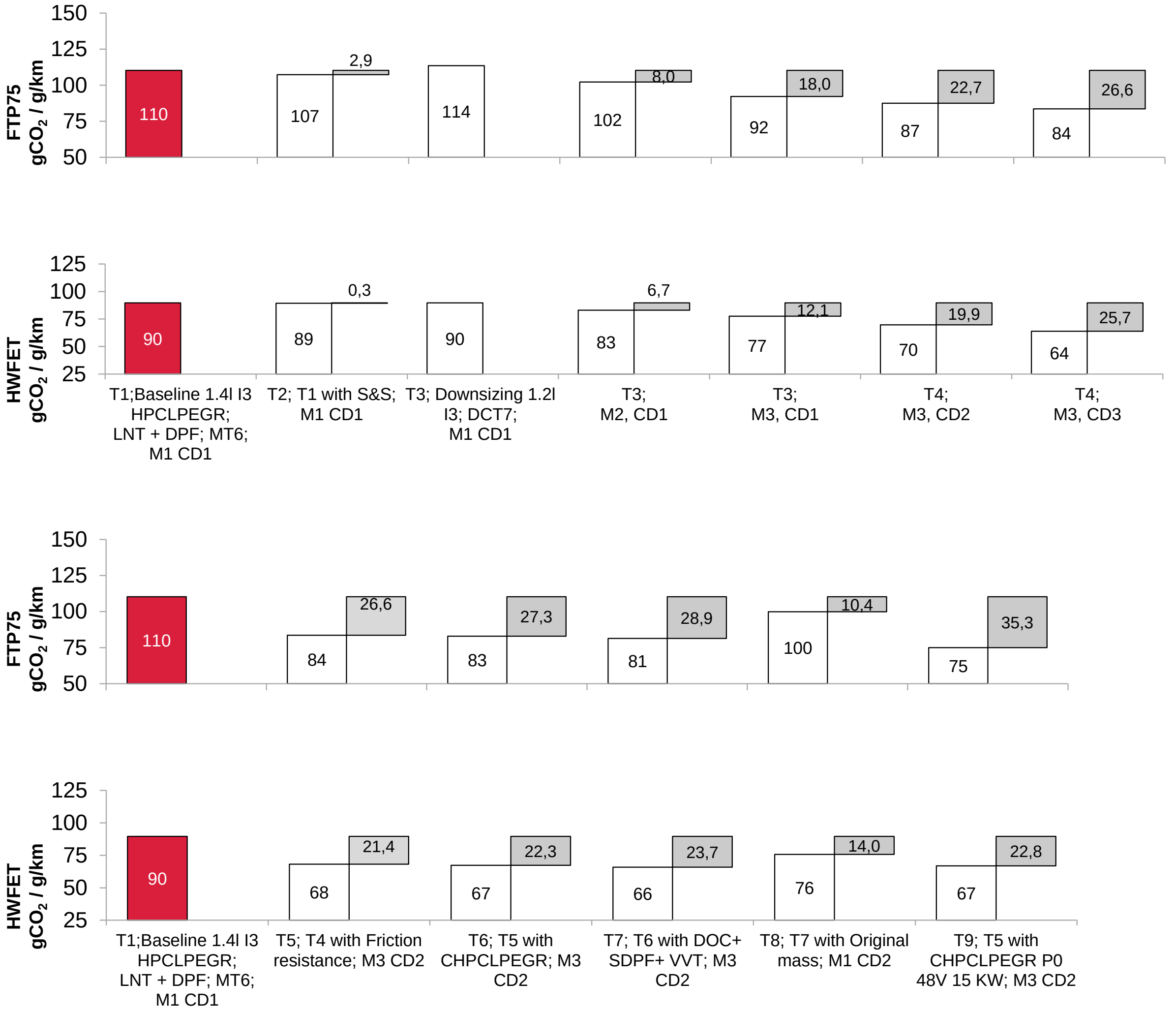
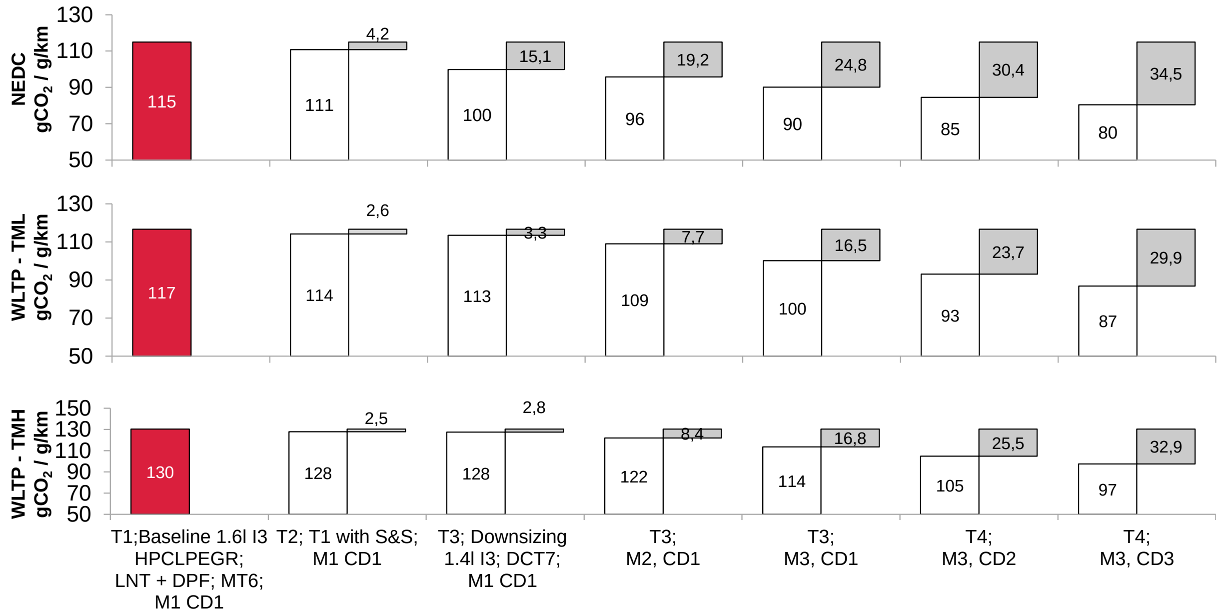


Figure 7.2 Overview of results for Segment B (US cycles).

7.1.1.2 Segment C

Segment C		EGR	T/C	VVT	CR	Engine friction	Hybridization	Transmission	EATS	Coast down set	REF / kg	TML / kg	TMH / kg
T1	Baseline	HP-CLP	single	not variable	not variable	baseline	without	MT6	LNT+CDPF	CD1	1459	1530	1679
T2	Start/Stop	HP-CLP	single	not variable	not variable	baseline	**start/stop**	MT6	LNT+CDPF	CD1	1459	1530	1679
T3	Downsizing/ Transmission/ M red Scenario I	HP-CLP	single	not variable	not variable	baseline	**Adv. start/stop**	**DCT7**	LNT+CDPF	CD1	**1469** **1326** **1182**	**1540** **1397** **1254**	**1689** **1546** **1402**
T4	M reduction Scenario II	HP-CLP	single	not variable	not variable	baseline	Adv. start/stop	DCT7	LNT+CDPF	**CD1-CD2-CD3**	**1182**	1254	1402
T5	Friction, resistance	HP-CLP	single	not variable	not variable	**-20% friction, el. coolant pump**	Adv. start/stop	DCT7	LNT+CDPF	CD2	1182	1254	1402
T6	Engine technologies	**CHP-CLP**	single	not variable	**2-step VCR**	-20% friction, el. coolant pump	Adv. start/stop	DCT7	LNT+CDPF	CD2	1182	1254	1402
T7	Aftertreatment	CHP-CLP	single	**variable**	2-step VCR	-20% friction, el. coolant pump	Adv. start/stop	DCT7	**DOC+SDPF**	CD2	1197	1269	1417
T8	Without weight reduction	CHP-CLP	single	variable	2-step VCR	-20% friction, el. coolant pump	Adv. start/stop	DCT7	DOC+SDPF	CD2	**1484**	1555	1704
T9	Hybrid, P0 , 48V	CHP-CLP	single	variable	Not variable	-20% friction, el. coolant pump	Adv. start/stop	DCT7	DOC+SDPF	CD2	1205	1276	1425

Table 7.2 Summary of variants for Segment C.



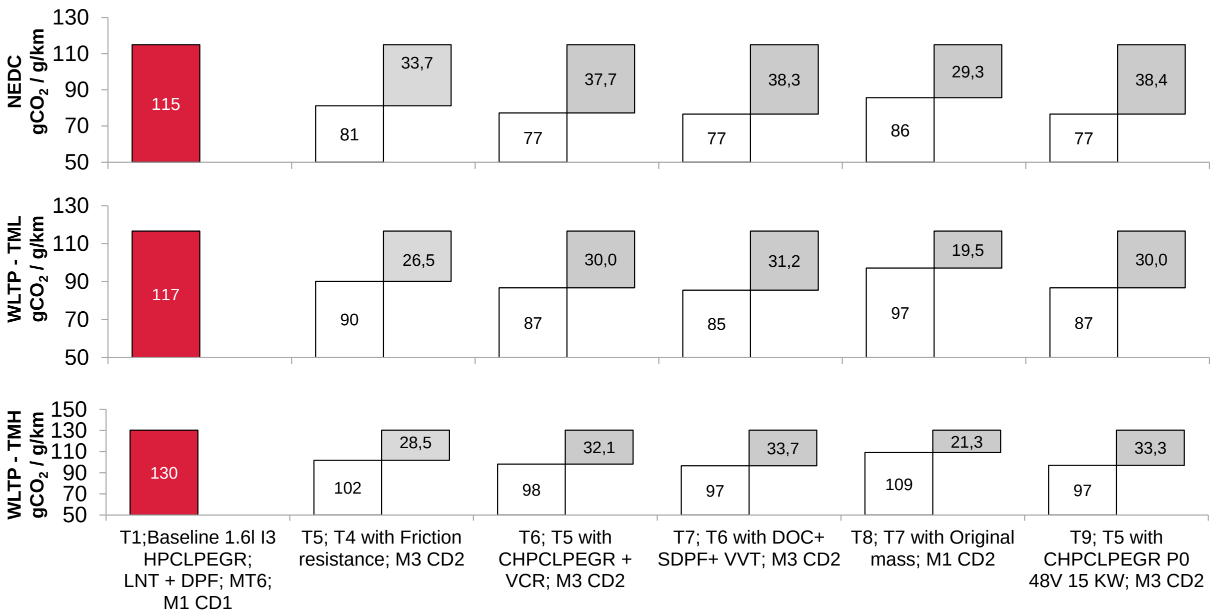


Figure 7.3 Overview of results for Segment C (EU cycles).

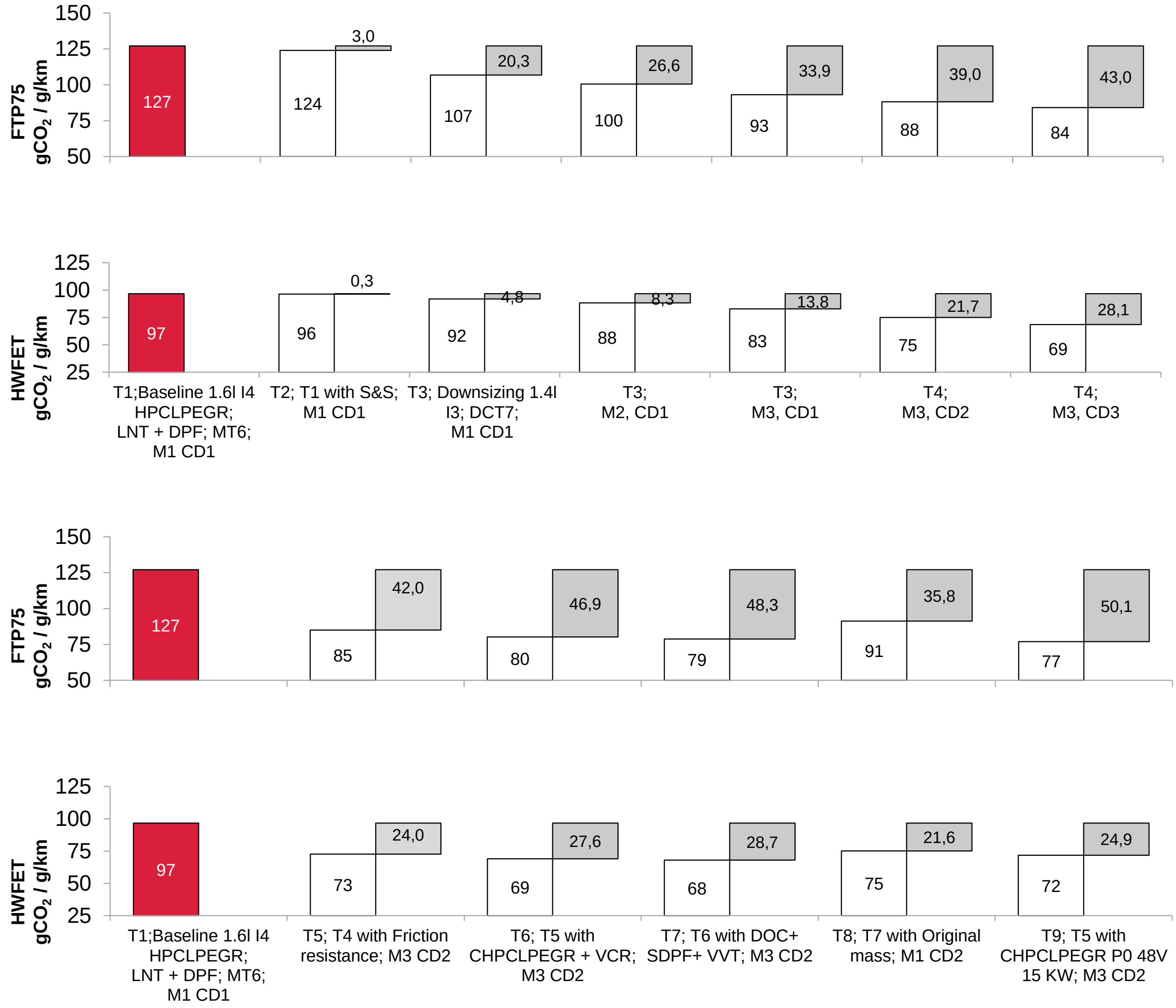


Figure 7.4 Overview of results for Segment C (US cycles).

7.1.1.3 Segment D

Segment D		EGR	T/C	VVT	CR	Engine friction	Hybridization	Transmission	EATS	Coast down set	REF / kg	TML / kg	TMH / kg
T1	Baseline	Cooled HP	single	not variable	not variable	baseline	without	MT6	DOC+SDPF	CD1	1650	1721	1913
T2	Start/Stop	Cooled HP	single	not variable	not variable	baseline	**start/stop**	MT6	DOC+SDPF	CD1	1650	1721	1913
T3	Downsizing/ Transmission/ M red Scenario I	Cooled HP	single	not variable	not variable	baseline	Adv. start/stop	**DCT10**	DOC+SDPF	CD1	**1645 1483 1320**	**1716 1554 1391**	**1908 1745 1583**
T4	M reduction Scenario II	Cooled HP	single	not variable	not variable	baseline	Adv. start/stop	DCT10	DOC+SDPF	**CD1- CD2- CD3**	1320	1391	1583
T5	Friction, resistance	Cooled HP	single	not variable	not variable	**-20% friction, el. coolant pump**	Adv. start/stop	DCT10	DOC+SDPF	CD2	1320	1391	1583
T6	Engine technologies	**CHP-CLP**	single	**variable**	not variable	-20% friction, el. coolant pump	Adv. start/stop	DCT10	DOC+SDPF	CD2	1320	1391	1583
T7	Aftertreatment	CHP-CLP	single	variable	**2-step VCR**	-20% friction, el. coolant pump	Adv. start/stop	DCT10	DOC+SDPF	CD2	1320	1391	1583
T8	Without weight reduction	CHP-CLP	single	variable	2-step VCR	-20% friction, el. coolant pump	Adv. start/stop	DCT10	DOC+SDPF	CD2	1645	1716	1908
T9	Hybrid P2 350 V	CHP-CLP	single	Variable	Not variable	20% friction, el. coolant pump	Adv. start/stop	**DCT10**	DOC+SDPF	CD2	1398	1469	1661

Table 7.3 Summary of variants for Segment D.

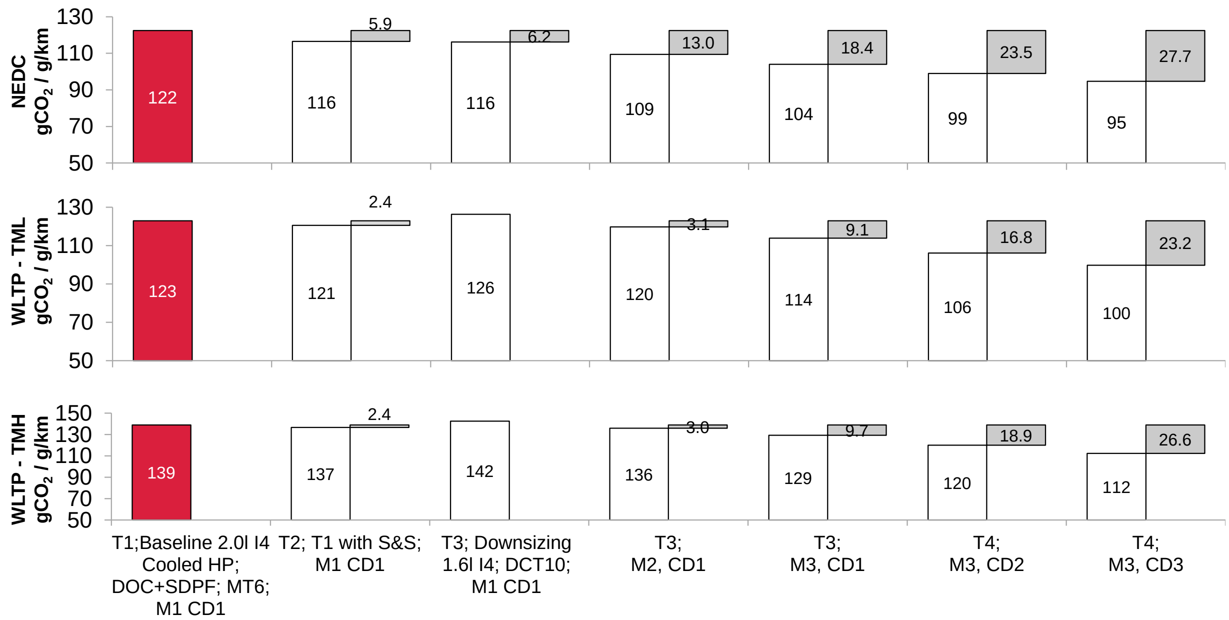
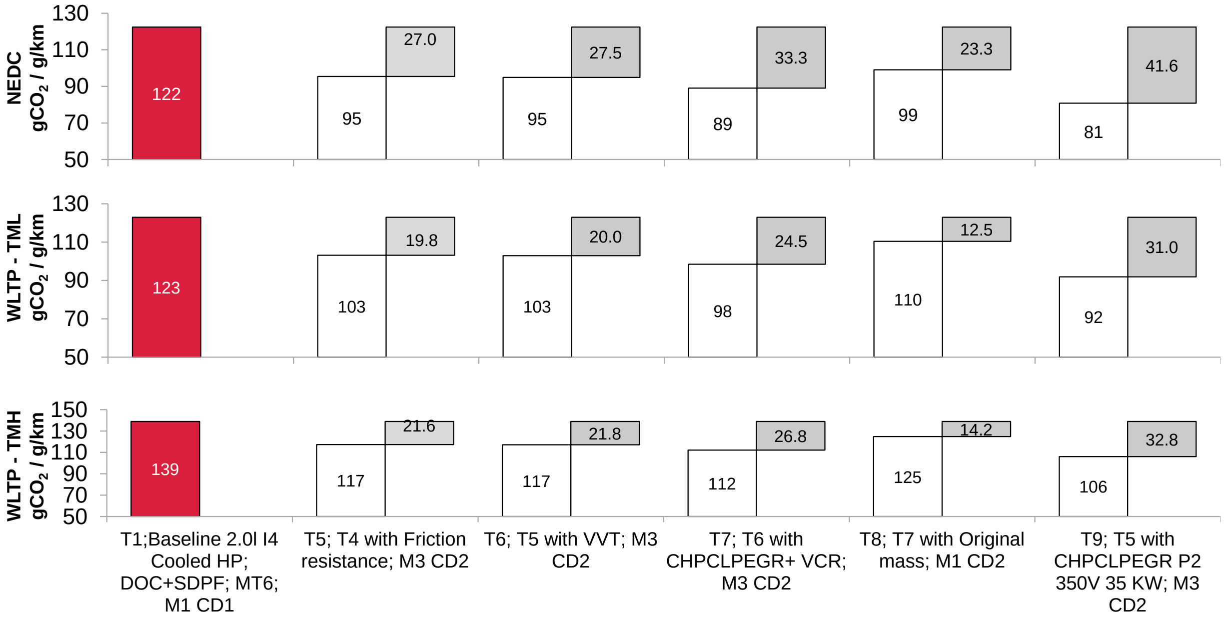
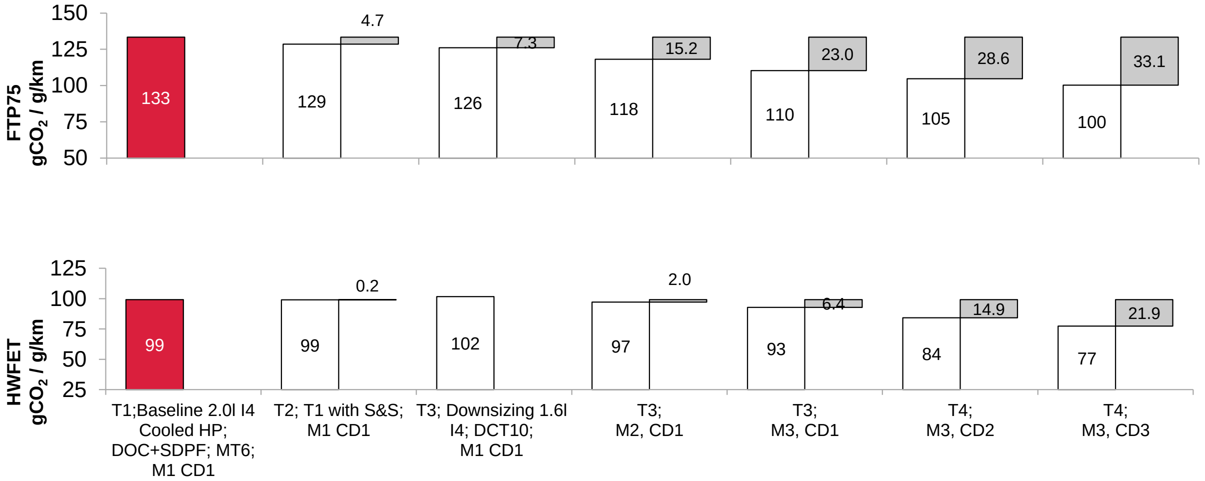


Figure 7.5 Overview of results for Segment D (EU cycles).


NEDC gCO2 / g/km
130
110
90
70
50
122
95
27.0
95
27.5
89
33.3
99
23.3
81
41.6
WLTP - TML gCO2 / g/km
130
110
90
70
50
123
103
19.8
103
20.0
98
24.5
110
12.5
92
31.0
WLTP - TMH gCO2 / g/km
150
130
110
90
70
50
139
117
21.6
117
21.8
112
26.8
125
14.2
106
32.8
T1;Baseline 2.0l I4 Cooled HP; DOC+SDPF; MT6; M1 CD1
T5; T4 with Friction resistance; M3 CD2
T6; T5 with VVT; M3 CD2
T7; T6 with CHPCLPEGR+ VCR; M3 CD2
T8; T7 with Original mass; M1 CD2
T9; T5 with CHPCLPEGR P2 350V 35 KW; M3 CD2


FTP75 gCO2 / g/km
150
125
100
75
50
133
129
4.7
126
7.3
118
15.2
110
23.0
105
28.6
100
33.1
HWFET gCO2 / g/km
125
100
75
50
25
99
99
0.2
102
97
2.0
93
6.4
84
14.9
77
21.9
T1;Baseline 2.0l I4 Cooled HP; DOC+SDPF; MT6; M1 CD1
T2; T1 with S&S; M1 CD1
T3; Downsizing 1.6l I4; DCT10; M1 CD1
T3; M2, CD1
T3; M3, CD1
T4; M3, CD2
T4; M3, CD3

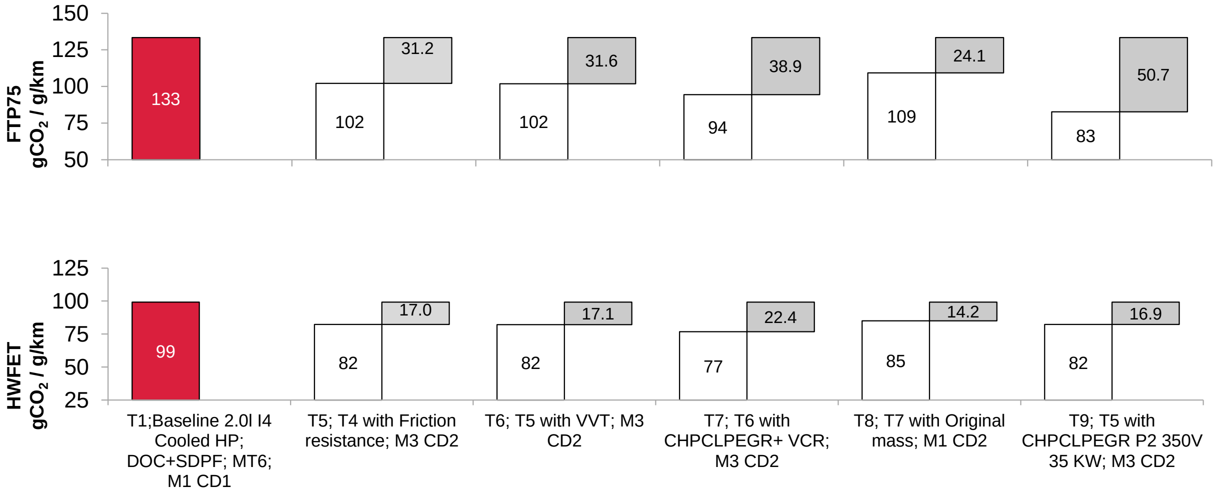
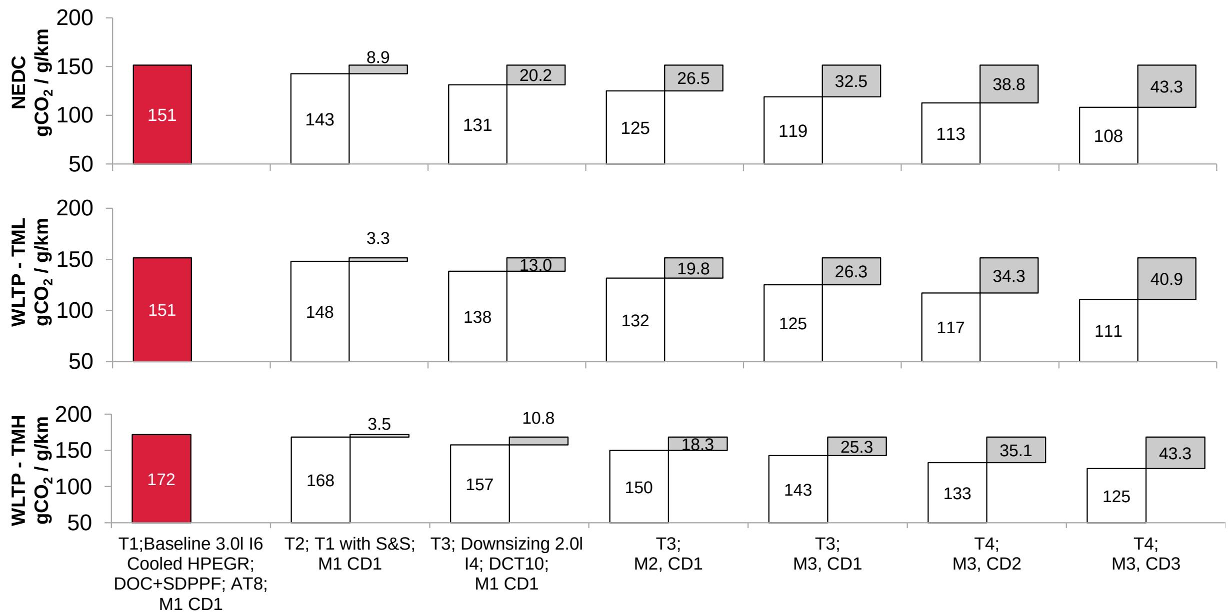


Figure 7.6 Overview of results for Segment D (US cycles).

7.1.1.4 Segment E

Segment E		EGR	T/C	VVT	CR	Engine friction	Hybridization	Transmission	EATS	Coast down set	REF / kg	TML / kg	TMH / kg
T1	Baseline	Cooled HP	single	not variable	not variable	baseline	without	AT8	DOC+SDPF	CD1	1863	1936	2169
T2	Start/Stop	Cooled HP	single	not variable	not variable	baseline	**start/ stop**	AT8	DOC+SDPF	CD1	1863	1936	2169
T3	Downsizing /Transmission/ M red Scenario I	Cooled HP	double	not variable	not variable	baseline	Adv. start/stop	**DCT10**	DOC+SDPF	CD1	**1793** **1609** **1425**	**1866** **1682** **1498**	**2099** **1916** **1732**
T4	M reduction Scenario II	Cooled HP	double	not variable	not variable	baseline	Adv. start/stop	**DCT10**	DOC+SDPF	**CD1-** **CD2-** **CD3**	1425	1498	1732
T5	Friction, resistance	Cooled HP	double	not variable	not variable	**-20% friction, el. coolant pump**	Adv. start/stop	**DCT10**	DOC+SDPF	CD2	1425	1498	1732
T6	Engine technologies	CHP-CLP	double	variable	not variable	-20% friction, el. coolant pump	Adv. start/stop	**DCT10**	DOC+SDPF	CD2	1425	1498	1732
T7	Aftertreatment	CHP-CLP	double	**variable**	2-step VCR	-20% friction, el. coolant pump	Adv. start/stop	**DCT10**	DOC+SDPF	CD2	1425	1498	1732
T8	Without weight reduction	CHP-CLP	double	variable	2-step VCR	-20% friction, el. coolant pump	Adv. start/stop	**DCT10**	DOC+SDPF	CD2	1793	1866	2099
T9	Hybrid P2 350 V	CHP-CLP	double	Variable	Not variable	20% friction, el. coolant pump	Adv. start/stop	**DCT10**	DOC+SDPF	CD2	1503	1576	1810

Table 7.4: Summary of variants for Segment E



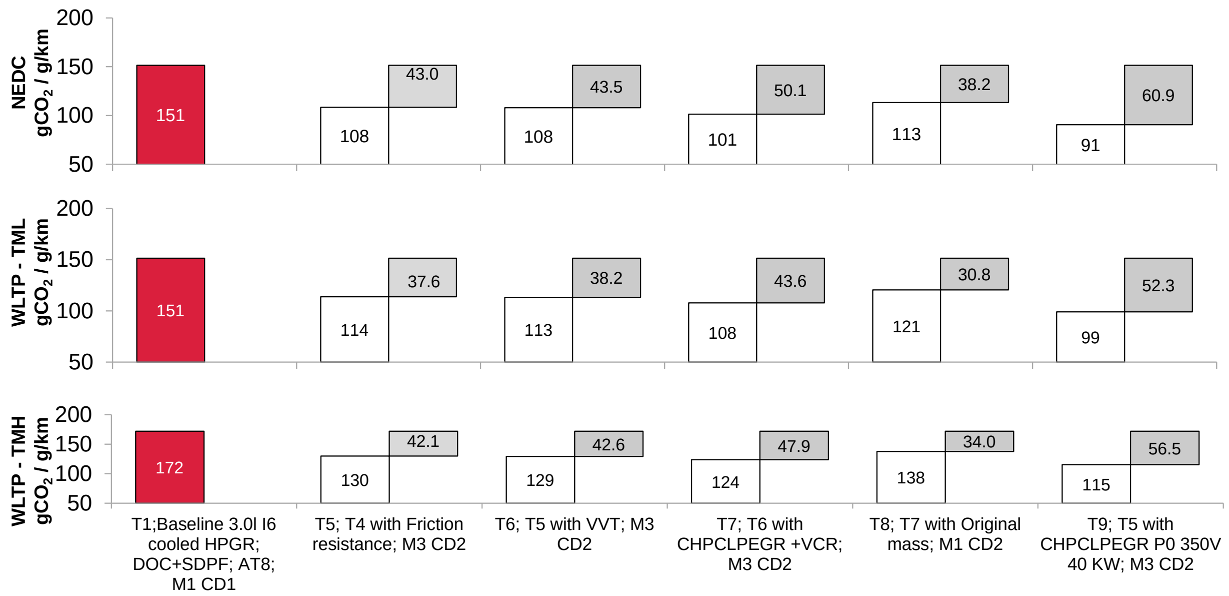


Figure 7.7 Overview of results for Segment E (EU cycles).

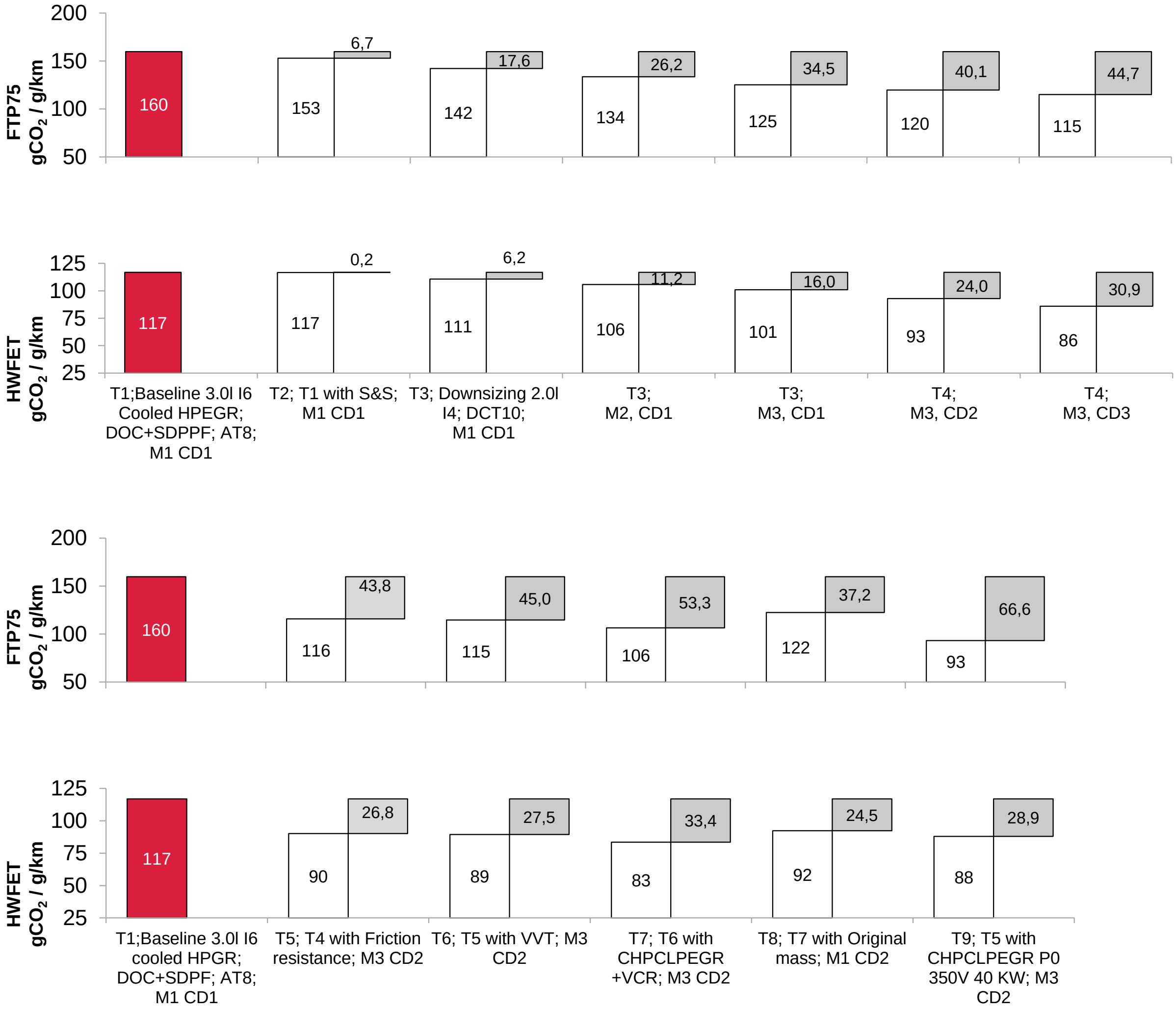


Figure 7.8 Overview of results for Segment E (US cycles).

7.1.1.5 Segment SUV

Segment SUV		EGR	T/C	VVT	CR	Engine friction	Hybridization	Transmission	EATS	Coast down set	REF / kg	TML / kg	TMH / kg
T1	Baseline	Cooled HP	single	not variable	not variable	baseline	without	AT8	DOC+SDPF	CD1	1713	1785	2019
T2	Start/Stop	Cooled HP	single	not variable	not variable	baseline	**start/stop**	AT8	DOC+SDPF	CD1	1713	1785	2019
T3	Downsizing/ Transmission/ M red Scenario I	Cooled HP	**double**	not variable	not variable	baseline	Adv. start/stop	**DCT10**	DOC+SDPF	CD1	**1643** **1474** **1305**	**1715** **1547** **1378**	**1949** **1780** **1611**
T4	M reduction Scenario II	Cooled HP	double	not variable	not variable	baseline	Adv. start/stop	DCT10	DOC+SDPF	**CD1-CD2-CD3**	1305	1378	1611
T5	Friction, resistance	Cooled HP	double	not variable	not variable	**-20% friction, el. coolant pump**	Adv. start/stop	DCT10	DOC+SDPF	CD2	1305	1378	1611
T6	Engine technologies	CHP-CLP	double	**variable**	not variable	-20% friction, el. coolant pump	Adv. start/stop	DCT10	DOC+SDPF	CD2	1305	1378	1611
T7	Aftertreatment	CHP-CLP	double	**variable**	**2-step VCR**	-20% friction, el. coolant pump	Adv. start/stop	DCT10	DOC+SDPF	CD2	1305	1378	1611
T8	Without weight reduction	CHP-CLP	double	variable	2-step VCR	-20% friction, el. coolant pump	Adv. start/stop	DCT10	DOC+SDPF	CD2	1643	1715	1949
T9	Hybrid P2 350 V	CHP-CLP	double	Variable	Not variable	20% friction, el. coolant pump	Adv. start/stop	**DCT10**	DOC+SDPF	CD2	1383	1456	1689

Table 7.5 Summary of variants for Segment SUV

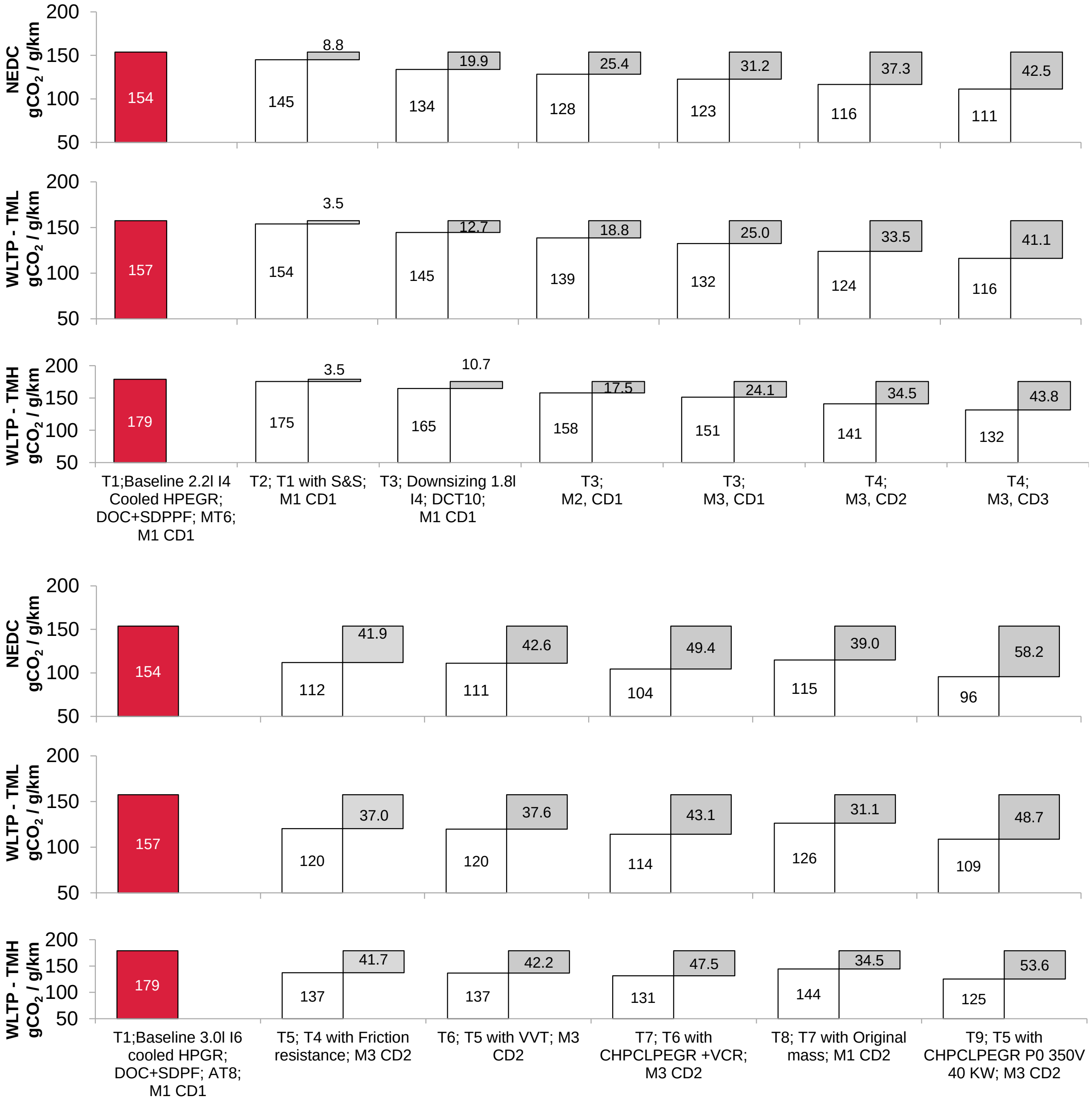


Figure 7.9 Overview of results for Segment SUV (EU cycles)

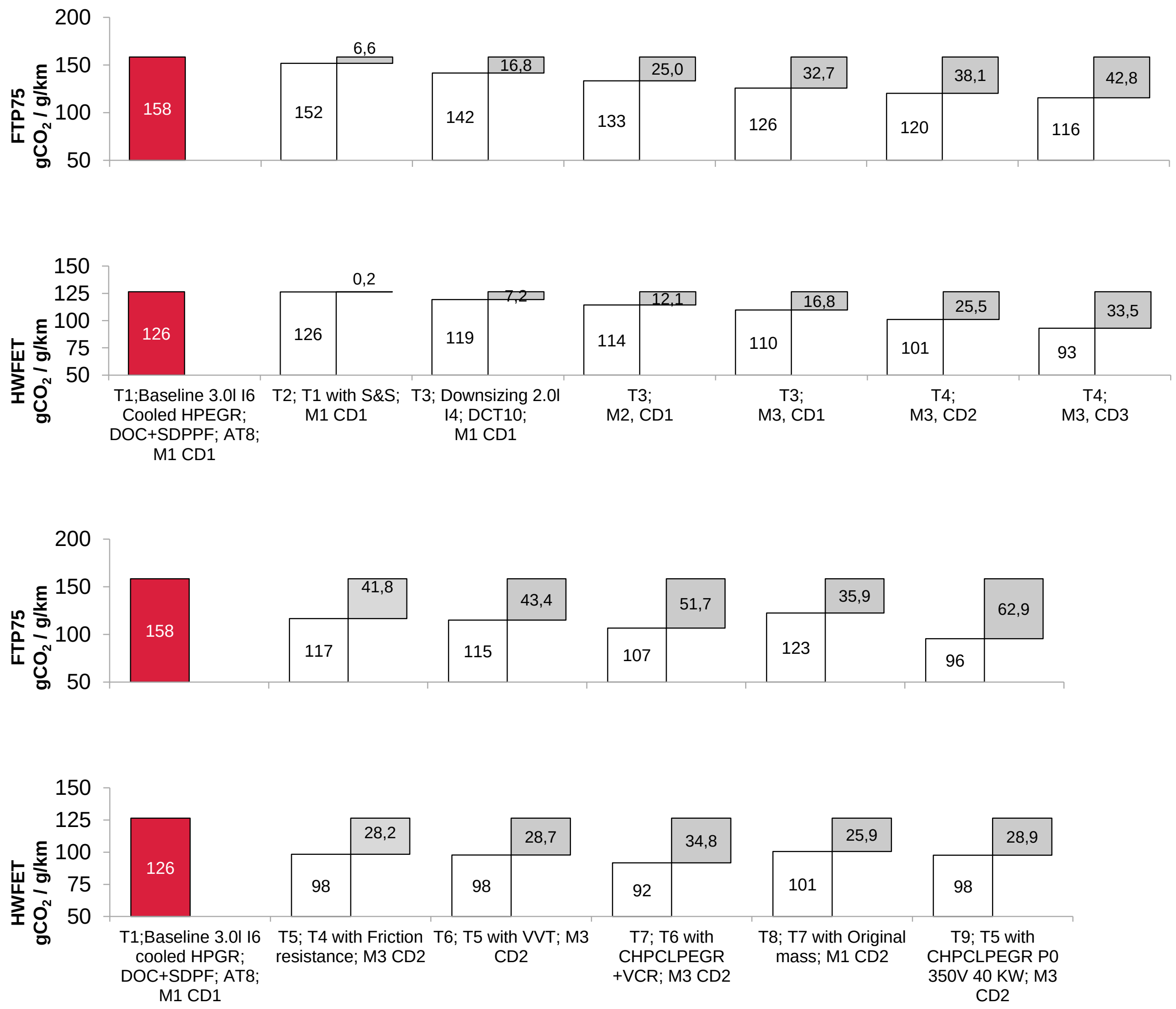


Figure 7.10 Overview of results for Segment SUV (US cycles).

7.1.1.6 Segment LCV

Segment SUV		EGR	T/C	VVT	CR	Engine friction	Hybridization	Transmission	EATS	Coast down set	REF / kg	TML / kg	TMH / kg
T1	Baseline	Cooled HP	single	not variable	not variable	baseline	without	MT6	DOC+SDPF	CD1	2051	2261	2419
T2	Start/Stop	Cooled HP	single	not variable	not variable	baseline	**start/stop**	MT6	DOC+SDPF	CD1	2051	2261	2419
T3	Downsizing/ Transmission/ M red Scenario I	Cooled HP	double	not variable	not variable	baseline	Adv. start/stop	**DCT10**	DOC+SDPF	CD1	**2056** **1853** **1651**	**2266** **2063** **1861**	**2424** **2222** **2019**
T4	M reduction Scenario II	Cooled HP	double	not variable	not variable	baseline	Adv. start/stop	**DCT10**	DOC+SDPF	**CD1-CD2-CD3**	1651	1861	2019
T5	Friction, resistance	Cooled HP	double	not variable	not variable	**-20% friction, el. coolant pump**	Adv. start/stop	**DCT10**	DOC+SDPF	CD2	1651	1861	2019
T6	Engine technologies	CHP-CLP	double	variable	not variable	-20% friction, el. coolant pump	Adv. start/stop	**DCT10**	DOC+SDPF	CD2	1651	1861	2019
T7	Aftertreatment	CHP-CLP	double	**variable**	2-step VCR	-20% friction, el. coolant pump	Adv. start/stop	**DCT10**	DOC+SDPF	CD2	1651	1861	2019
T8	Without weight reduction	CHP-CLP	double	variable	2-step VCR	-20% friction, el. coolant pump	Adv. start/stop	**DCT10**	DOC+SDPF	CD2	2056	226	2424
T9	Hybrid P2 350 V	CHP-CLP	double	Variable	Not variable	20% friction, el. coolant pump	Adv. start/stop	**DCT10**	DOC+SDPF	CD2	1729	1939	2097

Table 7.6 Summary of variants for Segment LCV

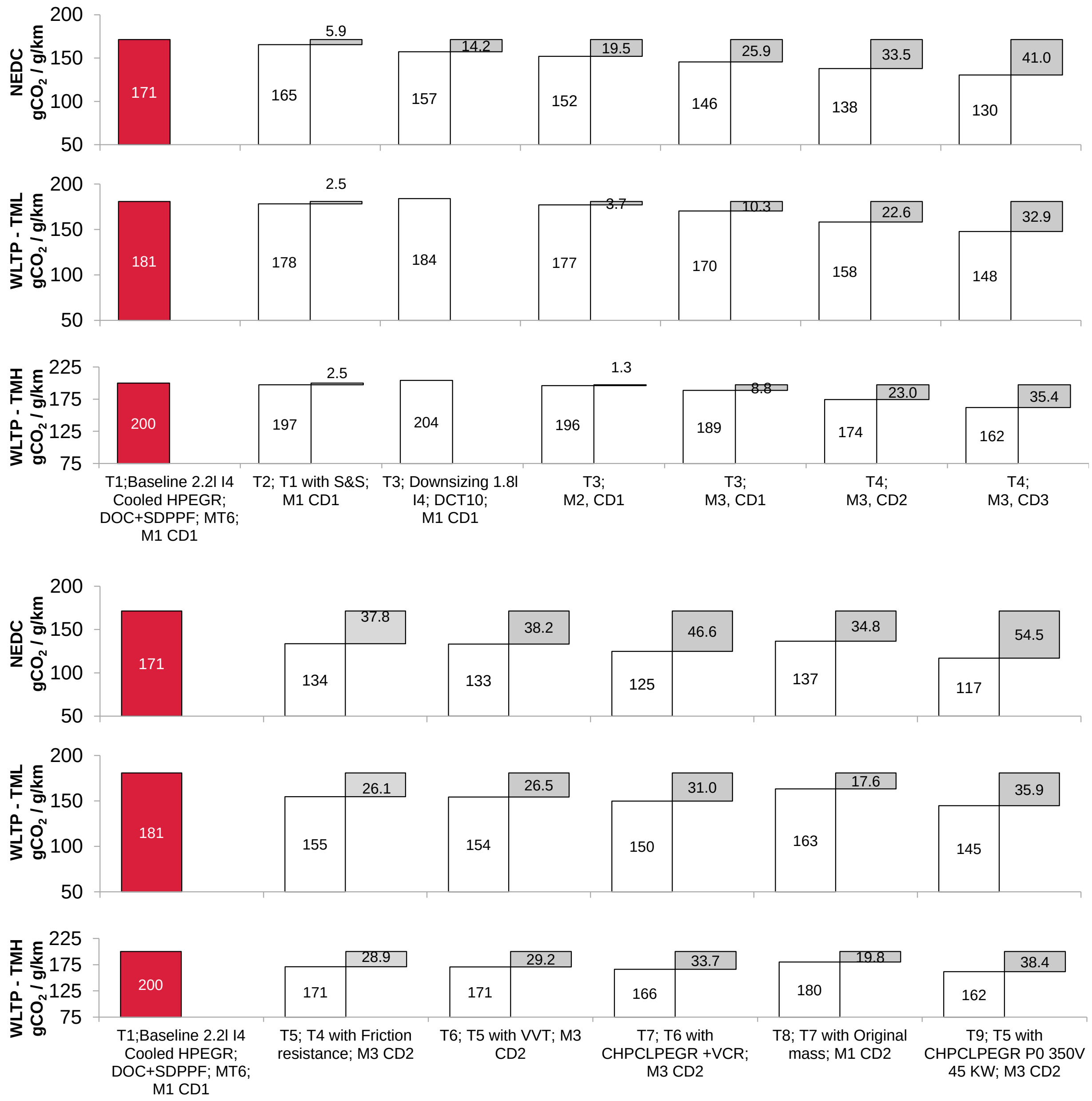


Figure 7.11 Overview of results for Segment LCV (EU cycles).

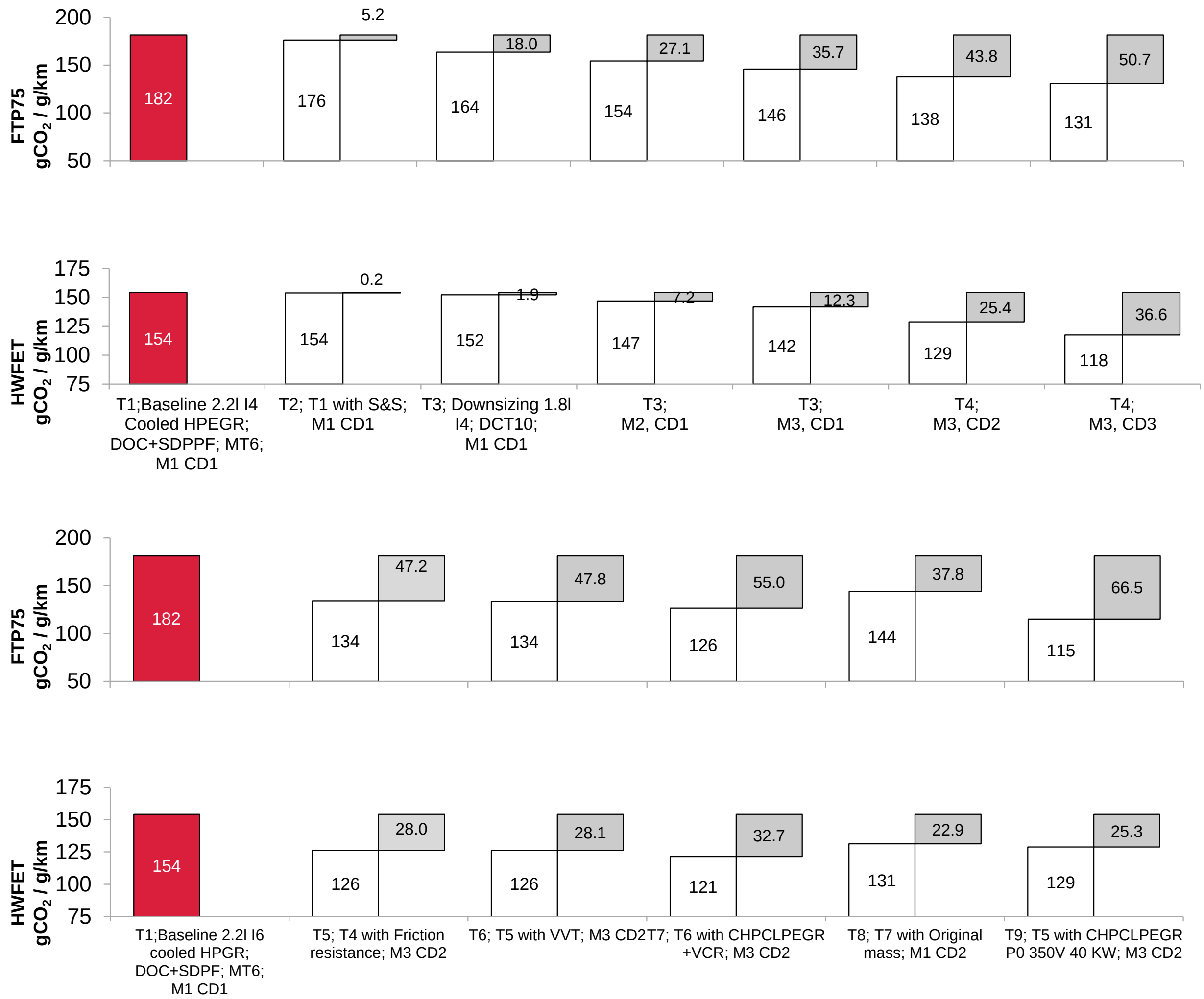


Figure 7.12 Overview of results for Segment LCV (US cycles).

7.2 CO_2 potential with respect to final cost for all segments

7.2.1 Segment B

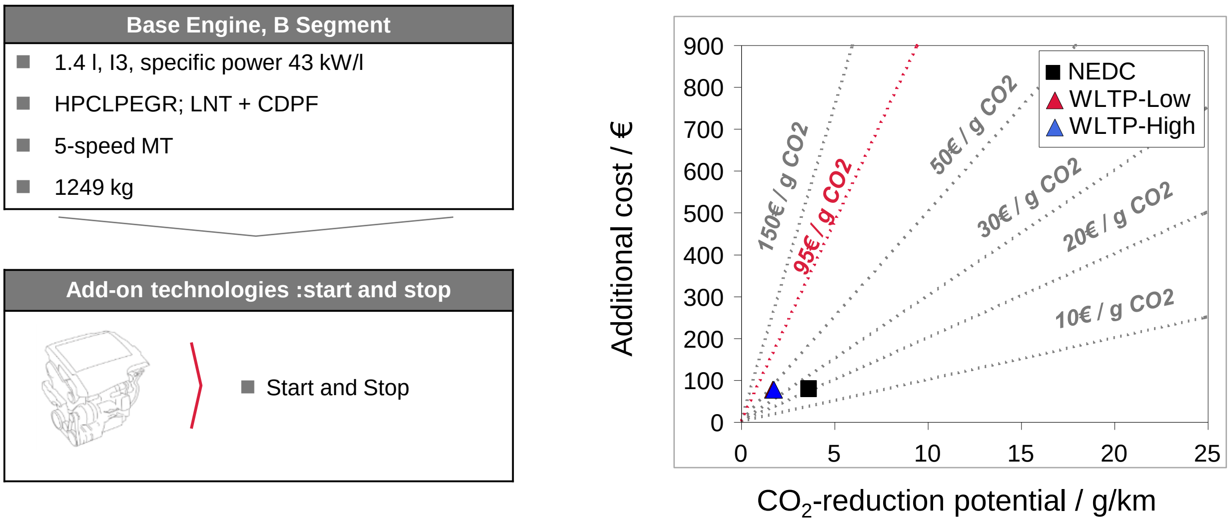


Figure 7.13 Segment B cost analysis for start and stop technology

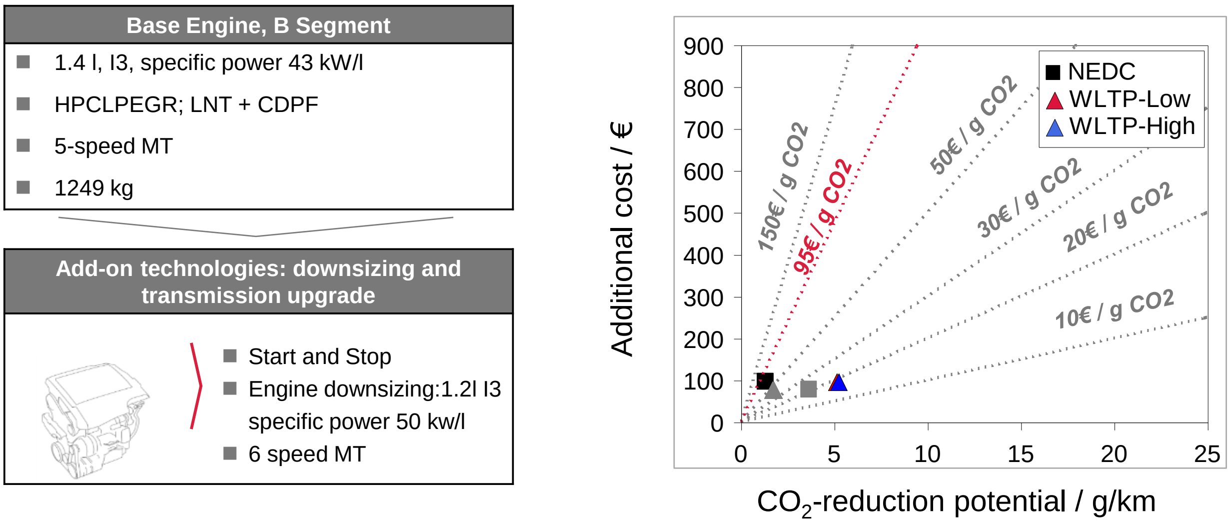


Figure 7.14 Segment B cost analysis for engine downsizing and transmissions upgrade

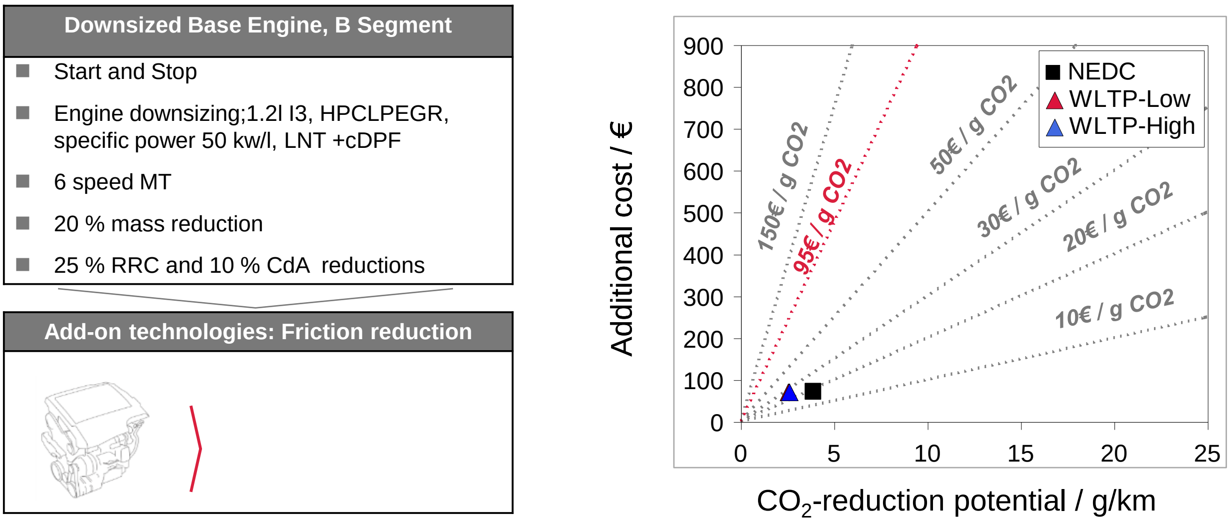


Figure 7.15 Segment B cost analysis for frictional upgrade

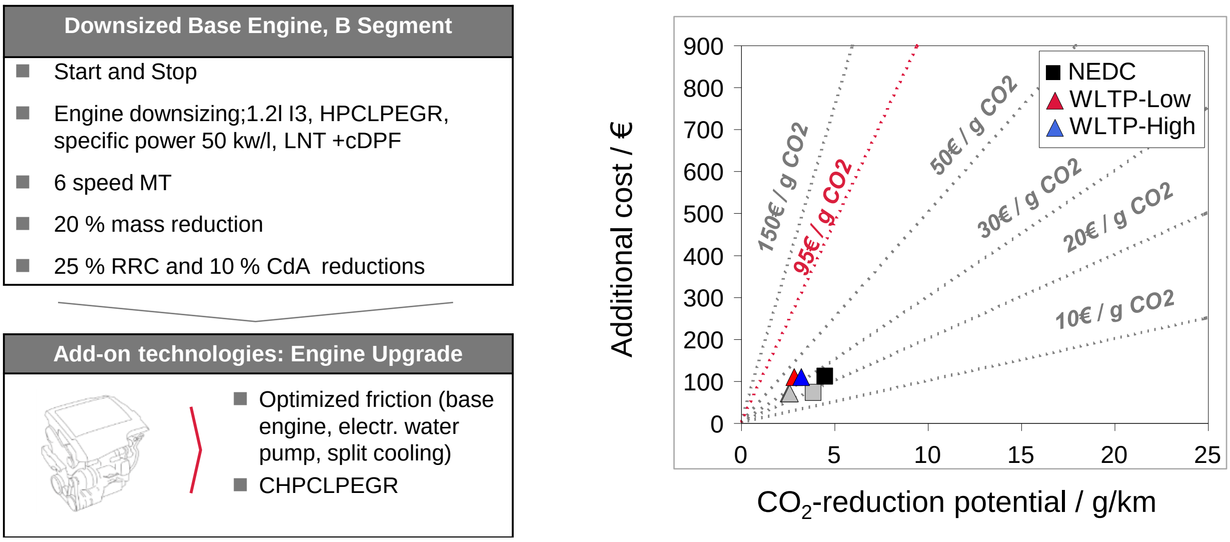


Figure 7.16 Segment B cost analysis for engine upgrade

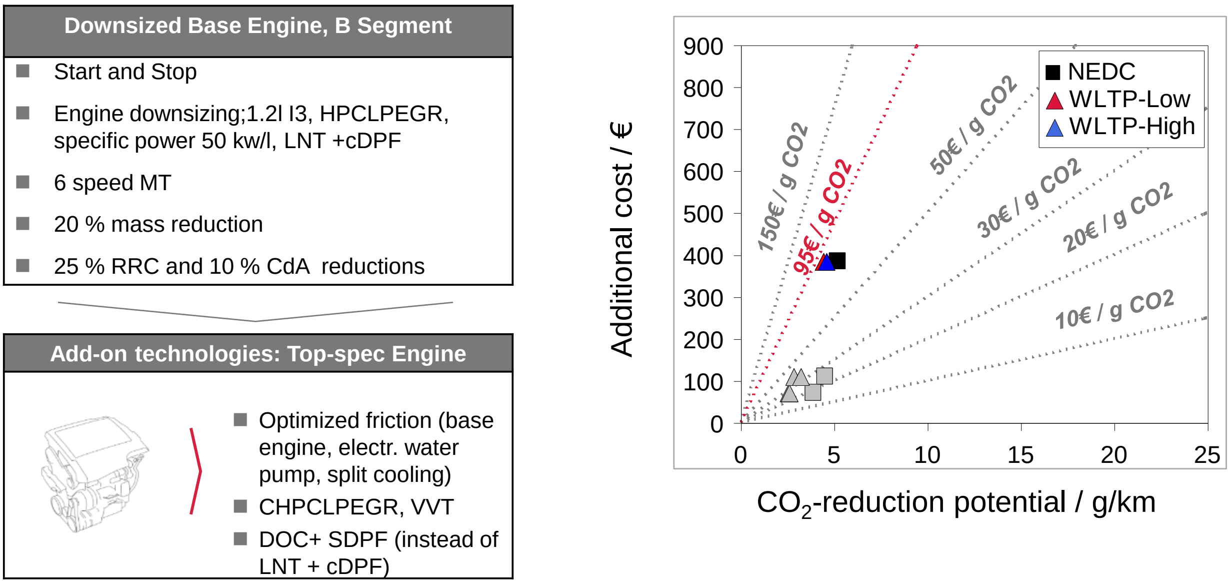


Figure 7.17 Segment B cost analysis for top-spec

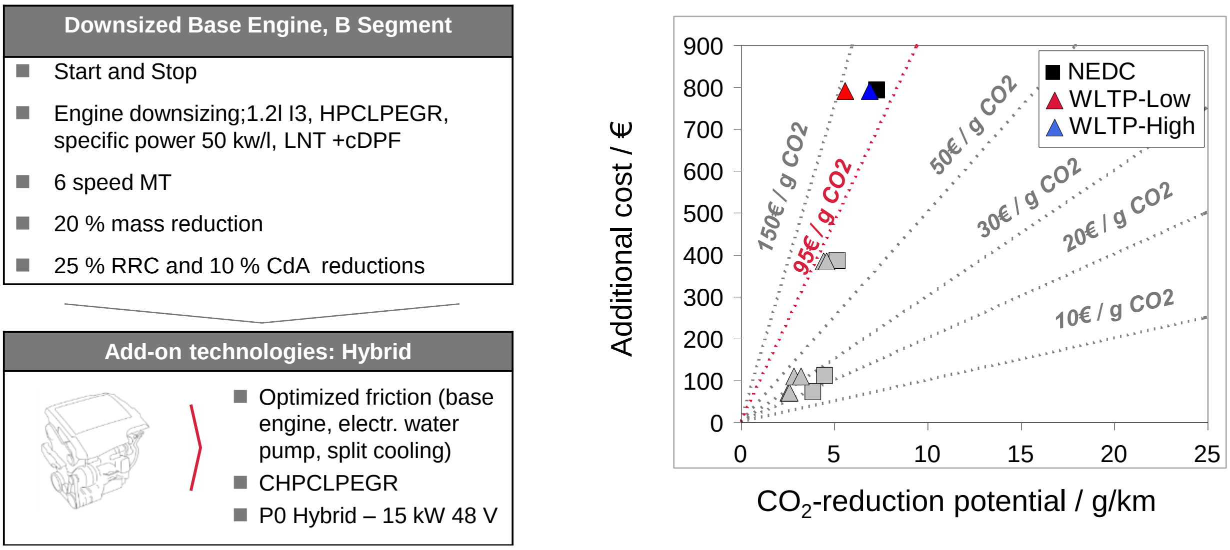


Figure 7.18 Segment B cost analysis for hybrid

7.2.2 Segment C

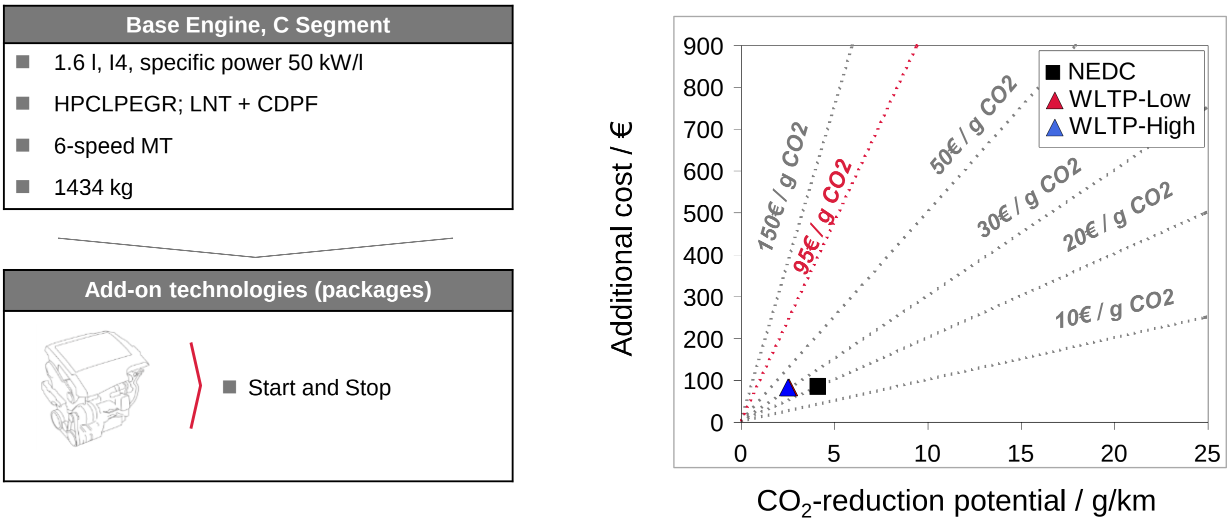


Figure 7.19 Segment C cost analysis for 4 technological start and stop

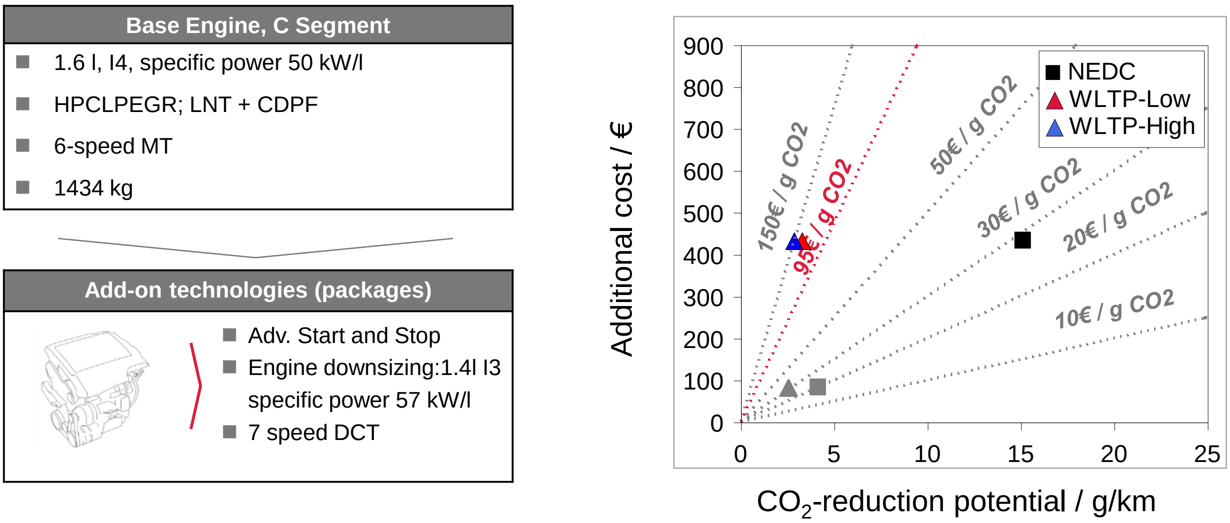


Figure 7.20 Segment C cost analysis for engine downsizing and transmissions upgrade

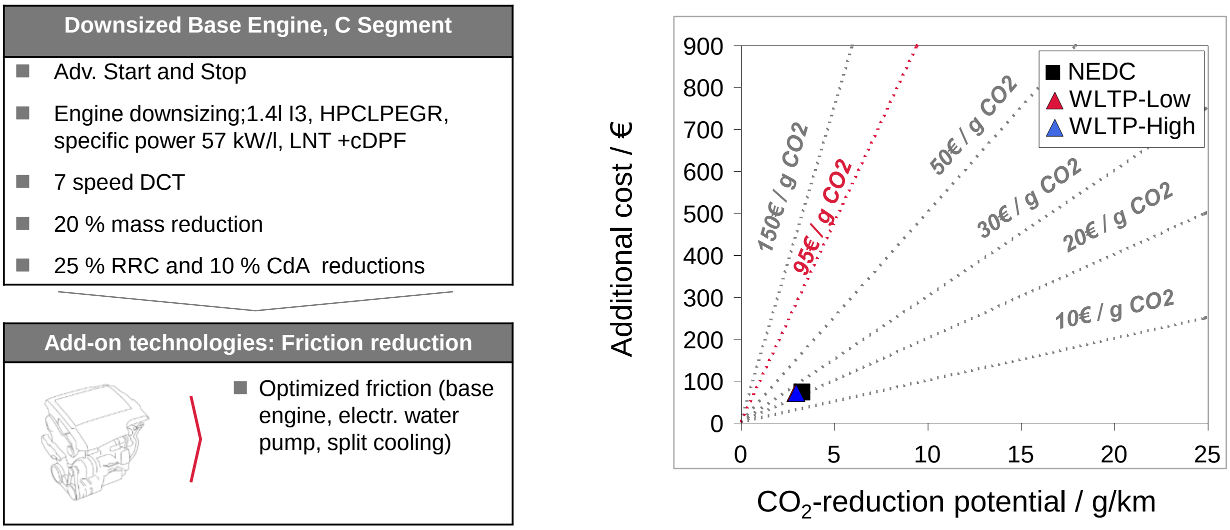


Figure 7.21 Segment C cost analysis for frictional upgrade

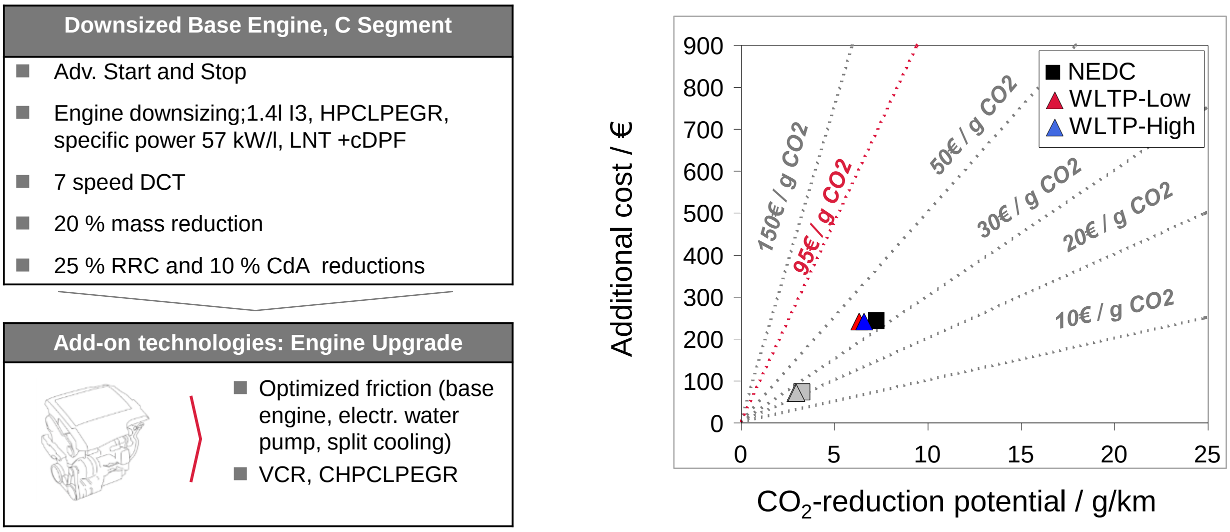


Figure 7.22 Segment C cost analysis for engine upgrade

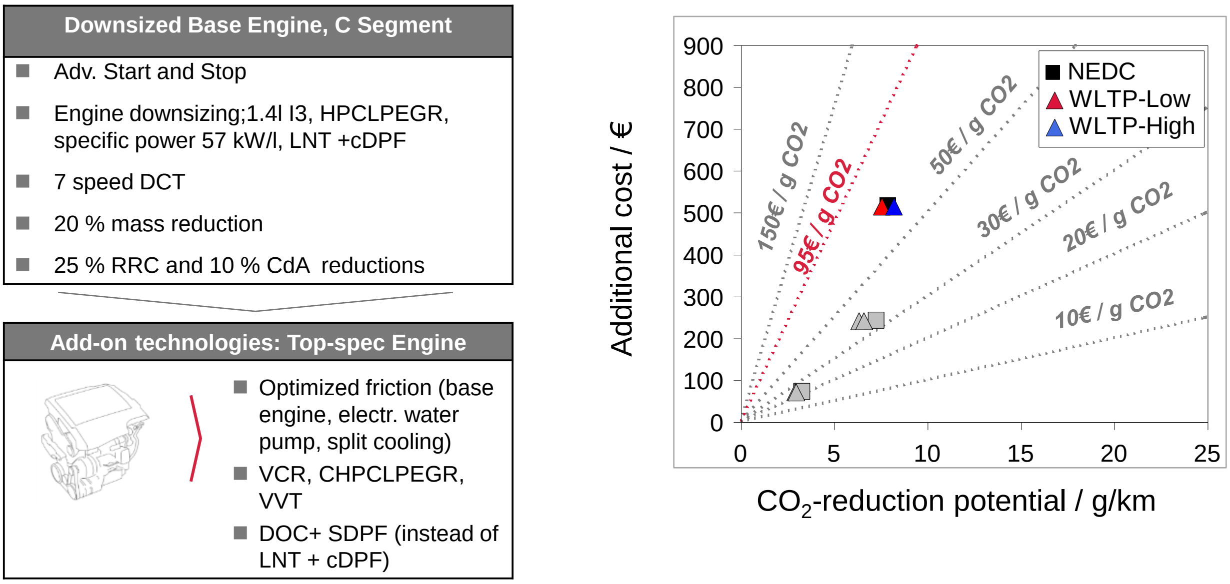


Figure 7.23 Segment C cost analysis for top-spec

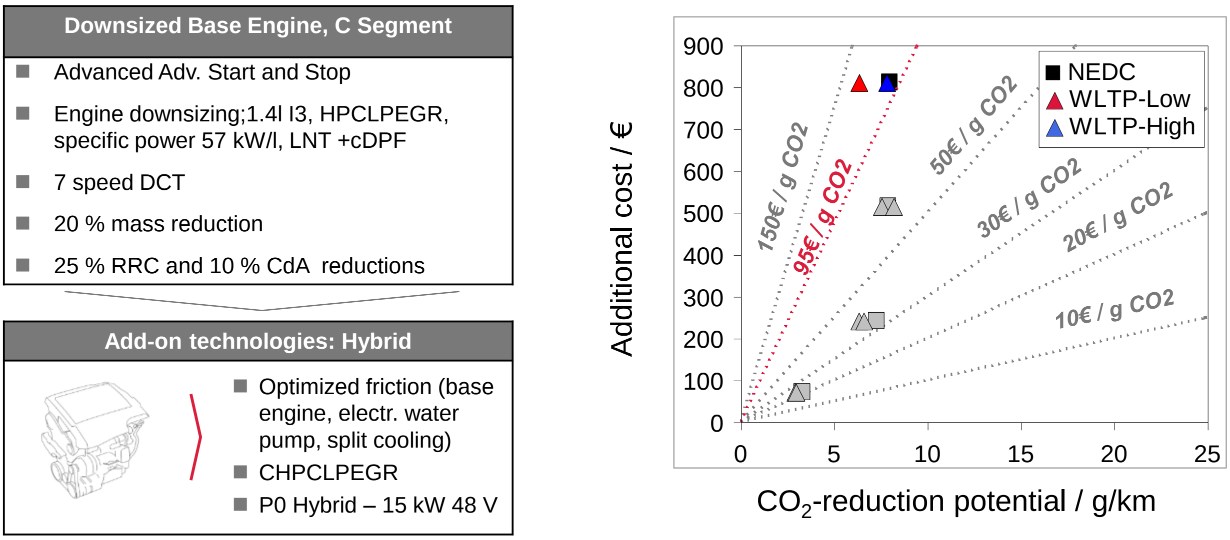


Figure 7.24 Segment C cost analysis for hybrid

7.2.3 Segment D

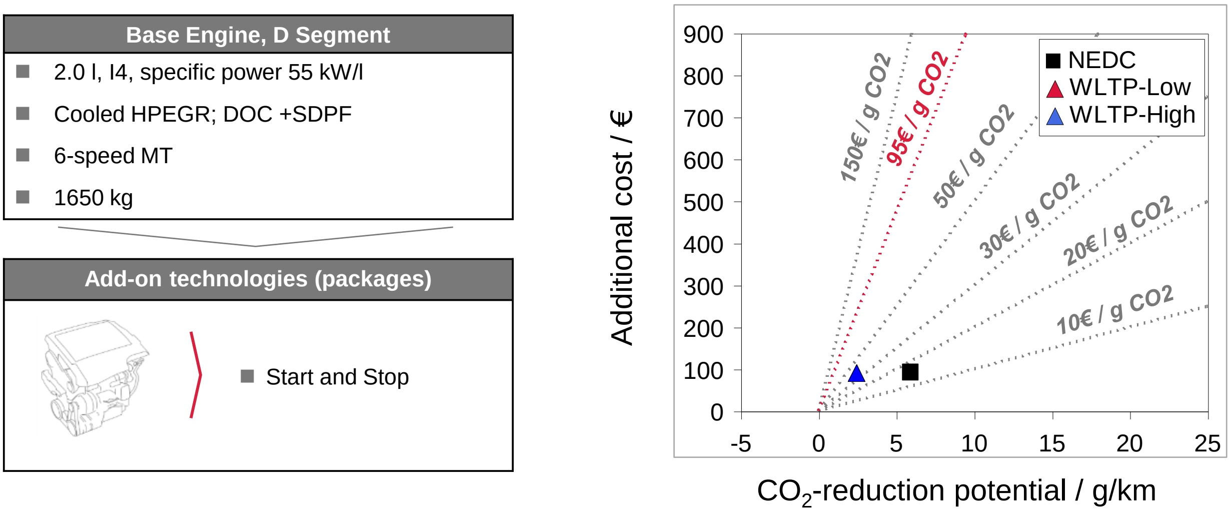


Figure 7.25 Segment D cost analysis for 4 technological start and stop

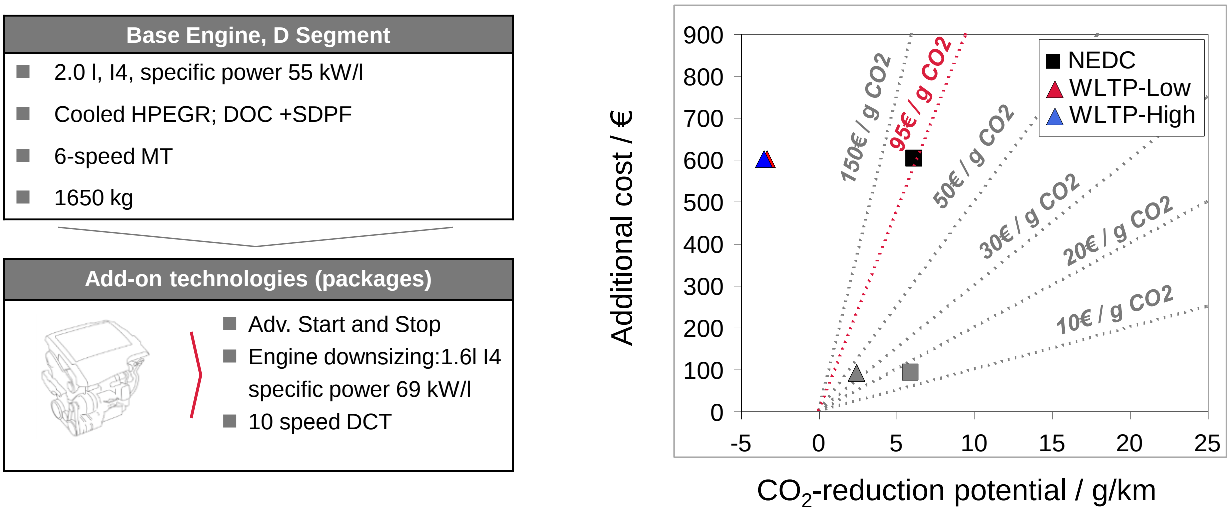


Figure 7.26 Segment D cost analysis for engine downsizing and transmissions upgrade

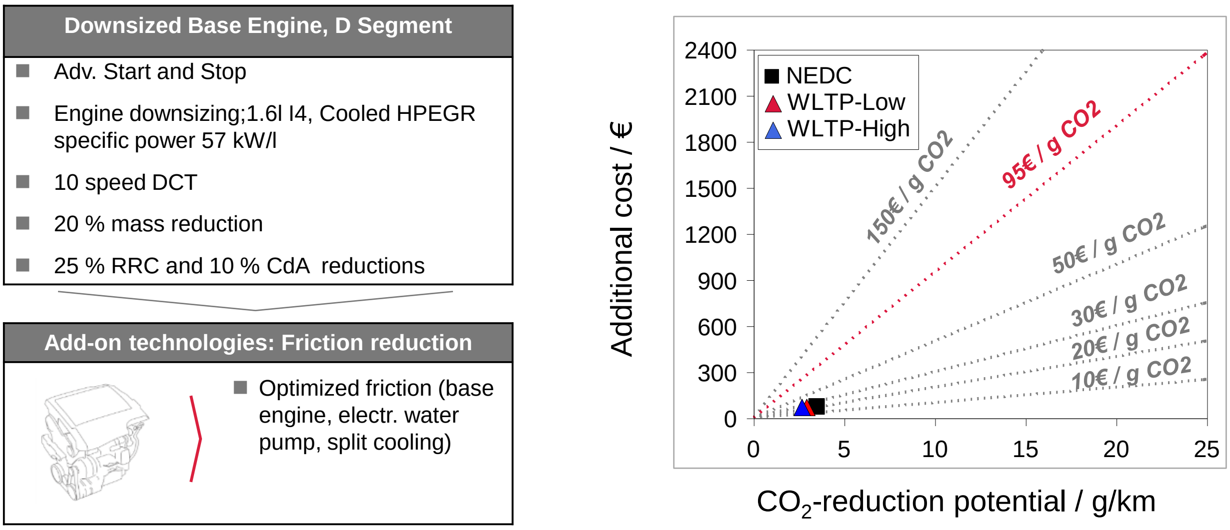


Figure 7.27 Segment D cost analysis for frictional upgrade

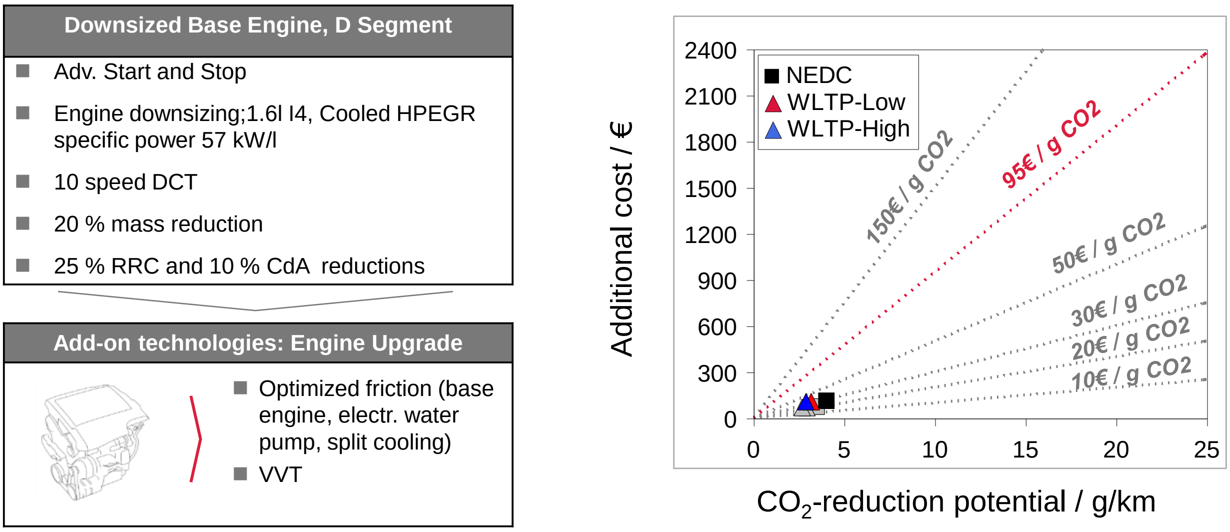


Figure 7.28 Segment D cost analysis for engine upgrade

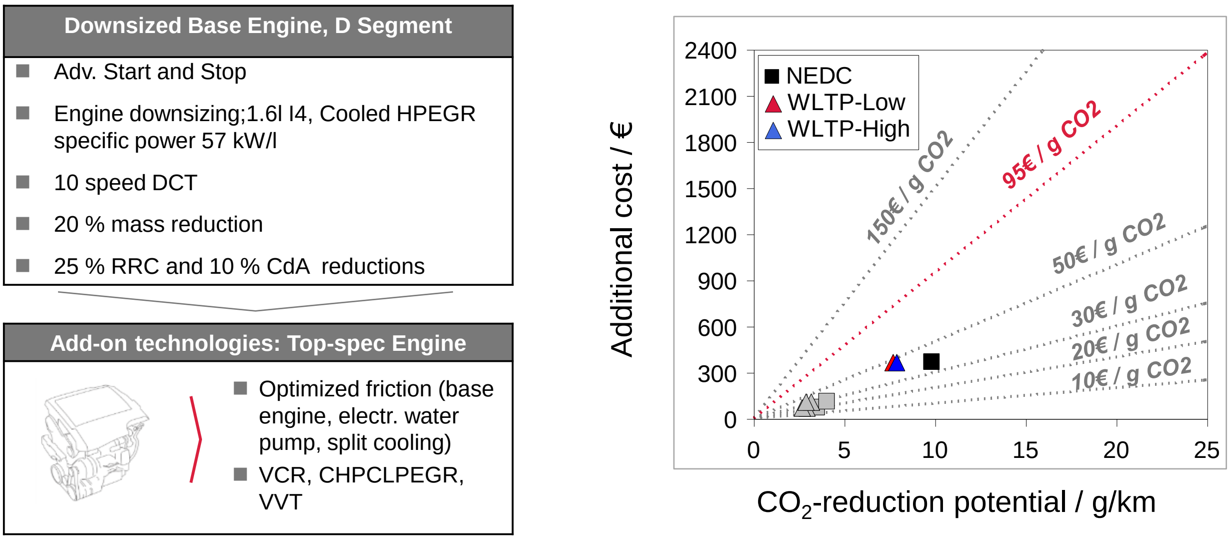


Figure 7.29 Segment D cost analysis for top-spec

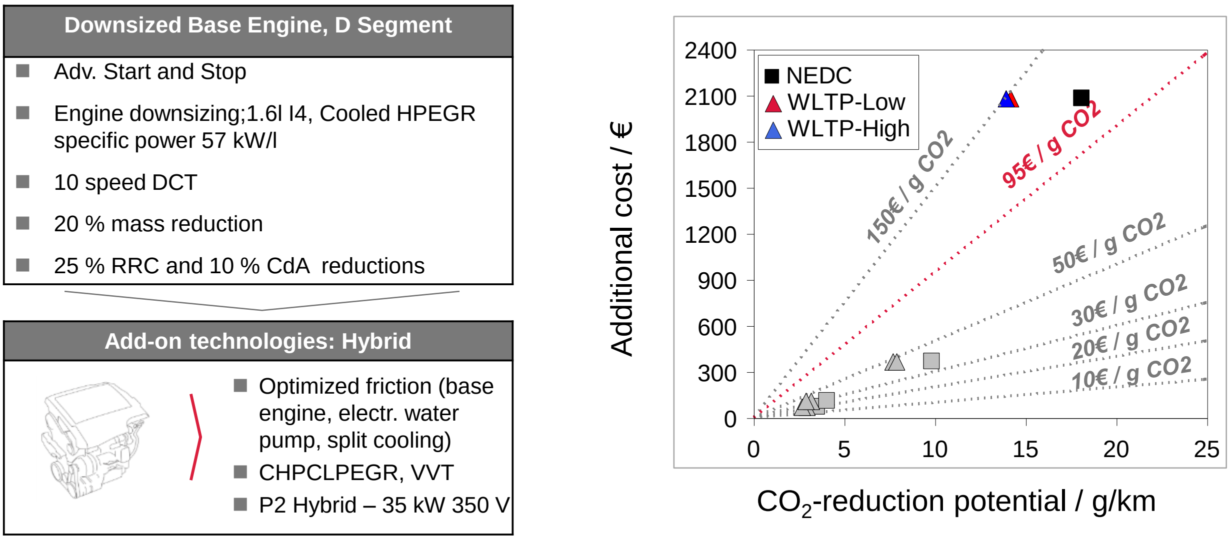


Figure 7.30 Segment D cost analysis for hybrid

7.2.4 Segment E

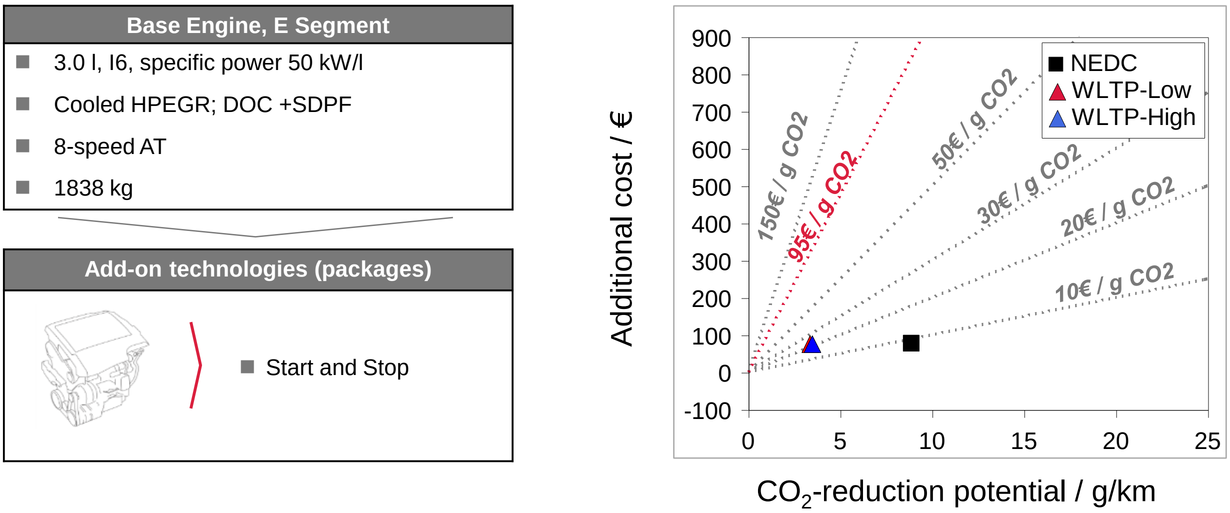


Figure 7.31 Segment E cost analysis for technological start and stop

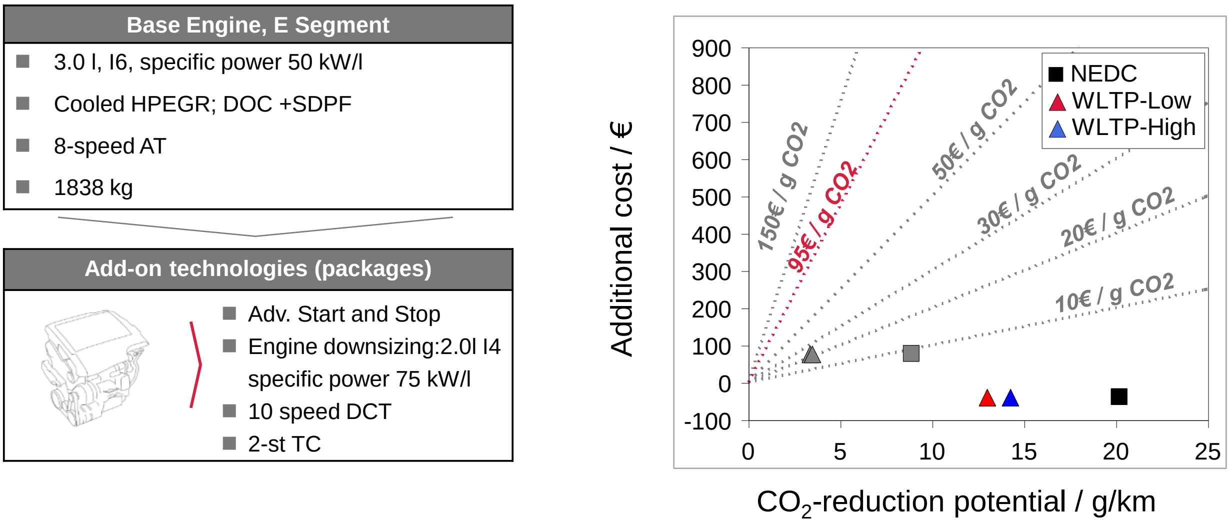


Figure 7.32 Segment E cost analysis for engine downsizing and transmissions upgrade

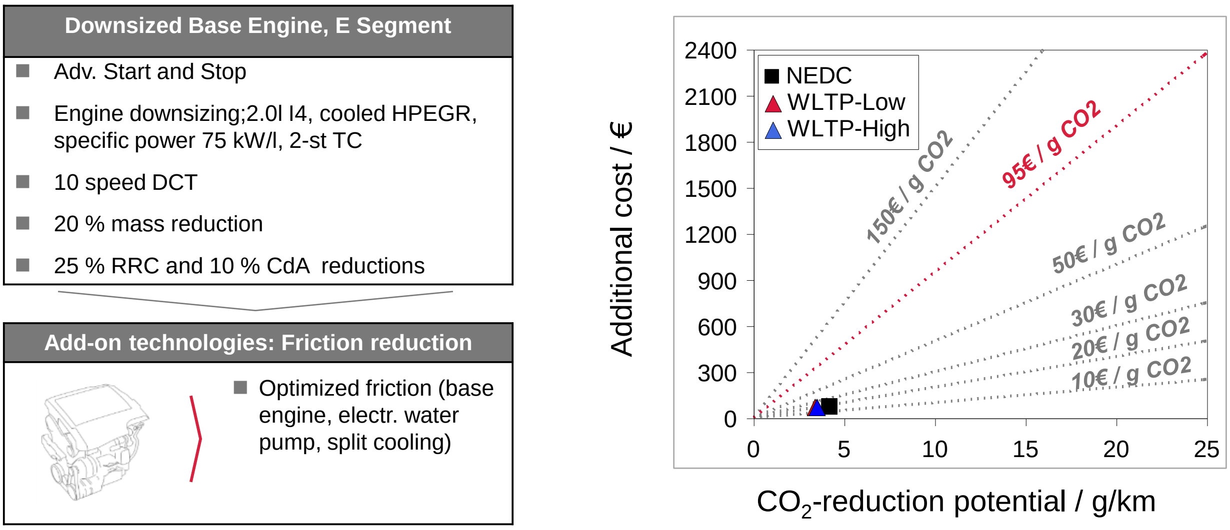


Figure 7.33 Segment E cost analysis for frictional upgrade

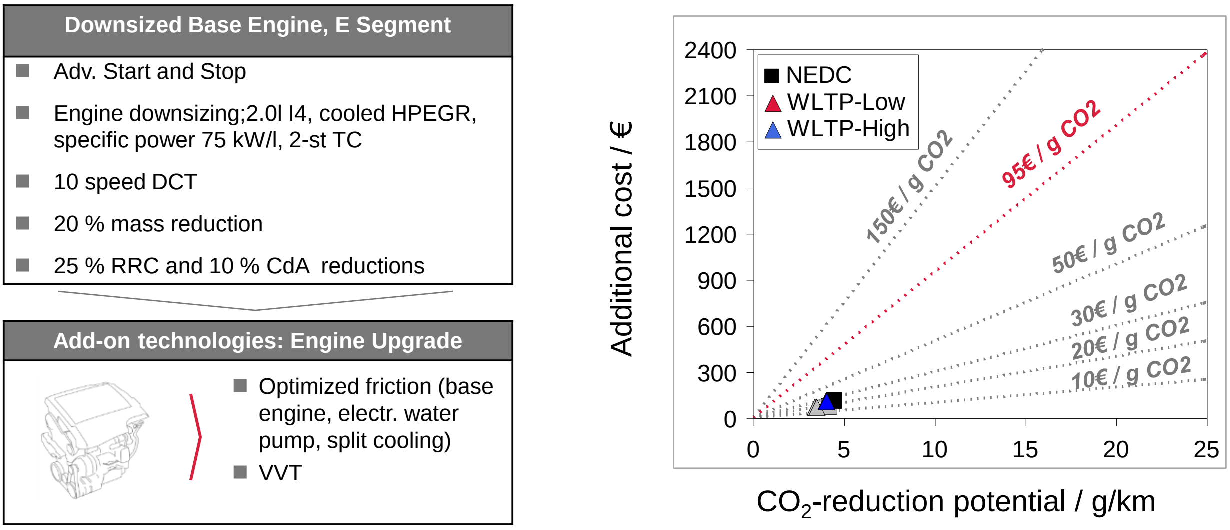
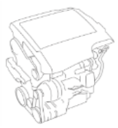


Figure 7.34 Segment E cost analysis for engine upgrade

Downsized Base Engine, E Segment

- Adv. Start and Stop
- Engine downsizing;2.0l I4, cooled HPEGR, specific power 75 kW/l, 2-st TC
- 10 speed DCT
- 20 % mass reduction
- 25 % RRC and 10 % CdA reductions

Add-on technologies: Top-spec Engine



- Optimized friction (base engine, electr. water pump, split cooling)
- VCR, CHPCLPEGR, VVT

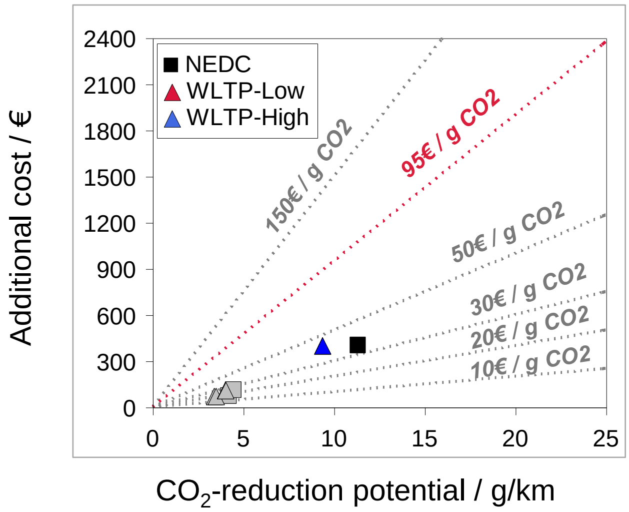
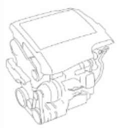


Figure 7.35 Segment E cost analysis for top-spec

Downsized Base Engine, E Segment

- Adv. Start and Stop
- Engine downsizing;2.0l I4, cooled HPEGR, specific power 75 kW/l, 2-st TC
- 10 speed DCT
- 20 % mass reduction
- 25 % RRC and 10 % CdA reductions

Add-on technologies: Hybrid



- Optimized friction (base engine, electr. water pump, split cooling)
- CHPCLPEGR, VVT
- P2 Hybrid – 40 kW 350 V

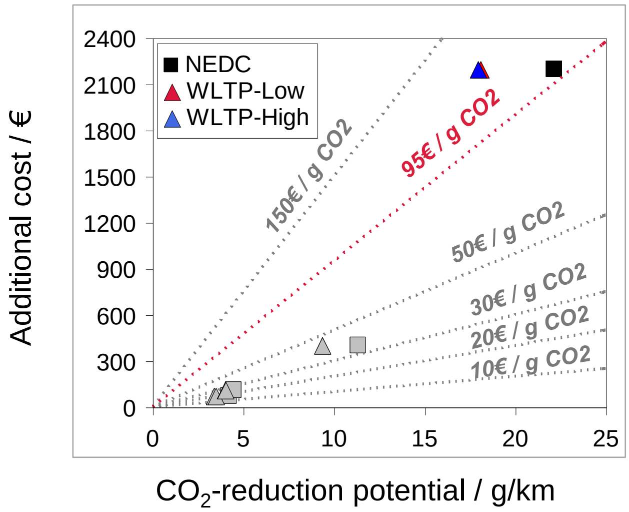


Figure 7.36 Segment E cost analysis for hybrid

7.2.5 Segment SUV

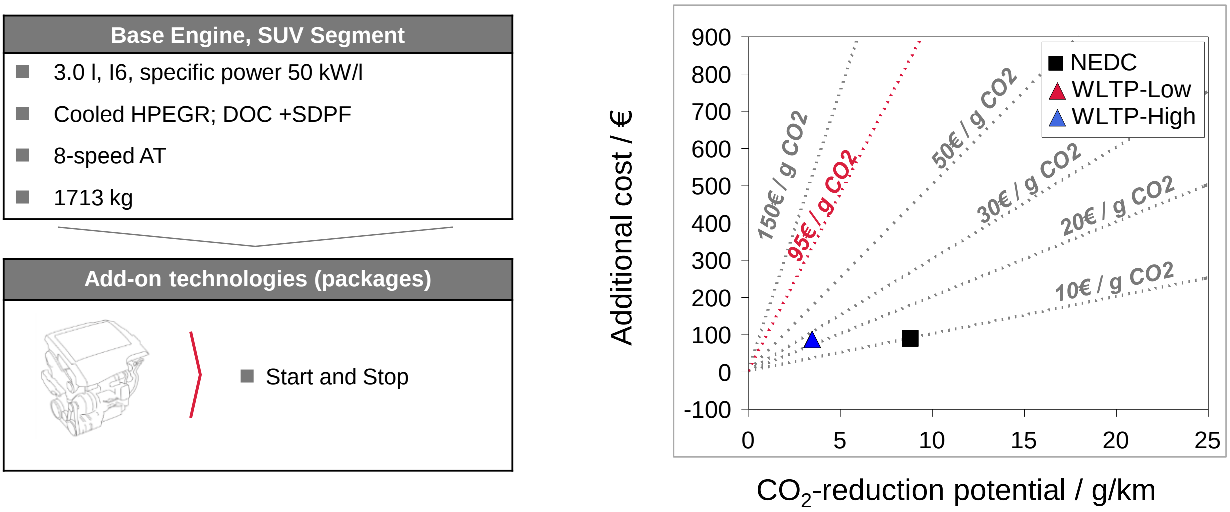


Figure 7.37 Segment SUV cost analysis for technological start and stop

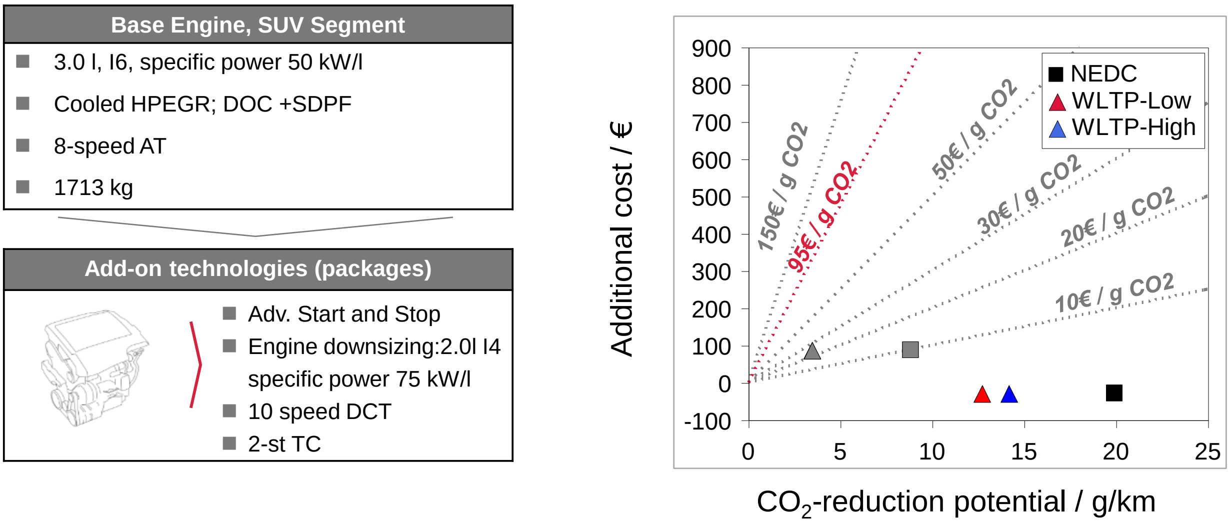


Figure 7.38 Segment SUV cost analysis for engine downsizing and transmissions upgrade

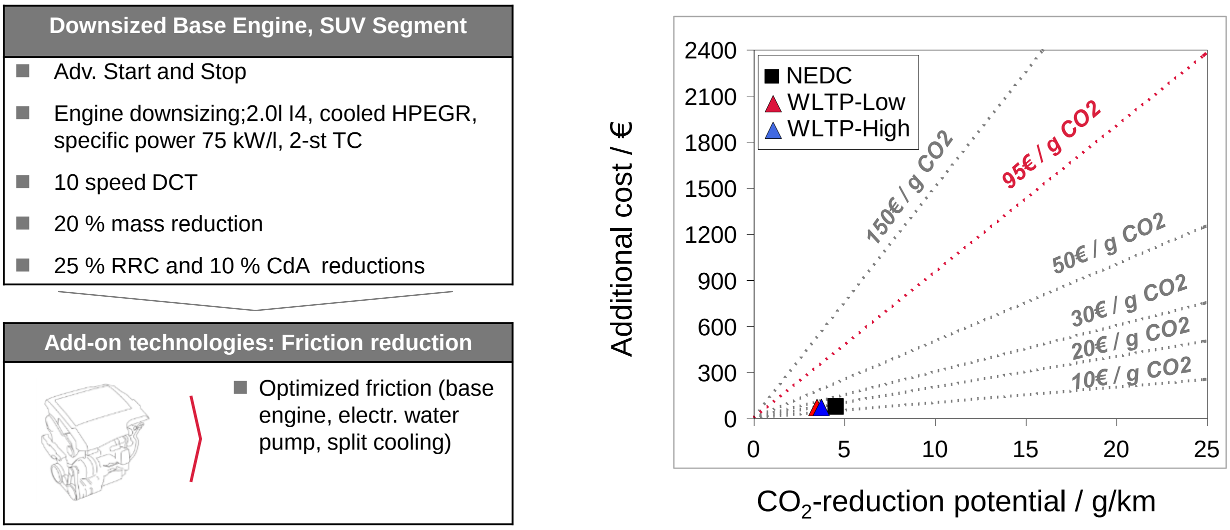


Figure 7.39 Segment SUV cost analysis for frictional upgrade

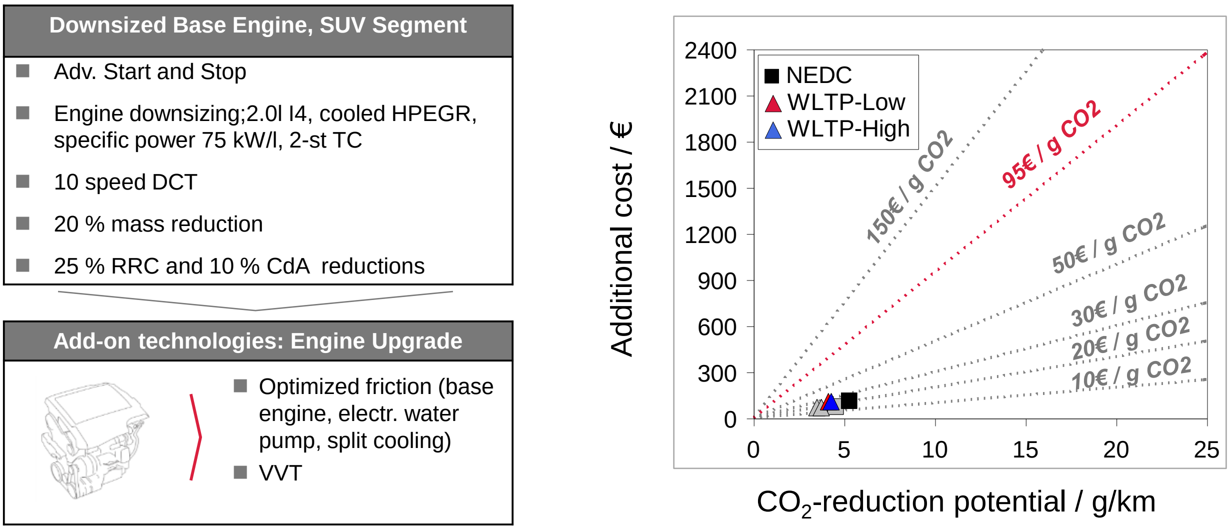
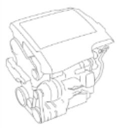


Figure 7.40 Segment SUV cost analysis for engine upgrade

Downsized Base Engine, SUV Segment

- Adv. Start and Stop
- Engine downsizing;2.0l I4, cooled HPEGR, specific power 75 kW/l, 2-st TC
- 10 speed DCT
- 20 % mass reduction
- 25 % RRC and 10 % CdA reductions

Add-on technologies: Top-spec Engine



- Optimized friction (base engine, electr. water pump, split cooling)
- VCR, CHPCLPEGR, VVT

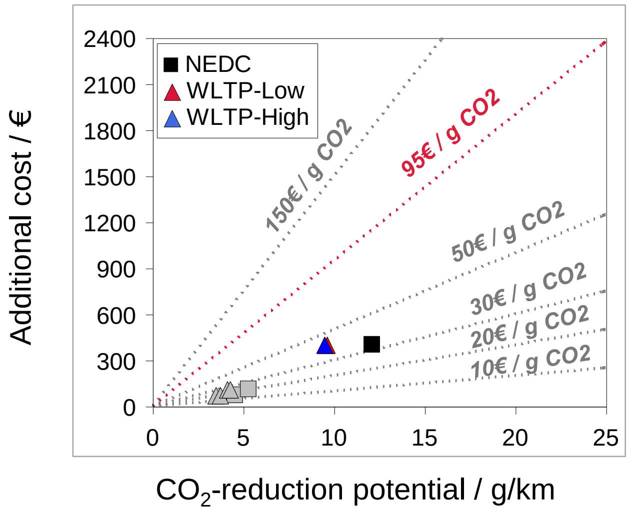
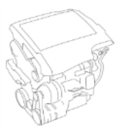


Figure 7.41 Segment SUV cost analysis for top-spec

Downsized Base Engine, SUV Segment

- Adv. Start and Stop
- Engine downsizing;2.0l I4, cooled HPEGR, specific power 75 kW/l, 2-st TC
- 10 speed DCT
- 20 % mass reduction
- 25 % RRC and 10 % CdA reductions

Add-on technologies: Hybrid



- Optimized friction (base engine, electr. water pump, split cooling)
- CHPCLPEGR, VVT
- P2 Hybrid – 40 kW 350 V

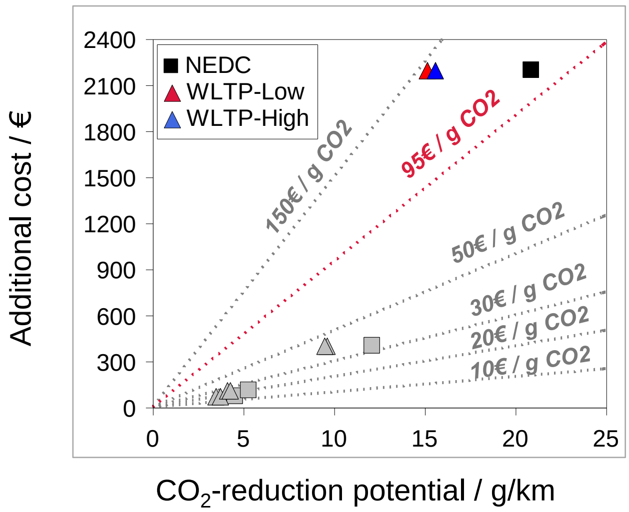


Figure 7.42 Segment SUV cost analysis for hybrid

7.2.6 Segment LCV

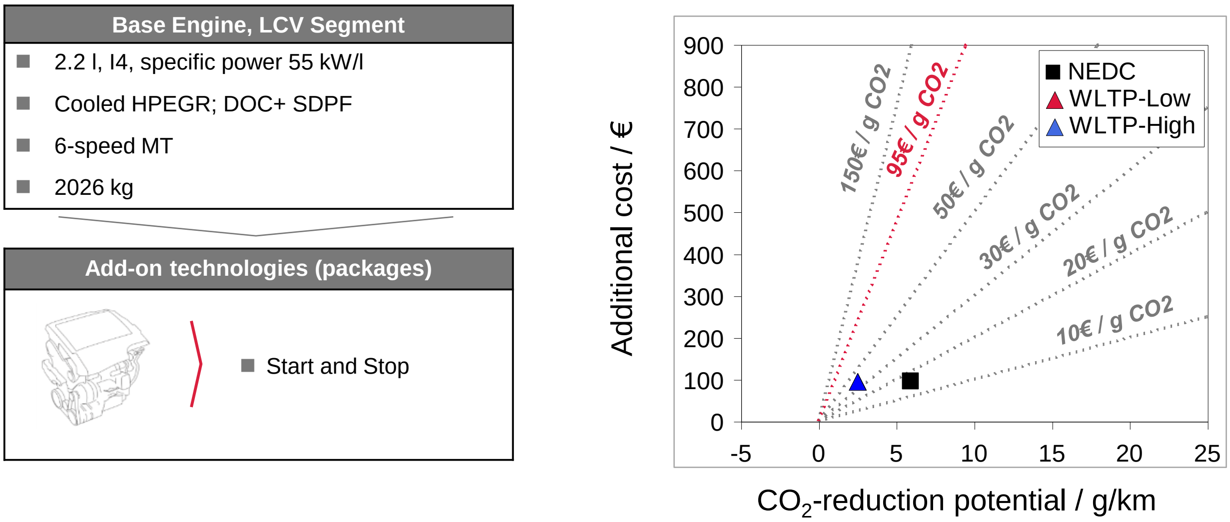


Figure 7.43 Segment LCV cost analysis for technological start and stop

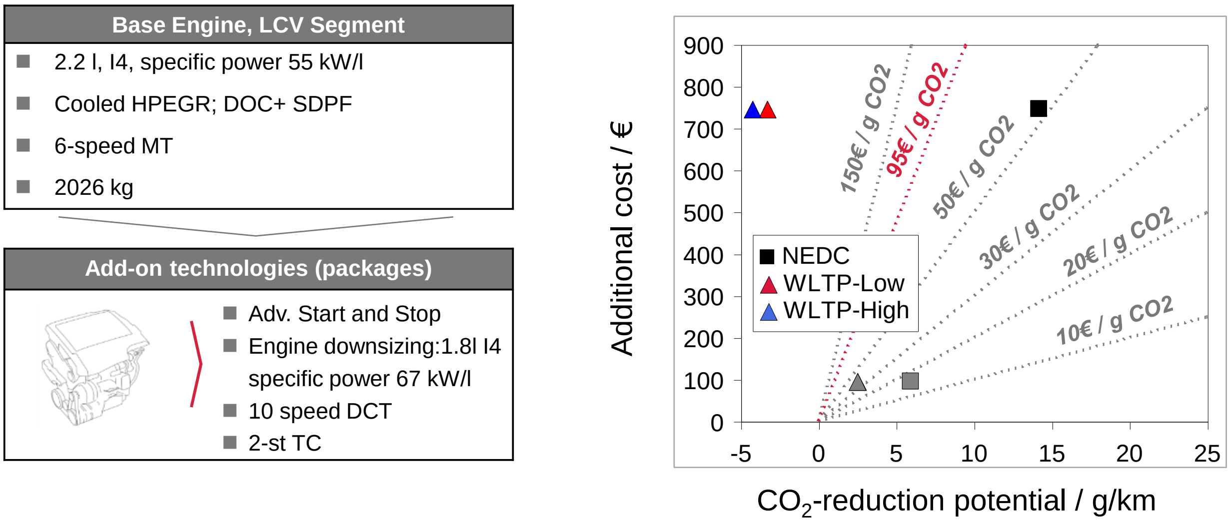


Figure 7.44 Segment LCV cost analysis for engine downsizing and transmissions upgrade

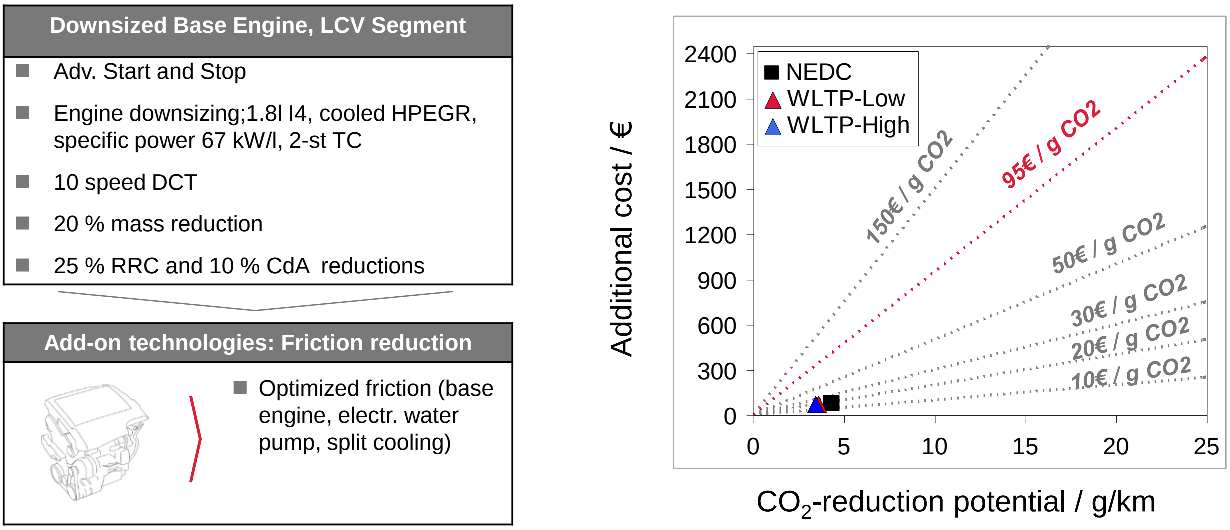


Figure 7.45 Segment LCV cost analysis for frictional upgrade

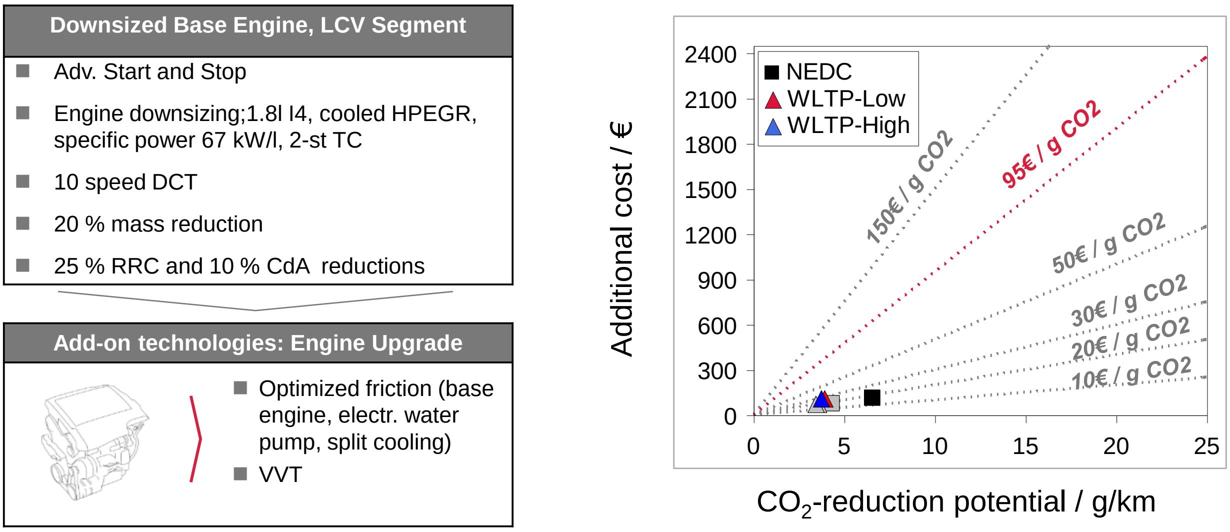


Figure 7.46 Segment LCV cost analysis for engine upgrade

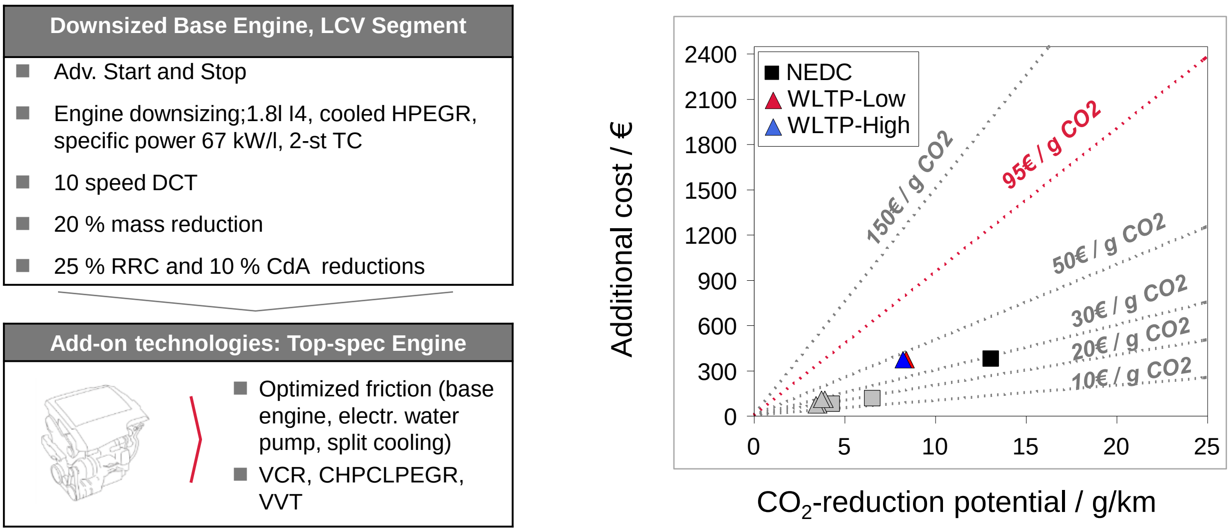


Figure 7.47 Segment LCV cost analysis for top-spec

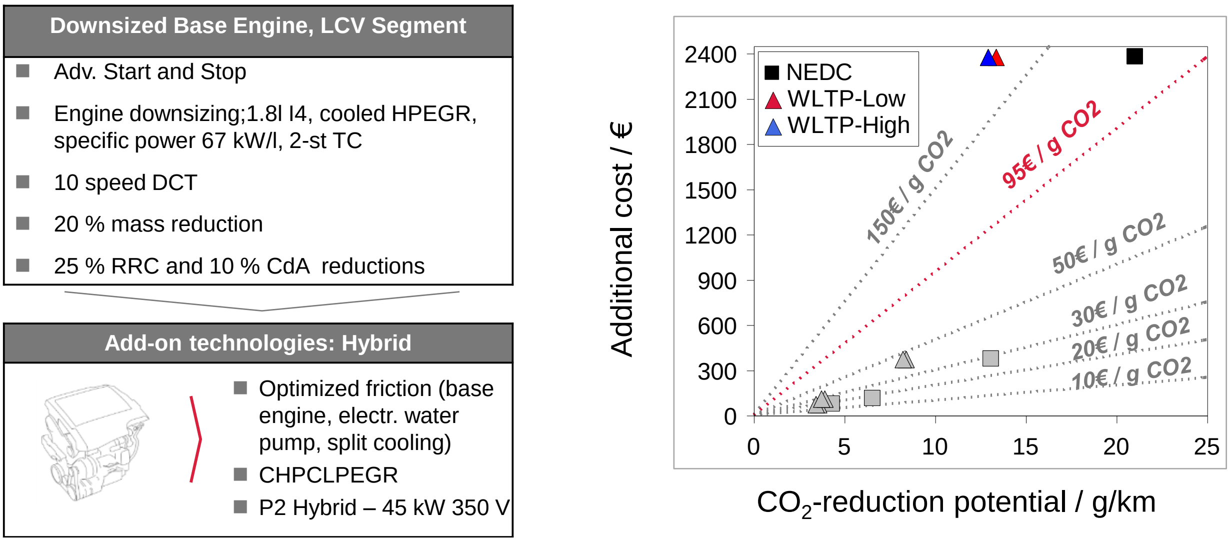


Figure 7.48 Segment LCV cost analysis for hybrid

8 Appendix: Gasoline Engines

8.1.1 Summary of average CO_2 emissions for all segments

8.1.1.1 Segment B

Segment B	Engine size / l	Cyl.	Spec.power / kW/l	EGR	T/C	Valvetrain	Compression Ratio	Engine friction	Hybridization	Transm.	Mass set	Coast down set
Baseline	1.3	4	50	w/o EGR	NA	DVVT	not variable	baseline	without	5-speed MT	M1	CD1
Start&Stop	1.3	4	50	w/o EGR	NA	DVVT	not variable	baseline	**start/stop**	5-speed MT	M1	CD1
Downsizing	**1.0**	**3**	**65**	**w/o EGR**	**single**	DVVT	not variable	baseline	**without**	5-speed MT	M1	CD1
Start&Stop	1.0	3	65	w/o EGR	single	DVVT	not variable	baseline	**start/stop**	5-speed MT	M1	CD1
Downsizing +Transmission upgrade +Weight reduction	**0.8**	3	**81**	w/o EGR	single	DVVT	not variable	baseline	start/stop	**6-speed MT**	**M2**	**CD1**
Mass red. Scenario II	0.8	3	81	w/o EGR	single	DVVT	not variable	baseline	start/stop	6-speed MT	**M3**	**CD1-CD2-CD3**
Friction package	0.8	3	81	w/o EGR	single	DVVT	not variable	**optimized**	start/stop	6-speed MT	M3	CD2
Engine technologies	0.8	3	81	w/o EGR	single	**DVVT +VVL**	not variable	optimized	start/stop	6-speed MT	M3	CD2
Engine technologies	0.8	3	81	w/o EGR	single	**DVVT +VVL +Miller**	not variable	optimized	start/stop	6-speed MT	M3	CD2
Engine technologies	0.8	3	81	**cooled LP**	single	**DVVT +VVL**	not variable	optimized	start/stop	6-speed MT	M3	CD2
Engine technologies	0.8	3	81	cooled LP	single	**DVVT +VVL +Miller**	not variable	optimized	start/stop	6-speed MT	M3	CD2
w/o weight reduction	0.8	3	81	cooled LP	single	DVVT +VVL +Miller	not variable	optimized	start/stop	6-speed MT	**M1**	**CD2**
DCT version to compare with hybrid	0.8	3	81	cooled LP	single	DVVT +VVL +Miller	not variable	optimized	start/stop	**7-speed DCT**	**M3**	**CD2**
Top spec engine + Hybrid	0.8	3	81	cooled LP	single	DVVT +VVL +Miller	not variable	optimized	**P0 with 48V**	7-speed DCT	M3	CD2

Table 8.1 Summary of variants for Segment B

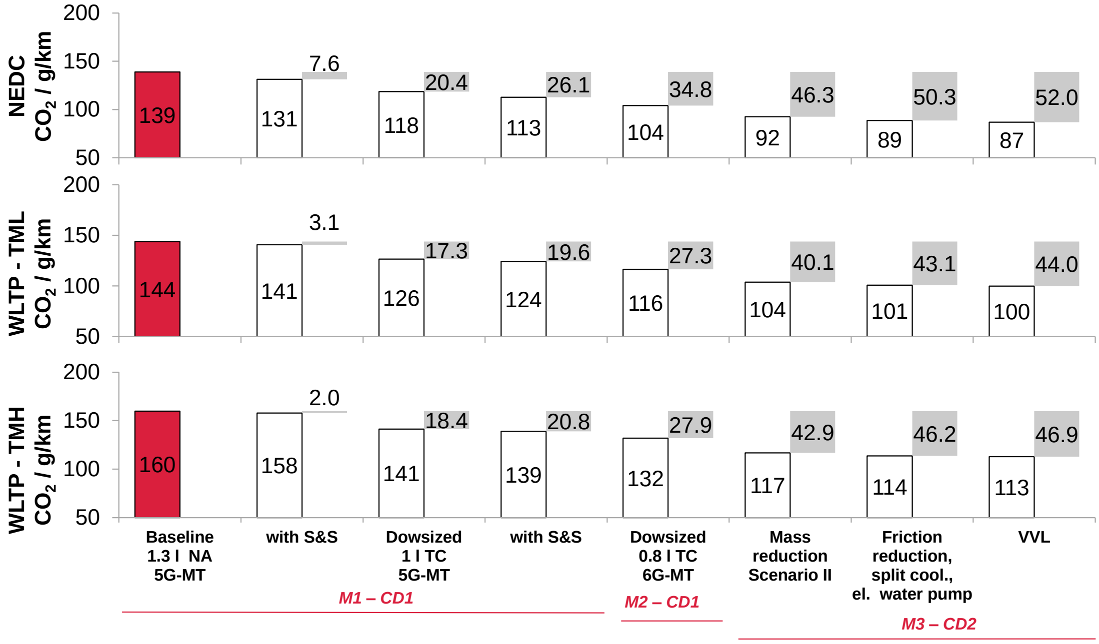


Figure 8.1 Overview of results for segment B in the EU cycles (part 1/2)

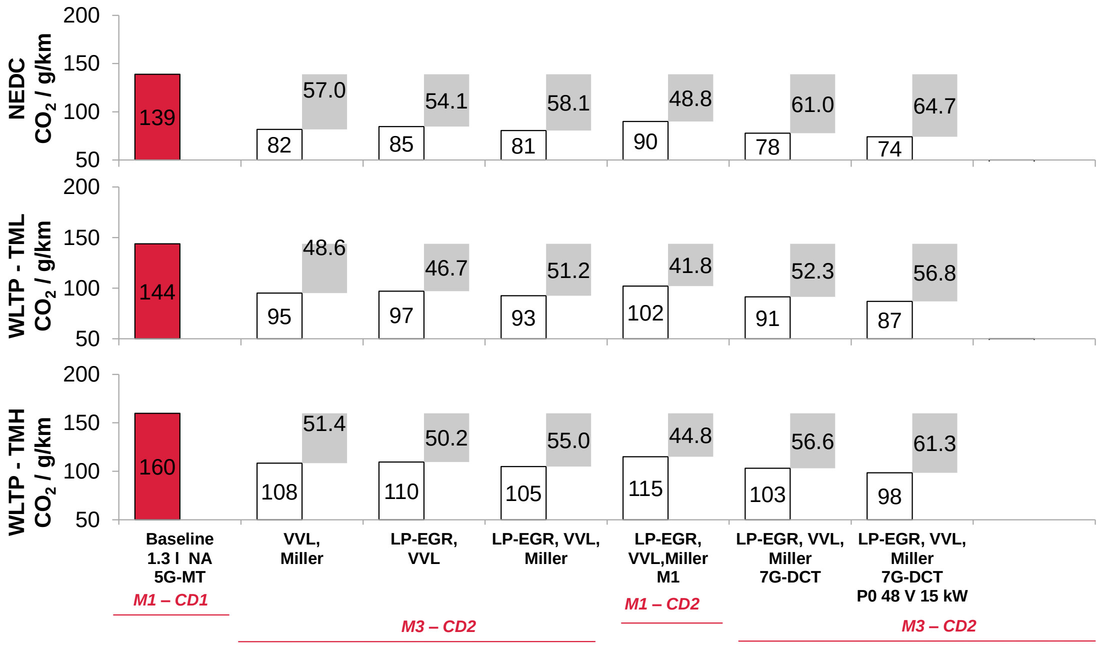


Figure 8.2 Overview of results for segment B in the EU cycles (part 2/2)

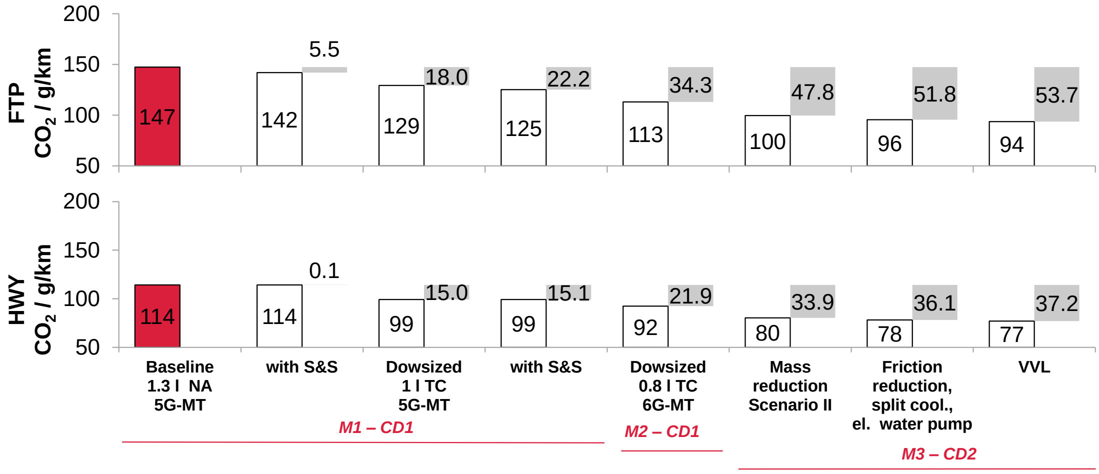


Figure 8.3 Overview of results for segment B in the US cycles (part 1/2)

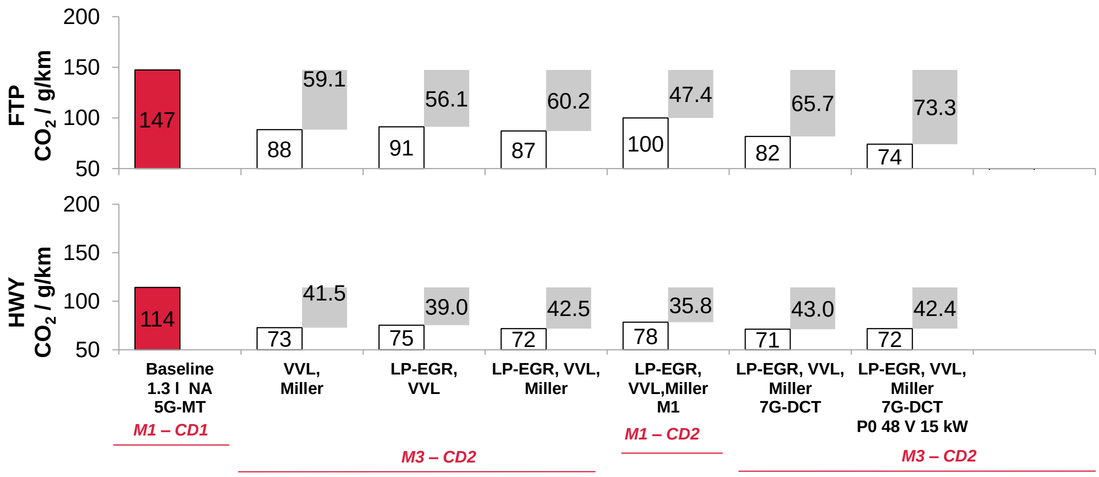


Figure 8.4 Overview of results for segment B in the US cycles (part 2/2)

8.1.1.2 Segment C

Segment C	Engine size / l	Cyl.	Spec.power / kW/l	EGR	T/C	Valvetrain	Compression Ratio	Engine friction	Hybridization	Transm.	Mass set	Coast down set
Baseline	1.8	4	53	w/o EGR	NA	DVVT	not variable	baseline	without	5-speed MT	M1	CD1
Start&Stop	1.8	4	53	w/o EGR	NA	DVVT	not variable	baseline	**start/stop**	5-speed MT	M1	CD1
Downsizing	**1.4**	4	**68**	w/o EGR	**single**	DVVT	not variable	baseline	**without**	5-speed MT	M1	CD1
Start&Stop	1.4	4	68	w/o EGR	single	DVVT	not variable	baseline	**start/stop**	5-speed MT	M1	CD1
Downsizing +Transmission upgrade +Weight reduction	**1.0**	**3**	**95**	w/o EGR	single	DVVT	not variable	baseline	start/stop	**6-speed MT**	**M2**	**CD1**
Mass red. Scenario II	1.0	3	95	w/o EGR	single	DVVT	not variable	baseline	start/stop	6-speed MT	**M3**	**CD1/CD2/CD3**
Transmission	1.0	3	95	w/o EGR	single	DVVT	not variable	baseline	**advanced start/stop**	**7-speed DCT**	M3	CD2
Friction package	1.0	3	95	w/o EGR	single	DVVT	not variable	**optimized**	advanced start/stop	7-speed DCT	M3	CD2
Engine technologies	1.0	3	95	w/o EGR	single	**DVVT +VVL**	not variable	optimized	advanced start/stop	7-speed DCT	M3	CD2
Engine technologies	1.0	3	95	w/o EGR	**2-stage**	**DVVT +VVL +Miller**	not variable	optimized	advanced start/stop	7-speed DCT	M3	CD2
Engine technologies	1.0	3	95	**cooled LP**	2-stage	**DVVT +VVL**	**two-step VCR**	optimized	advanced start/stop	7-speed DCT	M3	CD2
Engine technologies	1.0	3	95	cooled LP	2-stage	**DVVT +VVL +Miller**	**not variable**	optimized	advanced start/stop	7-speed DCT	M3	CD2
w/o weight reduction	1.0	3	95	cooled LP	2-stage	DVVT +VVL +Miller	not variable	optimized	advanced start/stop	7-speed DCT	**M1**	**CD2**
Extreme downsizing	**0.8**	3	**119**	cooled LP	2-stage	**DVVT +VVL**	not variable	optimized	advanced start/stop	7-speed DCT	**M3**	**CD2**
Top spec engine + Hybrid	**1.0**	3	**95**	cooled LP	2-stage	**DVVT +VVL +Miller**	not variable	optimized	**P0 with 48V**	7-speed DCT	M3	CD2
Extreme downsizing + Hybrid	**0.8**	3	**119**	cooled LP	2-stage	**DVVT +VVL**	not variable	optimized	P0 with 48V	7-speed DCT	M3	CD2

Table 8.2 Summary of variants for Segment C

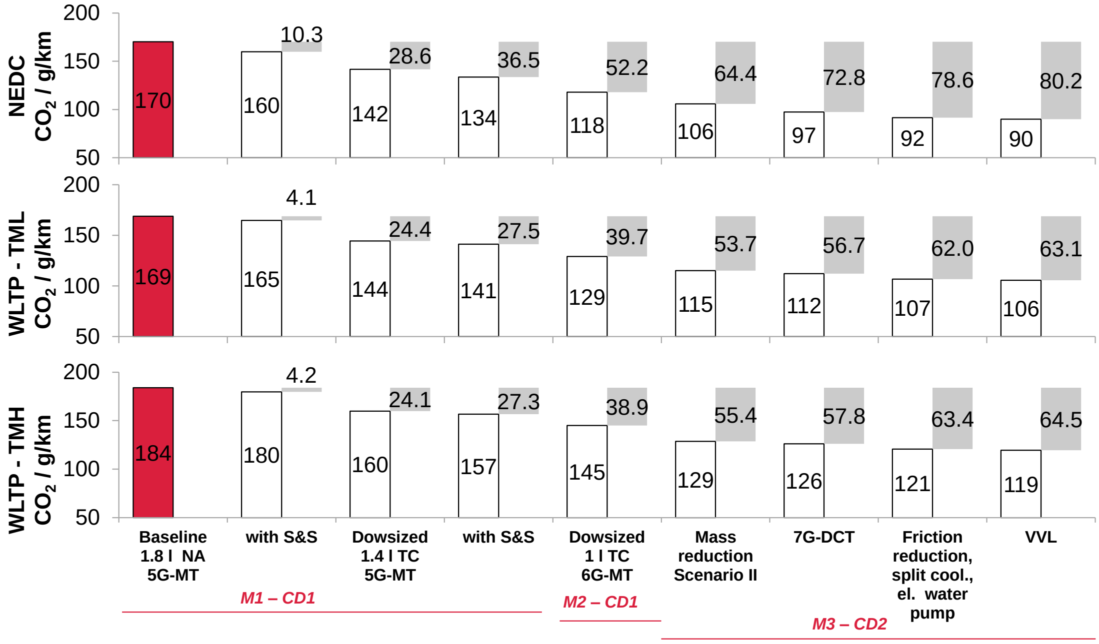


Figure 8.5 Overview of results for segment C in the EU cycles (part 1/2)

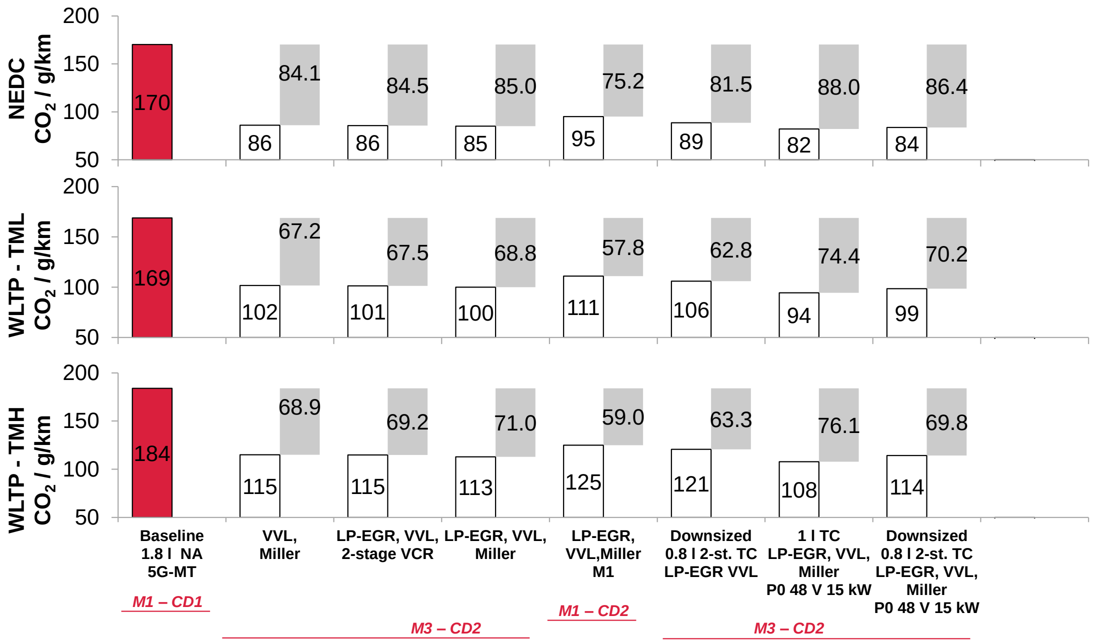


Figure 8.6 Overview of results for segment C in the EU cycles (part 2/2)

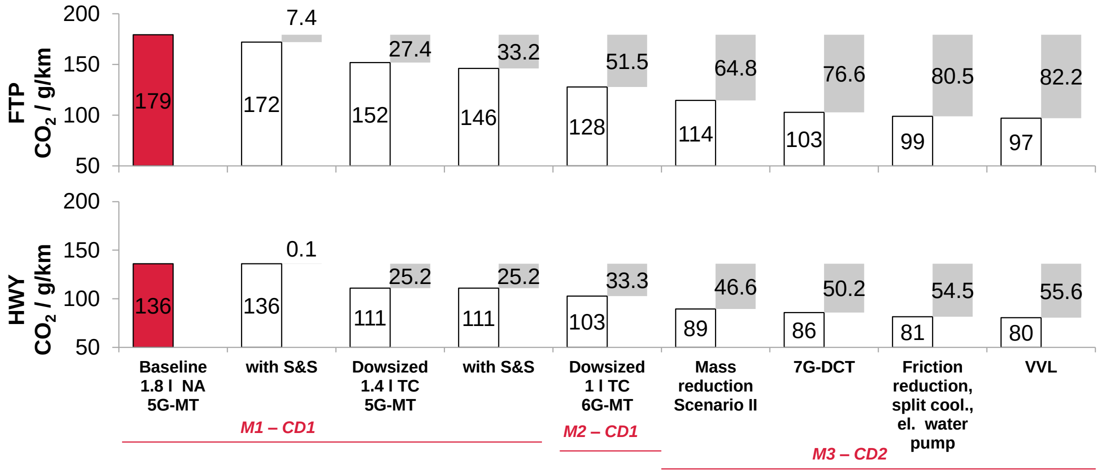


Figure 8.7 Overview of results for segment C in the US cycles (part 1/2)

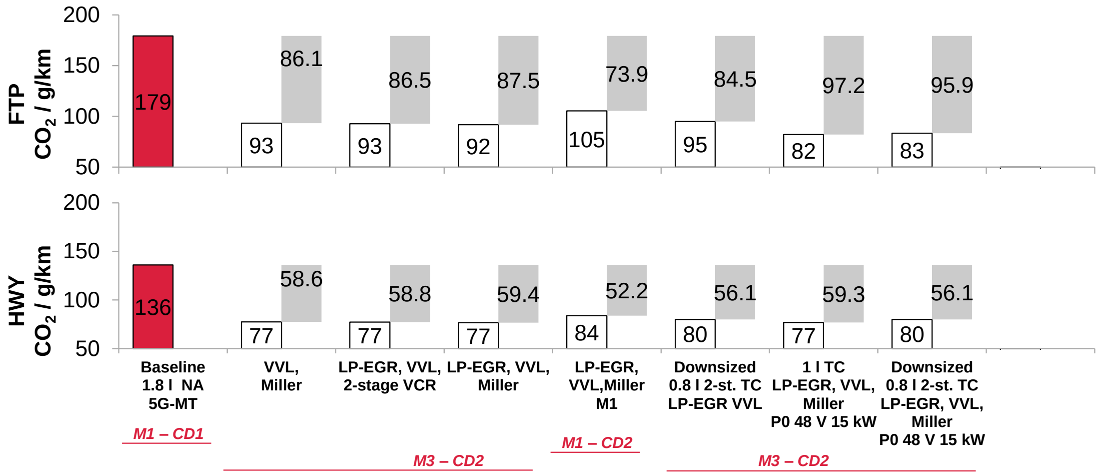


Figure 8.8 Overview of results for segment C in the US cycles (part 2/2)

8.1.1.3 Segment D

Segment D	Engine size / l	Cyl.	Spec.power / kW/l	EGR	T/C	Valvetrain	Compression Ratio	Engine friction	Hybridization	Transm.	Mass set	Coast down set
Baseline	2.4	6	56	w/o EGR	NA	DVVT	not variable	baseline	without	8-speed AT	M1	CD1
Start&Stop	2.4	6	56	w/o EGR	NA	DVVT	not variable	baseline	**start/stop**	8-speed AT	M1	CD1
Downsizing	**1.8**	**4**	**75**	w/o EGR	**single**	DVVT	not variable	baseline	**without**	8-speed AT	M1	CD1
Start&Stop	1.8	4	75	w/o EGR	single	DVVT	not variable	baseline	**start/stop**	8-speed AT	M1	CD1
Downsizing +Transmission upgrade +Weight reduction	**1.4**	4	**96**	w/o EGR	single	DVVT	not variable	baseline	**advanced start/stop**	**10-speed DCT**	**M2**	**CD1**
Mass red. Scenario II	1.4	4	96	w/o EGR	single	DVVT	not variable	baseline	advanced start/stop	10-speed DCT	**M3**	**CD1-CD2-CD3**
Friction package	1.4	4	96	w/o EGR	single	DVVT	not variable	**optimized**	advanced start/stop	10-speed DCT	M3	CD2
Engine technologies	1.4	4	96	w/o EGR	single	**DVVT +VVL**	not variable	optimized	advanced start/stop	10-speed DCT	M3	CD2
Engine technologies	1.4	4	96	w/o EGR	**2-stage**	**DVVT +VVL +Miller**	not variable	optimized	advanced start/stop	10-speed DCT	M3	CD2
Engine technologies	1.4	4	96	**cooled LP**	2-stage	**DVVT +VVL**	**two-step VCR**	optimized	advanced start/stop	10-speed DCT	M3	CD2
Engine technologies	1.4	4	96	cooled LP	2-stage	**DVVT +VVL +Miller**	**not variable**	optimized	advanced start/stop	10-speed DCT	M3	CD2
w/o weight reduction	1.4	4	96	cooled LP	2-stage	**DVVT +VVL +Miller**	not variable	optimized	advanced start/stop	10-speed DCT	**M1**	**CD2**
Engine technologies	**1.8**	4	**75**	**dedicated EGR**	**single**	**DVVT**	not variable	optimized	advanced start/stop	10-speed DCT	**M3**	**CD2**
Extreme downsizing	**1.0**	**3**	**135**	**cooled LP**	**2-stage**	**DVVT +VVL**	not variable	optimized	advanced start/stop	10-speed DCT	M3	CD2
Top spec engine + Hybrid	**1.4**	**4**	**96**	cooled LP	2-stage	**DVVT +VVL +Miller**	not variable	optimized	**full parallel hybrid**	10-speed DCT	M3	CD2
Extreme downsizing + Hybrid	**1.0**	**3**	**135**	cooled LP	2-stage	**DVVT +VVL**	not variable	optimized	full parallel hybrid	10-speed DCT	M3	CD2

Table 8.3 Summary of variants for Segment D

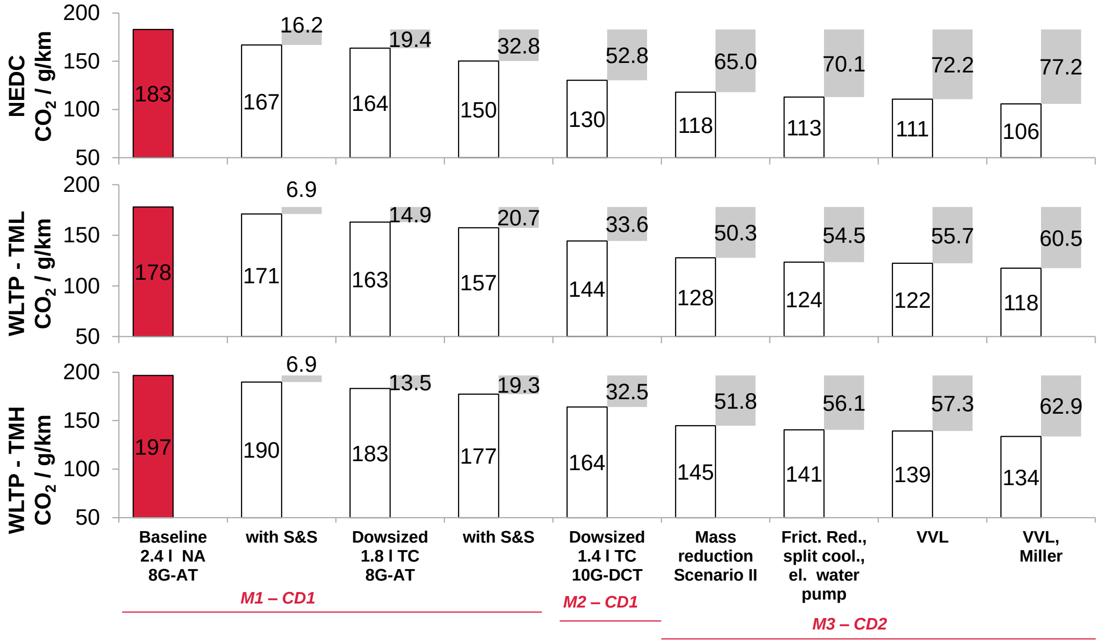


Figure 8.9 Overview of results for segment D in the EU cycles (part 1/2)

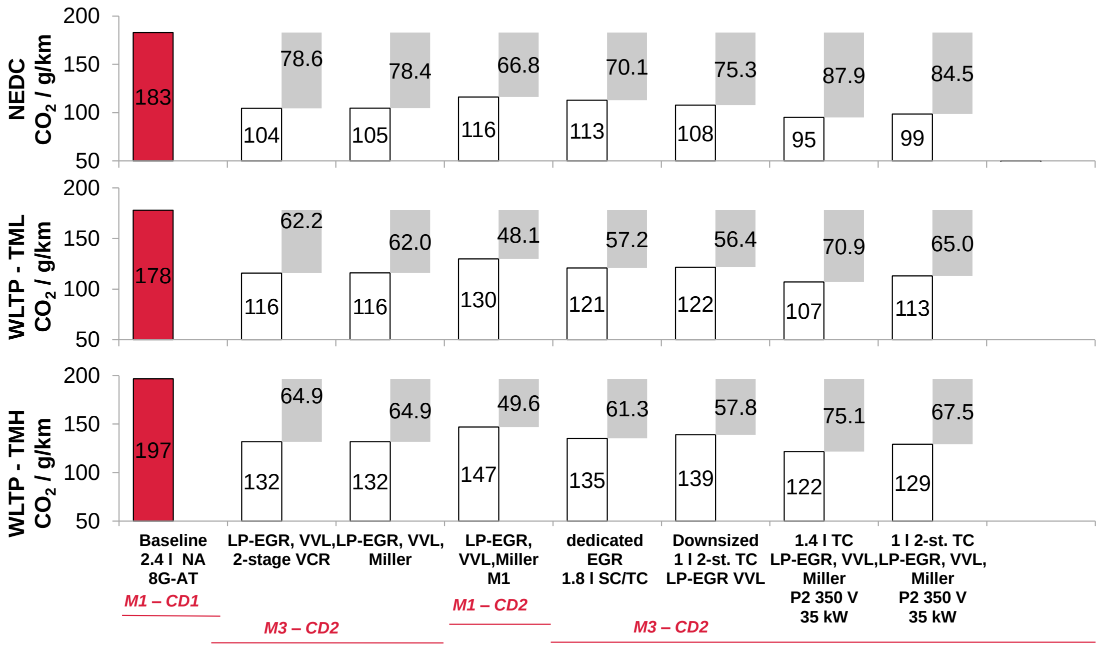


Figure 8.10 Overview of results for segment D in the EU cycles (part 2/2)

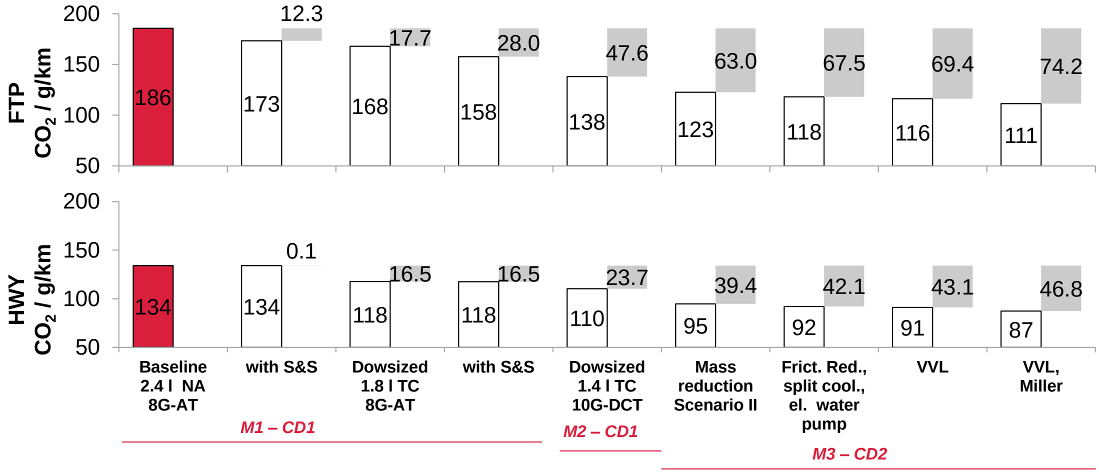


Figure 8.11 Overview of results for segment D in the US cycles (part 1/2)

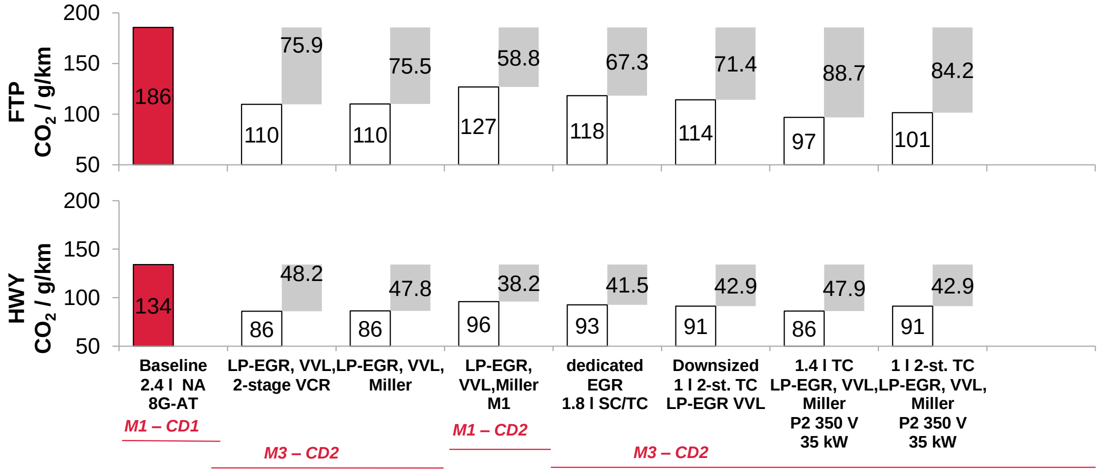


Figure 8.12 Overview of results for segment D in the US cycles (part 2/2)

8.1.1.4 Segment E

Segment E	Engine size / l	Cyl.	Spec.power / kW/l	EGR	T/C	Valvetrain	Compression Ratio	Engine friction	Hybridization	Transm.	Mass set	Coast down set
Baseline	3.0	6	60	w/o EGR	NA	DVVT	not variable	baseline	without	8-speed AT	M1	CD1
Start&Stop	3.0	6	60	w/o EGR	NA	DVVT	not variable	baseline	**start/stop**	8-speed AT	M1	CD1
Downsizing	**2.0**	**4**	**90**	w/o EGR	**single**	DVVT	not variable	baseline	**without**	8-speed AT	M1	CD1
Start&Stop	2.0	4	90	w/o EGR	single	DVVT	not variable	baseline	**start/stop**	8-speed AT	M1	CD1
Downsizing +Transmission upgrade +Weight reduction	**1.6**	4	**113**	w/o EGR	**single VTG**	DVVT	not variable	baseline	**advanced start/stop**	**10-speed DCT**	**M2**	**CD1**
Mass red. Scenario II	1.6	4	113	w/o EGR	single VTG	DVVT	not variable	baseline	advanced start/stop	10-speed DCT	**M3**	**CD1-CD2-CD3**
Friction package	1.6	4	113	w/o EGR	single VTG	DVVT	not variable	**optimized**	advanced start/stop	10-speed DCT	M3	CD2
Engine technologies	1.6	4	113	w/o EGR	single VTG	**DVVT +VVL**	not variable	optimized	advanced start/stop	10-speed DCT	M3	CD2
Engine technologies	1.6	4	113	w/o EGR	**2-stage**	**DVVT +VVL +Miller**	not variable	optimized	advanced start/stop	10-speed DCT	M3	CD2
Engine technologies	1.6	4	113	**cooled LP**	2-stage	**DVVT +VVL**	**two-step VCR**	optimized	advanced start/stop	10-speed DCT	M3	CD2
Engine technologies	1.6	4	113	cooled LP	2-stage	**DVVT +VVL +Miller**	**not variable**	optimized	advanced start/stop	10-speed DCT	M3	CD2
w/o weight reduction	1.6	4	113	cooled LP	2-stage	**DVVT +VVL +Miller**	not variable	optimized	advanced start/stop	10-speed DCT	**M1**	**CD2**
Engine technologies	**2.4**	4	**75**	**dedicated EGR**	**single**	**DVVT**	not variable	optimized	advanced start/stop	10-speed DCT	**M3**	**CD2**
Extreme downsizing	**1.2**	**3**	**150**	**cooled LP**	**2-stage**	**DVVT +VVL**	not variable	optimized	advanced start/stop	10-speed DCT	M3	CD2
Top spec engine + Hybrid	**1.6**	**4**	**113**	cooled LP	2-stage	**DVVT +VVL +Miller**	not variable	optimized	**full parallel hybrid**	10-speed DCT	M3	CD2
Extreme downsizing + Hybrid	**1.2**	**3**	**150**	cooled LP	2-stage	**DVVT +VVL**	not variable	optimized	full parallel hybrid	10-speed DCT	M3	CD2

Table 8.4 Summary of variants for Segment E

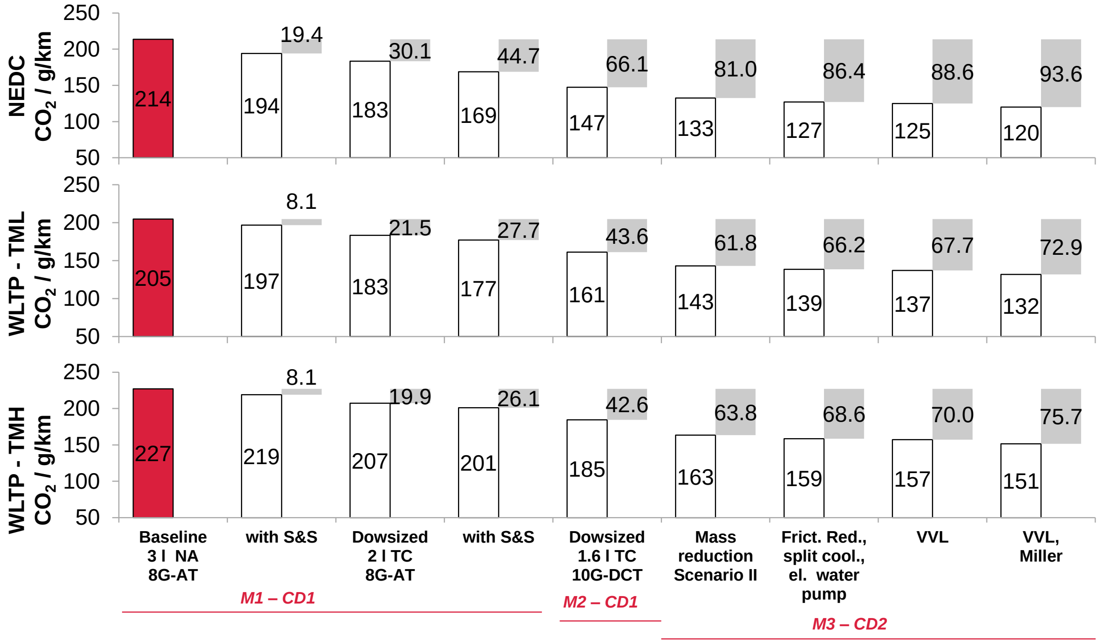


Figure 8.13 Overview of results for segment E in the EU cycles (part 1/2)

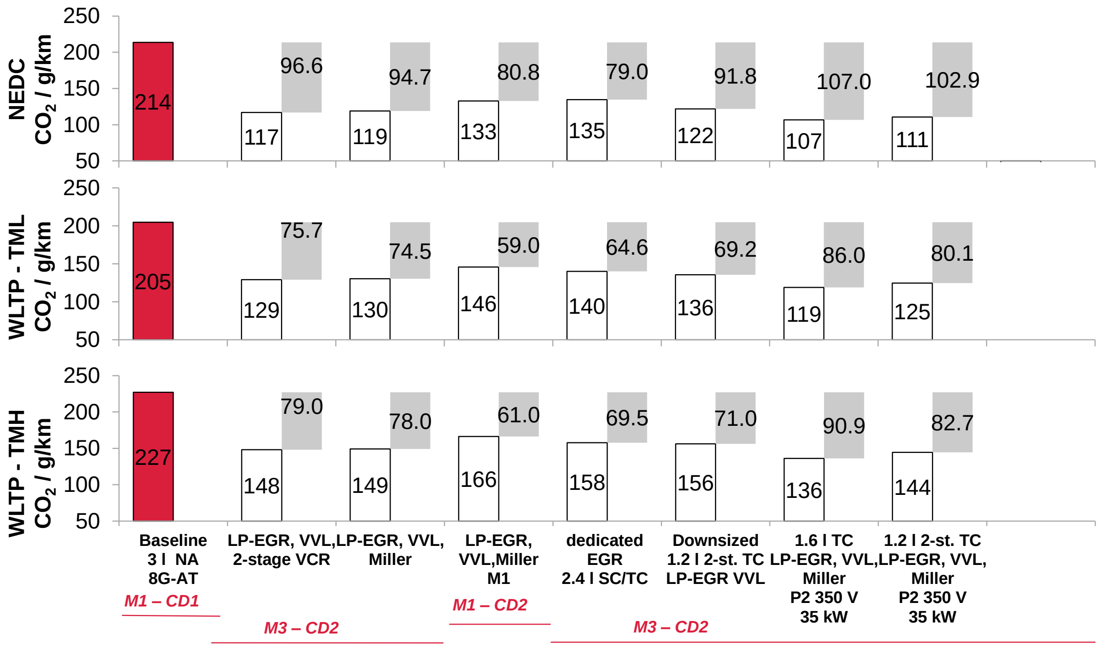


Figure 8.14 Overview of results for segment E in the EU cycles (part 2/2)

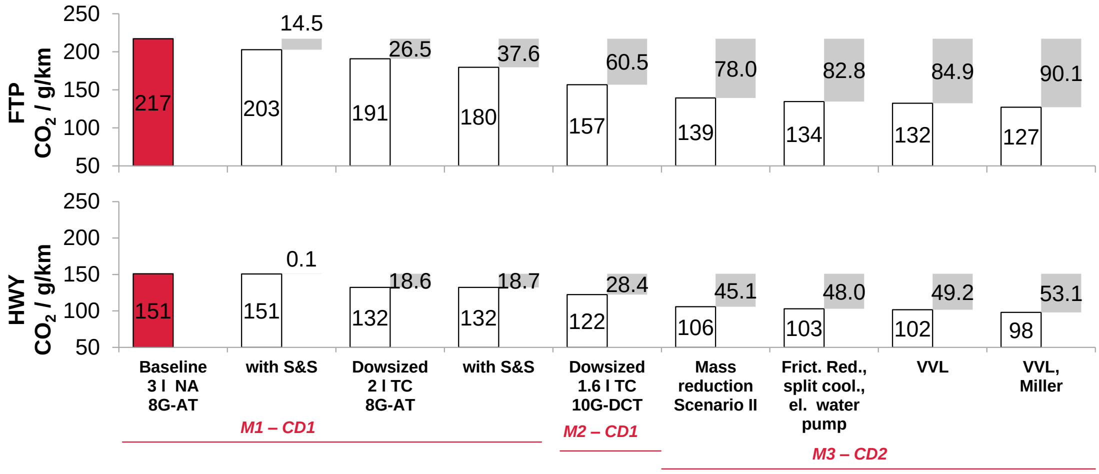


Figure 8.15 Overview of results for segment E in the US cycles (part 1/2)

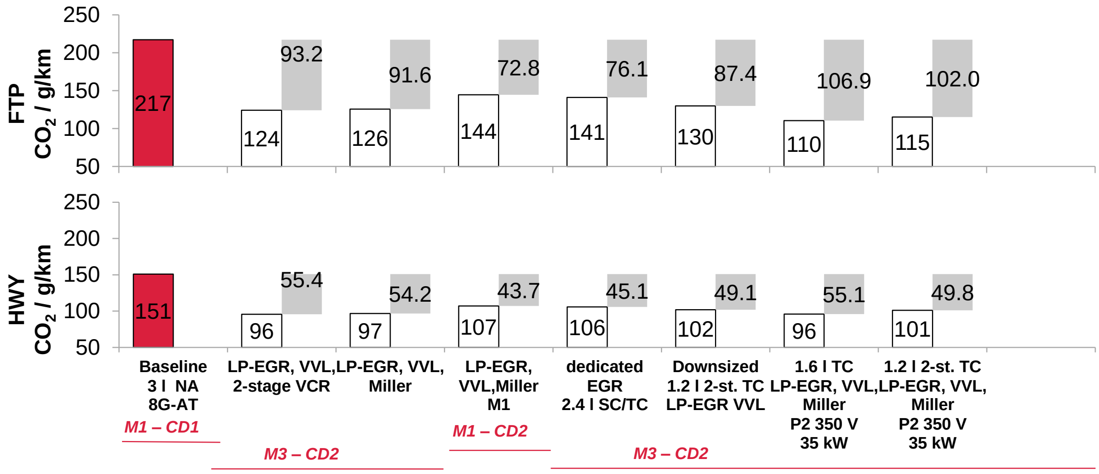


Figure 8.16 Overview of results for segment E in the US cycles (part 2/2)

8.2 CO_2 potential with respect to final cost for all segments

8.2.1 Segment B

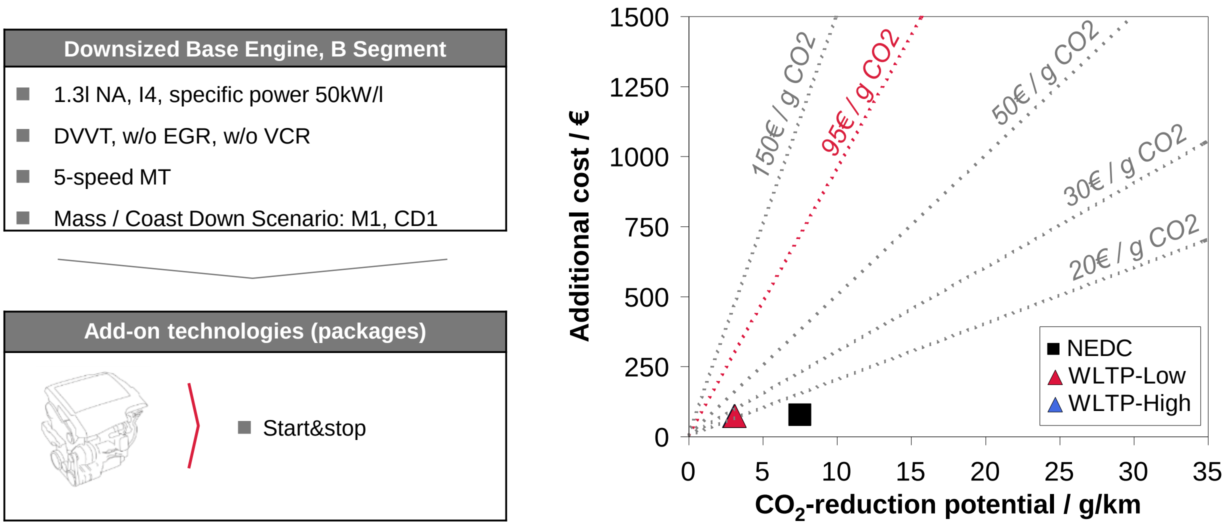


Figure 8.17 Segment B cost analysis for start and stop technology

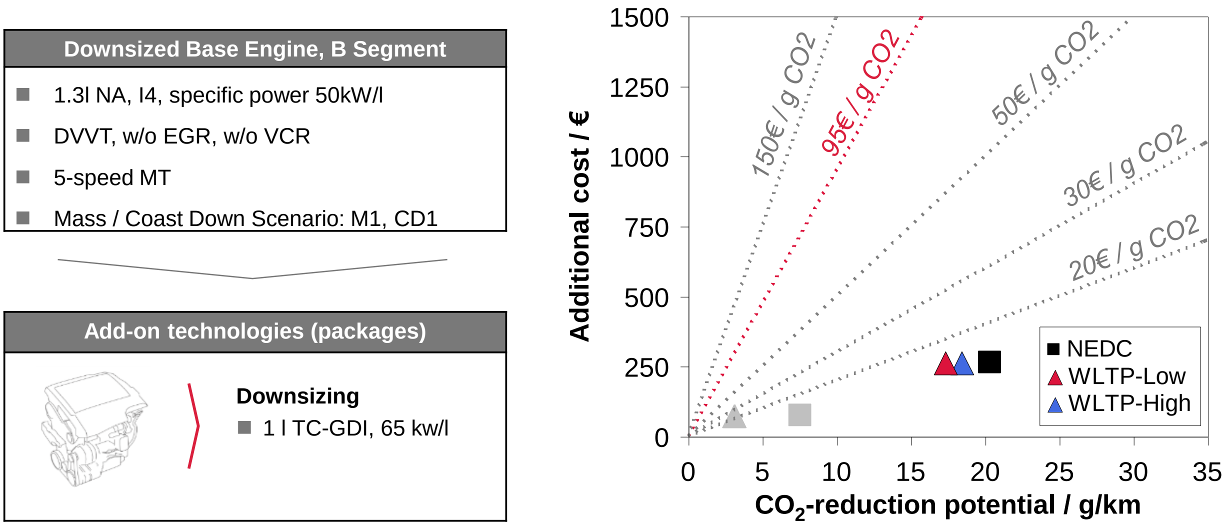


Figure 8.18 Segment B cost analysis for engine downsizing

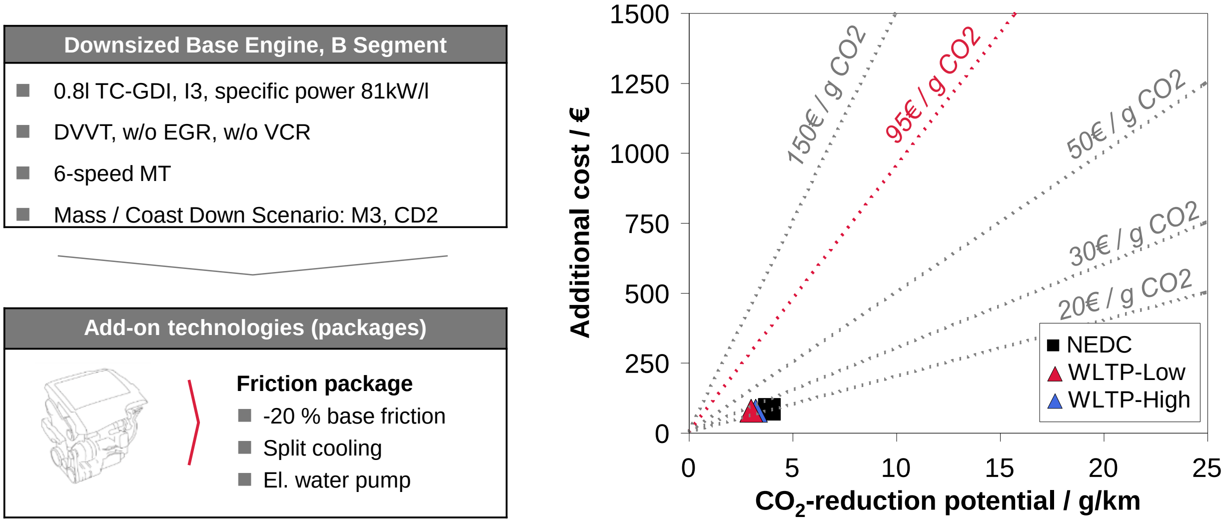


Figure 8.19 Segment B cost analysis for frictional upgrade

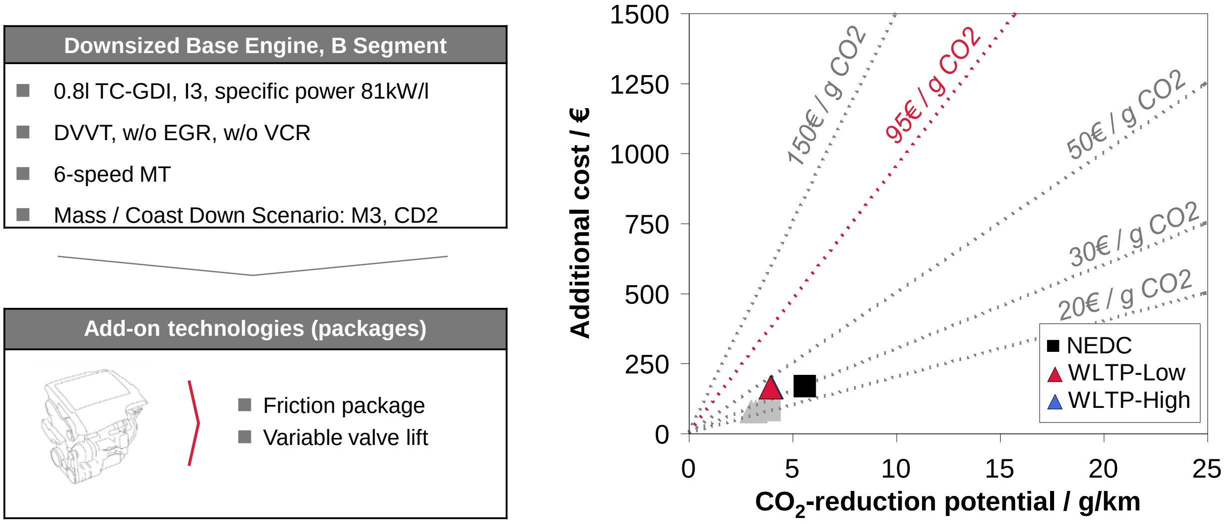


Figure 8.20 Segment B cost analysis for frictional upgrade and variable valve lift

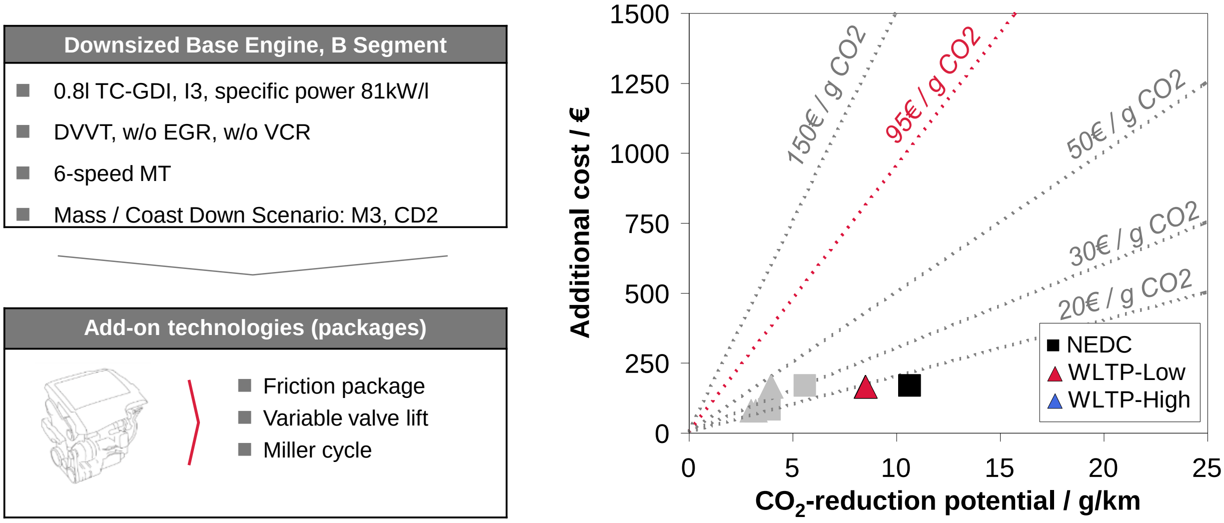


Figure 8.21 Segment B cost analysis for frictional upgrade, variable valve lift and Miller cycle

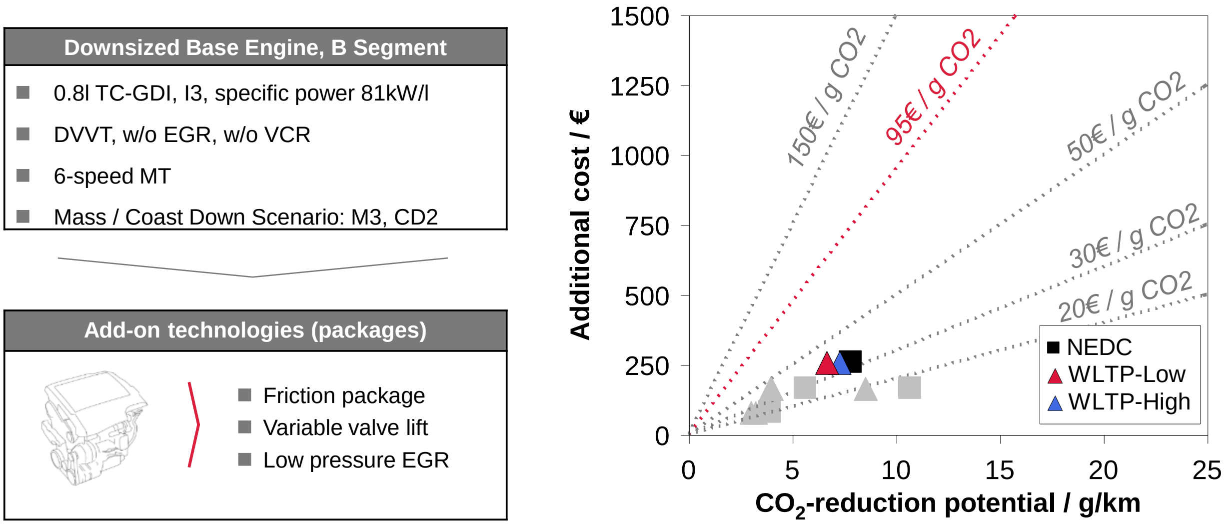


Figure 8.22 Segment B cost analysis for frictional upgrade, variable valve lift and low pressure EGR

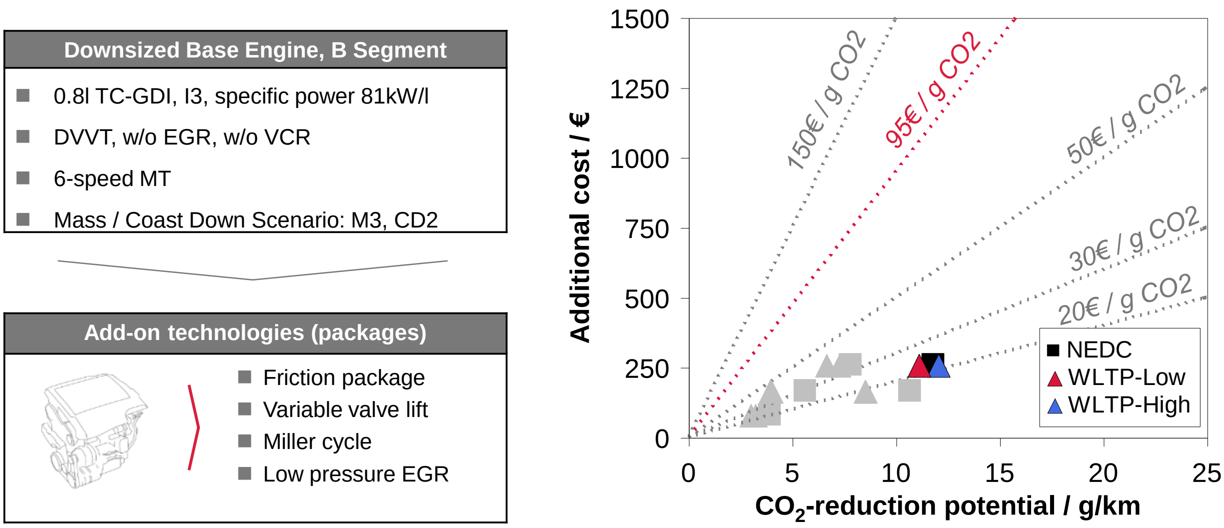


Figure 8.23 Segment B cost analysis for frictional upgrade, variable valve lift, Miller cycle and low pressure EGR

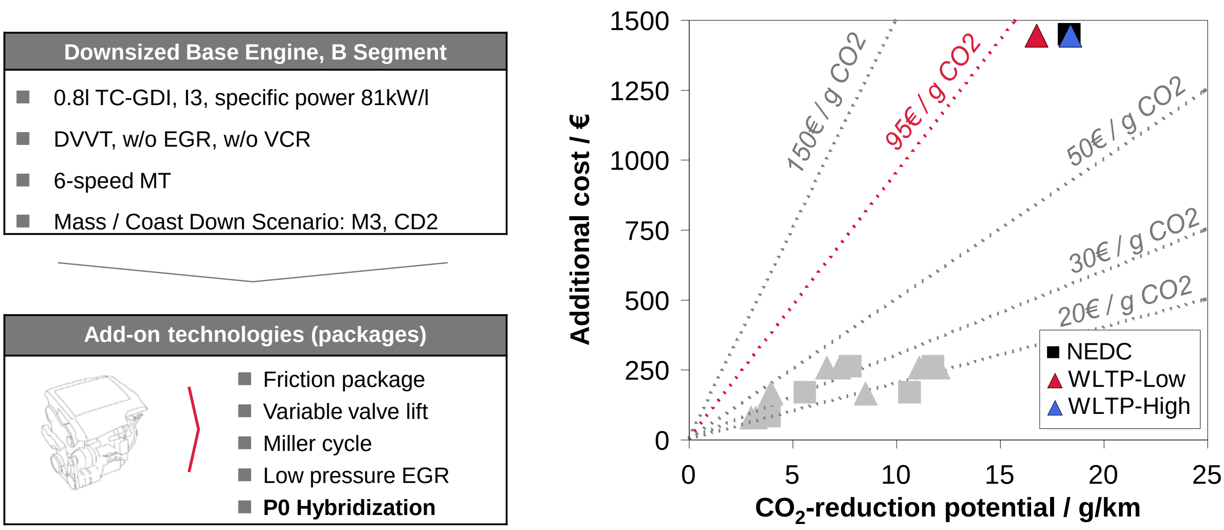


Figure 8.24 Segment B cost analysis for frictional upgrade, variable valve lift, Miller cycle, low pressure EGR and P0 hybridization

8.2.2 Segment C

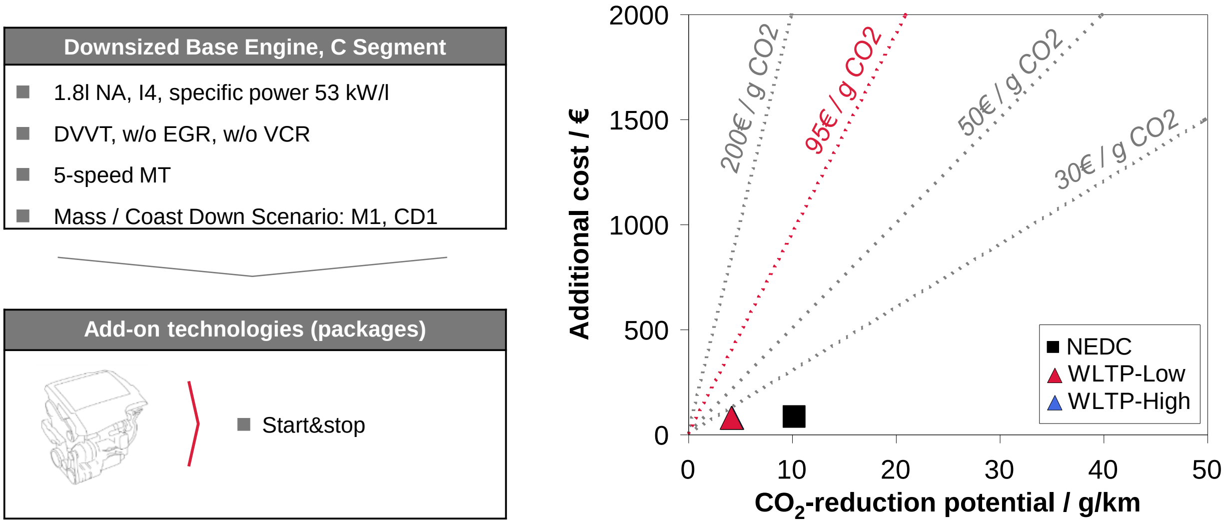


Figure 8.25 Segment C cost analysis for start&stop technology

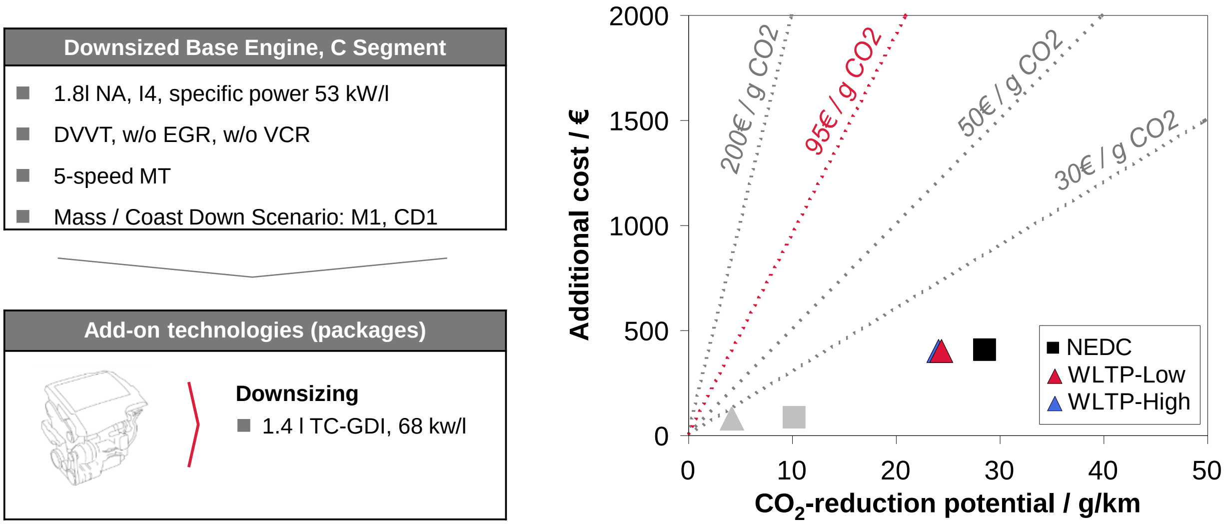


Figure 8.26 Segment C cost analysis for engine downsizing

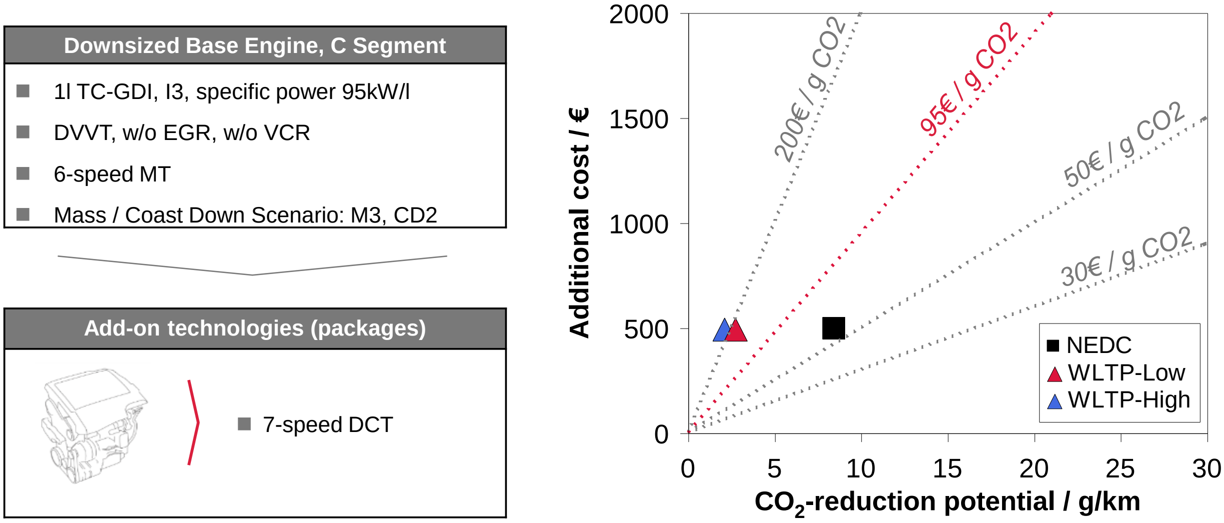


Figure 8.27 Segment C cost analysis for transmissions upgrade

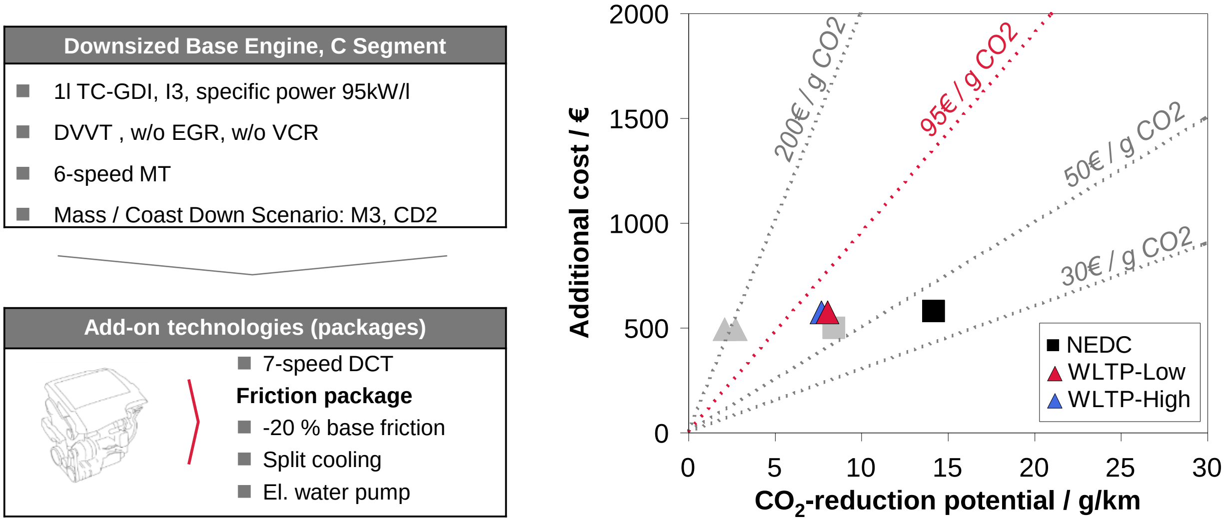


Figure 8.28 Segment C cost analysis for transmissions upgrade and frictional upgrade

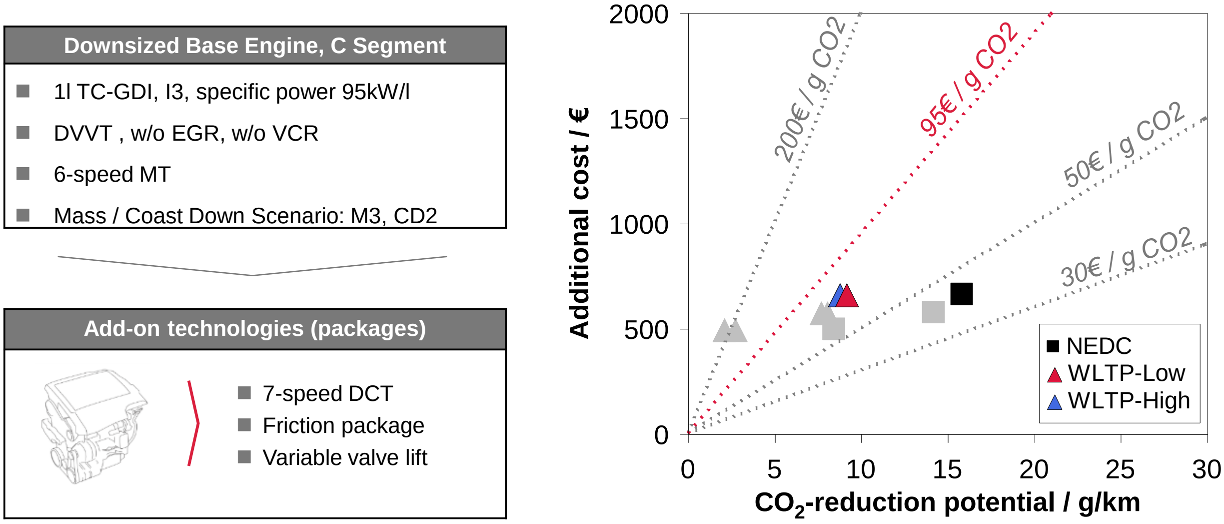


Figure 8.29 Segment C cost analysis for transmissions upgrade, frictional upgrade and variable valve lift

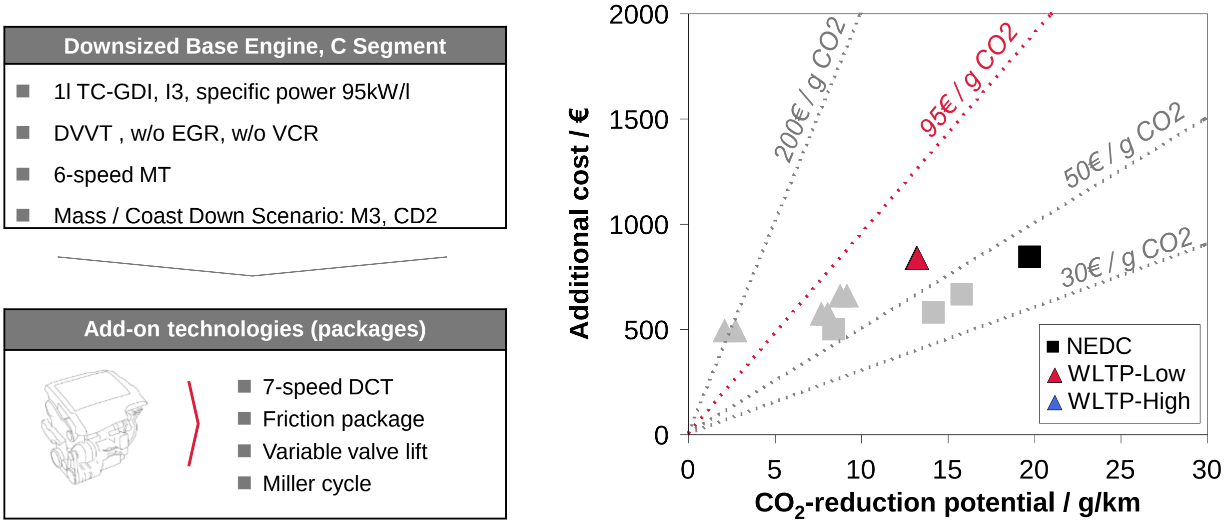


Figure 8.30 Segment C cost analysis for transmissions upgrade, frictional upgrade, variable valve lift and for Miller cycle

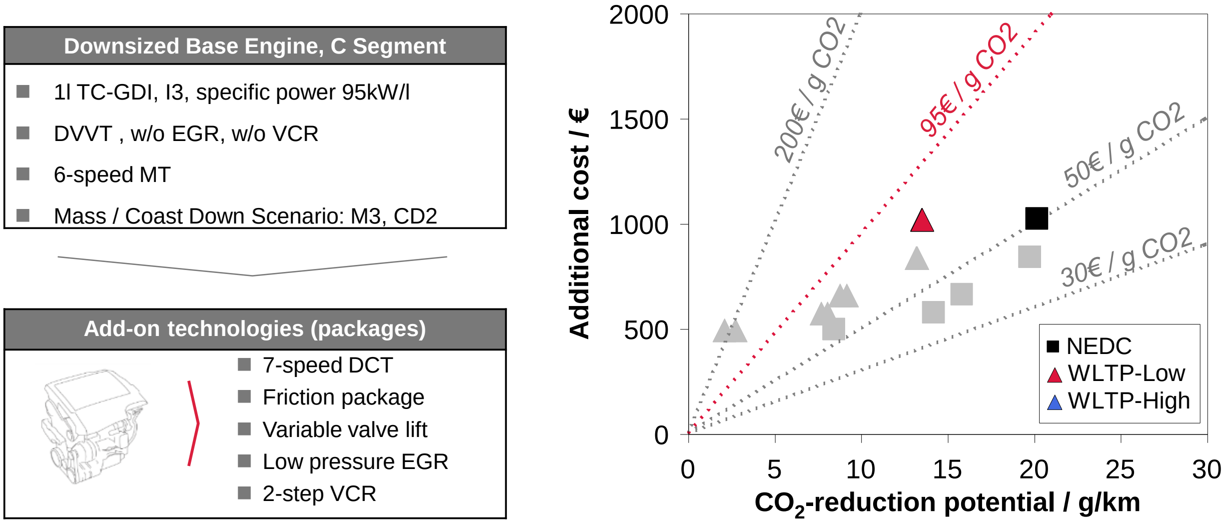


Figure 8.31 Segment C cost analysis for transmissions upgrade, frictional upgrade, variable valve lift, low pressure EGR and 2-step VCR

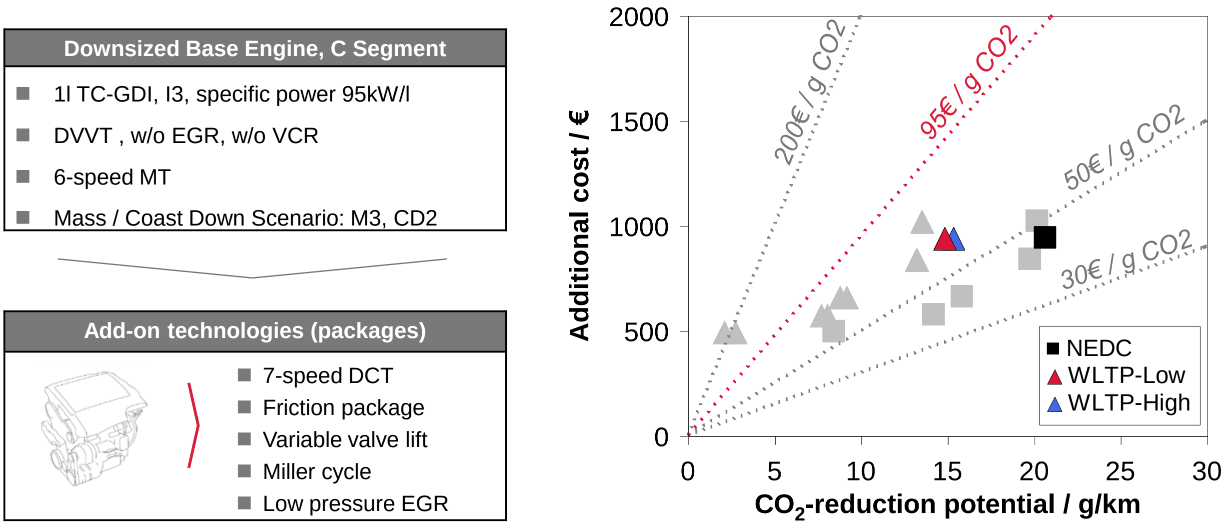


Figure 8.32 Segment C cost analysis for transmissions upgrade, frictional upgrade, variable valve lift, Miller cycle and low pressure EGR

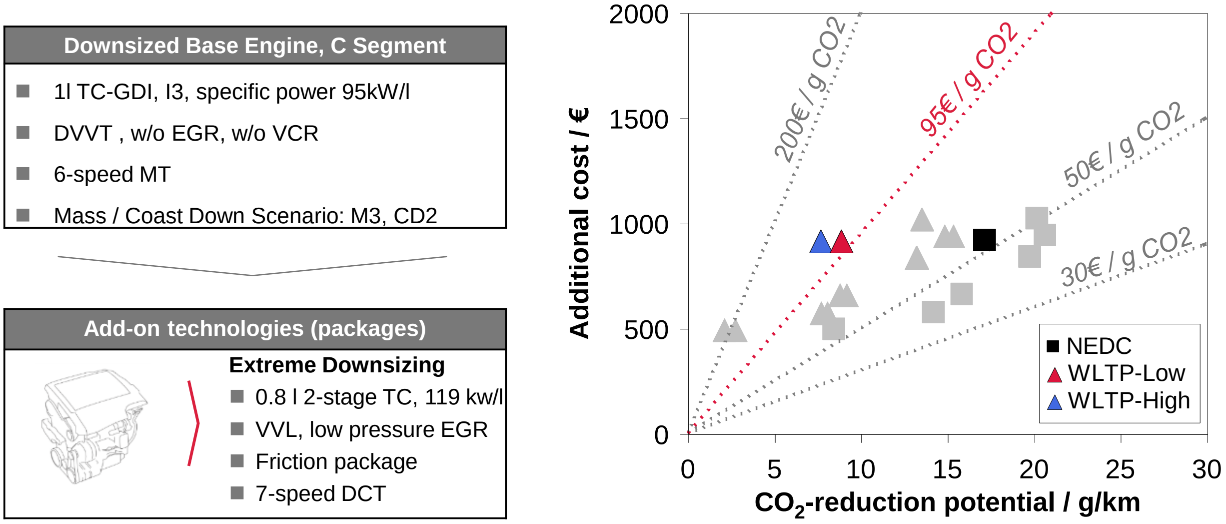


Figure 8.33 Segment C cost analysis for transmissions upgrade, frictional upgrade, variable valve lift, low pressure EGR and further engine downsizing

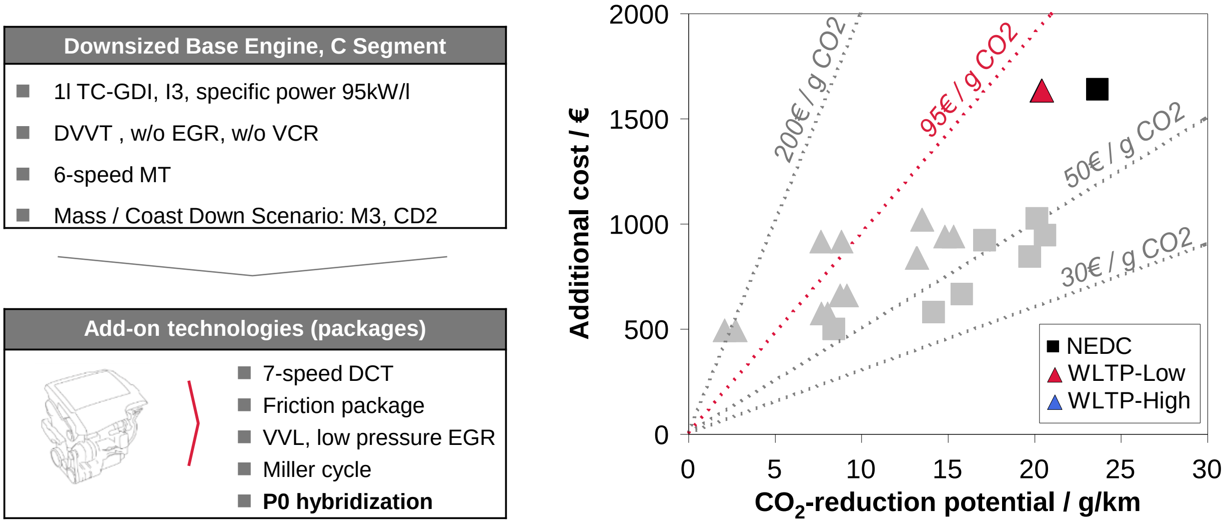


Figure 8.34 Segment C cost analysis for transmissions upgrade, frictional upgrade, variable valve lift, Miller cycle, low pressure EGR and P0 hybridization

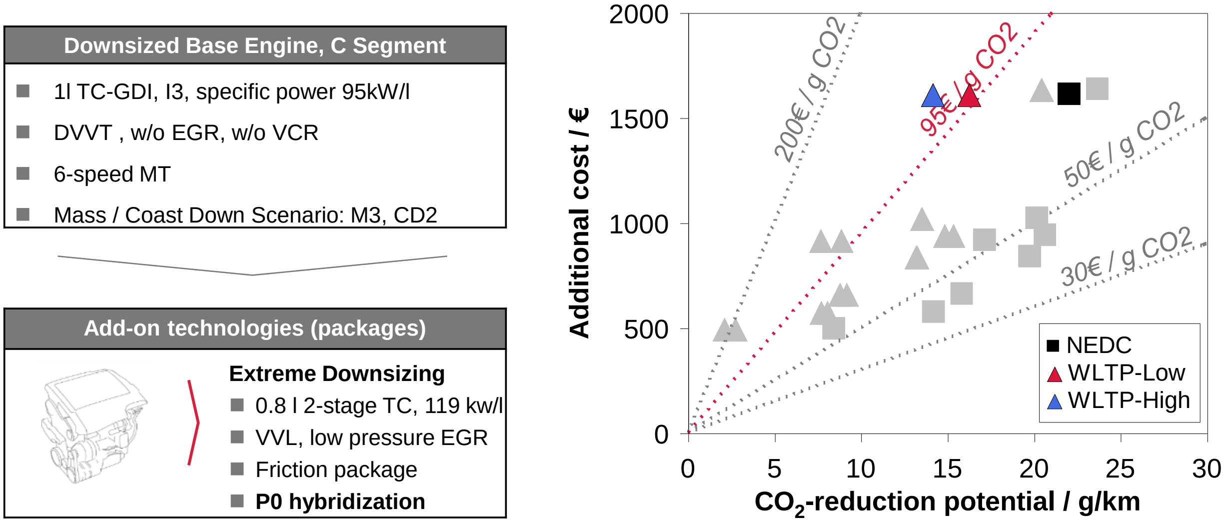


Figure 8.35 Segment C cost analysis for transmissions upgrade, frictional upgrade, variable valve lift, further engine downsizing, low pressure EGR and P0 hybridization

8.2.3 Segment D

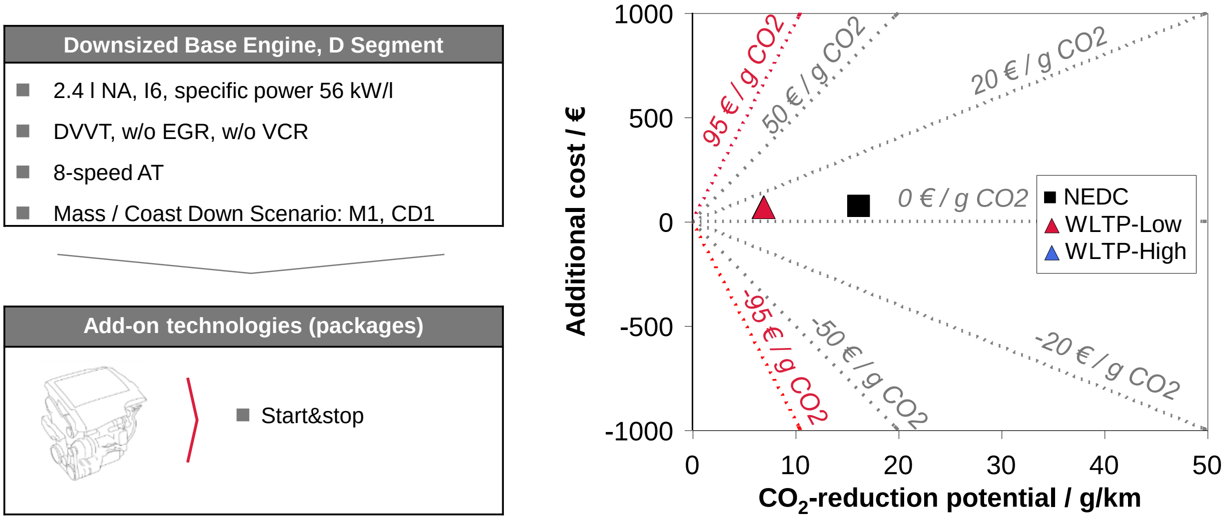


Figure 8.36 Segment D cost analysis for start&stop technology

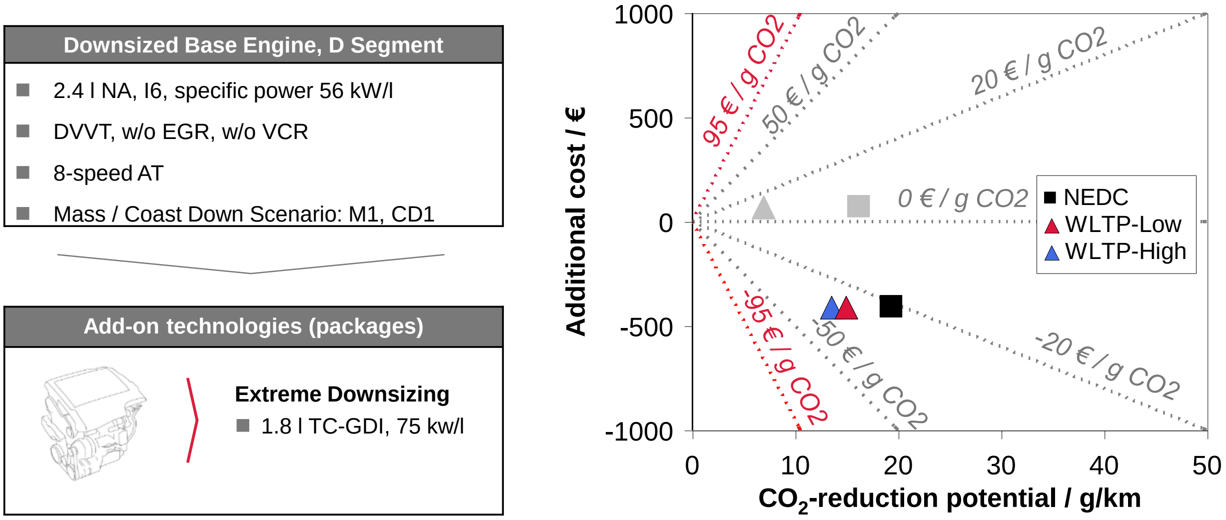


Figure 8.37 Segment D cost analysis for engine downsizing

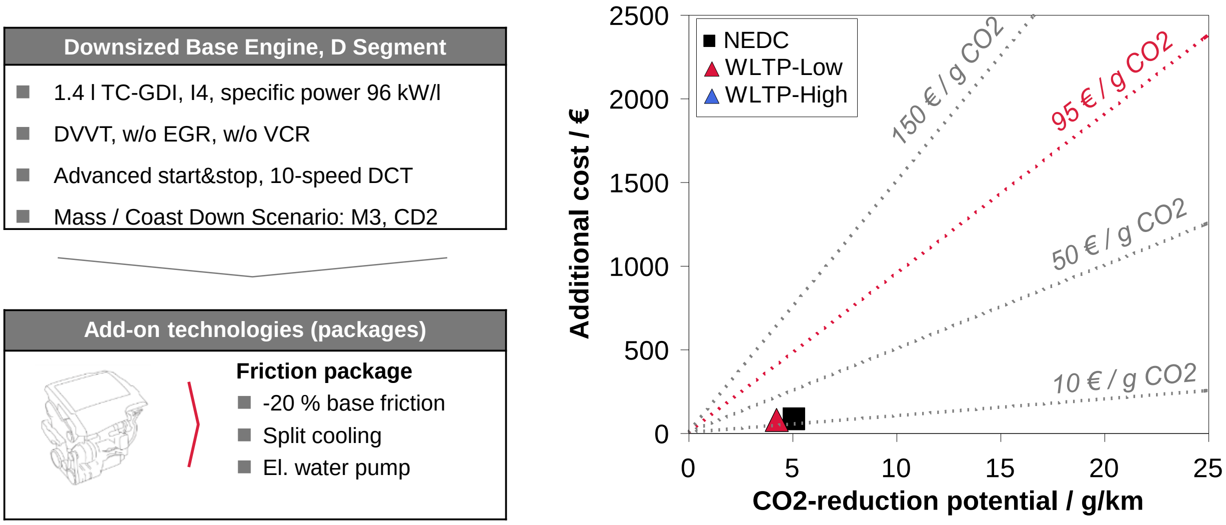


Figure 8.38 Segment D cost analysis for frictional upgrade

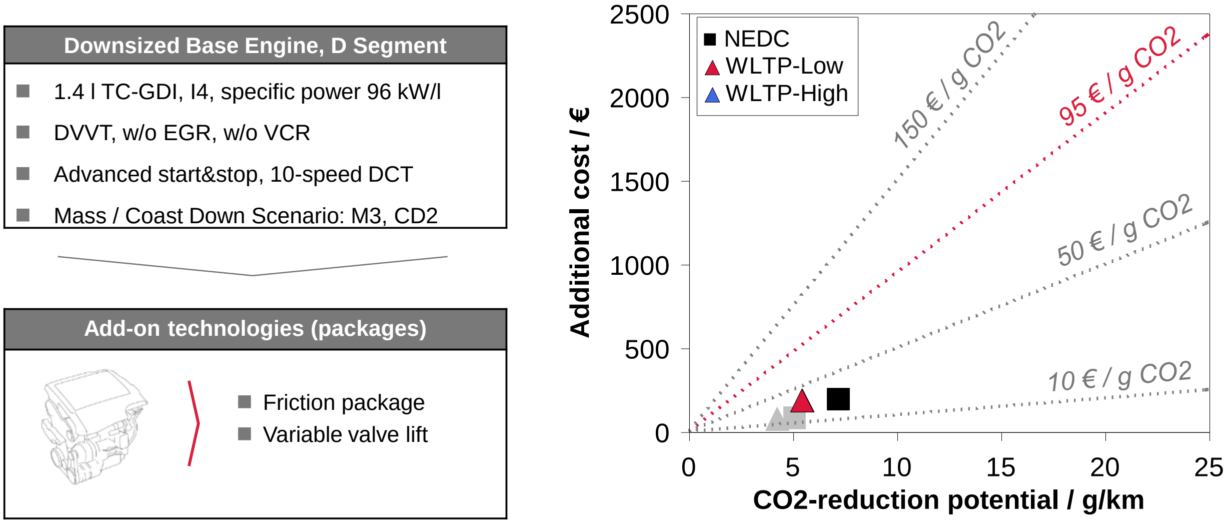


Figure 8.39 Segment D cost analysis for frictional upgrade and variable valve lift

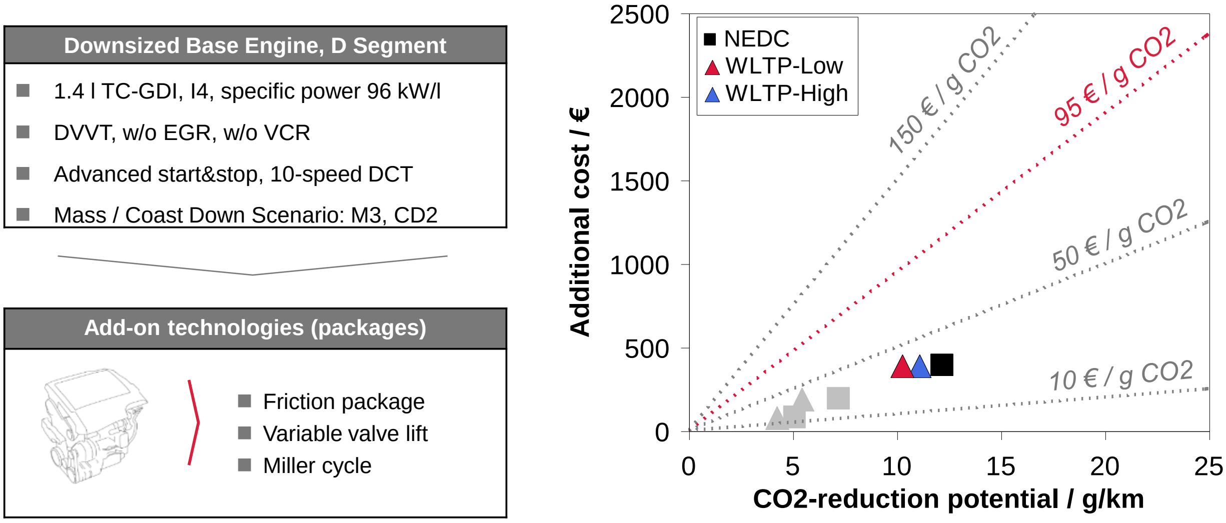


Figure 8.40 Segment D cost analysis for frictional upgrade, variable valve lift and Miller cycle

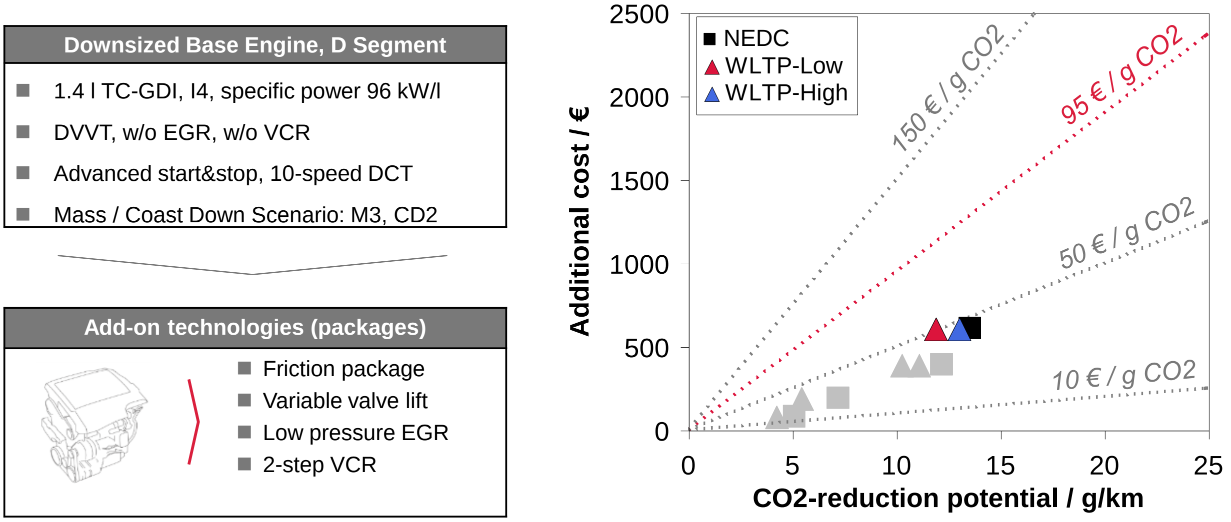


Figure 8.41 Segment D cost analysis for frictional upgrade, variable valve lift, low pressure EGR and 2 step VCR

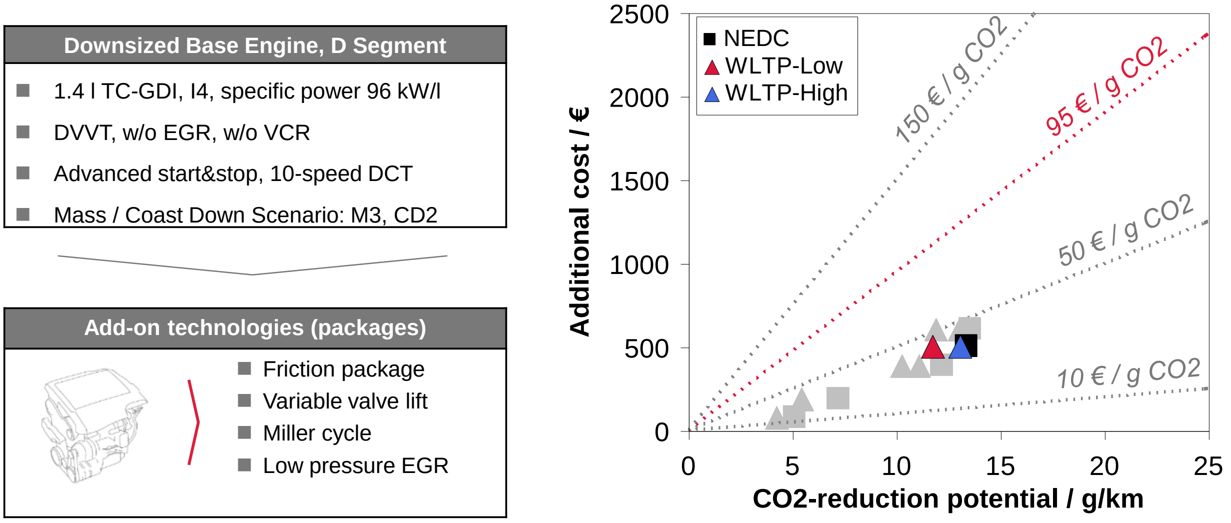


Figure 8.42 Segment D cost analysis for frictional upgrade, variable valve lift, Miller cycle and low pressure EGR

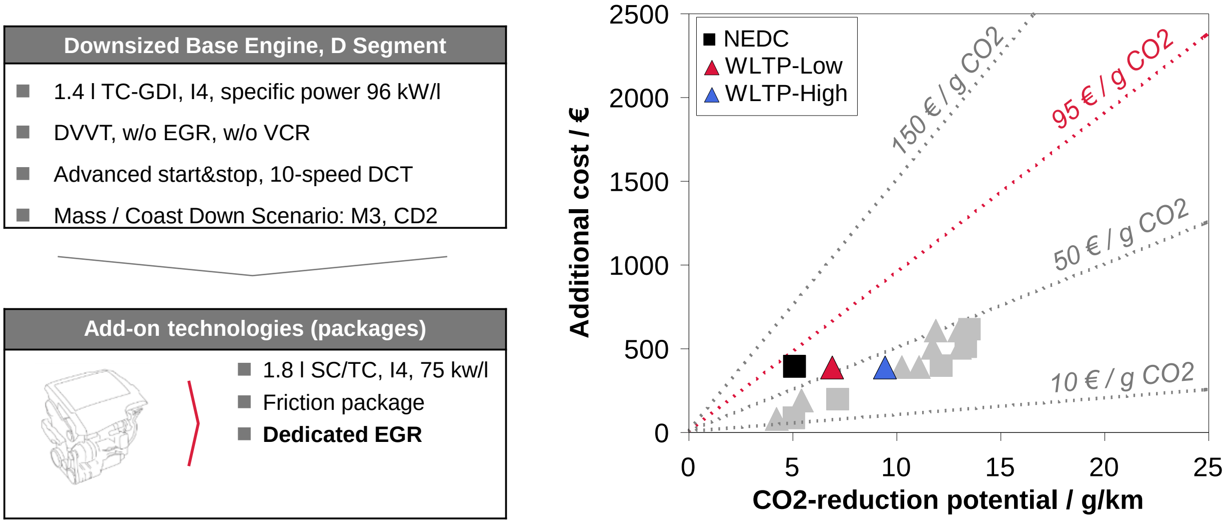


Figure 8.43 Segment D cost analysis for frictional upgrade and for dedicated EGR with a 1.8 l SC/TC I4 engine

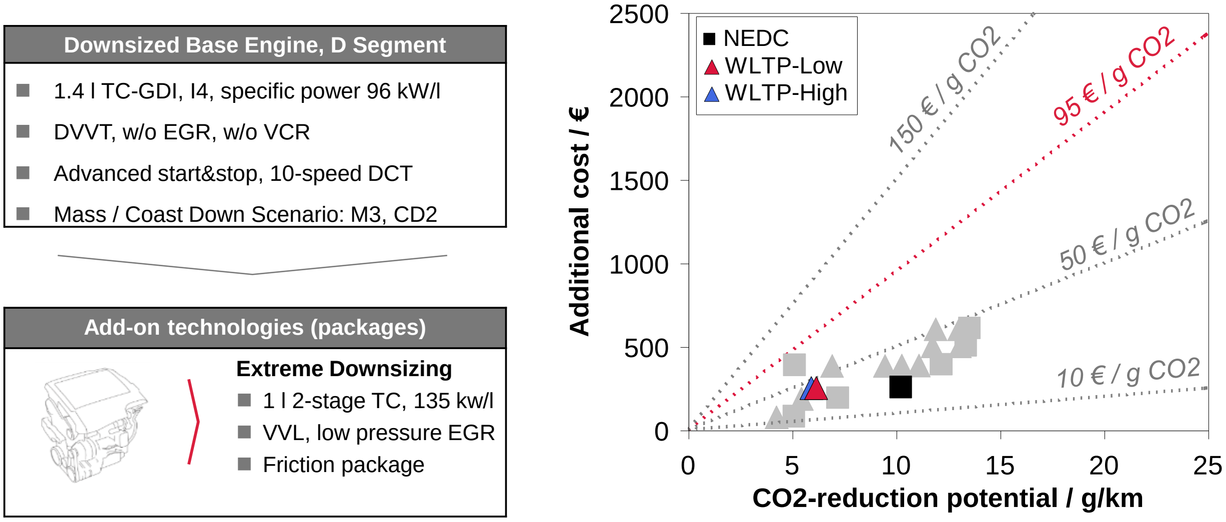


Figure 8.44 Segment D cost analysis for frictional upgrade, variable valve lift, low pressure EGR and further engine downsizing

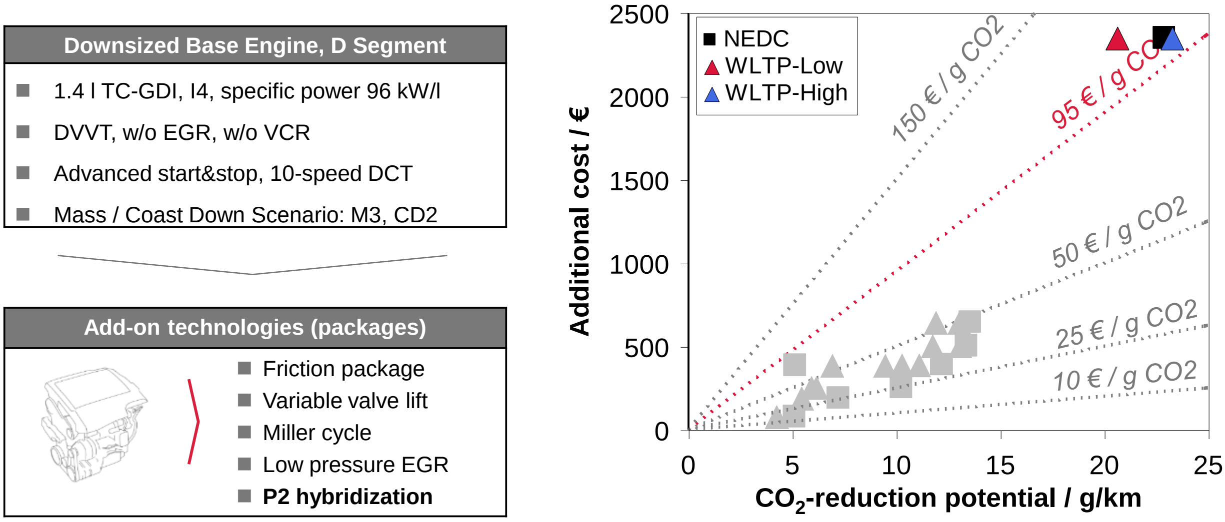


Figure 8.45 Segment D cost analysis for frictional upgrade, variable valve lift, Miller cycle, low pressure EGR and P2 hybridization

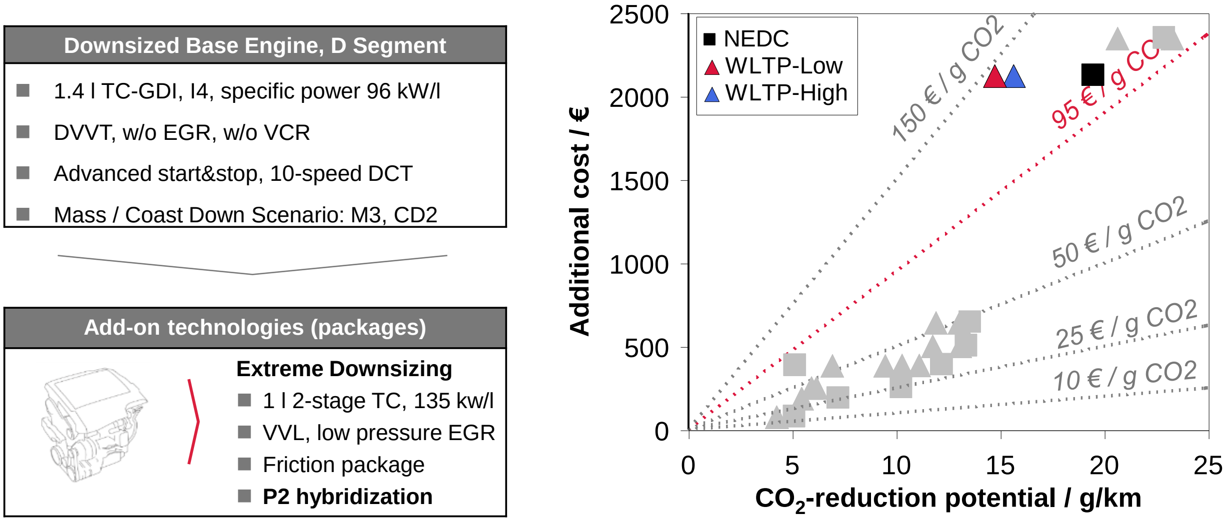


Figure 8.46 Segment D cost analysis for frictional upgrade, variable valve lift, low pressure EGR, further engine downsizing and P2 hybridization

8.2.4 Segment E

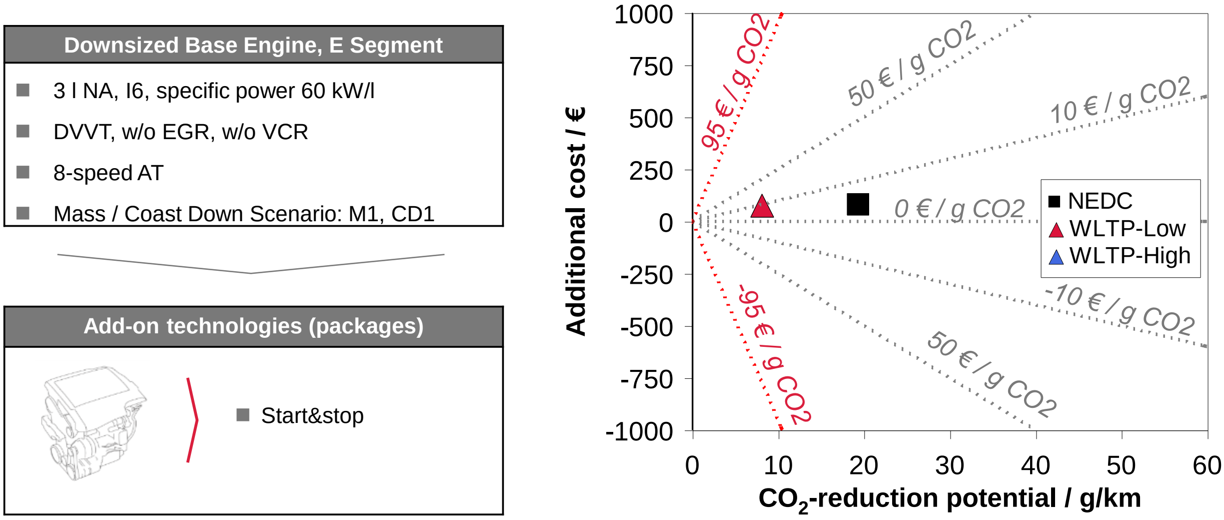


Figure 8.47 Segment E cost analysis for start&stop technology

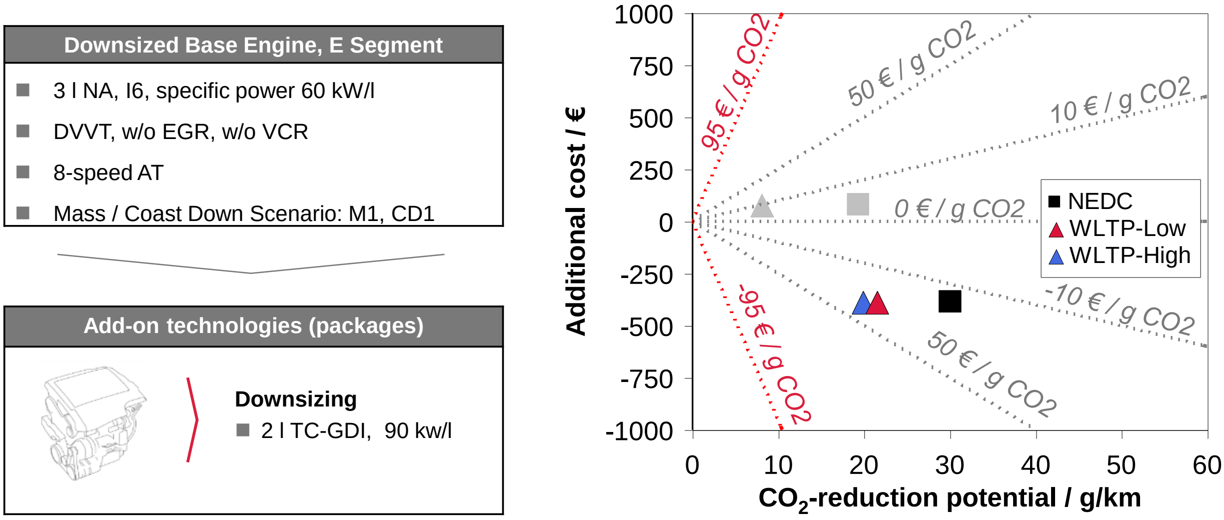


Figure 8.48 Segment E cost analysis for engine downsizing

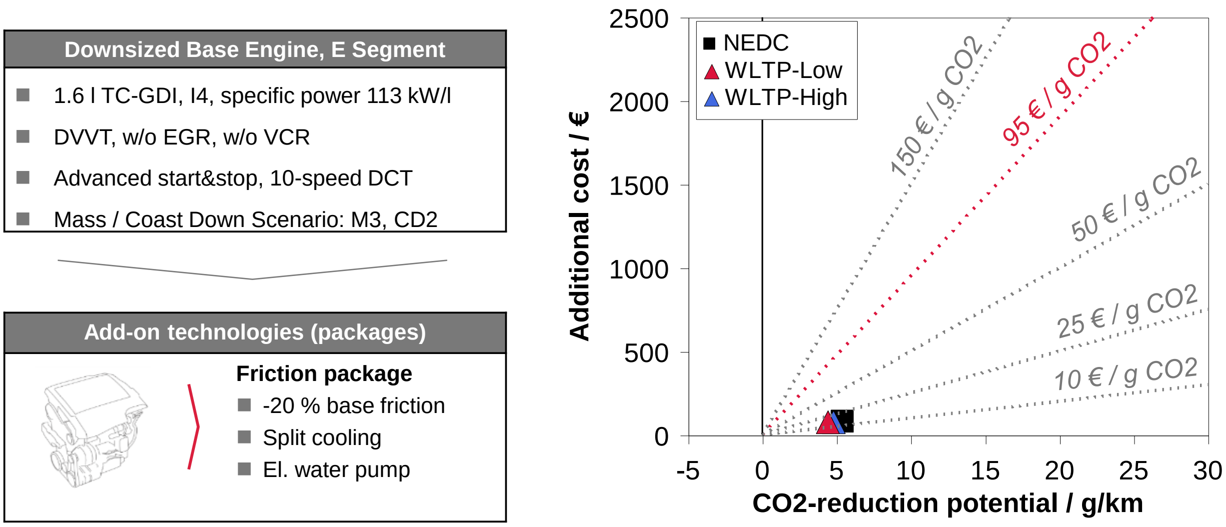


Figure 8.49 Segment E cost analysis for frictional upgrade

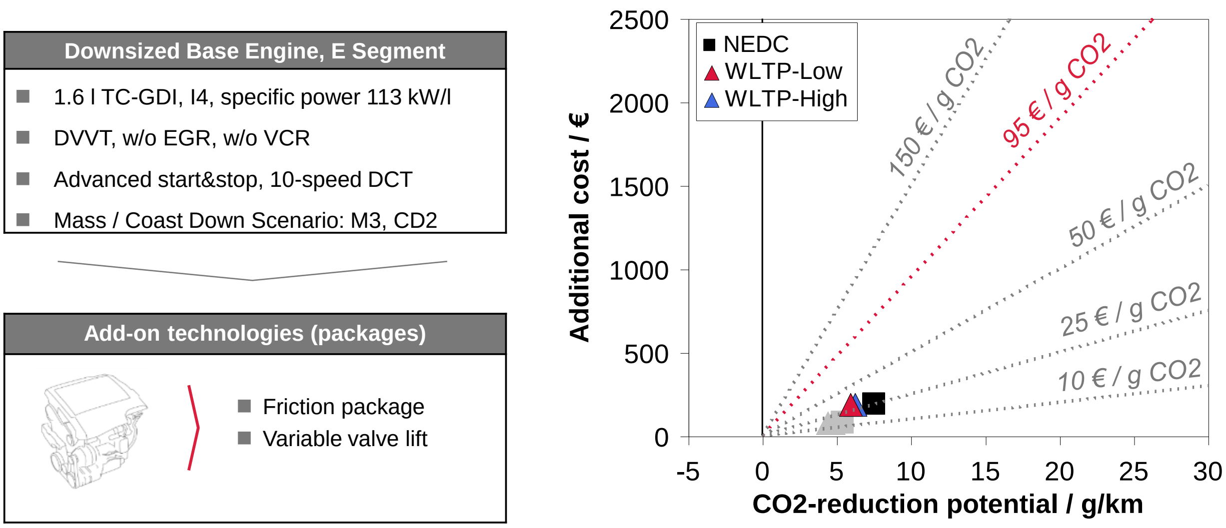


Figure 8.50 Segment E cost analysis for frictional upgrade and variable valve lift

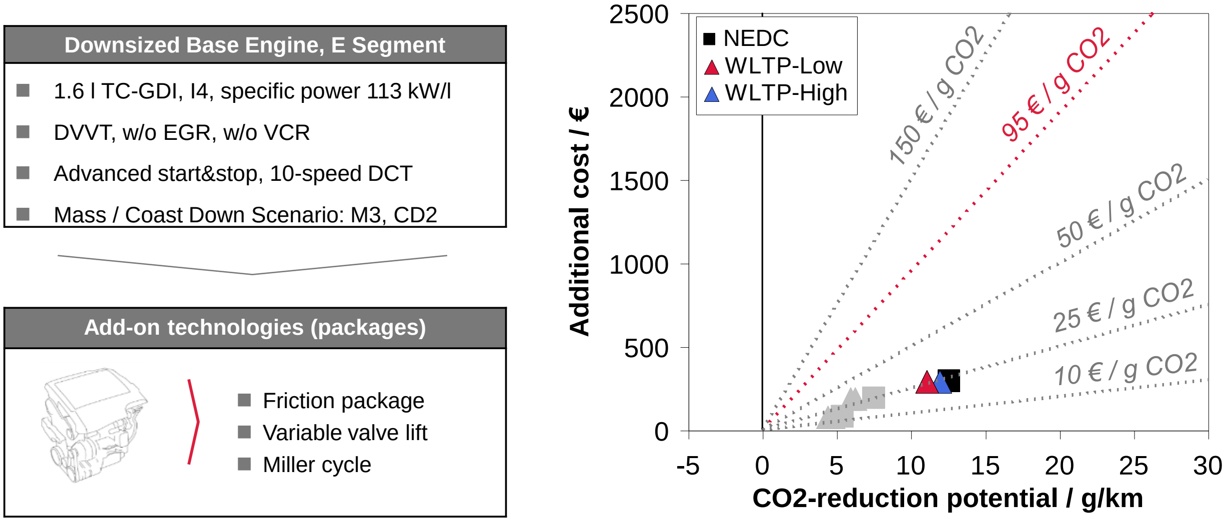


Figure 8.51 Segment E cost analysis for frictional upgrade, variable valve lift and Miller cycle

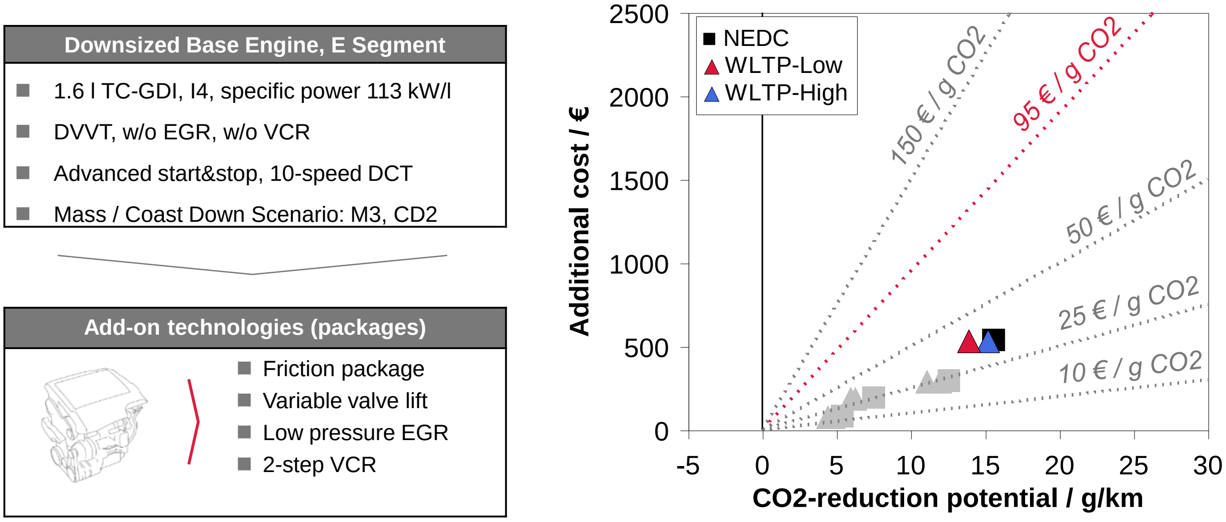


Figure 8.52 Segment E cost analysis for frictional upgrade, variable valve lift, low pressure EGR and 2-step VCR

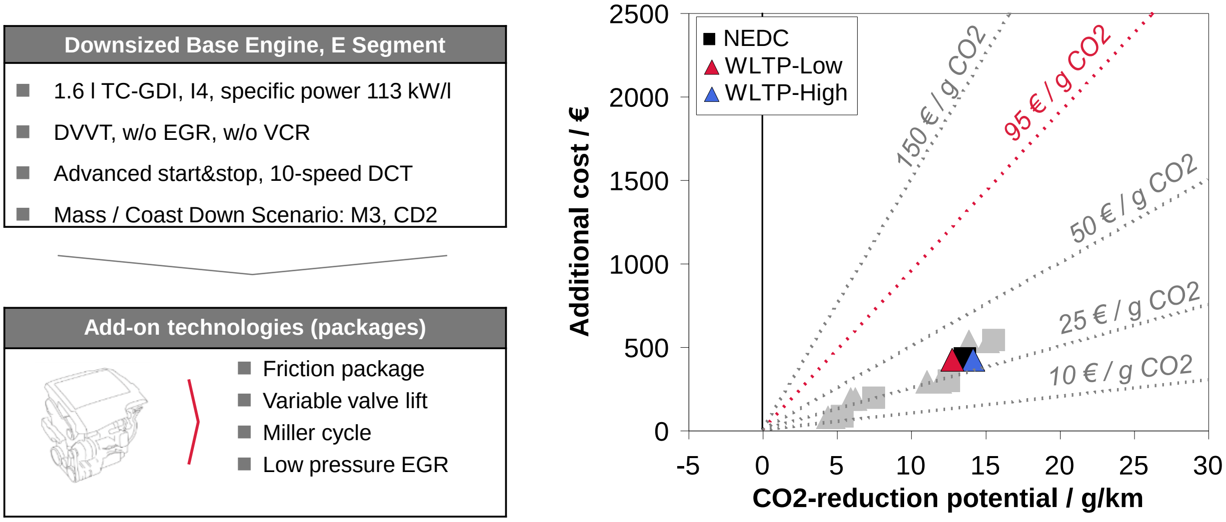


Figure 8.53 Segment E cost analysis for frictional upgrade, variable valve lift, Miller cycle and low pressure EGR

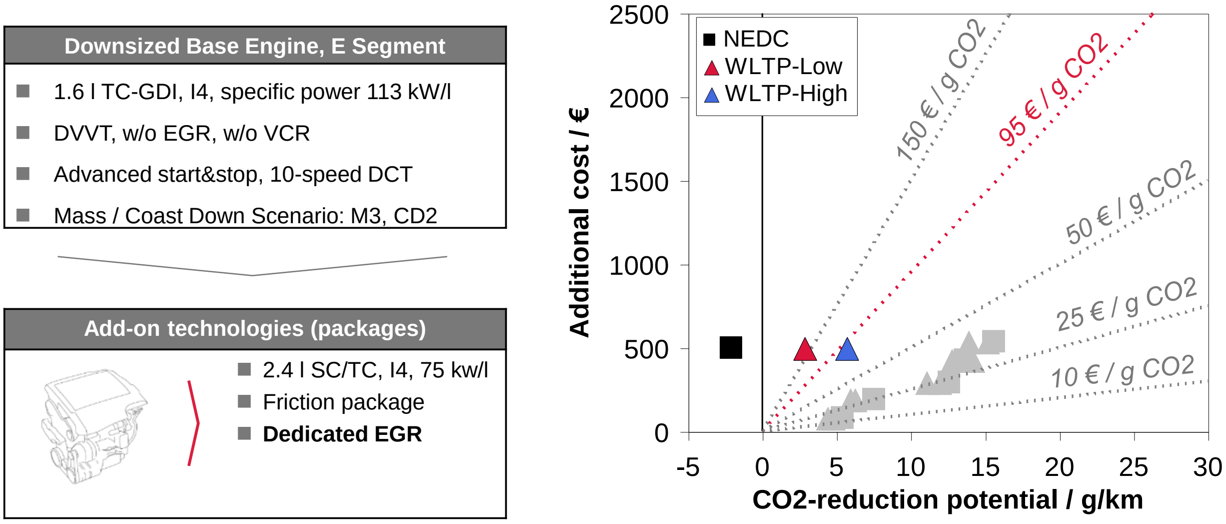


Figure 8.54 Segment E cost analysis for frictional upgrade and for dedicated EGR with 2.4 l SC/TC I4 engine

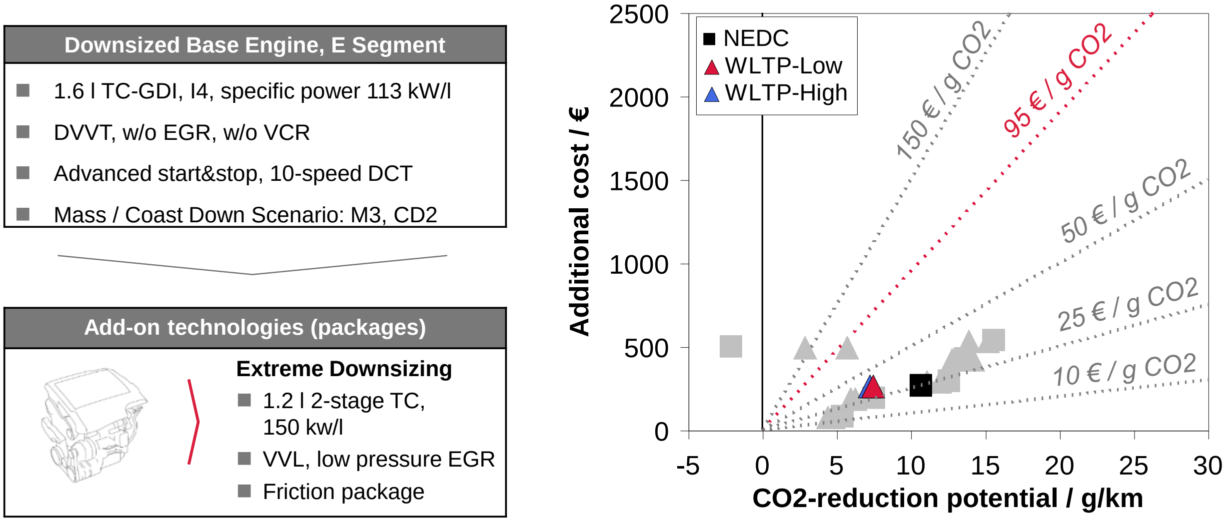


Figure 8.55 Segment E cost analysis for frictional upgrade, variable valve lift, low pressure EGR and further engine downsizing

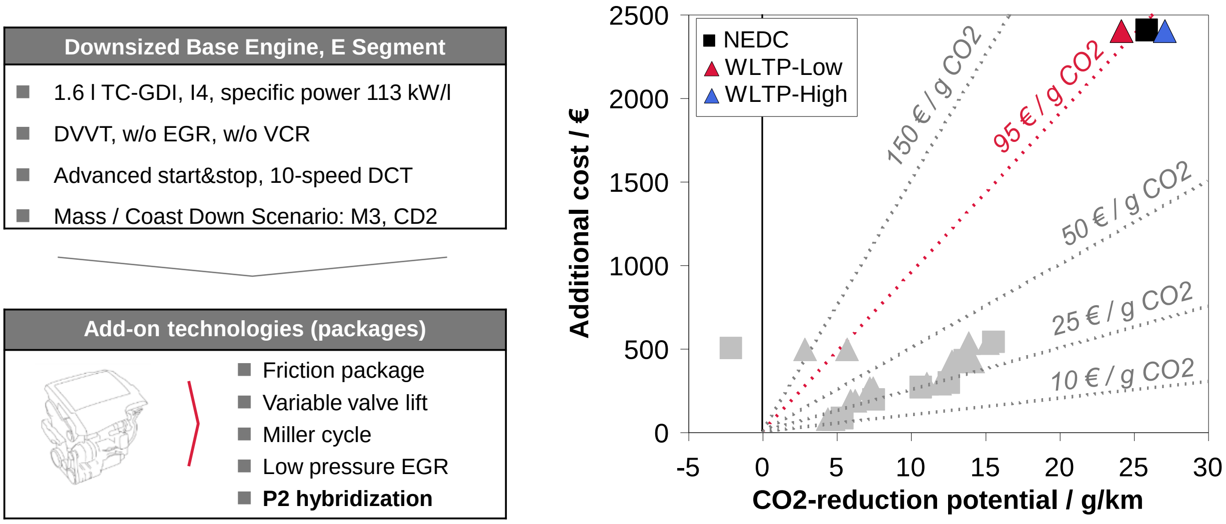


Figure 8.56 Segment E cost analysis for frictional upgrade, variable valve lift, Miller cycle, low pressure EGR and P2 hybridization

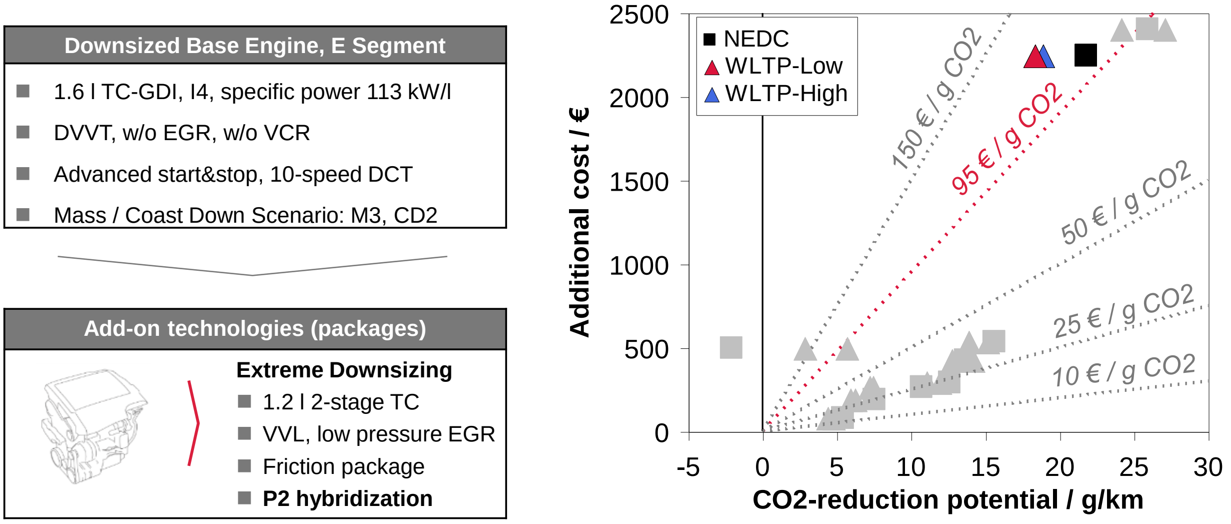


Figure 8.57 Segment E cost analysis for frictional upgrade, variable valve lift, low pressure EGR, further engine downsizing and P2 hybridization