

Evaluation of Valve Train Variability in Diesel Engines

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ABSTRACT

The continuously decreasing emission limits lead to a growing importance of exhaust aftertreatment in Diesel engines. Hence, methods for achieving a rapid catalyst light-off after engine cold start and for maintaining the catalyst temperature during low load operation will become more and more necessary. The present work evaluates several valve timing strategies concerning their ability for doing so. For this purpose, simulations as well as experimental investigations were conducted. A special focus of simulation was on pointing out the relevance of exhaust temperature, mass flow and enthalpy for these thermomanagement tasks. An increase of exhaust temperature is beneficial for both catalyst heat-up and maintaining catalyst temperature. In case of the exhaust mass flow, high values are advantageous only in case of a catalyst heat-up process, while maintaining catalyst temperature is supported by a low mass flow. Another focus of simulation was on analyzing the exhaust temperature gaining effects relevant for the considered alternative valve timings. Simulation results have shown that an early exhaust valve opening, a late intake valve closing and the deactivation of cylinders is of particular interest for exhaust thermomanagement. Besides the validation of simulation results, the main focus of measurements was on analyzing effects which are not covered by simulation (transient operation, cold engine conditions 298 K, emissions, test cycles), which is exemplarily shown here for early exhaust valve opening (EEVO). In addition to the methods based on alternative valve timings, conventional exhaust thermomanagement methods like a retarded combustion were also considered in simulations. Moreover, an electrical heating device was considered experimentally and compared to EEVO in the FTP75 with regard to SULEV30 for passenger cars.

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INTRODUCTION [1]

The market share of Diesel engines in passenger cars in Europe is well above 50%. The main reason for this can be found in the high efficiency and hence low fuel consumption which is also advantageous in terms of a reduction of fleet-average CO₂ emissions. However, not only the targets of fuel consumption and CO₂ but also pollutant regulations have become more and more challenging. Since the conformance to all requirements of the world's strictest emission limits - concerning HC, CO, soot and NO_x - is not conceivable by solely improving the combustion process, the Exhaust Aftertreatment System (EAS) has gained an essential relevance. Most of the aftertreatment components, however, need sufficiently high exhaust temperatures for acceptable conversions rates. Considering this, the Diesel engine induces special demands on aftertreatment thermomanagement since, compared to a conventional stoichiometric spark ignition engine, the exhaust temperatures are generally lower, due to the higher Air Fuel Ratio (AFR). Moreover, the EAS tends to be more complex, as Threeway-Catalyst (TWC) technology cannot be realized in lean operation.

Depending on the engine and EAS configuration, thermomanagement strategies for the following tasks are of special interest:

- Catalyst heat-up after engine cold start
- Maintaining catalyst temperature during low load operation
- Increase of the exhaust temperature for regeneration of a Diesel Particle Filter (DPF)
- Providing cylinder charge conditions appropriate for rich combustion mode as it is necessary for Lean NO_x Trap (LNT) regeneration (purging)

A major demand on the operation strategy is the achievement of high conversion rates in the EAS after engine cold start in order to fulfil lowest emission targets. Since the conversion of the catalyst is strongly dependent on the monolith temperature [2], a rapid heat-up of the catalyst is necessary. High exhaust temperatures are beneficial for this purpose, whereas the influence of high exhaust mass flow or high exhaust enthalpy, respectively, is dependent on the engine operation as well as the catalyst, due to the fact that the mass flow influences for instance the space velocity. Additionally, maintaining heat-up process may not be the same for all EAS components, as the

catalytic coating can support the heat-up process by means of exothermic reactions. In modern Diesel engine EAS concepts, the most upstream catalyst, represented by a Diesel Oxidation Catalyst (DOC) or a LNT, plays a key role in cold start behavior. Therefore, the interaction with these components is primarily considered here.

The focus of this work is the evaluation of valve train variability in terms of exhaust thermomanagement methods. Nevertheless, it is worth mentioning that there are alternative possibilities for the discussed purpose. Naturally, the exhaust temperature can be increased by the adaption of the combustion process in the cylinder, realized by a retarded combustion [3]. Another common solution is the restriction of the aspirated air by means of a throttle in the air path [4]. Furthermore, the implementation of an electric heater in the exhaust duct located upstream the EAS [5,6,7,2] can support the heating. Some of these methods will be used here as reference or for comparison with the evaluated Variable Valve Train (VVT) strategies.

Having mentioned the purpose of VVT for exhaust thermomanagement and alternative exhaust warm up strategies, the application field of VVT in a broader sense will be summarized. VVT is considered here as the addition of any degree of freedom compared to a conventional valve train. Besides the shifting of valve opening and closing via a simple cam phasing unit, full variability concerning valve timing and valve lift is also taken into account. Moreover, switching off the valve actuation allows the deactivation of cylinders. VVT is widely used in Spark Ignition (SI) engines of passenger cars. In SI engines, cam phasers and systems with the possibility of switching between several cam profiles [8,9,10,11] are state-of-the-art, as well as the more rarely used continuous variation of the valve lift. Regarding Diesel engines, only a few series-production engines are instrumented with VVT [12,13], which is also caused by the fact that the efficiency potential of VVT in SI engines is more promising in combination with the quantity control. Nevertheless, research investigations have also been conducted for VVT in Diesel engines for various objectives:

- Reduction of engine-out NO_x emissions [14,15,16,17]
- Generating in-cylinder charge motion [18]
- Optimization of volumetric efficiency at full load [18,16]
- Control of homogeneous charge compression ignition (HCCI) [19]
- Improve cold run stability [18]
- Increasing exhaust temperature [20,21,22,23,24]

The last item above is also the incitement for the underlying investigation, at which the influence on the fuel consumption, also referred to as fuel penalty, and the impact on the emissions is additionally considered.

In order to explain the influencing factors on the exhaust temperature, it is demonstrative to start with an energy balance of the engine and to consider the possibilities to increase exhaust enthalpy:

$$\dot{H}_{\text{exh } 41} = \dot{H}_{\text{in}} + \dot{Q}_{\text{Fuel}} - P_{\text{eff}} - \dot{Q}_{\text{Loss}} \quad (1)$$

The thermodynamic system described by Eq. (1) is graphically shown in Figure 1. $\dot{H}_{\text{exh } 41}$ is defined as the exhaust enthalpy flow downstream turbocharger and upstream the first catalytic component. $\dot{H}_{\text{in}}$ describes the enthalpy flow of the aspirated charge including air and low pressure EGR. Below, the reference temperature for the enthalpy calculation is set to ambient conditions, that is $T_{\text{amb}} = 298 \text{ K}$, so that $\dot{H}_{\text{exh } 41}(298 \text{ K}) = \dot{H}_{\text{in}}(298 \text{ K}) = 0$. $\dot{Q}_{\text{Fuel}}$ stands for the heat release resulting from combustion of the fuel and P_{eff} considers the mechanical power output. Finally, the entire heat loss over the system boundary is defined as $\dot{Q}_{\text{Loss}}$. Having explained $\dot{H}_{\text{exh } 41}$ in Eq. (1) it is apparent how the enthalpy flow can be influenced. Since the effective efficiency, defined by

$$\eta_e = \frac{P_{\text{eff}}}{\dot{Q}_{\text{Fuel}}}, \quad (2)$$

is dependent on these parameters, several strategies more or less influence the specific fuel consumption. An efficiency-neutral increase of exhaust enthalpy could be gained by reducing heat losses, which is only possible to a certain extent. For a constant power output P_{eff} and if the change in $\dot{H}_{\text{in}}$ is neglected, the exhaust enthalpy $\dot{H}_{\text{exh } 41}$ can additionally be raised by an increase of $\dot{Q}_{\text{Fuel}}$, which results in a decrease of efficiency. What has to be taken into account is that the enthalpy flow related to ambient conditions consists of mass flow, specific heat and temperature, as it can be seen in Eq. (3),

$$\dot{H}_{\text{exh } 41} = \dot{m}_{\text{exh}} \cdot c_p \cdot (T_{41} - T_{\text{amb}}). \quad (3)$$

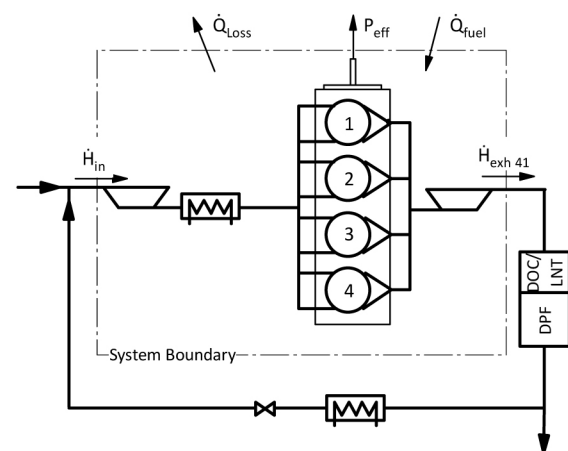


Figure 1. Energy balance of the entire engine

Generally, it can be stated that the EAS works optimally in a defined temperature window. Whether this temperature window is ideally reached by high temperature or high enthalpy of the exhaust gas, is not only dependent on operating conditions but also on EAS components. This is explained with an example displayed in Figure 2. The graphic shows the temperature response in the EAS after a sudden load step, and compares a basis engine operation to one with cylinder deactivation (CDA). Although the exhaust temperature rises with CDA, the temperature response of the EAS system with CDA is significantly delayed. This is the result of the reduced exhaust mass flow in CDA-mode and a catalyst which is far downstream the turbo charger,

as it is e.g. the case for an underfloor SCR system. If the position of the EAS system is close to the engine (closed couple), the disadvantage can be compensated. This leads to the assumption that for heat-up processes, exhaust enthalpy is more important than temperature, especially when the EAS system is distant from the engine.

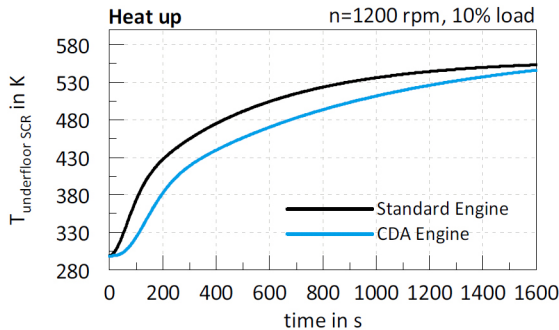


Figure 2. Heating up the exhaust aftertreatment system after engine start followed by a constant load point.

When it comes to maintaining the temperature of the catalyst substrate which is hotter than the exhaust gas, matters are distinct as it is shown in Figure 3. A high mass flow improves the heat transfer and thus the catalyst cools rapidly. Therefore, for maintaining catalyst temperature in low load points CDA is beneficial, although the enthalpy flow of the exhaust gas is not increased due to the mass flow reduction. Hence, mainly the high exhaust temperature is important in this operating regime. The interaction between exhaust enthalpy, exhaust temperature and the catalyst monolith temperature can be found in [23].

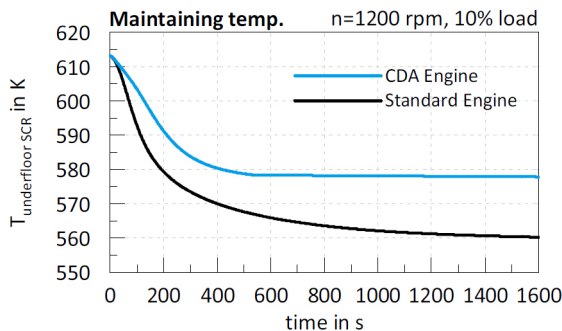


Figure 3. Maintaining temperature of the exhaust aftertreatment system after a step change in load down to 10%.

Having pointed out the different influences of exhaust enthalpy and temperature, fundamental thermomanagement methods are now distinguished. Besides the increase of both enthalpy and temperature, realized by

- Increase of injected fuel mass,
- Adaption of net rate of heat release,
- Installing a heating device and
- Reduction of effective expansion,

solely exhaust temperature can be increased by

- Increasing temperature of aspirated cylinder charge and by
- Declining heat capacity of aspirated cylinder charge.

The first item is already explained above. An adaption of the net rate of heat release can be realized by a retarded fuel injection. Additionally, a heating device, such as an electric heater or a burner can be installed, which is especially beneficial when it comes to heat losses, since the heating unit can be positioned directly upstream the catalyst. Exhaust temperature and enthalpy can also be raised by a reduction of the effective expansion. On the one hand, the geometrical compression ratio can be lowered, and on the other hand, an adaption of exhaust valve timing can result in a declined expansion, which is of particular interest here. Increasing the temperature of the aspirated cylinder charge generally increases the exhaust temperature, and if boost pressure is not raised this results in a smaller mass of cylinder charge and hence lower heat capacity. Decreasing heat capacity cannot only be realized by an increased charge temperature, but also by reduction of pressure and the deactivation of cylinders when the entire engine is considered.

Concerning VVT strategies for exhaust thermomanagement, the mentioned strategies can thus be classified in two major groups:

1. Methods that increase exhaust temperature without reduction of aspirated cylinder charge and hence constant exhaust mass flow. This results in an increase of exhaust enthalpy in the same way as temperature. Naturally, these methods involve either increased fuel consumption or a decrease of power output.
2. On the opposite, methods are based on a reduction of cylinder mass and thus lowered heat capacity. Exhaust temperature can be raised without the rise of fuel energy. However, the exhaust enthalpy flow cannot be increased in the same way.

METHODS

The evaluation of the valve train variability focusing on exhaust thermomanagement is represented in terms of simulation and measurement on an engine test bed. The use of simulation allows the consideration of a wide range of virtual hardware and an analysis of the influence of single parameters (e.g. residual gas concentration in cylinders or wall heat flow in exhaust ports), whereas the measurement enables for instance the evaluation of the actual impact on the emissions, which is demanding to predict accurately in a simulation model for all components.

Simulation

The simulation model can be divided in the engine model and a simplified catalyst model, which is not further explained here but can be found in [1].

The engine was modelled with the software package BOOST from AVL. The model is characterized by the one-dimensional consideration of gas flow pipes and ports of the intake and exhaust system. The cylinder and plenums are considered as zero-dimensional, depending only on time or the crank angle respectively. An explicit discussion of the underlying (differential) equations of pipes, plena and turbo charger can be found in [1]. The cylinder was described by the first law of thermodynamics, the conservation of mass and the equation of state for ideal gases. Therefore, the combustion was modelled by the definition of the Rate Of Heat Release (ROHR), derived from measurement data. The cylinder wall

heat flow was calculated based on a Newton's approach for convective heat flow and a heat transfer coefficient based on Woschni/Huber [25,26] was used. The wall heat flow in the intake and exhaust ports was modelled based on Zapf [27] and the turbo charger was modelled by using swallowing capacity and efficiency maps for compressor and turbine (full model).

Following VVT strategies are investigated by means of simulation:

- Early Exhaust Valve Opening (EEVO)
- Late Intake Valve Opening (LIVO)
- Late Intake Valve Closing (LIVC)
- Internal Exhaust Gas Recirculation
- Cylinder Deactivation (CDA)

Measurement

The measurements were executed on a transient engine test bed. The investigated passenger car common rail Diesel engine, which was used on the test bed, belongs to the 2 liter class and is schematically shown in Figure 4. It is equipped with a hybrid EGR system with an uncooled high pressure Exhaust Gas Recirculation (EGR) system for low load points and cooled low pressure EGR up to 3000 rpm and a BMEP of 16 bar. The valvetrain consists of Double Overhead Camshafts (DOHC), roller-type finger followers, hydraulic valve-clearance compensation and four-valve technology, as it is standard in modern Diesel engines. The engine is charged by a single-stage turbocharger with Variable Nozzle Turbine (VNT, VTG). The adapted EAS is characterized by a LNT that is located most upstream in the exhaust path, a DPF and an underfloor Selective Catalytic Reduction (SCR) system. The engine calibration allows to reach Euro6 NO_x -limits without a NO_x -EAS. However, the strictest emission legislations, as for example the US LEV3 SULEV30 legislation, are still challenging, especially when the emission limits have to be fulfilled with aged EAS components [2,28,29,30].

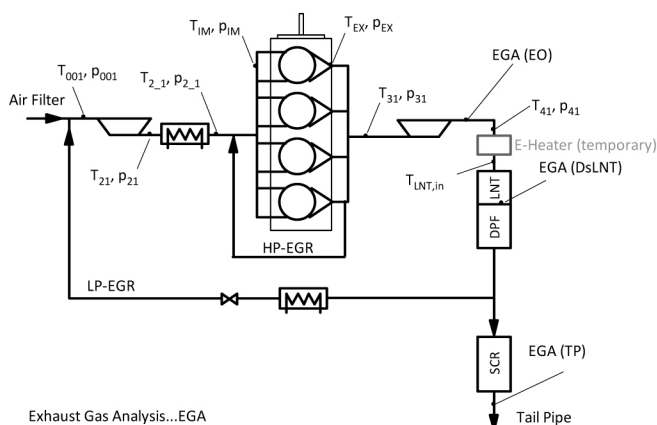


Figure 4. Engine test bed configuration

Measurement results of EEVO aiming to improve the cold start pollutant behavior are shown here. For this purpose, an exhaust cam shaft was designed with an alternative cam profile. One exhaust valve per cylinder opens regularly, which is assigned as *Basis*, whereas the second exhaust cam profile was adapted in a way that the exhaust valve opens 95°CA earlier, but closes at the same time as the *Basis*.

This configuration is referred to as *EEVO1*. In simulation, the opening of only one valve resulted to be beneficial due to lower exhaust port wall heat losses [1], which can be explained by the reduced overall port surface. Moreover, such a system can be used in combination with a rotated valve layout, which allows the variability of intake and exhaust valve with only one camshaft [12]. A second valve lift curve, named *EEVO2*, is realized by changing the installed position of the cam *EEVO1*. This results in an Exhaust Valve Opening (EVO) that is another 13 °CA earlier than *EEVO1*, but also leads to a shifted Exhaust Valve Closing (EVC). Due to the advanced closing, residual-gas compression is to be expected. The valve lift of the exhaust valves can be seen in Figure 5 and EVO, as well as EVC is summed up in Table 1. By means of measurements, the results of EEVO are compared with a configuration including an E-heater, which is positioned upstream the LNT as can be seen in Figure 4.

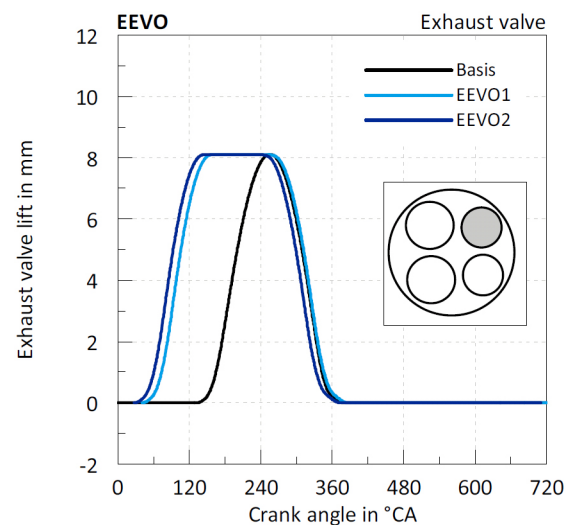


Figure 5. Valve lift curves of the Basis and the advanced opening of EEVO1 and EEVO2 realized on one exhaust valve only (colored in gray)

Table 1. Valve train specifications

Number of Valves per Cylinder	2x2
Exhaust Valve Open (Basis)	152° ATDC @ 0.25 mm lift
Exhaust Valve Open (EEVO1)	57° ATDC @ 0.25 mm lift
Exhaust Valve Open (EEVO2)	44° ATDC @ 0.25 mm lift
Exhaust Valve Close (Basis)	0° BTDC @ 0.25 mm lift
Exhaust Valve Close (EEVO1)	0° BTDC @ 0.25 mm lift
Exhaust Valve Close (EEVO2)	13° BTDC @ 0.25 mm lift

RESULTS AND DISCUSSIONS

Overview by Means of Simulation

In this section, an overview of the VVT thermomanagement strategies is given and the valve lift curves are schematically displayed in Figure 6. The gray arrows exemplarily show the advance of EVO or the retardation of IVO and IVC for the investigated strategies. A more detailed look is then taken on EEVO and cylinder deactivation. For the purpose of a general comparison, the BSFC is shown as a function of the reached temperature of the exhaust gas turbocharger turbine, T_{41} , for a stationary load point. Additionally, the

exhaust enthalpy flow is investigated. The results for an engine speed of 1500 rpm and a BMEP of 2 bar are displayed in Figure 7. By regarding Figure 7, where the extent of the alternative valve timing $\Delta\phi$ is varied, it can be seen immediately that some of the strategies lead to an increase of temperature without an increase in enthalpy. This effect is explainable by the reduction of mass flow. Variable intake valve timings enable the increase of exhaust temperature by a reduction of cylinder charge heat capacity related to fuel energy. Two possible strategies, namely a Late Intake Valve Opening (LIVO) and Late Intake Valve Closing (LIVC), are considered and compared to the reduction of mass of cylinder charge realized with a throttle in the intake manifold. LIVO, LIVC and throttling all reduce exhaust mass flow and hence exhaust enthalpy cannot be raised in the same way as temperature. However, LIVC shows good results concerning BSFC. Considering these strategies under the boundary of same exhaust temperatures, an analysis of the cylinder pressure curves during the gas exchange phase has shown that LIVO results in a significant drop of cylinder pressure in the intake period [24]. This resembles an evacuation process and induces high pumping work which even exceeds the one caused by throttling. Compared to these, LIVC does not suffer from an extremely negative Pumping Mean Effective Pressure (PMEP) which explains the relatively low fuel penalty.

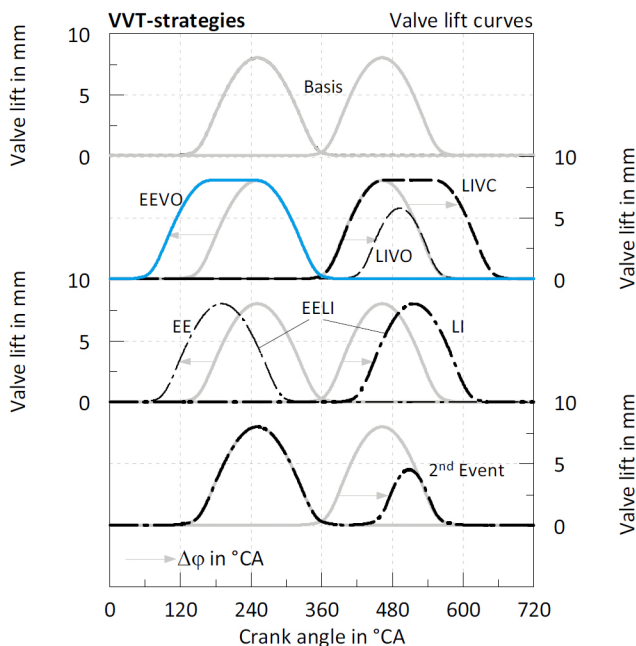


Figure 6. VVT strategies

When the exhaust enthalpy flow in Figure 7 is regarded, it can be seen that only EEVO leads to a significant increase. For throttling and LIVC, the reduction of mass flow even overcompensates the increase of temperature, leading to a decrease of exhaust enthalpy. This confirms that methods achieving an increase of exhaust temperature by a reduction of Charge Fuel Ratio (CFR) are not adequate when an increase of exhaust enthalpy is desired. The CFR is defined as

$$\text{CFR} = \frac{m_{\text{SHP}}}{m_{\text{Fuel}}} = \frac{m_{\text{Air}} + m_{\text{EGR,ext}} + m_{\text{EGR,int}}}{m_{\text{Fuel}}}, \quad (4)$$

where m_{SHP} is the mass of cylinder charge at the start of the high pressure loop, including air mass, internal and external EGR mass.

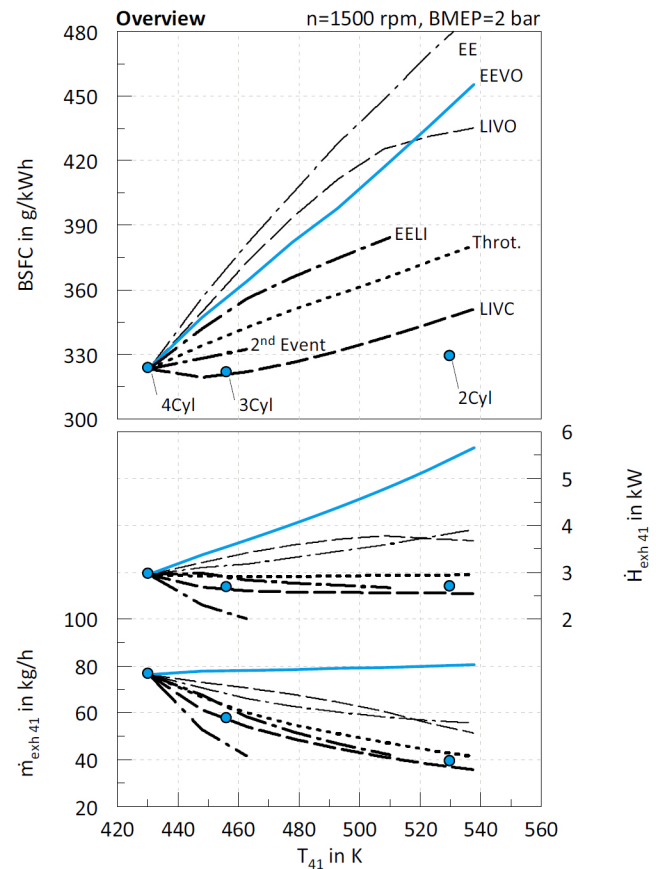


Figure 7. Overview of the impact of VVT strategies on BSFC and exhaust enthalpy over temperature downstream turbocharger. This is realized by increasing the extent of the alternative valve timing $\Delta\phi$ or by CDA.

It might be assumed that an increase of exhaust temperature is ideally realized by LIVC, since the BSFC does not increase tremendously.

However, a big drawback has to be mentioned. The pressure and temperature at the start of combustion drop significantly for a retarded closing of the inlet valves. This leads to critical conditions in terms of ignition and combustion stability, in particular during cold start where the cylinder walls are cold. The low final compression pressure also leads to problematic THC raw emissions.

A further method of reducing CFR, that increases the charge temperature at EVO, can be achieved by replacing cooled low pressure EGR by hot internal EGR. A theoretical investigation of the influence of CFR on the temperature can be found in [1]. Cylinder charge density can be lowered with internal EGR and hence mass and CFR can be reduced. For realization, three different VVT strategies are considered. Firstly, internal EGR can be realized by an advanced closing of the exhaust valves by phasing (Early Exhaust...EE). Secondly, the intake valves can additionally be opened later by phasing (Early Exhaust Late Intake...EELI), which is based on the same principle. It should be noted, that these strategies interfere with an early exhaust valve opening and also lead to residual gas compression. Thirdly, a second valve lift of the exhaust valve during the intake stroke, compare to [38], leads to internal EGR (2nd Event).