

Investigations on a Catalyst Heating Strategy by Variable Valve Train for SI Engines

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ABSTRACT

The objective of this investigation was to evaluate the effects of a variable intake and exhaust valve timing in terms of opening, closing, opening duration, lift curve and number of active valves per pair on a four cylinder direct-injecting SI engine for the catalyst heating idling phase at the beginning of an NEDC emission test procedure. The first step evaluated the engine behavior at a reference point of operation. Its parameters in valve timing were adjusted to match the valve timing of the base production engine. The second step investigated the effects of an earlier exhaust valve opening while the exhaust valve closing time was kept and the exhaust valve opening duration was extended. The third step was to answer the question for the optimum number of exhaust valves in order to minimize the wall heat losses inside the cylinder head. The optimum 3V exhaust valve timing has been defined as the basis for exhaust valve timing for steps four and five. The fourth step contained the variation of intake valve opening and closing. The group of selected optimum valve timing / ignition timing combinations mainly consists of late intake valve opening and decreased intake valve opening durations. The fifth and final step was to evaluate the optimum number of intake valves in order to find out whether it makes sense to add another source of charge motion. It can be stated that a combination of both a late intake valve opening and intake valve deactivation must be excluded for this evaluated catalyst heating point of operation. From this point of view, two alternative VVT strategies with comparable potential can be seen: either late intake valve opening with decreased intake valve opening duration, or intake valve deactivation with standard intake valve opening duration and a slight valve overlap. Both of these intake valve strategies have at least one thing in common: a valve train with the digital ability to change the cams is necessary for their realization. Their general potential can be numbered in a 3-13 % increase of exhaust gas and catalyst temperature, a 5-30 % decrease of gaseous emissions output, and a 80-95% decrease of FSN. It also has the potential of a 5-25 % decrease in related standard deviation of imep.

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1. INTRODUCTION

Advanced SI engines for passenger cars are characterized by direct fuel injection, downsizing and turbo charging. Additionally, valve trains show a growing flexibility for the improvement of fuel consumption, emissions behavior and

driving performance. With the background of international emissions regulations becoming continuously more stringent ([Table 1-A](#)), it is necessary to investigate the additional potential of VVT features for the enhancement of items such as the catalyst heating phase at the beginning of a emission test procedure.

Table 1-A. Recent and future emissions regulations for EU [1]

Legislation level	Euro 5	Euro 5+	Euro 6
Month / Year	09 / 2009	09 / 2011	09 / 2014
CO [g/km]	1,000	1,000	1,000
THC [g/km]	0,100	0,100	0,100
NMHC [g/km]	0,068	0,068	0,068
NO _x [g/km]	0,060	0,060	0,060
PM [g/km]	0,005	0,0045	0,0045
PN [1/km]	-	-	tbd until 09/2014

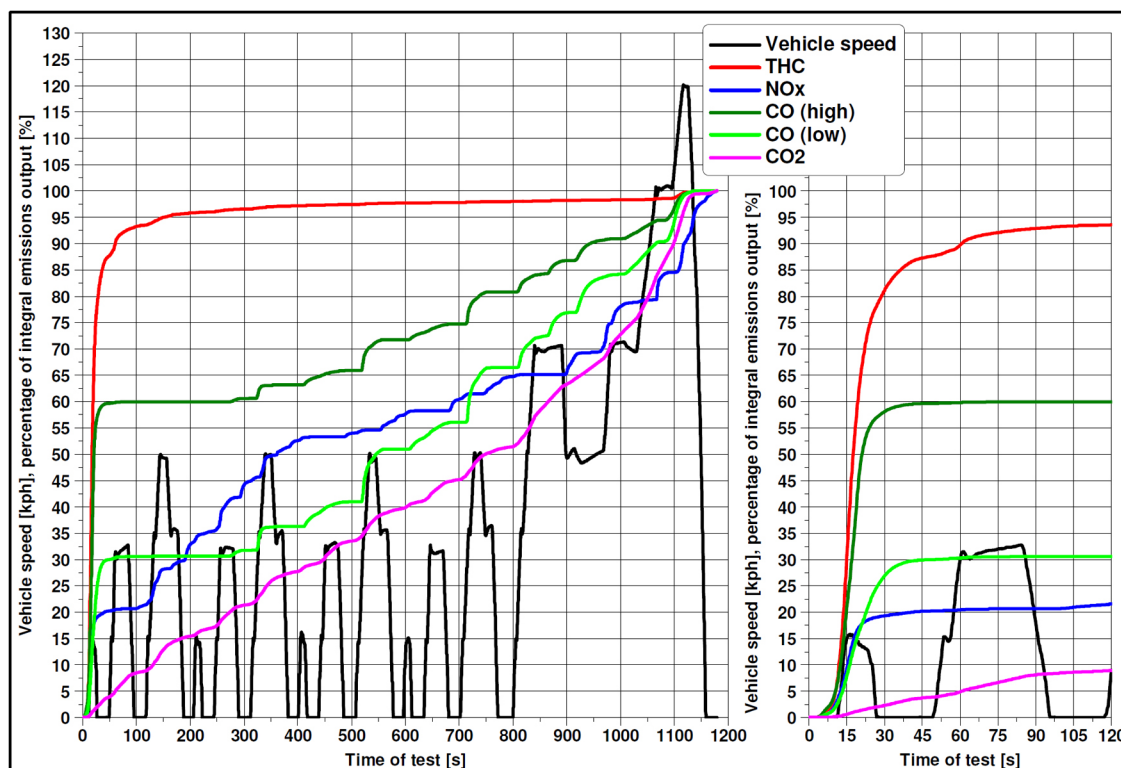


Figure 1-1. Integral emissions output over entire NEDC and during start & catalyst heating phase

The catalyst heating phase is of high importance for the entire emissions output during the test procedure (Figure 1-1) and belongs to the points of operation with the most critical thermodynamic conditions for mixture formation, ignition, combustion and gas exchange. In terms of the equipment of the engine with an appropriate subsystem, there are three major alternative technologies: an electrical heating inside or outside of the catalyst body, a secondary air system with additional air access to the exhaust valve ducts, or a multiple DI injection internal lean-burn strategy. The multiple DI injection internal lean-burn strategy offers the best compromise of the cost efforts to the fuel economy and emissions behavior of the engine. Advanced DI gasoline engines usually offer double or triple multiple injection strategies which can also be applied to other points of operation.

The motivation for investigating began with the need to understand the potential of variable valve trains out of the steady-state operation at standard temperature operation conditions (2). Among these sensible points of operation are engine start, idling, idling at low engine temperatures, engine warm-up and the catalyst heating phase. The targets for these investigations are obvious: the exhaust gas temperature and the exhaust gas mass flow are to be increased in order to reduce the duration of catalyst heating phase at the beginning of the test cycle. The combustion stability must be improved in relation to engine roughness and comfort, and the fuel economy and emissions output must be enhanced, especially for THC and CO (Figure 1). It is the objective for the catalyst

heating phase to conclude which SI valve train configurations offer the best ratio of technological and cost efforts versus engine behavior enhancements.

2. TEST EQUIPMENT

The test engine is based on a series production downsizing four stroke four cylinder gasoline engine with turbocharging (Table 2-A). The engine was equipped with a direct injection system with lateral injector position, a multi-hole spray pattern and a peak injection pressure of 130 bars. In order to match the efficiency targets in connection with advanced valve train strategies, the geometrical compression ratio was significantly increased. The test bench connected the engine to an independent cooling water and intake air conditioning. The test bench media conditioning equipment was able to continuously keep both temperatures at NEDC start temperature of 21°C. The measurement equipment was set for the acquisition of gaseous emissions, Filter Smoke Number (FSN), fuel mass flow, pressure and temperature data based on standard measurement techniques.

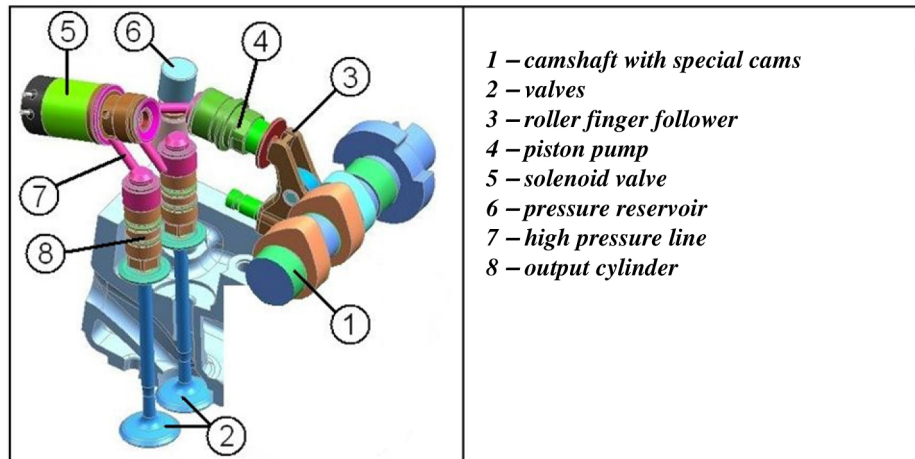


Figure 2-1. Layout of "UniAir" variable valve train system (3)

Table 2-A. Test engine data

Displaced volume	1390 ccm
Number of cylinders	4
Stroke	75.6 mm
Bore	76.5 mm
Connecting Rod	144 mm
Geometric compression ratio	13:1
Number of Valves	4
Injection system	lateral multi-hole DI
Boosting technology	Turbocharger (waste gate)

The major advanced engine feature is a complete electro-hydraulic variable valve train based on the FIAT / Schaeffler "UniAir" technology (3; 4). Illustrations of the major design and features are shown in Figure 2-1 and Figure 2-2.

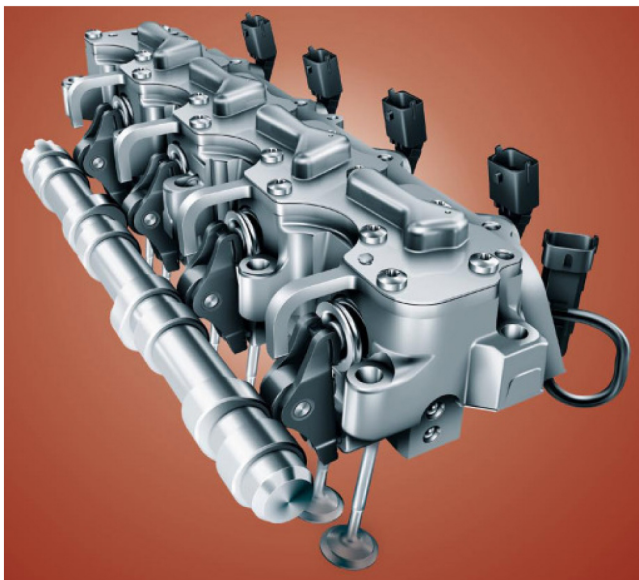


Figure 2-2. design of "UniAir" variable intake valve train system for a four cylinder SI production engine (4)

In addition to the functional principle in Figure 2-1 and Figure 2-2, the valve train of the test engine has been equipped with an actuation which is individual to each of the 16 valves to investigate valve or cylinder deactivation strategies. Extensions within the valve train equipment are the intake camshaft and exhaust camshaft phasers with an operational range of 60°CA each. The operational features of this variable valve train are shown in Figure 2-3, Figure 2-4, Figure 2-5, Figure 2-6.

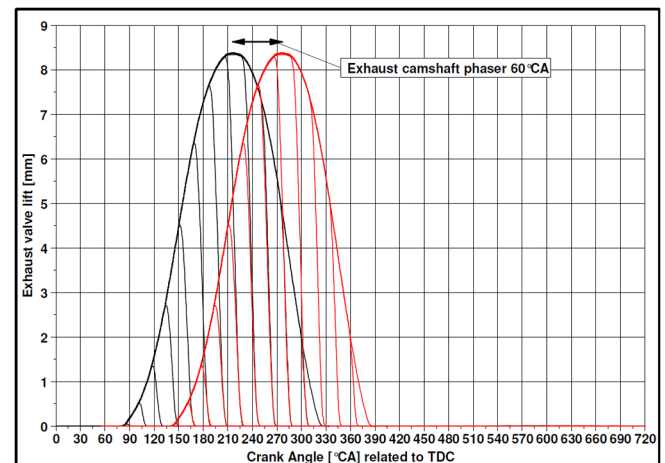


Figure 2-3. Variable exhaust valve timing with exhaust camshaft phaser and "Late Valve Opening" mode

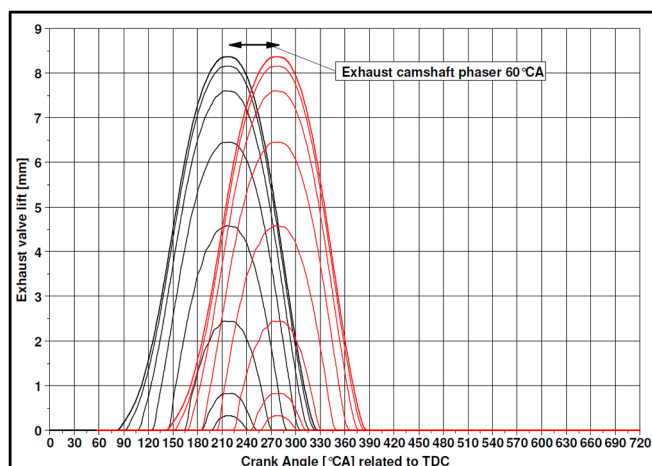


Figure 2-4. Variable exhaust valve timing with exhaust camshaft phaser and "Early Valve Closing" mode

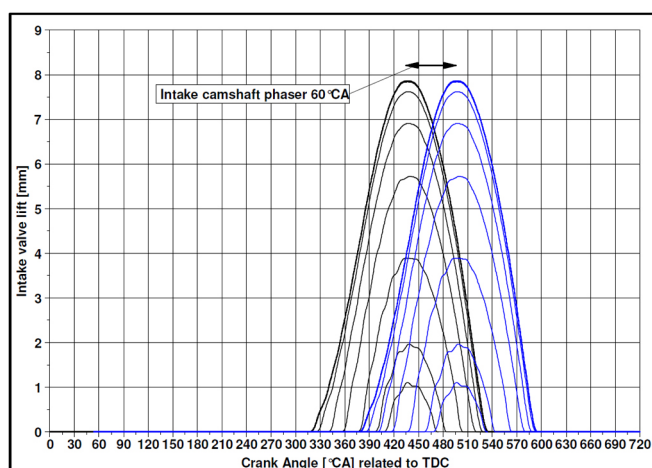


Figure 2-5. Variable intake valve timing with intake camshaft phaser and "Late Valve Opening" mode

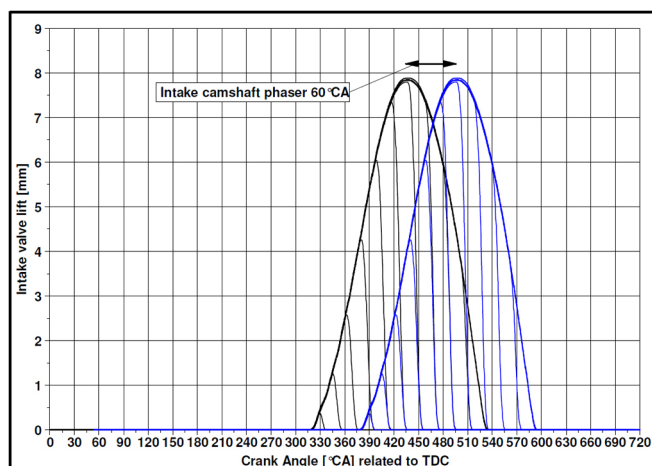


Figure 2-6. Variable intake valve timing with intake camshaft phaser and "Early Valve Closing" mode

One of the major features of this valve train equipment additional to the variety of valve timings itself is the capability to separate the effective compression ratio from the geometric compression ratio. By means of a widely variable Intake Valve Closing (IVC), the effective compression ratio can be adjusted in a wide range. Because of the relation of IVC, effective compression ratio and effective cylinder displacement, Figure 2-7, the best efficiency or stability must be evaluated at each point of operation.

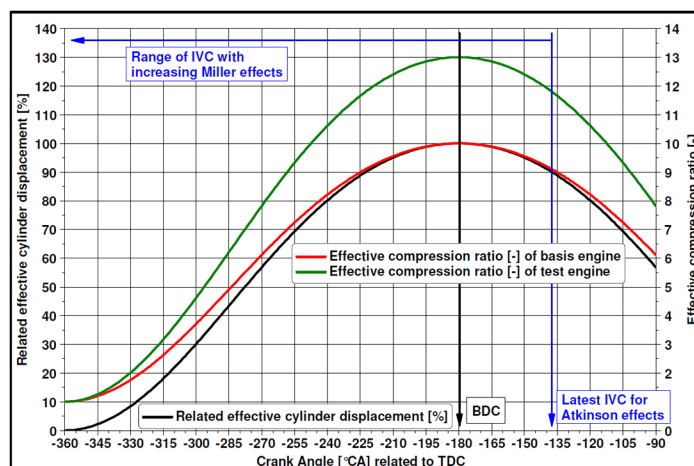


Figure 2-7. Relation of related effective cylinder displacement, effective compression ratio and intake valve closing position

3. METHODOLOGY OF EXPERIMENTS AND ANALYSIS

One of the essential questions of the experimental methodology regarding the wide valve timing ranges and combinations was how to operate the engine in terms of engine temperature and possible transient behavior. The major focus of these investigations was not to find a solution for certain emissions legislations, but to determine the dominant effects of preselected valve train strategies in terms of emissions, heat output and combustion stability. This target was to be combined with a satisfactory productivity of the experimental procedure on the test bench in order to evaluate many valve train effects in detail.

Therefore, the decision was made to create a steady-state but low temperature operation. Thus the best compromise of experimental progress, calibration efforts, reliability and comparability of the results was given in order to identify promising valve train strategies. Using this method, the results were free from dynamic inaccuracies, especially in the case of lambda.

The next step was to determine the operational point of the engine. The usual engine idling speed for catalyst heating on four cylinder DI engines is $n = 1200$ rpm. This level of engine speed was also taken for the steady-state low temperature investigation. The level of indicated mean

Table 3-A. General data of the point of operation

Engine speed [rpm]	1200
Indicated mean effective pressure [bar]	1.7
Engine and intake air temperature [°C]	21
Injection strategy	triple
Fuel mass split [%]	60 : 20 : 20
Injection timing of 1 st injection event	SOI ₁ = 270°CA bTDC
Injection timing of 2 nd injection event	EOI ₂ = 120°CA bTDC
Injection timing of 3 rd injection event	EOI ₃ = 70°CA bTDC
Fuel rail pressure [MPa]	6.0

effective pressure at the engine temperature of 21°C was taken from experimental data from a number of comparable four cylinder DI engines and defined by imep = 1.7 bar. This value illustrates the increase of engine friction due to the low operational temperature. The frictional mean effective pressure nearly doubled compared to the standard operational conditions.

Another determination to be made was the level of the air-to-fuel-ratio or lambda. In comparison to the transient catalyst heating phase at the beginning of NEDC, a slightly lean mixture was defined by $\lambda = 1.02$. Due to the steady-state operation, the pre-catalyst lambda control could already be in use and thus contributed to accurate and comparable measurements.

Like the carry-over definition of engine speed and indicated mean effective pressure from several comparable DI engines, the injection strategy was comparable. Thus, a triple injection strategy with one injection event during the suction stroke and two injection events during the compression stroke was applied. Due to the comparatively high static flow rate of the applied type of injectors and the low fuel mass of the second or third injection events, it was necessary to reduce the fuel rail pressure to 60 bars = 6.0 MPa in order to avoid the injectors to work at the border to the ballistic operational range of actuation time.

The complete general data of the catalyst heating point of operation is given in [Table 3-A](#). In order to illustrate the injection strategy, the injection pattern is given in [Figure 3-1](#) in comparison to the cylinder pressure and fuel rail pressure traces of the reference operational point of catalyst heating.

The most urgent task was to choose the promising valve timing strategies or elements of them to influence the engine behavior during the catalyst heating phase. This task was important because of the large number of valve timing combinations which were generally possible on the test engine but could not be considered in this investigation. Therefore, general assumptions have been made about the probable effects of valve timing to the engine combustion behavior on the thermodynamically sensible point of a catalyst heating idling operation:

1. It is likely that a low or even no valve overlap reduces the internal exhaust gas recirculation rate and thus improves the ignition and combustion conditions.

2. A retarded or even late intake valve opening would generate an increased intake flow velocity by an elevated pressure difference across the intake valves. The increase in intake flow velocity would lead to a higher level of in-cylinder turbulence even at delayed ignition timing.

3. It was the assumption that the effective compression ratio should be kept as close as possible to the geometric value by adjusting the intake valve closing near the BDC.

4. A possibly alternate or additional way to generate an intensive charge motion inside the cylinders could be the complete deactivation of one of the two intake valves.

5. The wall heat losses inside the cylinder head i.e. by the walls of exhaust ducts can be reduced by deactivation of one of the two exhaust valves.

6. It is likely that an earlier exhaust valve opening will produce an increased level of exhaust gas pressure and temperature. The simultaneous loss of indicated mean effective pressure must be compensated by an increase of air and fuel mass per cycle, which improves the ignition and combustion stability.

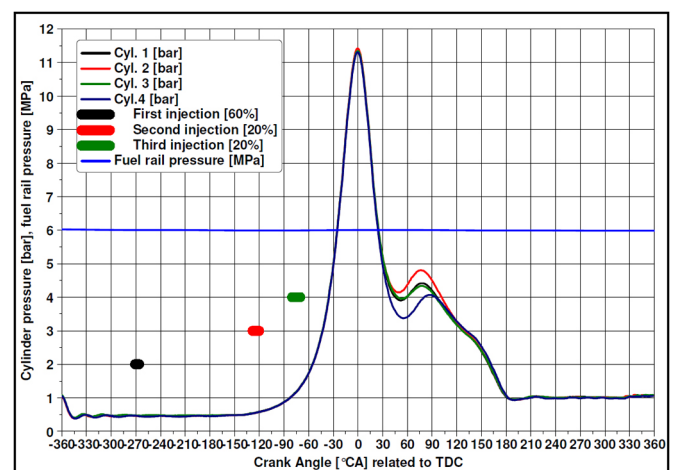
**Figure 3-1. Triple injection strategy for catalyst heating**

Figure 3-2 illustrates these considerations in detail:

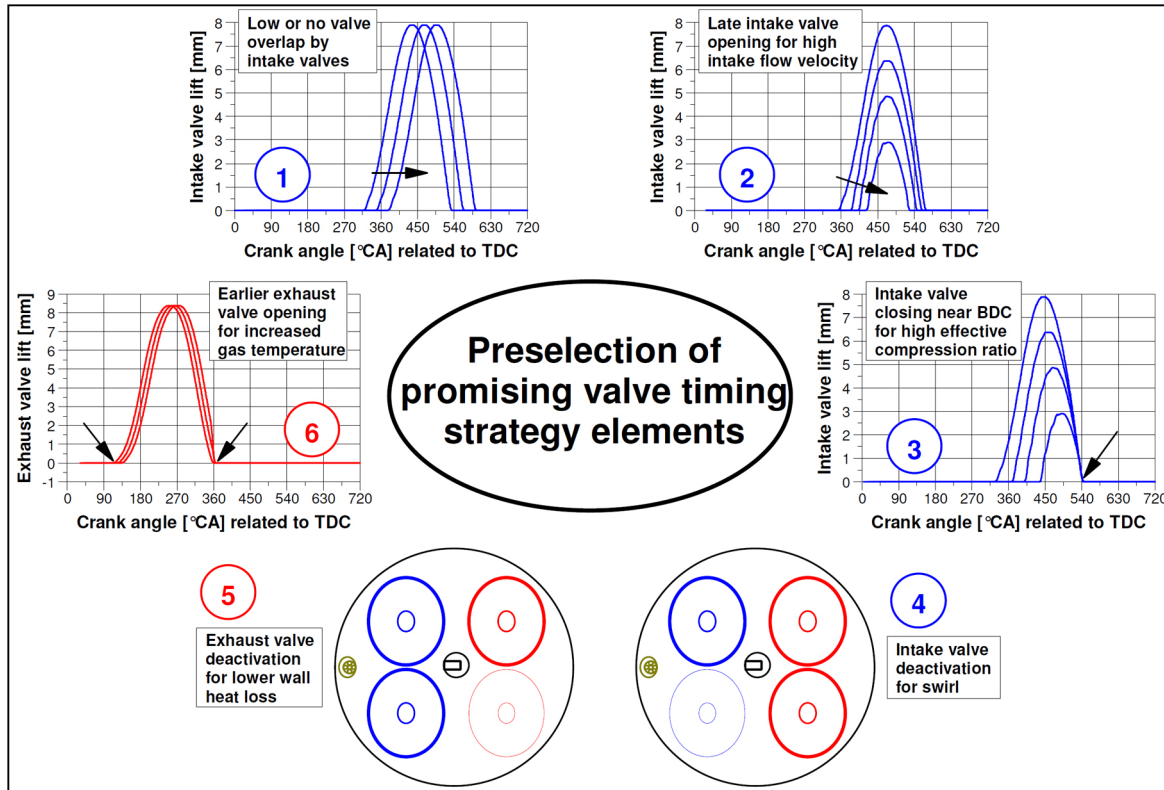


Figure 3-2. Pre-selection of promising valve timing strategy elements

As mentioned, it was the general objective to investigate the effects and related potential of several valve timing strategies. These potentials should be evaluated in relation to the standard catalyst heating strategy without VVT effects.

Due to this definition, a standard catalyst heating point strategy was applied to the test engine using nearly full or full valve lifts curves and adjusting the camshaft phasers to the appropriate position in order to get a reference valve timing. The results of this standard catalyst heating point of operation were taken as the reference engine behavior. The results of every measured point of operation of several VVT strategies have been compared to them and calculated or displayed in a related percentage form.

In order to ensure a high level of comparability and to exclude any unacceptable influences from differences of engine and intake air temperature conditioning from one test bench session to the next, it was defined to carry through an individual measurement of the reference operational point of catalyst heating with standard valve timing for each test bench session. Thus, there was an opportunity to compare every single point of VVT operation to the results of the appropriate reference operational point.

In order to illustrate the analytic post-processing of experimental results, the major calculations are explained in detail in the below formulas. First, it was necessary to equalize the tolerances of adjustable indicated mean pressure

within a certain parameter series. Therefore the average values of indicated mean effective pressure were calculated.

$$imep_{average} = average[imep_1, imep_n] \quad (3-I)$$

The second step contained the normalization of all mass flow or energetically related results as i.e. mass flows of emissions or exhaust gas temperatures.

$$Value_{1,norm} = Value_1 \cdot \frac{imep_{average}}{imep_1} \quad (3-II)$$

$$Temperature_{1,norm} [^{\circ}C] = (Temperature_1 [^{\circ}C] + 273.15) \cdot \frac{imep_{average}}{imep_1} - 273.15 \quad (3-III)$$

In order to express the related potential in comparison to the reference operational point, the third step was to calculate the ratio of the appropriate value of the operational point and the reference operational point.

$$\Delta Value_1 = 100\% \cdot \frac{Value_{1,norm}}{Value_{1,norm,ref}} \quad (3-IV)$$

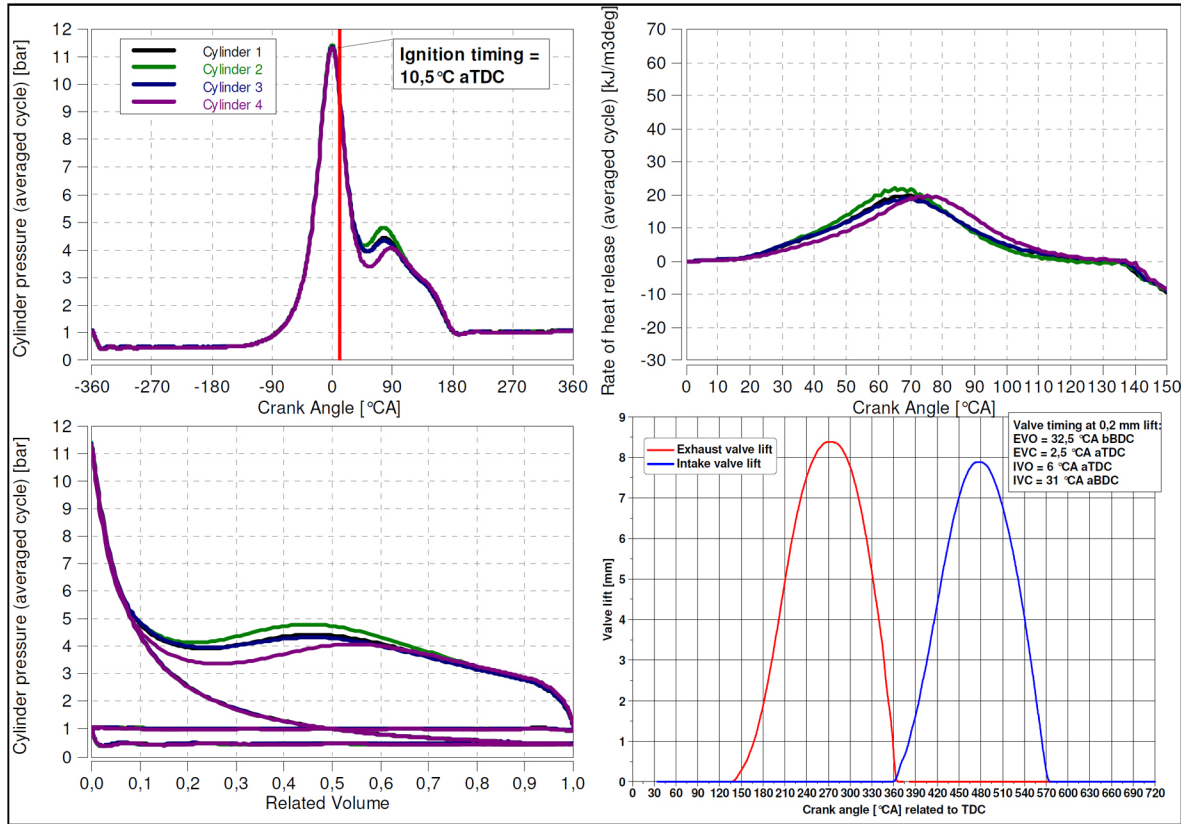


Figure 4.1-1. Valve lift & timing and thermodynamic properties of the reference point of operation

$$\Delta Temperature_{1,norm} [\%] = 100\% \cdot \left(\frac{Temperature_{1,norm} [^{\circ}C] + 273.15}{Temperature_{1,norm,ref} [^{\circ}C] + 273.15} \right) \quad (3-V)$$

The fourth step was to calculate an integrating number of the emissions output. A nominated duration of the catalyst heating phase has been defined by 26.5 seconds in reference to the NEDC velocity profile. The equation 3-X is used to calculate its result $\Sigma EmissionsSum = 1$ for the reference catalyst heating point of operation. Thus, the potential of any VVT point of operation can be easily evaluated in its difference to 1.

$$\dot{m}_{CO,Euro5} = \frac{m_{CO,Euro5}}{t_{NEDC}} \quad (3-IX)$$

$$\Sigma EmissionsSum_1 = \left(\frac{\dot{m}_{THC,1}}{\dot{m}_{THC,Euro5}} + \frac{\dot{m}_{NOx,1}}{\dot{m}_{NOx,Euro5}} + \frac{\dot{m}_{CO,1}}{\dot{m}_{CO,Euro5}} \right) \cdot \frac{t_{CatHeatPhase}}{3600s} \cdot Ratio_{Emissions,CatHeatPhase} \quad (3-X)$$

4. EXPERIMENTAL RESULTS

4.1. REFERENCE CATALYST HEATING POINT OF OPERATION

This point of operation was to define the base line of engine behavior. As mentioned, this point of operation was adjusted with nearly full valve lifts and appropriate camshaft timing to reproduce the valve timing of the production engine. The major properties are given in Figure 4.1-1.

$$Ratio_{Emissions,CatHeatPhase} = 60\% \quad (3-VI)$$

$$\dot{m}_{THC,Euro5} = \frac{m_{THC,Euro5}}{t_{NEDC}} \quad (3-VII)$$

$$\dot{m}_{NOx,Euro5} = \frac{m_{NOx,Euro5}}{t_{NEDC}} \quad (3-VIII)$$

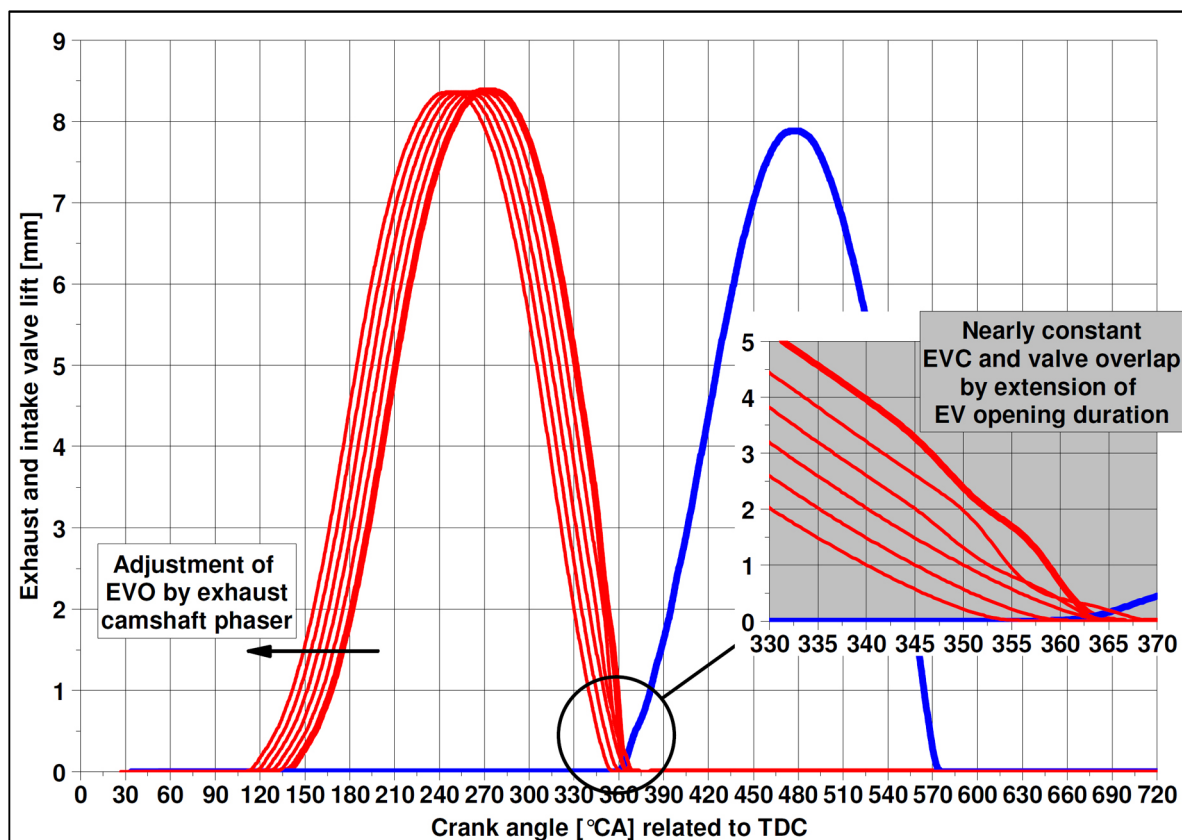


Figure 4.2-1. Valve lift & timing for the variation of exhaust valve opening

4.2. VARIATION OF EXHAUST VALVE OPENING

The first experimental task was to investigate the variation of an exhaust valve opening. This parameter variation was based on two major features of the valve train: the exhaust camshaft phaser and the variable exhaust valve opening duration by the UniAir “Early exhaust valve closing” mode. The objective was to adjust the exhaust valve opening earlier in steps while keeping the exhaust valve closing as exact as possible to the reference value. This combined parameter variation was useful because the valve overlap was nearly constant and its slightly variable internal exhaust gas recirculation influence did not effect the ignition and combustion behavior in a severe way. In order to illustrate the variation of exhaust valve timing, the lift curves are presented in Figure 4.2-1. As displayed, this variation is based on the fixed intake valve timing taken from the reference point. The exhaust valve timing variation was combined with a variation of ignition timing in order to find the best thermodynamic solution. The results can be presented in an exhaust valve opening-ignition timing map.

The expectation of results in a variation of the exhaust valve opening was a gain in exhaust gas temperature and thus in catalyst temperature. Additionally, an improved level of cyclic variations in terms of related standard deviation of

indicated mean effective pressure and a decrease in unburned hydrocarbon emissions were expected. There were concerns about a THC- NO_x -trade off because of the increasing in-cylinder peak temperatures and growing NO_x production (5) due to the increasing air and fuel masses in the cylinder in order to compensate the loss of indicated mean effective pressure due to the earlier exhaust valve opening. An increase of indicated specific fuel consumption was also taken into account. As displayed in Figure 4.2-2 and Figure 4.2-3, these estimations did not completely match the actual engine behavior. Due to the several trade-off situations in the results, the priority was to identify one exhaust valve timing configuration to be the basis for the coming intake valve timing variations. It can be seen that the potential in catalyst temperature, indicated specific fuel consumption and related standard deviation of indicated mean effective pressure depends much more on the ignition timing than on the exhaust valve opening position. Based on the estimation that an intensified catalyst heating phase is able to compensate losses in indicated specific fuel consumption, it can be seen that the gain in catalyst temperature is nearly proportional to the loss in indicated specific fuel consumption. Considering the related standard deviation of indicated mean effective pressure, it is obvious that there is only a minor number of configurations that offer an improved engine stability combined with a generally earlier exhaust valve opening.