

A New Approach for Modeling Cycle-to-Cycle Variations within the Framework of a Real Working-Process Simulation

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ABSTRACT

For a reliable and accurate simulation of SI engines reproduction of their operation limits (misfiring and knock limit) and in this context the knowledge of cyclic combustion variations and their influence on knock simulation are mandatory. For this purpose in this paper a real working-process simulation approach for the ability to predict cycle-to-cycle variations (ccv) of gasoline engines is proposed. An extensive measurement data base of four different test engines applying various operation strategies was provided in order to gain a better understanding of the physical background of the cyclic variations. So the ccv initiated by dilution strategies (internal EGR, lean operation), the ccv at full load and at the knock limit could be investigated in detail. Finally, the model was validated on the basis of three further engines which were not part of the actual development process. In order to obtain a thorough understanding of the ccv phenomenology additionally the so-called residual gas feedback effect had to be examined. Therefore a one dimensional CFD simulation was implemented since in this case gas dynamic processes are essential. Basing on the final ccv model it was possible to evaluate the influence of a fluctuating combustion on knock simulation. A zero-dimensional knock model approach was implemented within the ccv model in order to ascertain the benefit maintained by single cycle resolved knock simulations. However, in correspondence with the measurement data analysis of knocking single cycles, it comes to a conclusion that an accurate prediction of knock phenomena requires the consideration of further influences beyond the mere influence of cyclic variations.

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INTRODUCTION

For over one hundred years engineers and scientists around the world have been preoccupied with the continuous further development of a single machine - the combustion engine. It is therefore easy to understand that future progress will increasingly diminish in scope and will also be much more difficult to achieve. However, previous niche areas, additionally driven by the recent increase in the possibilities of simulative engine development, are becoming increasingly more attractive. For example, combustion stability is playing an increasingly important role in the design of present-day gasoline engine concepts - the more stable the combustion, the further the engine operating limits can be exhausted or the closer these can be approached. Against this backdrop there are above all two limits that restrict engine operation. In the lower part-load range one should mention the misfiring limit which comes into play primarily in the course of high degrees

of charge dilution (residual gas, air) and the associated increase in cyclic fluctuations. At full load, on the other hand, the knock limit restricts optimum-efficiency combustion control. For illustrative purposes [Figure 1](#) shows the cycle-to-cycle variations (ccv) by reference to the interrelationships of indicated mean effective pressure and 50% MFB timing.

Accordingly, knowledge of ccv is accompanied by a clear improvement in accuracy in the consumption-relevant engine map range and thus increases the predictive capability of the simulation. For this reason, the development of a simulation model in order to predict the cyclic fluctuations of gasoline engines reflects to a large extent the spirit of current engine development.

In this paper we present a phenomenological model approach with which it is possible to model the gasoline engine cyclic fluctuations using a quasi-dimensional combustion model [8] within the framework of a real

working process calculation [9]. After adapting just two main adjustment parameters by reference to a few operating points, it is possible to calculate predictively the cycle-to-cycle variations over the complete engine map range. This concerns, in addition to the variation of engine speed, load, air fuel ratio and residual gas content, to a certain extent also the variation of valve timing and intake valve lifts or of measures which change the charge movement and turbulence in the combustion chamber. Also, starting out from the simulation of the ccv in combination with current knock model approaches [7, 4], we examine a new, extended way of knock modeling which is based on the varying pressure and temperature characteristics caused by the ccv.

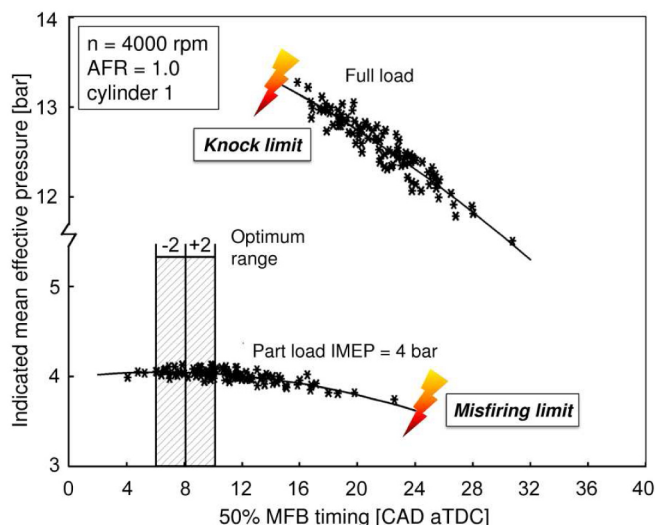


Figure 1. Representation of knock and misfiring limit [3]

STATE OF THE ART

Physical Background of CCV

In view of the fact that the subject of cyclic combustion fluctuations has been the focus of research time and time again in the past decades, there exist already numerous publications that address the subject. These publications have examined in detail the causes of and backgrounds to the ccv [6, 10, 11, 12, 13, 15]. In the following a brief summary shall be outlined.

Figure 2 shows a schematic overview of the causes of ccv. Their origin can be found in the stochastic fluctuations of the gas exchange which affect the ccv of in-cylinder flow and mixture formation around the spark plug. In combination with turbulence fluctuations, which are correlated to the ccv of in-cylinder flow, these are the main reasons for a fluctuating ignition and inflammation phase. Due to the fact that the progressing combustion depends heavily on the early inflammation phase - a retarded cycle will continue to burn slowly and an advanced cycle will continue to burn faster [2] - the inflammation phase is of major importance for ccv. Admittedly, during the further flame propagation there is an

effect of the fluctuations in turbulence or mixture homogeneity on ccv. But, due to averaging of local turbulence or mixture fluctuations over the flame front their influences are likely to be of lower priority [2, 6, 13, 15].

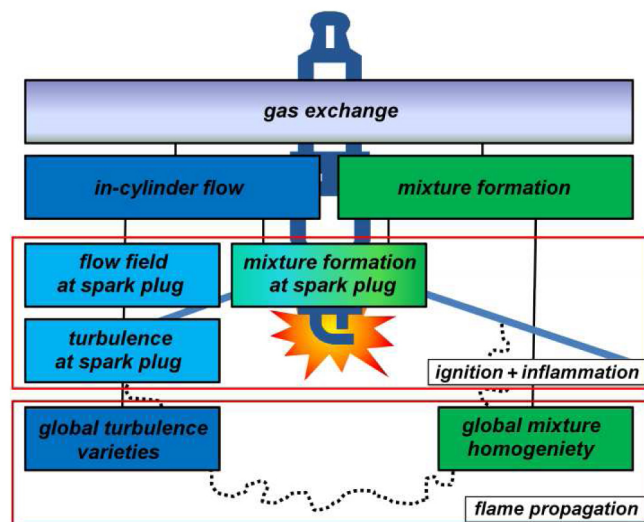


Figure 2. Derivation of the causes of ccv

Beyond that there are influencing factors which increase the impact of the above mentioned stochastic fluctuations on the cyclic variations of combustion. Those are, for instance, the degree of charge dilution, the global turbulence level or the position of combustion (e.g. 50% MFB timing). In other words, the cyclic variability - for example observed in the COV IMEP - is always a result of the combination of the ccv causes depicted in Figure 2 and the engine operation mode (lean operation, exhaust gas recirculation, retarded ignition due to knock etc.). Reaching misfiring limit another influence, the so called residual gas feedback effect [16, 25], becomes apparent which also will be discussed later in this paper.

At this point it should be emphasized that there are ccv causing and ccv increasing factors which have to be considered separately when modeling ccv. With regard to a phenomenological modeling an implemented parameter variation has to be dependent on these increasing factors while representing the stochastic fluctuations which are originally causing ccv.

Quasi-Dimensional Combustion Modeling

The quasi-dimensional combustion model used in this study is sufficiently explained in [8, 9]. Since the presented ccv model affects various correlations regarding this combustion model its principle idea should be outlined in the following. Based on hemispheric flame propagation the combustion chamber is divided in a burned and an unburned zone. These are separated by the flame front which thermodynamically is not considered (see Figure 3).

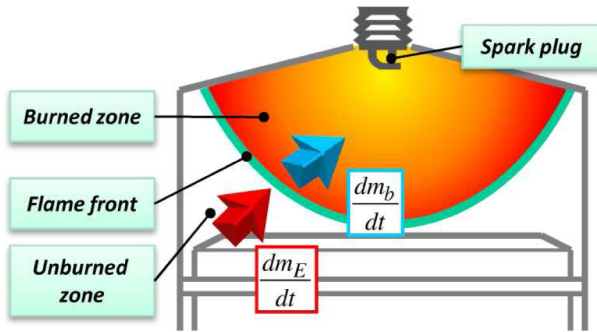


Figure 3. Schematic flame propagation [8]

The burn rate dm_b is calculated by means of the mass m_F entrained into the flame front and the characteristic burn-up time τ_L (Equation 1).

$$\frac{dm_b}{dt} = \frac{m_F}{\tau_L} \quad (1)$$

The characteristic burn-up time depends on the laminar flame speed s_L and the Taylor length l_T (Equation 2).

$$\tau_L = \frac{l_T}{s_L} \quad (2)$$

The laminar flame speed is calculated in accordance with [23] while the Taylor length can be computed by knowledge of the global length scale l , turbulence speed u_{turb} , turbulent kinetic viscosity ν_T and the Taylor factor χ_{Taylor} which is assumed to be 15 (Equation 3) [23].

$$l_T = \sqrt{\chi_{Taylor} \frac{\nu_T \cdot l}{u_{turb}}} \quad (3)$$

The turbulence speed u_{turb} is computed by a homogeneous, isotropic k- ϵ turbulence model [24] and influences - besides Taylor length l_T or respectively characteristic burn-up time τ_L - also the entrainment velocity u_E (Equation 4) which in turn defines the entrainment mass flow dm_E into the flame front (see Equation 5). The density in the unburned zone ρ_{ub} is based on the corresponding mass and volume while the flame surface A_{fl} is calculated according to [8].

$$u_E = u_{turb} + s_L \quad (4)$$

$$\frac{dm_E}{dt} = \rho_{ub} \cdot A_{fl} \cdot u_E \quad (5)$$

Previous Modeling of CCV

Various formulations for the simulative description and prediction of combustion fluctuations have already been

published - an extract from those can be found in [17, 18, 19, 20, 21, 22, 26, 27]. Depending on the purpose there are different approaches of calculating ccv. Most of them are based on a quasi-dimensional combustion model with the aim of simulating ccv predictively [1, 18, 19, 26]. Other studies investigate ccv by means of LES [17, 27] or adopt new simulation methods by using detailed chemical kinetics [21] or knock prediction models [22]. Special feature of the ccv model approach proposed in this paper is its easy and universal applicability due to the amount and variety of measurement data investigated in the scope of this study. Thus, in combination with its small computational effort (about 2-8 seconds per operating point) the ccv model provides a highly practical value.

ANALYSIS OF CYCLE-TO-CYCLE VARIATIONS

Measurement Data Pool

Four test engines were available for the development of the model approach. However, they did differ greatly in terms of the analyzed operating range (full load, part load, complete engine map), the engine concept (downsizing, natural gas engine) or the examined effects (residual gas variation, AFR variation, ignition point variation towards knocking operation). In all, there was a highly comprehensive pool of measurement data covering all the effects relevant to complete ccv modeling. The following engines were explicitly available for the analyses: the Daimler M271, the Mahle downsizing engine, the MTU Series 4000 and the Opel Z22YH - the relevant geometry and operation data are set out in Table 1. The measurement data range is schematically set out in Figure 4, where it must be noted that in the case of the Daimler M271 - due to various timing and valve lift variations (see Figure 5) - the scope of measurement data is much greater than this form of representation would lead one to assume. The same applies to the range of the MTU Series 4000, which through AFR and ignition point variations made a valuable contribution to model development.

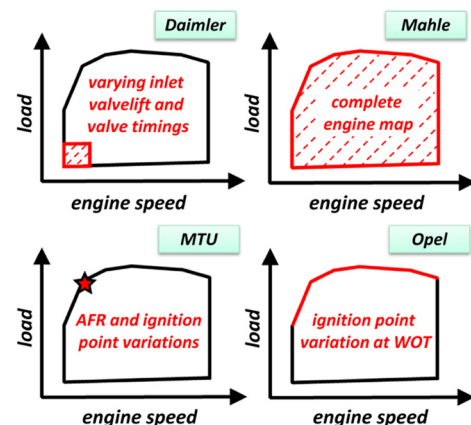


Figure 4. Measurement data ranges (schematic)

Table 1. Geometry and operation data of the investigated test engines

Test engine	Daimler	Mahle	MTU	Opel
Type [-]	R4	R3	V12/one-cyl.	R4
Displacement [cm ³]	1796	1200	57199	2198
Stroke [mm]	85	74	210	95
Bore [mm]	82	83	170	86
Geom. compression [-]	10.0	9.75	12.9	11.9
Max. power [kW] at engine speed [rpm]	135 5500	144 6500	1200 1500	114 5600
Fuel	Gasoline	Gasoline	Natural gas	Gasoline
No. of operating points [-]	> 1000	270	17	44

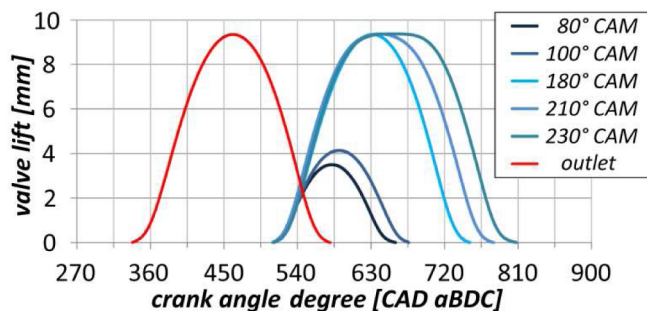


Figure 5. Valve lift curves of different cam profiles of the Daimler M271

Physical Backgrounds to the Cycle-to-Cycle Variations

Influences of charge dilution

Figure 6 shows for the standard camshaft (opening 180° crank angle, measured at 2 mm valve lift) at 1600 rpm and 3 bar IMEP the interrelationship of residual gas content and cyclic fluctuations - here in the form of the standard deviation of IMEP (STDDEV IMEP). For very high residual gas contents (>40%) the STDDEV IMEP must be considered with the reservation that the inflammability or the laminar flame propagation and additionally due to the high ccv level also the accuracy of the gas exchange analysis are questionable. It can be clearly seen that knowledge of the residual gas content on its own is not enough to be able to make a concrete statement about the relevant ccv. Because particularly with increasing residual gas contents - above all in the misfiring limit range - the correlation between cyclic

fluctuations and inert gas content is becoming increasingly more indifferent. For example, for an EGR rate of around 33% there are standard deviations of the indicated mean effective pressure of around 0.1 to 0.3 bar - this corresponds to a COV IMEP range of approximately 3.3 to 10%.

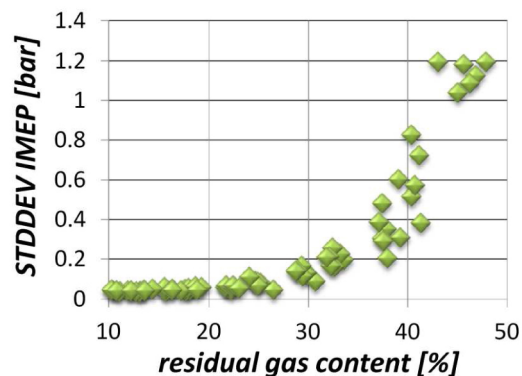


Figure 6. STDDEV IMEP against EGR rate, Daimler M271 (180° camshaft, 1600 rpm, 3 bar)

Clearly further effects must exist whose influence on the ccv is not covered by the residual gas content. This is verified by, among others, Figure 7. Using the 230° camshaft of the Daimler M271, the cyclic fluctuations are shown here as a function of the residual gas content and the IVC timing by way of example. It can be clearly seen that with identical residual gas content there are clear differences with regard to the IVC timing. For this reason, in-cylinder flow and its effects on combustion are assumed to play a further role when investigating ccv.

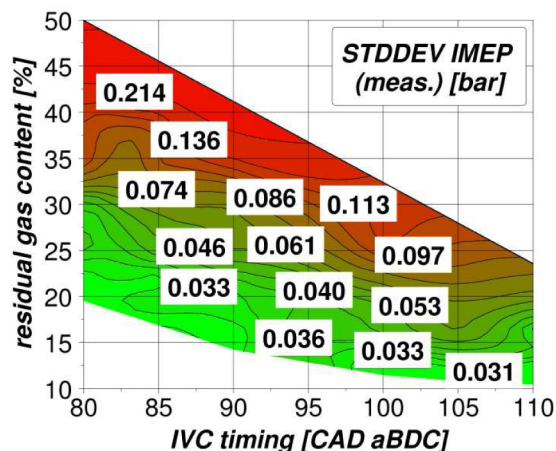


Figure 7. STDDEV IMEP as a function of EGR rate and IVC timing (230° camshaft, 1600 rpm, 3 bar)

To examine more closely the influence of charge dilution, it thus seems sensible to consider the burn duration. Here, great significance is attached to the early combustion phase with regard to the cyclic fluctuations in general [2]. Against this background, initially 0-10% burn duration is examined. Figure 8 shows the typically exponential interrelationship

between the STDDEV IMEP and the 0-10% burn duration. In this case the 0% MFB point corresponds to the ignition point applied in the ECU. Consequently the apparent ignition delay - i.e. the time from the ignition point to a first perceptible pressure increase in the combustion curve - is also modeled in the 0-10% burn duration. The different burn durations are obtained from a "scanning" of intake and exhaust timings which lead as the result of a varying valve overlap to different EGR rates. Long 0-10% burn durations tend to result from high residual gas contents, where here also the influence of the valve timings or the charge movement and turbulence is felt. Compared with the representation against the residual gas content a somewhat clearer correlation is indeed presented, from a burn duration of 50° crank angle - which in this case corresponds to a residual gas content of 30 to 40% - but likewise only limited statements can be made about the extent of the cyclic fluctuations.

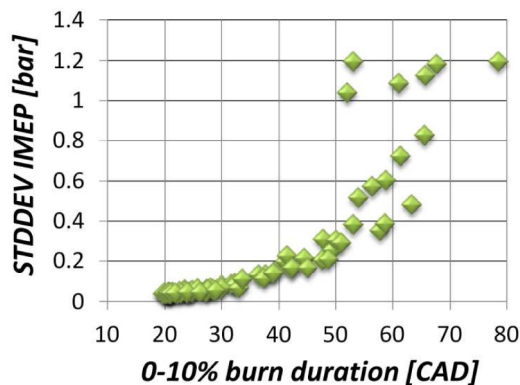


Figure 8. STDDEV IMEP against 0-10% burn duration, Daimler M271 (180° camshaft, 1600 rpm, 3 bar)

It is also possible to investigate the influences of charge dilution on the example of the MTU Series 4000 (see Figure 9). In contrast to the Daimler M271, however, this examination is performed by increasing the AFR up to the lean misfiring limit. Varying the AFR to the lean misfiring limit brings about, along similar lines to residual gas dilution in the Daimler M271, an increase in the burn duration and consequently an increase in the ccv. A combined AFR and ignition point variation on the other hand, with a longer initial inflammation phase, shows no clear increase in the cyclic fluctuations. In this case the negative effects of a delayed inflammation phase are compensated by advancing the ignition timing or respectively by shifting the point of 50% MFB in the optimum direction. This shift brings about a stabilization of combustion manifesting in the form of a reduced fluctuation of the indicated pressure-volume work and thus also of the indicated mean effective pressure (COV IMEP). For both variations it can be clearly seen that leaning in the range of the misfiring limit entails substantially longer 0-10% burn durations. Provided solely the air fuel ratio is increased, this rise in 0-10% burn duration behaves almost linearly to the leaning. Simultaneously advancing the point of 50% MFB does in fact bring about a reduction in the cyclic

fluctuations, but the associated lower combustion temperatures lead to an overproportional retardation of the 0-10% burn duration.

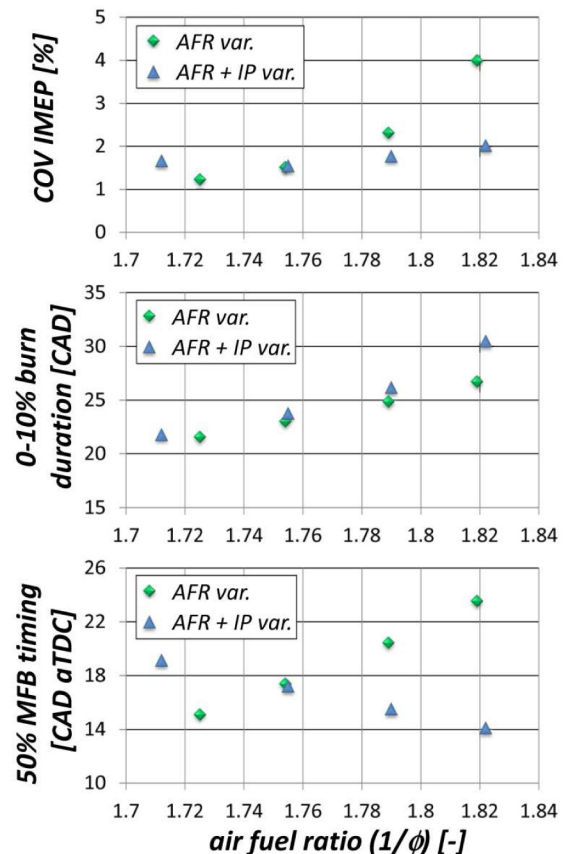


Figure 9. COV IMEP, 0-10% burn duration and 50% MFB timing over AFR, MTU gas engine (one-cylinder, 1500 rpm, 21-23 bar)

To summarize, it must be stressed that sole knowledge of the 0-10% burn duration is not sufficient to determine the ccv satisfactorily. Further effects such as a change in the charge movement and turbulence or adjustment of the ignition point can influence the combustion fluctuations sometimes very significantly. E.g. this is also demonstrated when examining the combined AFR and ignition point variation of the MTU engine.

Influence of 50% MFB timing on the cycle-to-cycle variations

In Figure 10 a comparison is made on the example of the Mahle downsizing engine for every 30 burning functions of two operating points at different engine load. It can be clearly seen that both operating points differ substantially in the extent of the COV IMEP, but only slightly in the fluctuation range of the 10% or 50% MFB points. The stability of the combustion is thus dependent in the case of moderate degrees of dilution on the position of combustion. The behavior of the increasing COV IMEP with a more retarded 50% MFB timing can be clearly explained by way of the parabolic

correlation between thermal efficiency and point of 50% MFB [3].

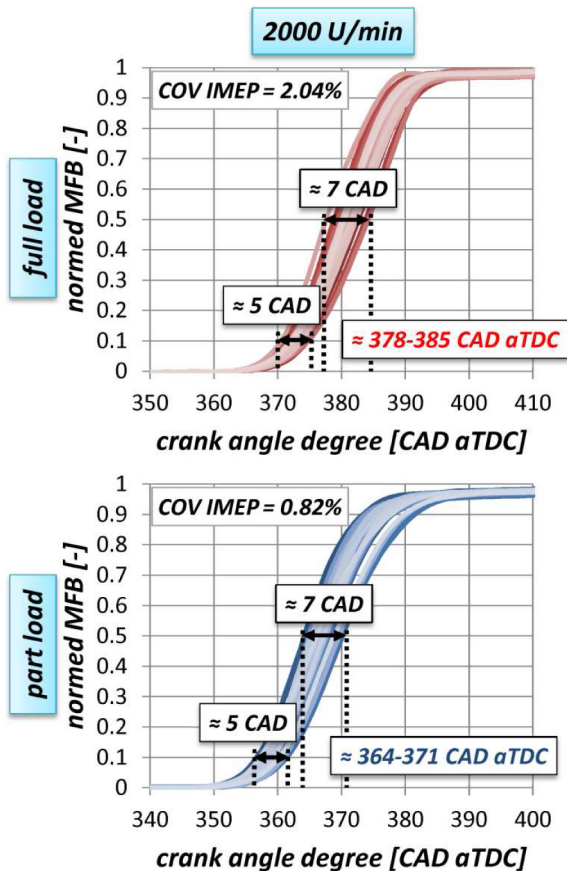


Figure 10. Normalized mass fraction burned against crank angle, Mahle downsizing engine (8/25 bar, 2000 rpm, 30 combustion cycles)

For this purpose Figure 11 shows schematically a constant fluctuation at two operating points which differ in their point of 50% MFB. Depending on the 50% MFB timing of the respective operating point, a wide difference in the fluctuations of efficiency is obtained for an identical fluctuation range. Operating points with virtually optimum-efficiency 50% MFB timing show a very small fluctuation of efficiency, since the slope of the parabola changes only slightly in this range. However, the behavior in the case of retarded or advanced points of 50% MFB is quite different: due to the steep drop in efficiency a varying 50% MFB timing results in unequally higher fluctuations of efficiency. Against this background it becomes clear that the rise in ccv is only superficially connected with the engine load and is caused primarily by the shift in combustion accompanying this load increase. The point of 50% MFB is brought into play in the following to number and evaluate this shift. The dependence of the cyclic fluctuations on the combustion position is of fundamental importance in terms of understanding the backgrounds and causes of the cyclic fluctuations.

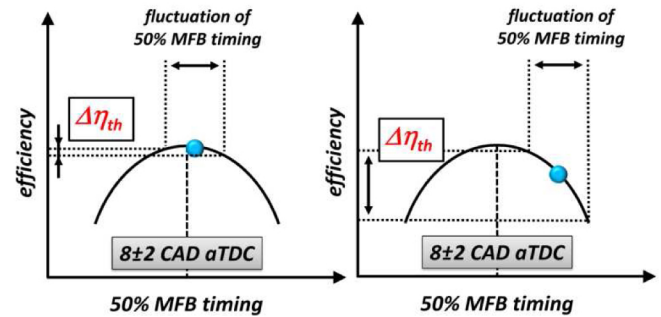


Figure 11. Influence of 50% MFB timing on the efficiency fluctuation

Compared with the Mahle downsizing engine, an ignition point variation at full load was available for the Opel Z22YH. Here the ignition points were successively advanced for every engine speed in order to induce knocking combustion. With regard to the analysis of the dependence of the ccv on the combustion position, this ignition point variation proves to be advantageous in that the previous results and findings can be validated in this way. Figure 12 shows the fluctuations of the indicated mean effective pressures plotted against engine speed and 50% MFB timing.

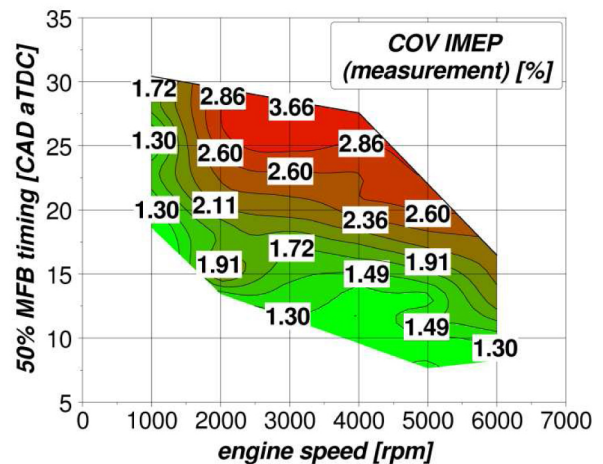


Figure 12. COV IMEP against engine speed and 50% MFB timing, Opel Z22YH

It can be clearly seen that, as with the Mahle engine, the ccv increase with a retarded point of 50% MFB. Furthermore, with a constant point of 50% MFB for middle to high engine speeds there is a certain dependence on speed on the part of the cyclic fluctuations. What is striking here is that for constant 50% MFB timings at low engine speeds lower ccv occur. As well as the interrelationship between the position and fluctuation of combustion, it is consequently possible to observe an engine speed dependence of the ccv in the higher load range. However, with regard to modeling, it is necessary to distinguish quite clearly which effects bring about this speed dependence so as not for instance in the low-load range to confuse the influence of engine speed on internal exhaust