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IMPLICATION OF CYCLE-TO-CYCLE VARIABILITY IN SI ENGINES

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ABSTRACT
The paper deals with utilization of an adaptive combustion model in order to simulate cycle-to-cycle combustion variability of SI engines. The used empirical adaptive combustion model consists of two parts: the first part for ignition delay prediction and the second part for in-cylinder combustion process description. There is proved mutual independence of these two phases and shown their characteristics in terms of cycle-to-cycle variability. The practical utilization of the cycle-to-cycle variability simulation is demonstrated by computational analysis of various variability levels at different engine operational points in order to assess its impact on engine fuel consumption. The calculation results are generalized for SI gasoline engines independent of both engine load and combustion rate as well.
KEYWORDS: SI ENGINE, CYCLE-TO-CYCLE VARIABILITY, ADAPTIVE COMBUSTION MODEL, SPARK TIMING, IGNITION DELAY, VARIABILITY FACTOR, FUEL CONSUMPTION.

SHRNUTÍ
Tento příspěvek se zabývá použitím adaptivního modelu hoření za účelem simulace mezicyklové variability hoření ve válci zážehového spalovacího motoru. Použitý numerický adaptivní model hoření sestává ze dvou částí: první část pro predikci průtahu zážehu a druhá část pro popis vlastního spalovacího procesu ve válci motoru. Je zde ukázána vzájemná nezávislost těchto dvou částí a jejich vlastnosti s ohledem na mezicyklovou variabilitu. Praktické využití simulace mezicyklové variability je demonstrováno na výpočtové analýze různých úrovní variability v různých pracovních bodech motoru s cílem ohodnotit dopad mezicyklové variability na spotřebu paliva. Výsledky výpočtů jsou zobecněny pro zážehové benzínové motory a lze je využít nezávisle jak na zatížení motoru, tak i na rychlosti hoření.
KLÍČOVÁ SLOVA: ZÁŽEHOVÝ MOTOR, MEZICYKLOVÁ VARIABILITA, ADAPTIVNÍ MODEL HOŘENÍ, PŘEDSTIH ZÁŽEHU, PRŮTAH ZÁŽEHU, FAKTOR VARIABILITY, SPOTŘEBA PALIVA.

1. INTRODUCTION
The cycle-to-cycle combustion variability is a well-known phenomenon among engineers dealing with engine development. The dispersion in combustion process is generally caused by three factors: the variation in turbulent gas motion in a cylinder during combustion; the variation in the amounts of fuel, air, and burned gas present in a given cylinder during each cycle; variations in mixture composition within the cylinder near the spark plug – due to variations in mixing between air, fuel, recirculated exhaust gas and residual gas [3]. The strongest impact on resulting cycle-to-cycle combustion variability has the onset of the combustion process (flame kernel development) which is particularly essential in case of standard SI engines. The pre-flame period is negatively affected by non-homogeneous mixture in vicinity of the spark plug and too weak or too intensive charge movement [1], [4], [9]. The further flame propagation is strongly influenced by the combustion chamber topology and the in-cylinder turbulence intensity, while the most dominant factor in terms of cycle-to-cycle variation is the turbulence intensity which varies substantially among individual cycles [10].

Various measures of the cycle-to-cycle combustion variability are widely used. They can be defined in terms of variations in the cylinder pressure between different cycles or in terms of variations in the parameters of the burning process. The most used quantities of pressure-related parameters are the maximum cylinder pressure, the crank angle at which this maximum pressure occurs, the maximum rate of pressure rise and the indicated mean effective pressure (IMEP). The burn-rate-related parameters are the maximum heat-release rate or mass burning rate, the crank angle at 50% of mass burned fraction, the flame development angle and the rapid burning angle [3]. The indicated mean effective pressure is very sensitive to measurement accuracy, especially when non water-cooled sensors with high cyclic temperature drift are used [7].

The main task of this paper is to investigate cycle-to-cycle variability in terms of combustion process differences, i.e. without turbulence and other effects on changes in mass of charge trapped in the cylinder. The investigation of cylinder-to-cylinder variation sakes is also not a scope of this paper. Therefore, the most convenient measure for the cycle-to-cycle variability assessment is the standard deviation (SDEV) in crank angle at 50% of mass burned fraction $MBF50\%$ which reflects the phasing of combustion process and is nearly independent of charging variations

$$SDEV\,(MBF50\%\,) = \sqrt{\frac{\sum \left(MBF50\% \,-\, \overline{MBF50\%}\right)^2}{n\,-\,1}} \tag{1}$$

Coefficient of variation (COV) can be used as an alternative measure of the maximum cylinder pressure $PMAX$ in form:

$$COV\,(PMAX\,) = \frac{100}{\overline{PMAX}}\sqrt{\frac{\sum \left(PMAX \,-\, \overline{PMAX}\right)^2}{n\,-\,1}} \tag{2}$$

The frequently applied limit value for smooth engine run is $COV(PMAX) < 15\%$. However, this parameter has to be used carefully, especially when the spark timing is not set to its optimum. Typical values of coefficient of variation of maximum cylinder pressure and standard deviation in crank angle at 50% of mass burned fraction at optimum ignition timing are shown in Figure 1. All displayed engine operational points in these maps are used for further computational analysis of the potential for a fuel consumption reduction in this paper. At very low engine load at $IMEP < 1$ bar (not displayed in Figure 1) the cycle-to-cycle variability is usually increasing due to lower compression pressure and temperature and higher content of residual gas in the cylinder.

The relationship between two above mentioned measures in Figure 2 is shown. The correlation between these two quantities is for maximum brake torque (MBT) spark timing very close. Engine

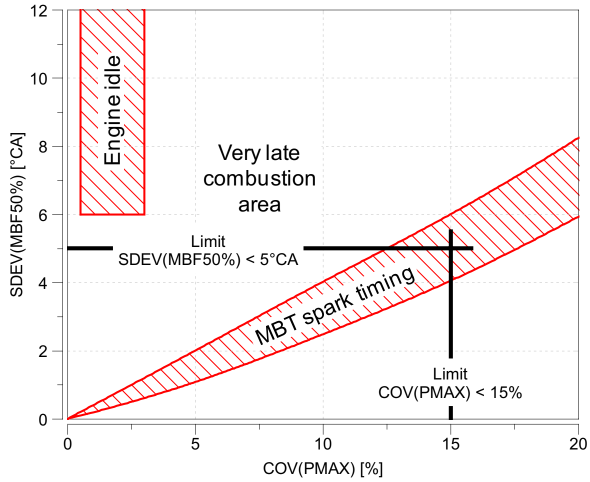

FIGURE 2: Relationship between coefficient of variation of maximum cylinder pressure and standard deviation in crank angle at 50% of mass burned fraction.

OBRÁZEK 2: Vzájemná korelace mezi variačním koeficientem maximálního tlaku a směrodatnou odchylkou úhlu natočení klikového hřídele při 50% vyhoření náplně válce.

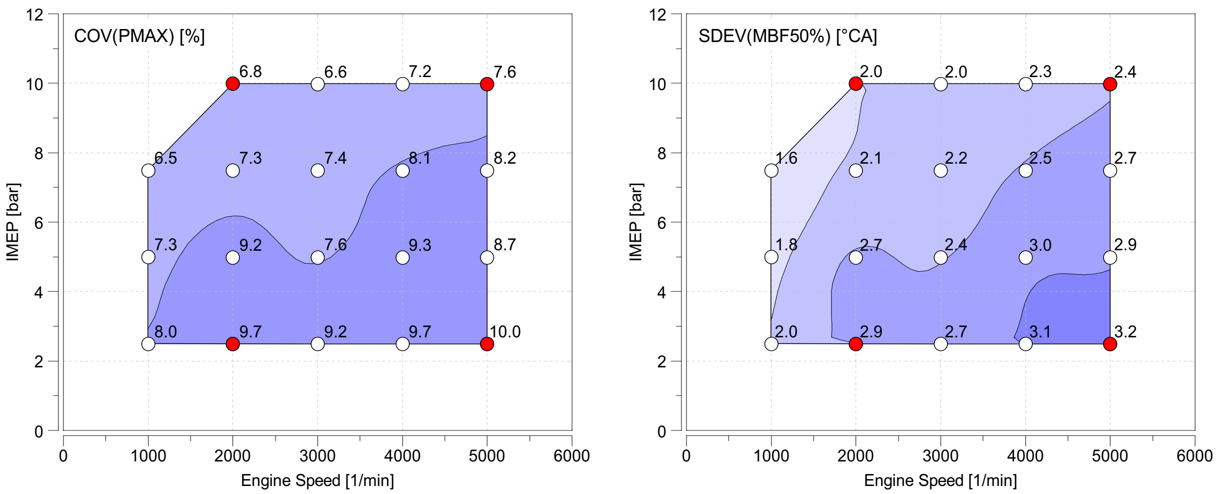

FIGURE 1: Typical values of coefficient of variation of maximum cylinder pressure and standard deviation in crank angle at 50% of mass burned fraction. An example for naturally aspirated four-cylinder SI gasoline engine of swept volume 1.6 dm^3. Red colored points mark the engine operational points where a deeper computational analysis has been carried out.

OBRÁZEK 1: Typické hodnoty variačního koeficientu maximálního tlaku a směrodatné odchylky úhlu natočení klikového hřídele při 50% vyhoření náplně válce. Příklad pro nepřeplňovaný čtyřválcový zážehový benzínový motor o zdvihovém objemu 1,6 dm^3. Červené body označují režimy motoru, při kterých byla provedena detailnější výpočtová analýza.

operational points with exceptional spark timing lie outside this region, typically engine idle, as has been mentioned above.

2. DESCRIPTION OF ADAPTIVE COMBUSTION MODEL

For the simulation of different conditions during combustion process the empirical adaptive combustion model has been developed [4]. This model offers very simple implementation of variability control because it consists of two independent parts. The first part predicts the ignition delay Dj$_{ign}$ which is defined as a crank angle difference between spark ignition and very first sign of the heat release, as graphically shown in Figure 3. The second part is the flame propagation process. These two parts have been proven as an independent processes [5].

The crank angle duration of the ignition delay is mathematically

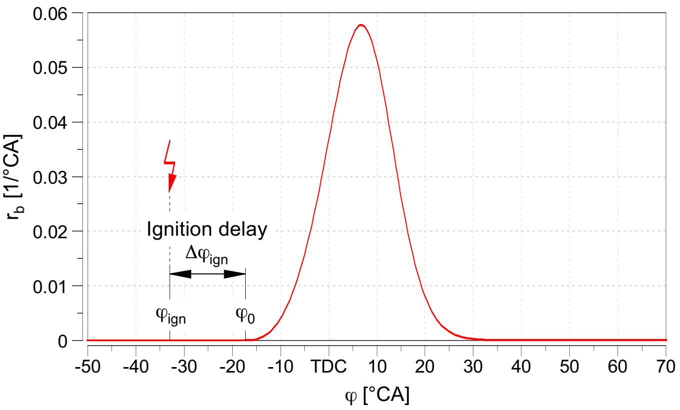

FIGURE 3: Normalized burn rate r_b, definition of the ignition delay Dj$_{ign}$.
OBRÁZEK 3: Jednotková rychlost hoření r_b, definice průtahu zážehu Dj$_{ign}$.

given by improved empirical formula originally coming from [4], [5]. The investigation of more SI engines allowed to achieved new form

$$\Delta\varphi_{ign} = A_{ign} A_{ign,v} 5\times10^{4} n^{0.7} p_{ign}^{-0.2} T_{ign}^{-2} \left[0.2 + \left(\lambda_{mix} - 0.8\right)^{2}\right] \times \times \left(1 + x_{r,st}\right)^{5} \qquad (3)$$

$$[\text{°CA}, 1, 1, 1/\text{min}, \text{bar}, \text{K}, 1, 1]$$

The formula improvement consists in change of multiplier and single exponents based on regression analysis of extended experiments. Besides calibration factor A_{ign} there is also multiplier $A_{ign,v}$ which is used for generation of the cycle-to-cycle variability. The ignition delay depends on the engine speed n, in-cylinder pressure p_{ign} and mean in-cylinder temperature T_{ign} at the moment of the spark ignition. Relative air/fuel ratio in unburned mixture l$_{mix}$ is given by relative air/fuel ratio l and by residual mass fraction x_r in the cylinder

$$\lambda_{mix} = \lambda + x_r \frac{\lambda - 1}{1 - x_r} \qquad \forall \quad \lambda > 1$$
$$\lambda_{mix} = \lambda \qquad \forall \quad \lambda \le 1 \qquad (4)$$

Mass fraction of stoichiometric residual gases $x_{r,st}$ is given by formula

$$x_{r,st} = x_r \frac{1 + L_{st}}{1 + \lambda L_{st}} \qquad \forall \quad \lambda > 1$$
$$x_{r,st} = x_r \qquad \forall \quad \lambda \le 1 \qquad (5)$$

where L_{st} denotes stoichiometric wet air/fuel ratio.

The main combustion phase is described by the change of mass fraction of burned gas x_b during combustion. Improved empirical formula covering wider range of SI engines has form

$$\frac{dx_b}{d\varphi} = A_b A_{b,v} 15\times10^{-5} n^{-0.5} p^{0.3} T^{0.4} \left(\frac{V_c}{V}\right)^{1.5} \times$$
$$\times \left[0.8 - \left(\lambda_{mix} - 0.83\right)^{2}\right]^{4.4} \left(1 - x_{r,st}\right)^{4.4} x_{mix}^{1.1} \left(\varphi - \varphi_0\right)^{2.5} \qquad (6)$$

$$[1/\text{°CA}, 1, 1, 1/\text{min}, \text{bar}, \text{K}, \text{m}^3, 1, 1, 1, \text{°CA}]$$

where p, T and V denotes instantaneous in-cylinder pressure, mean in-cylinder temperature and cylinder volume respectively. V_c is the compression volume at piston top dead center (TDC) and x_{mix} is the instantaneous unburned mixture mass fraction in the cylinder. The multiplier $A_{b,v}$ which extends the calibration factor A_b is used for the cycle-to-cycle variability simulation. The meaning of the angles j and j0 follows from Figure 3. The change of the mass of burned fuel m_{fuel} is then

$$\frac{dm_{fuel}}{d\varphi} = \frac{m_c}{1 + \lambda L_{st}} \frac{dx_b}{d\varphi} \qquad (7)$$

where m_c is the total cylinder mass.

2.1 DETERMINATION OF CALIBRATION AND VARIABILITY FACTORS

For the determination of calibration and variability factors, it is necessary to evaluate sufficient volume of measured data – at least the sequence of 200 cycles. Indicated pressure traces have to undergo thermodynamic analysis in terms of getting mean in-cylinder temperature and normalized burn rate r_b which is crucial parameter. The determination of the single cycle-based factor for the ignition delay comes from rearrangement of equation (3)

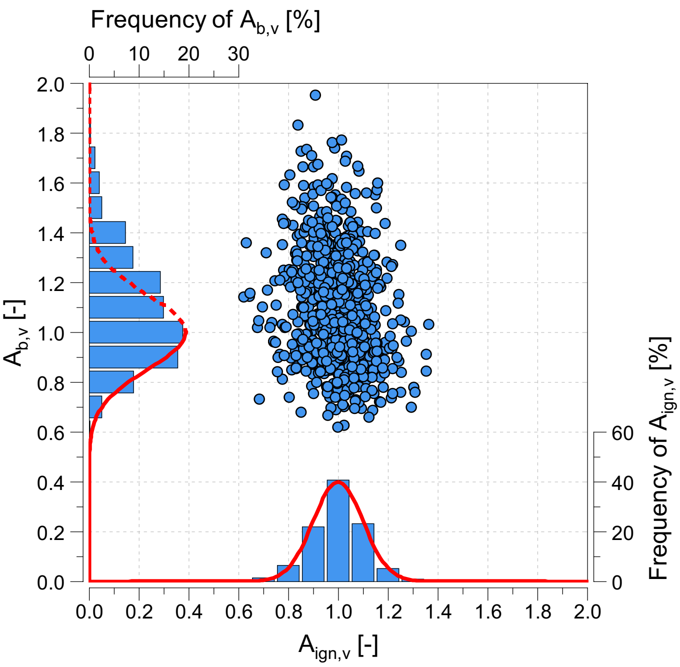

FIGURE 4: Typical distribution of variability factors for ignition delay $A_{ign,v}$ and main combustion phase $A_{b,v}$ over 870 successive cycles. Traces of the normal distribution, dashed line represents a range where the normal distribution doesn't coincide with measurement.

OBRÁZEK 4: Typické rozdělení faktorů pro variabilitu průtahu zážehu $A_{ign,v}$ a variabilitu vlastního procesu hoření $A_{b,v}$ během 870 po sobě následujících pracovních cyklů. Křivky reprezentující normální rozdělení, čárkovaná čára představuje oblast, kdy se normální rozdělení neshoduje s měřením.

$$A_{ign}\,A_{ign,v} = \\ = \frac{\Delta\varphi_{ign}}{5\times10^{4}\,n^{0.7}\,p_{ign}^{-0.2}\,T_{ign}^{-2}\left[0.2+\left(\lambda_{mix}-0.8\right)^{2}\right]\left(1+x_{r,st}\right)^{5}} \tag{8}$$

where $D\varphi_{ign}$ is the measured ignition delay for each single cycle. The product $A_{ign}\,A_{ign,v}$ consists of the main calibration factor A_{ign} which is constant for the sequence of all cycles and of the variability factor $A_{ign,v}$ which deviates from value 1 in the current cycle

$$A_{ign} = \mathrm{median}\left(A_{ign}A_{ign,v}\right) \\ A_{ign,v} = \frac{A_{ign}A_{ign,v}}{\mathrm{median}\left(A_{ign}A_{ign,v}\right)} \tag{9}$$

A similar procedure has to be done for the main combustion phase. For the determination of the single cycle-based factor $A_b\,A_{b,v}$ used in equation (6), the most important region at the rate of burning curve (see Figure 3) is the vicinity of its maximum.

Therefore, this factor can be determined based on measured in-cylinder conditions at 50% of mass burned fraction. Thus, from equation (6) can be derived

Similarly to the previous case, the calibration factor A_b is constant over the sequence of all cycles and variability factor $A_{b,v}$ deviates from value 1 according to the current cycle

$$A_b = \mathrm{mode}\left(A_bA_{b,v}\right) \doteq \mathrm{median}\left(A_bA_{b,v}\right) \\ A_{b,v} = \frac{A_bA_{b,v}}{\mathrm{mode}\left(A_bA_{b,v}\right)} \doteq \frac{A_bA_{b,v}}{\mathrm{median}\left(A_bA_{b,v}\right)} \tag{11}$$

Here, the mode function should be used for an accurate evaluation because of asymmetrical frequency distribution of the factor $A_b\,A_{b,v}$. The mode of a set of data values is the value that appears most often. Since the mode determination can cause some difficulties, the median has been used instead for an approximation. Typical distribution of both mentioned variability factors is shown in Figure 4. It is obvious that factors $A_{ign,v}$ and $A_{b,v}$ are mutually almost independent, although a cross-correlation between ignition delay and combustion rate can be observed [8], [11]. While the frequency distribution of $A_{ign,v}$ is quite symmetrical and corresponds to normal distribution, the distribution of $A_{b,v}$ is asymmetrically outspread to its higher values.

A decision whether a distribution is symmetrical or not is possible by means of the adjusted Fisher-Pearson standardized moment coefficient also called skewness [2]. The magnitude of skewness describes how symmetrical a distribution is about its mean. A positive value indicates a leaning to the right of mean and a negative value indicates a leaning to the left. The skewness is defined as

$$SKEW(x) = \frac{n}{(n-1)(n-2)}\sum\left(\frac{x-\bar{x}}{SDEV(x)}\right)^{3} \tag{12}$$

From the thermodynamic analysis of a measured in-cylinder pressure can be observed that quantities like indicated mean effective pressure, maximum heat-release rate, maximum cylinder pressure or the crank angle at 50% of mass burned fraction follow almost normal distribution because their skewness for 200 cycles is within the 90 percent range ±0.28 [2]. Sequences of calculated variability factors according to equations (9) and (11) with both measured and by means of simulation restored chosen quantities are depicted in Figure 5.

$$A_b\,A_{b,v} = \frac{r_{b,50\%}}{15\times10^{-5}\,n^{-0.5}\,p_{50\%}^{0.3}\,T_{50\%}^{0.4}\left(\dfrac{V_c}{V_{50\%}}\right)^{1.5}\left[0.8-\left(\lambda_{mix}-0.83\right)^{2}\right]^{4.4}\left(1-x_{r,st}\right)^{4.4}\left(1-x_r\right)^{0.1}0{,}5^{1.1}\left(\varphi_{50\%}-\varphi_0\right)^{2.5}} \tag{10}$$

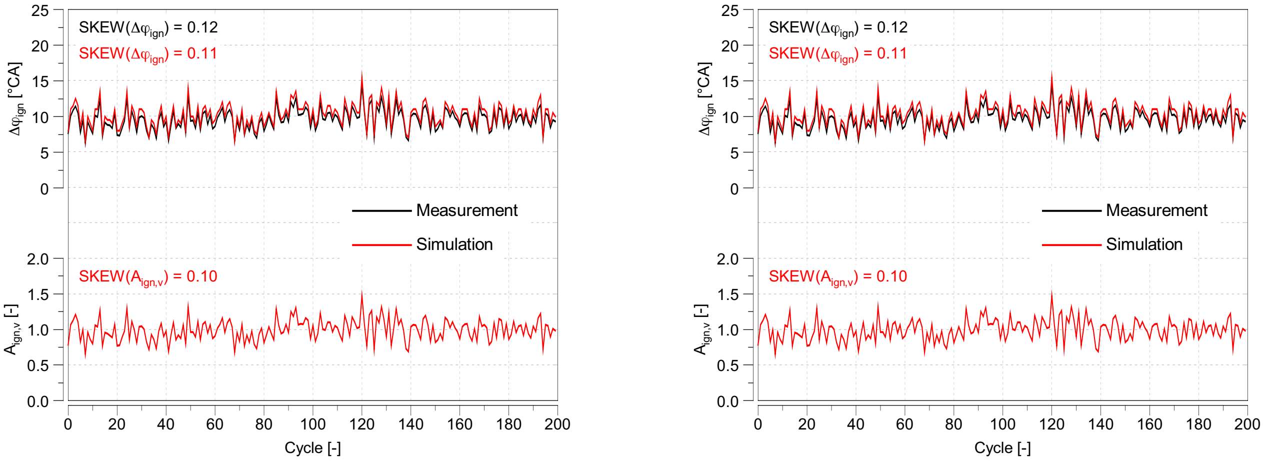

FIGURE 5: Sequence of measured and simulated ignition delay for 200 successive cycles, used variability factors for ignition delay $A_{ign,v}$ and main combustion phase $A_{b,v}$, values of maximum cylinder pressure and crank angle at 50% of mass burned fraction. Engine speed 3000 min-1, $IMEP$ = 5 bar.
OBRÁZEK 5: Sekvence naměřených a simulovaných průtahů zážehu pro 200 po sobě následujících cyklů, použité hodnoty faktorů pro variabilitu průtahu zážehu $A_{ign,v}$ a variabilitu vlastního procesu hoření $A_{b,v}$, maximální spalovací tlaky a úhel natočení klikového hřídele při 50% vyhoření náplně válce. Otáčky motoru 3000 min-1, $IMEP$ = 5 bar.

For purely simulation purposes, some artificial patterns of variability factors are desirable. As the distribution of the variability factor for ignition delay $A_{ign,v}$ is symmetrical, one can write it in symbolic form

$$A_{ign,v} = 1^{+\sigma}_{-\sigma} \tag{13}$$

where s represents a random positive number, which deviates from zero and follows normal distribution with its median in 0. Due to the asymmetrical frequency distribution of the main combustion variability factor $A_{b,v}$ the more detail analysis had to be carried out in computational tool [6] assuming calculation with the median in equation (11). In order to achieve almost normal frequency distribution of the maximum cylinder pressure

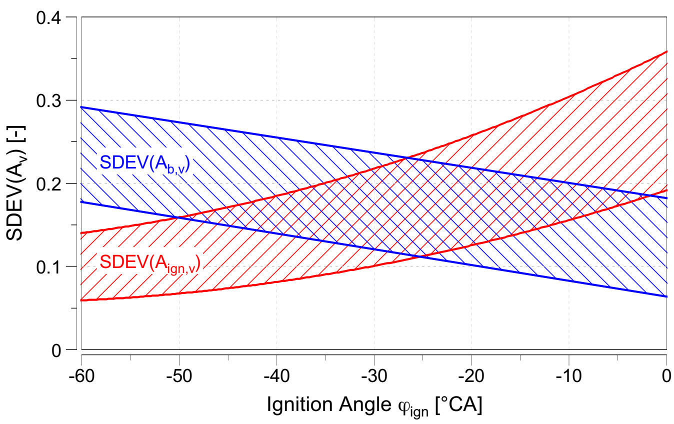

FIGURE 6: Typical values of standard deviation in variability factors $A_{ign,v}$ and $A_{b,v}$ as a function of ignition angle, bands of occurrence.
OBRÁZEK 6: Typické hodnoty směrodatných odchylek faktorů pro variabilitu průtahu zážehu $A_{ign,v}$ a variabilitu vlastního procesu hoření $A_{b,v}$ v závislosti na předstihu zážehu, pásma výskytu.

and other above-mentioned quantities the variation of the combustion variability factor has to follow relation

$$A_{b,v} = 1^{+\frac{\sigma}{1-\sigma}}_{-\sigma} \tag{14}$$

where s represents a random positive number, which deviates from zero and follows normal distribution with its median in 0, s value in equation (14) is independent of that used in equation (13). Note that the number of values lower and higher than 1 is equal, thereby the frequency distribution of the function (14) doesn't exactly coincide with the distribution in Figure 4 but the use of this relation for simulation purposes is quite sufficient.

2.2 MAGNITUDE OF VARIABILITY FACTORS

As a measure of the cycle-to-cycle variability level can be considered the standard deviation in variability factors $A_{ign,v}$ and $A_{b,v}$ whose sequence centric values correspond to the median of value 1, see equations (9) and (11). This approach allows separate assessment of preflame and main combustion phase. The calculation for ignition delay is simple but the standard deviation in combustion variability factor $A_{b,v}$ has to be evaluated for values lower or equal to 1 only due to its asymmetrical distribution

$$SDEV(A_{ign,v}) = \sqrt{\frac{\Sigma\left(A_{ign,v} - 1\right)^2}{n - 1}}$$

$$SDEV(A_{b,v}) = \sqrt{\frac{\Sigma\left(A_{b,v} - 1\right)^2}{n - 1}} \quad \forall \quad A_{b,v} \le 1 \tag{15}$$

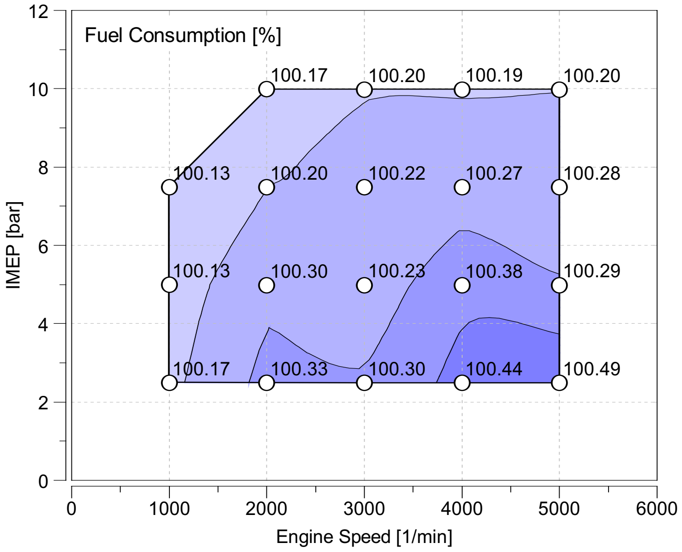

FIGURE 7: Calculated fuel consumption increase due to the cycle-to-cycle variability levels related to Figure 1.
OBRÁZEK 7: Vypočtené navýšení spotřeby paliva vlivem mezicyklové variability dané parametry dle Obrázku 1.

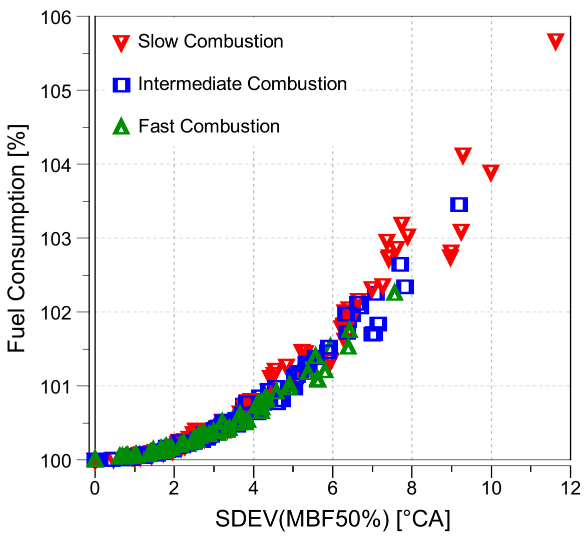

FIGURE 8: Fuel consumption as a function of standard deviation in crank angle at 50% of mass burned fraction, calculation results for red colored engine operational points from Figure 1 considering different combustion rates at optimum spark timing.
OBRÁZEK 8: Spotřeba paliva v závislosti na směrodatné odchylce úhlu natočení klikového hřídele při 50% vyhoření náplně válce, výsledky výpočtů pro červeně označené body v Obrázku 1 s uvažováním různých rychlostí hoření při optimálním předstihu zážehu.

Typical values of standard deviation in variability factors as a function of ignition angle which has been found as a strong influencing parameter are shown in Figure 6. Other parameters like an engine speed, mean indicated pressure, combustion duration, etc. don't show so close correlation. In general, advancing the spark increases the magnitude of ignition delay and also its standard deviation. The shortest ignition delay can be observed at ignition angles around TDC when both in-cylinder pressure and temperature reach the highest values – see the relation in equation (3). The variability factor $A_{ign,v}$ induces relative changes in ignition delay, therefore $SDEV(A_{ign,v})$ is increasing with decreasing of mean value of the ignition delay itself. If the ignition delay is close to zero the changes in the variability factor $A_{ign,v}$ have no impact on the cycle-to-cycle variability level at all.

3. SIMULATION OF CYCLE-TO-CYCLE VARIABILITY

The practical use of the cycle-to-cycle variability simulation is demonstrated on investigation of fuel saving potential by variability level reduction. The reference conditions have been given by cycle-to-cycle variability levels of the SI gasoline engine from Figure 1. All measured engine operational points have undergone thermodynamic analysis in order to get calibration and variability factors using equations (8) – (11). The calibration factors and sequences of variability factors have been then used for fuel consumption calculation at optimum ignition timing for all engine operational points. This calculation has been carried out in modified calculation software [6] for 200 successive cycles. Afterwards, the variability level had been suppressed to zero ($A_{ign,v} = 1$, $A_{b,v} = 1$) and calculated hypothetical fuel consumption at optimum ignition timing was compared with previous results. The outcome of this computational analysis is presented in Figure 7. Displayed values represent fuel consumption increase due to the cycle-to-cycle variability levels related to Figure 1. One can see the maximum fuel consumption saving of magnitude 0.49%. However, this value is not even entirely achievable in practice due to the ultimate minimum variability level.

3.1 GENERALIZATION OF RESULTS

The main goal of the results generalization was to assess a wide range of engine operational points including different combustion processes. A detailed simulation analysis has been carried out at four engine operational points – red colored points in Figure 1. In order to evaluate entire scale of combustion rates it was necessary to generate artificial sequences of the variability factors of different variability levels as an input for the calculation. The standard deviations in variability factors calculated according to (15) have been chosen in range of á0, 0.4ñ with step 0.1. Thereby, the calculation input was formed by the grid of 5x5 variability factor patterns. The artificial generation of the variability factors with respect of relations (13) and (14) has been done in LabVIEW 2010 environment by using function *Discrete Random*. Additionally, in order to consider the influence

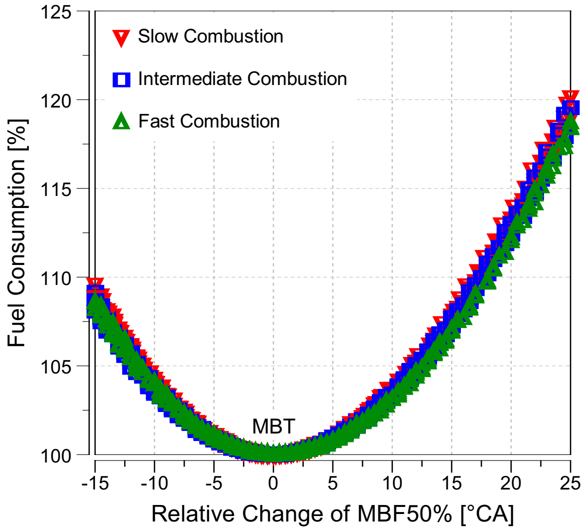

FIGURE 9: Fuel consumption as a function of relative change of crank angle at 50% of mass burned fraction.
OBRÁZEK 9: Spotřeba paliva v závislosti na relativní změně úhlu natočení klikového hřídele při 50% vyhoření náplně válce.

of different combustion rates the calculation was carried out for three different calibration factors A_b = 0.5, 1 and 2 which represent slow, intermediate and fast combustion respectively. The calibration factor for ignition delay was invariable $A_{ign} = 1$. The simulation of 75 events at each of four engine operational points was performed in modified calculation software [6] for 200 successive cycles. The calculation results are summarized in graphical form in Figure 8. The relative fuel consumption 100% is assigned to uniform combustion without any cycle-to-cycle variability implication. From the practical point of view it doesn't matter how the resulting variability level has been achieved – by variation of the ignition delay or of the main combustion phase. Therefore, the impact of the cycle-to-cycle variability on fuel consumption can be simply expressed as a function of standard deviation in crank angle at 50% of mass burned fraction which is easy detectable parameter. One can see that the influence of the combustion rate is marginal. Assuming optimum spark timing and usual values of $SDEV(MBF50\%)$ = 3°CA a theoretical potential for fuel consumption improvement in terms of cycle-to-cycle variability reduction can be estimated to 0.4% only. This value corresponds to the results in Figure 7.
Above mentioned marginal influence of the combustion rate can be explained by following computational analysis at four red colored engine operational points depicted in Figure 1. The engine fuel consumption in steady state mode was calculated with various spark timing for three different combustion rates. The results after recalculation to relative change of the crank angle at 50% of mass burned fraction are shown in Figure 9. The minimum fuel consumption corresponds to maximum brake torque spark timing (MBT), as expected. A leaning of the angle $MBF50\%$ either to lower or higher values leads to fuel consumption increase almost independent of the engine load or combustion rate magnitude like that in Figure 8. Therefore, the chart in Figure 8 can be considered as generally valid for SI gasoline engines with optimum spark timing. If the ignition timing is far from its optimum, the cycle-to-cycle variability level doesn't influence engine fuel consumption so much, because the relationship between relative change of the angle $MBF50\%$ and fuel consumption is nearly linear as shown in Figure 9. The location of the mean value of the angle $MBF50\%$ affects the fuel consumption almost an order of magnitude greater than a cycle-to-cycle variability. If a knocking is considered the situation is more complex. Under knocking conditions, the spark advance has to be retarded, i.e. the angle $MBF50\%$ is shifted from MBT point to higher values, which leads to significantly higher fuel consumption. Since the knock event is related to stochastic nature of combustion the reduction of cycle-to-cycle variability suppresses a knock occurrence and thus allows advancing combustion [8].

4. CONCLUSION

The cycle-to-cycle variability is often discussed topic related to overall engine efficiency and thereby a subject for optimization. The simulation of the cycle-to-cycle variability is possible by using empirical adaptive combustion model extended by variability factors. Individual patterns of variability factors for ignition delay and main combustion phase, which are independent of each other, can be either measured or artificially generated according to required variability level. This approach leads to very good agreement with real engine behavior.
Although it is technically possible to reduce cycle-to-cycle variability the carried-out sensitivity analysis without knock limitation at optimum spark timing shows that achievable engine fuel consumption reduction is up to 0.4% only. On the other hand, with an increasing variability the engine efficiency deterioration is more progressive, thus should be avoided. The right spark timing regarding the mean value of the crank angle at 50% of mass burned fraction has much stronger effect on engine fuel consumption than a cycle-to-cycle variability.

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