

Knock in an Ethanol Fueled Spark Ignition Engine: Detection Methods with Cycle-Statistical Analysis and Predictions Using Different Auto-Ignition Models

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ABSTRACT

Knock is studied in a single cylinder direct injection spark ignition engine with variable intake temperatures at wide open throttle and stoichiometric premixed ethanol-air mixtures. At different speeds and intake temperatures spark angle sweeps have been performed at non-knocking conditions and varying knock intensities. Heat release rates and two zone temperatures are computed for both the mean and single cycle data.

The in-cylinder pressure traces are analyzed during knocking combustion and have led to a definition of knocking conditions both for every single cycle as well as the mean engine cycle of a single operating point. The timing for the onset of knock as a function of degree crank angle and the mass fraction burned is determined using the "knocking" heat release and the pressure oscillations typical for knocking combustion.

A detailed chemical kinetic model for ethanol combustion is used to compute ignition delay times (IDT) for stoichiometric ethanol-air mixtures at pressures and temperatures relevant to the conditions measured on the engine test bench. A multi-step Arrhenius type correlation has been fit to the data including the observed flattening of the IDT for ethanol at relatively low temperatures (<850K) and compared to other data available in literature.

Empirical knock prediction models available in literature are tested against the available measurement data and improvements to the models are formulated. The importance of accurate IDT values as well as a model for the reducing probability of knock towards the end of combustion for the precision of a knock model is illustrated.

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INTRODUCTION

Many studies have focused on the benefits of ethanol compared to gasoline as a fuel for spark ignition engines. The higher latent heat of vaporization and the excellent knock resistance allow dedicated ethanol engines to reach efficiencies of over 40% [1]. Even though the knock resistance of ethanol is excellent, the future design of ethanol engines will benefit greatly of the thorough understanding of knock phenomena and the availability of accurate knock prediction methods for these engines.

Knocking combustion is characterized by high frequency pressure oscillations resulting from auto-ignition of the unburned mixture in front of the flame front of the main combustion in spark-ignition engines. Because of the sensitivity to pressure and temperature, knock is often related to

advances in spark angle, high compression ratios or turbo-charging. When the conditions in the unburned zone are favorable, the unburned mixture can auto-ignite. This happens typically at so called hot-spots in the end gas: inhomogeneities in the temperature and mixture distribution in front of the flame affect also these auto-ignition locations as well as other mechanisms such as lube-oil droplets and hot surfaces.

In production engines knock detection is typically done using an acceleration sensor mounted to the engine block. For research or development purposes in-cylinder pressure sensors are mainly used to see the characteristic high frequency oscillations. Numerous studies have focused on the detection of knock and have correlated the measured quantities to the knock intensity. Most of these studies however have used gasoline or a gasoline surrogate. For ethanol knock detection little publications can be found

Measurements using a wide range of operating parameters can be used to evaluate and calibrate predictive knock models. A knock model should correctly predict the onset of knock timing for a given engine cycle. Livengood and Wu [2] used the assumption that the auto-ignition chemistry for knock is cumulative. Under this assumption the timing of the onset of knock (SOK) can be predicted by integrating the inverse of the IDT. Accurate ignition delay time correlations are thus of crucial importance for the correct prediction of knock. In this paper overviews of available knock models and ignition delay time correlations are given and the knock models are tested against measurements on a single cylinder engine test bench using stoichiometric ethanol/air mixtures with partially poor agreement. Improvements concentrating on the ignition delay time correlation for ethanol at engine relevant conditions as well as the influence of the thermal boundary layer of the walls on the probability of knock are suggested.

The paper is organized as follows. First the measurements are presented and serve as the basis for the definition of a knock detection method for both single and mean cycles. A cycle-statistical analysis on the knock intensity of the single cycles is carried out. Next, chemical kinetic calculations for the IDT of ethanol are carried out and lead to a new correlation for the IDT. It is shown in a last step that the new formulation for the IDT improves significantly the accuracy of the given knock models. Finally a knock model is calibrated for the single cycles. Using the cylinder geometry data, the importance of using a formulation for the reducing probability of knock towards the end of the cycle due to the vicinity of the walls is illustrated.

MEASUREMENTS AND EXPERIMENTAL SETUP

Experimental Setup

Measurements have been carried out on a Swissauto Wenko 250cc single cylinder 4 stroke spark ignition engine with the specifications given in Table 1. The engine is equipped with a direct injection system using a Bosch HDEV 5 injector located in between the intake valves. The start of injection is timed at EVC to allow enough time for evaporation and mixing. The combustion chamber has a pent roof with the spark plug located between the four valves, slightly off center. A sketch of the combustion chamber with the piston at top dead center can be seen in Figure 1, including the propagation of a spherical flame front.

The intake air temperature is controlled with an electric heater in the intake path and allows temperatures up to 120°C. The in-cylinder pressure profiles (144 cycles per operating point) are recorded every 0.2°CA with a piezo-electric pressure sensor (Kistler 6052). Intake and exhaust pressure profiles are recorded as absolute pressures with a Kistler 4075 and 4049 respectively. Fuel consumption is measured using a coriolis mass flow meter (Endress+Hauser Cubemass). The engine is connected to an APIcom FP75 eddy current brake. The engine

control unit is a Bosch MS4 Sport. All temperatures are measured using radiation protected thermocouples. A sketch of the engine test bench including the sensor positions is shown in Figure 2.

Table 1. Engine specifications.

Displaced volume	250 cc
Stroke	56.5 mm
Bore	75 mm
Connecting Rod	95 mm
Compression ratio	12.5:1
Number of Valves	4
Number of Cylinders	1
Fuel Injection	DI (110 bar)

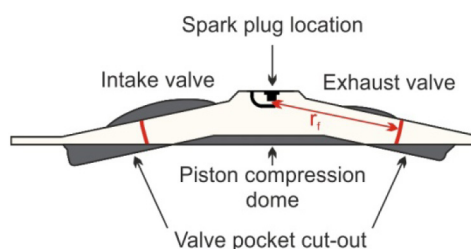


Figure 1. Sketch of the combustion chamber with the piston at top dead center and a spherical flame front with radius, r_f . The view from the side wall is cut through the symmetry plane containing intake and exhaust pipe and spark plug.

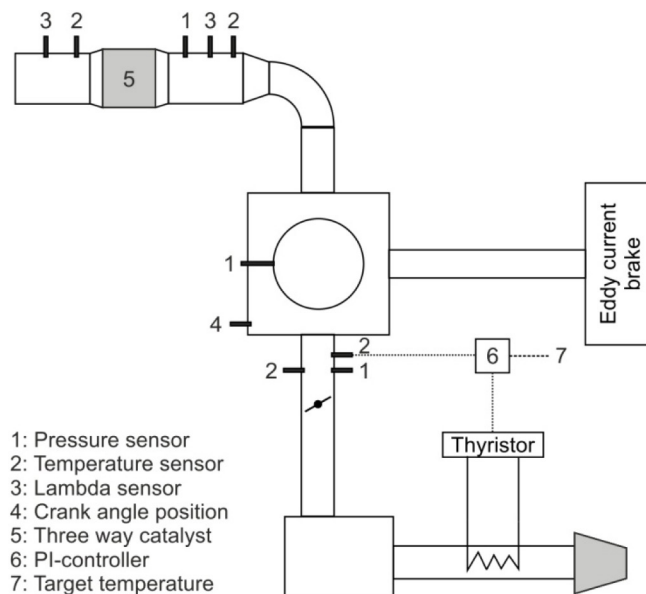


Figure 2. Sketch of the engine test bench with the main sensor positions.

Hydrous ethanol (6% water content) is chosen as a fuel. All operating points have a stoichiometric (lambda-controlled, Bosch LSU 4.9) homogeneous mixture. The engine is, for the current purpose, operated at wide open throttle position and at steady state. A series of measurements has been carried out for the operating points listed in Table 2.

Table 2. Measurement operating points.

Speed [rpm]	T_{in} [°C]	Spark advance [°CAbTDC]
3000	80	45.00 – 60.00
	100	35.25 – 60.00
	120	30.00 – 50.25
4000	80	33.75 – 44.25
	100	30.00 – 43.50
	120	33.75 – 47.25
4500	80	35.25 – 53.25
	100	33.75 – 44.25
	120	31.00 – 41.25
5000	80	33.75 – 41.25
	100	33.75 – 43.50
	120	33.75 – 41.25
5500	80	30.00 – 47.25
	100	30.00 – 41.25
	120	28.75 – 33.75

At each combination of speed and intake temperature a spark advance variation is carried out starting from very late (non-knocking condition) up to strong knock at earlier spark timings.

Heat Release Rate Calculation

To understand and study the knocking behavior in a spark ignition engine, the temperature of the unburned mixture is an important quantity. It is important to take the correct assumptions for the cylinder heat transfer and wall temperatures.

For the heat release rate and unburned temperature calculation the commercial software GT-Power by Gamma Technologies is used. A 1D model consisting of the intake, exhaust and cylinder geometry as well as the valve lift profiles is built up. The recorded intake and exhaust pressures and temperatures are imposed at the boundaries of the model to adequately capture the gas dynamics in the intake and exhaust path. A thermal model is solved imposing the heat transfer coefficients on the outside of the intake/exhaust path, the coolant-liner, coolant-head, oil-liner and oil-piston interfaces. Conduction between the cylinder head and intake/exhaust path is allowed. For the heat transfer coefficient inside the cylinder the Woschni model is applied. In this way the wall temperatures are results of the calculation. The injection profile is imposed. For the evaporation of the fuel a model based on the 50% evaporation time, CA_{50} , is used (Eq. (1)). The evaporation rate increases with temperature and speed. To model wall impingement a factor for the heat taken from the wall is included.

$$Evaporation\ rate = \frac{4.16 \cdot 4000rpm}{CA_{50}} \left(\frac{T}{600K} \right) \left(\frac{n}{4000rpm} \right) \quad (1)$$

The heat transfer coefficients and evaporation profile coefficients are the unknown parameters that have to be estimated. The cooling circuit of the engine is driven by an external pump and the flow is independent of operating condition. The evaporation time constant has little influence on the unburned temperature. To calibrate the heat transfer coefficient and the wall impingement factor two criteria are used: the convergence of the heat release rate calculation and the calculated temperature in the model for the cylinder head close to the intake valves as illustrated in Figure 3.

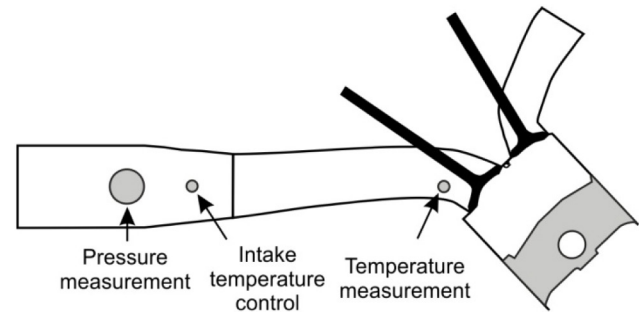


Figure 3. Intake temperature measurement locations.

The simulation performs both a 'forward' and 'backward' run. The backward run starts at IVC and calculates the heat release rate from the experimental filtered (low pass 5 kHz) pressure profile, $\hat{p}$ using a 2-zone model. It uses the results from the forward run for the conditions at IVC and only covers the high pressure cycle. The forward run uses the heat release rate computed in the backward run and covers the complete 720°CA. The calculation converges when the calculated pressure trace from the forward run matches the measured profile and the predicted amount of fuel burned equals the available fuel mass. Additional to these convergence criteria, the temperature in front of the intake valves must match the measured value. A summary of the calculation method and assumptions can be seen in Figure 4. The heat release rate and unburned temperature calculations are carried out for both the mean cycle (averaged over 144 cycles for one operating point) as well as each single cycle.

ANALYSIS OF MEASUREMENT DATA

Spark angle variations have produced measurement data with different knock intensities. A definition is needed to distinguish between knocking and non-knocking cycles and operating points. A definition for the timing of the onset of knock is also needed. This chapter first gives an overview of the existing knock detection methods and consequently describes the definitions used in this work. In the last sub-chapter a cycle-statistical analysis is carried out.

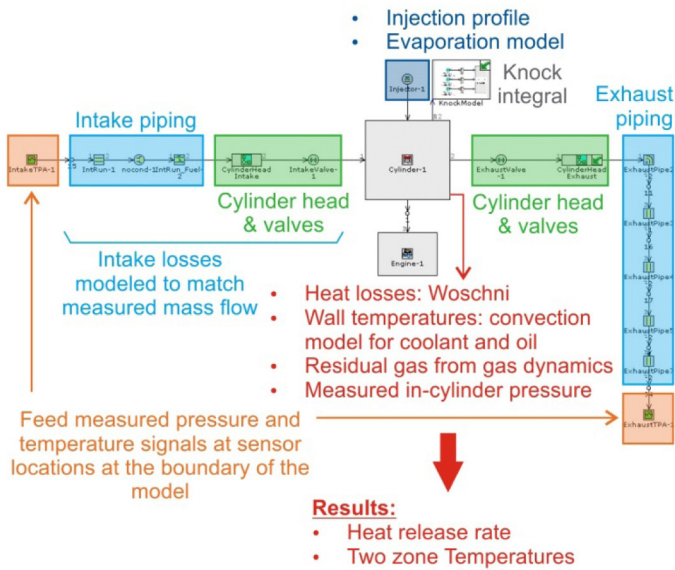


Figure 4. Heat release rate calculation: method and assumptions.

Knock Detection Methods

Knock detection can typically be done using the pressure signal trace using an in-cylinder pressure sensor. Different knock indices for the detection of knock have been proposed. Most of these are based either on the in-cylinder pressure signal or on the heat release rate. The heat release rate is the result of calculations with the pressure signal as an input and both are of course related. In this description they are however treated separately.

Knock is characterized by the typical pressure oscillations caused by the auto-ignition of the unburned mixture in front of the flame. Often a high-pass or band-pass filter in the region of about 4-25 kHz is applied to the in-cylinder pressure signal, obtaining just the knock induced oscillations in the signal, $\tilde{p}$. The actual frequencies of these oscillations depend on the size of the engine as the waves travel with the local speed of sound [3]. In our case a high pass filter of 5 kHz has been applied to all in-cylinder pressure measurement data to extract all the relevant information corresponding to the characteristic pressure oscillations during knocking combustion.

Most often the maximum amplitude of the pressure oscillations (MAPO) is used as a measure for knock intensity (Eq. (2)) [4]. Next to the maximum amplitude, the integral of the modulus of pressure oscillations (IMPO) is used as shown in Eq. (3) with θ the crank angle, θ_0 the spark timing and ζ the integration window [5]. This measure considers both the amplitude and the time duration of the oscillations.

$$MAPO = \max(|\tilde{p}|) \quad (2)$$

$$IMPO = \int_{\theta_0}^{\theta_0+\zeta} |\tilde{p}| d\theta \quad (3)$$

In [6] the signal energy of the pressure oscillations (SEPO) or also the resonance energy, E_{res} (Eq. (4)), is used to compute an experimental knock indicator as the logarithm of the ratio of the signal energy of the pressure oscillations to the motored pressure curve.

$$E_{res} = \int_{\theta_0}^{\theta_0+\zeta} \tilde{p}^2 d\theta \quad (4)$$

Another way to detect knock is to integrate the power density spectrum of the Fourier transform over a certain frequency band [7]. The method used by Wayne et al. [8] is the value of the third derivative of the pressure signal. The crank angle for the onset of knock was determined by a first value crossing. The knock intensity was determined by the amount of energy left in the cylinder.

A knock detection method or knock index can also be based on the heat release rate. In this case the heat release rate for each single cycle is evaluated. In [9] a knock detection method based on the filtered heat release rate $d\tilde{Q}/d\theta$ is proposed. The high frequency content of the heat release rate is evaluated and a value similar to E_{res} is computed. In this case it can be called the signal energy of the heat release rate oscillations (SEHRRO). The onset of knock is detected by comparing the value of SEHRRO before and after auto-ignition. Corti et al. [10] find that the pressure oscillations are the 'mechanical signature' of the knocking combustion. The underlying cause of the pressure oscillations is the heat release in front of the mean flame front. Non-knocking cycles are approximated using two Wiebe functions. When large oscillations in the heat release rate exist, a third Wiebe function is fitted to model the knocking combustion. The knock intensity is related to either the maximum value or the integral of the third Wiebe function. Also in [11] it is found that the high frequency content is characteristic for knocking combustion. It is however the result of the rapid heat release and it is dependent on cylinder geometry. The authors propose a knock detection method based on the frequency content of the unfiltered heat release rate of the single cycles trying to identify the heat release originating from knocking combustion. The frequency band between 30 and 200 oscillations/cycle is found to yield a good indication of knock intensity and is used as the knock detection method.

The methods based on the cylinder pressure are applied to the measurements in this study. For all operating points and each individual cycle the values for MAPO, IMPO, E_{res} and

$d^3\tilde{p}/d\theta_{max}^3$ are computed. A comparison of the different values is shown in Figure 5. The correlation coefficients, r , that shows the linear correlation between two values, are high, which shows that the values correlate well. The different detection techniques are thus robust and comparable to each other. It was decided not to include the available detection mechanisms based on the heat release rate here. The methods based on the high frequency content of the heat release rate are similar to those based on the high frequency part of the pressure signal.

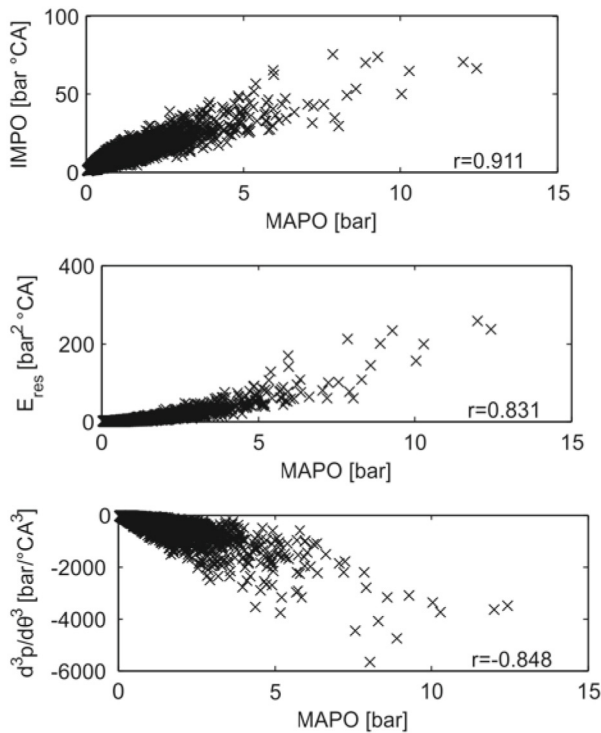


Figure 5. Comparison of knock indices used in literature and their correlation coefficients: $MAPO$ (Eq. (2)), $IMPO$ (Eq. (3)), E_{res} (Eq. (4)) and the maximum of the third derivative of the pressure oscillations.

The practical feature of knock, causing damage to the engine is the pressure oscillation and corresponding vibration. Therefore in this paper the $MAPO$ value is used to determine whether a cycle knocks or not. For the purpose of this work it is not necessary to determine the severity of knock, but only the timing when a cycle knocks.

To rule out the effect of the operating point on the knock detection method, the $MAPO$ values are relative to the mean value of the operating point with the latest spark timing for that speed and intake temperature as in [4]. After an analysis of the data, it was found that $MAPO_{rel} > 4$ is a good criterion for a knocking cycle, because it differentiates from the signal noise and clearly identifies knocking combustion.

Onset of Knock Timing Detection

Only when a cycle is detected as a knocking cycle, the timing for the onset of knock can be determined. The start of the pressure oscillations does not determine the timing for the onset of knock. It is rather the auto-ignition and corresponding heat release preceding the oscillations that trigger the knocking combustion. This can be seen when $\tilde{p}$ and the heat release rate for a single cycle are compared. Figure 6 shows the high pass filtered pressure referenced to the value for late spark timings $\tilde{p}_{rel}$ and the heat release rate for the cycle based on the low pass pressure signal, $d\hat{Q}/d\theta$. During the descending of the HRR a second increase in the HRR is seen around 5°CAaTDCf. This increase is due to the auto-ignition of the mixture in front of the main flame front. It can be seen very clearly, how the “knocking” HRR occurs before the pressure oscillations start. This is only true for cycles that have strong pressure oscillations. Cycles on borderline knock do not have a second increase in HRR. An example is given in Figure 7. The start of knock (SOK) indices based on the pressure oscillations “max” (overall maximum value) and “lim” (first time higher than 4) are close together. In the HRR a flattening can be observed around 4°CAaTDCf. This can be attributed to the knocking combustion. The typical fast decrease in HRR after auto-ignition can be observed as well. The SOK location based on the HRR is therefore defined as the local maximum in the derivative of the HRR.

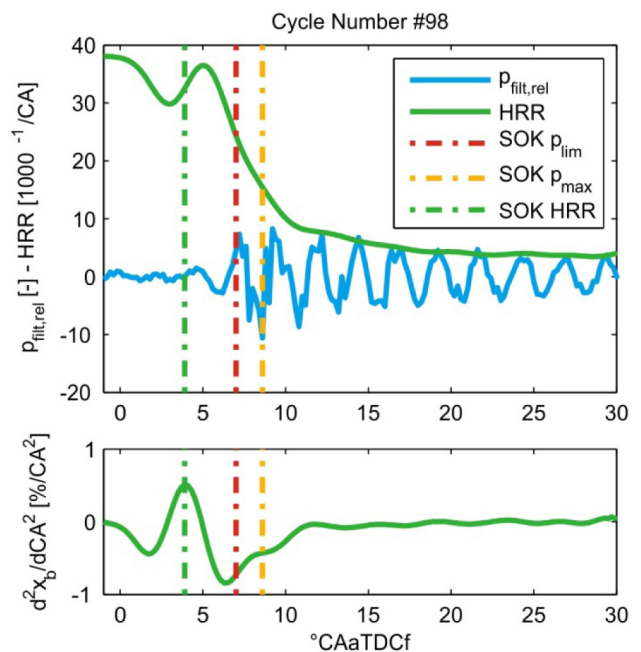


Figure 6. Heat release rate and first derivative, pressure fluctuation amplitude and different start of knock criteria for knocking cycle - strong knock. Measurement operating point: 3000rpm, $T_{in}=100^{\circ}\text{C}$, 55.50°CA spark advance.