

Formulation of a Knock Model for Ethanol and Iso-Octane under Specific Consideration of the Thermal Boundary Layer within the End-Gas

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ABSTRACT

Knock is often the main limiting factor for brake efficiency in spark ignition engines and is mostly attributed to auto-ignition of the unburned mixture in front of the flame. In order to study knock in a systematic way, spark angle sweeps with ethanol and iso-octane have been carried out on single cylinder spark ignition engine with variable intake temperatures at wide open throttle and stoichiometric premixed fuel/air mixtures.

Much earlier and stronger knock can be observed for iso-octane compared to ethanol at otherwise same engine operating conditions due to the cooling effect and higher octane number of ethanol, leading to different cycle-to-cycle variation behavior.

Detailed chemical kinetic mechanisms are used to compute ignition delay times at conditions relevant to the measurements and are compared to empirical correlations available in literature. The different correlations are used in a knock model approach and are tested against the measurement data. The importance of using accurate ignition delay time expressions in predicting the correct timing for the onset of knock is illustrated for both ethanol and iso-octane.

The probability of the occurrence of knock is significantly reduced towards the end of the cycle. A new model approach for the thermal boundary layer close to the cylinder walls is included in the knock integral to take into account its effect on the knock probability thus improving significantly the accuracy of the knock prediction. The formulation of the knock model can be derived from the geometry of the combustion chamber and includes its specific shape.

CITATION: Steurs, K., Blomberg, C., and Boulouchos, K., "Formulation of a Knock Model for Ethanol and Iso-Octane under Specific Consideration of the Thermal Boundary Layer within the End-Gas," *SAE Int. J. Engines* 7(4):2014, doi:10.4271/2014-01-2607.

INTRODUCTION

The automotive industry is experiencing a serious trend towards ever more economic and ecologic vehicles. The demand for cars with very low fuel consumption and CO₂ emissions is increasing and the regulations on pollutant emissions are becoming more and more stringent. The use of alternative renewable fuels is an attractive option to meet these targets. Ethanol has gained interest over the last years with increased global production [1], mainly due to its excellent knock resistance and the possibility to be produced in a renewable way. Currently ethanol is mainly produced from corn (USA) or sugar (Brazil) and is in conflict with the food market, but predictions foresee an increase in the production of ethanol of the 2nd generation, based on lignocellulosic materials [2].

Many researchers have tested SI engines using different ethanol blends and compared the operation to regular gasoline operation in terms of fuel consumption, power output and brake

efficiency. There are three common ways of comparing data that yield different statements. Comparisons at wide open throttle in which case an increase in torque output is usually observed at higher ethanol contents [3, 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15]. Due to the excellent knock resistance of ethanol, maximum brake torque (MBT) timing can often be sustained at wide open throttle (WOT), where gasoline is knock limited. Because of this and the overall lower combustion temperatures, there is no need for fuel enrichment. Achieving MBT timing with ethanol where this is not possible with gasoline yields a significant increase in both brake power and brake thermal efficiency (BTE). On top of that, the compression ratio for the ethanol fueled engine can be increased to obtain even higher BTE. The efficiency increase stagnates when MBT can no longer be reached at full load. A second comparison that can be made, is the case where the brake torque is kept constant [16]. This is a good comparison for part load behavior. Due to the higher torque production of ethanol, pumping losses increase and lower efficiency benefits are observed. A last

comparison is made when there is no closed loop mixture control [17, 18, 19, 20, 21]. In this case, the air-to-fuel ratio changes, due to constant injection timing for different ethanol blends. Ethanol has a lower energy density and for increasing ethanol content the mixture becomes leaner. Due to the leaning effect a decrease in torque is observed. The volumetric efficiency and the BTE improve.

The aforementioned studies mainly use port fuel injection (PFI). The charge cooling effect of ethanol is much more pronounced when using direct injection (DI) [22, 23] and consequently there are possibilities for gains in brake efficiency and power, but also in improvement of the cold start behavior [24, 25, 26, 27, 28, 29, 30, 31, 32, 33, 34, 35]. The low volatility of ethanol can cause improper mixing and evaporation and the production of soot particles. The specific properties of ethanol allow high degrees of turbocharging and exhaust gas recirculation (EGR). Turbocharging can be used due to the excellent knock resistance of ethanol to increase power output or allow a high degree of downsizing. In order to reduce combustion temperatures and consequently avoid knock and high NO_x emissions, EGR can be used. Several concepts have been introduced that combine high compression ratios, turbocharging and EGR to obtain peak brake efficiencies higher than 40% for high ethanol fuels [36, 37, 38].

From the presented overview it is apparent that there is a large interest in ethanol fueled engines. The main benefits of ethanol include the potential in terms of energy efficiency and CO₂ savings. The excellent knock resistance of ethanol allows for high efficiency concepts at high compression ratios. In order to fully exploit the specific properties of ethanol as a fuel, dedicated ethanol engines need to be designed. These engines can offset, to some extent, the lower energy density with higher brake efficiency, compared to gasoline. Dedicated ethanol engines should consider the knock limits of ethanol. For the design of these engines, a fundamental understanding of the behavior of ethanol under knocking conditions is of crucial importance.

The subject of this work is therefore the study of knock in an ethanol fueled engine. The results are compared to the behavior of iso-octane under the same conditions. Iso-octane is a primary reference fuel, a common surrogate for gasoline and relatively well studied in terms of chemical kinetic models and measurements. Extra attention is given to the cycle-to-cycle variations within one operating point. A cycle-resolved analysis shows in which cycles auto-ignition is initiated. 0D/1D simulation tools are often used in the preliminary design phase of spark ignition engines. These tools rely heavily on empirical and phenomenological models to predict the performance of a simulated engine. Fast and reliable models for the prediction of knock are of crucial importance for the accuracy of the results. The data gathered in this study are therefore used to assess the quality of the results of the knock models available in literature. Improvements to the formulations are presented that include the observations that are made specific to the influence of the combustion chamber geometry on the knock

phenomenon. The current study refrains from using expensive 3D calculations and hardware with optical access and shows that with standard equipment accurate knock predictions for different fuels can be made.

The structure of the paper is such that each chapter first gives a literature review on the subject and explains the state of the art. Then the methodology that is used in this paper is presented. The results corresponding to the methodology are discussed at the end of each chapter and focus on the comparison between ethanol and iso-octane. The different topics that are discussed include:

- In a first chapter the experimental setup and the operating conditions for the measurements are introduced.
- In a second chapter the algorithm for the heat release rate and two-zone temperature calculation is explained and a sensitivity study is carried out.
- In the third chapter a knock detection criterion is selected including a definition for the timing for the onset of knock. The criterion is applied to the available measurement data and a cycle-statistical analysis is presented.
- Chapter four presents a literature review on the available data for the ignition delay times. Chemical kinetic calculations are carried out and compared to published correlations.
- In a last chapter the different correlations are tested in knock integral formulations to illustrate the importance of detailed ignition delay time data for knock modeling. The necessity to include a model for the influence of the combustion chamber walls is observed. A new model approach, based on the mass fraction of unburned gas in the thermal boundary layer, taking into account the specific geometry of the combustion chamber is introduced.

EXPERIMENTAL SETUP, FUEL SELECTION AND MEASUREMENTS

Experimental Setup

The engine under consideration for this study is a single cylinder 4-stroke spark ignition engine manufactured by Swissauto Wenko AG with the specifications given in [table 1](#). In the original configuration, using gasoline PFI, the engine produces a maximum of 13.67 kW at 6000 r/min and 22.31 Nm at 5000 r/min.

Originally the engine is equipped with a PFI system using an injection pressure of 4 bar. To avoid problems with the evaporation of ethanol and to fully benefit of its high latent heat of vaporization, a direct injection system is designed and installed on the engine. A Bosch HDEV5 six-hole injector is located in between the intake valves and fuel is injected with a pressure of 110 bar. The fuel injection for DI starts at the timing of the exhaust valves closing. In this way no fuel is lost directly into the exhaust and maximum time is available for mixing and evaporation. The intake valves are opened during the injection and the incoming air interacts and mixes with the fuel spray.

Because the PFI injector is located well before the intake valves, the timing for the end of injection is fixed to 320°CAbTDCf. The fuel pressure is supplied externally in both cases using a Bosch FP100 pump for PFI and a three cylinder radial piston pump for the high pressure in the DI system.

Table 1. Test engine data.

Displacement	250 cc
Stroke	56.5 mm
Bore	75 mm
Connecting rod	95 mm
Number of valves	4
Number of cylinders	1
Compression ratio	12.5:1
Injection mode	DI (Ethanol) – PFI (Iso-octane)

The engine test bench is designed and equipped to meet the requirements for the investigation of knock. A sketch of the test bench setup as well as the main sensor and component positions can be seen in figure 1. The in-cylinder pressure is recorded with a piezo-electric pressure sensor (Kistler 6052) located between the intake and exhaust valves on the chain side of the cylinder head. The intake and exhaust pressure profiles are recorded using piezo-resistive pressure sensors, Kistler 4075A and the cooled Kistler 4049A respectively, for absolute pressure measurements. The fuel pressure in the DI system is measured using a Kistler 4065A. The piston position, as well as the different pressures, are recorded every 0.2°CA.

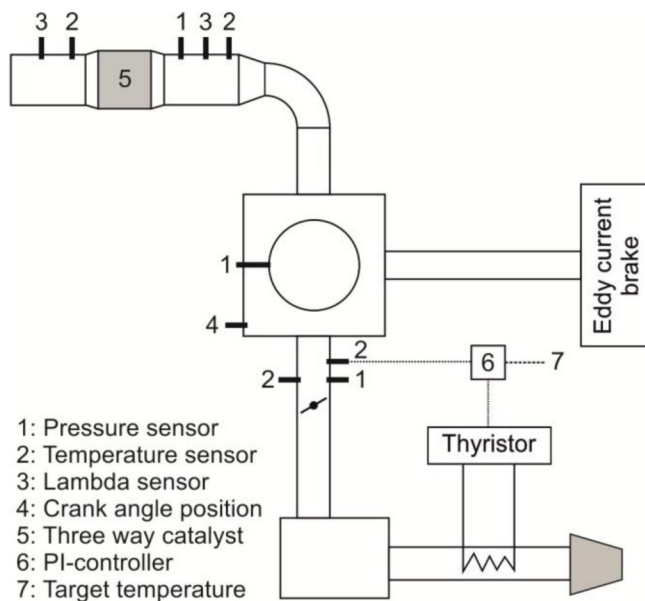


Figure 1. Schematics of the engine test bench and the main sensor positions.

The injection and ignition events are controlled using the engine control unit (ECU) Bosch MS4 Sport. A Bosch LSU 4.9 lambda-probe is installed in the exhaust for mixture control. A three way catalyst (TWC) reduces the pollutant exhaust emissions and a muffler reduces the exhaust noise. Fuel consumption is

measured using a coriolis mass flow meter. Multiple thermocouples are placed in the intake and exhaust paths and in the cooling water circuit for temperature measurements.

An air-box and heater are installed in the intake system to control the temperature of the air entering the cylinder. A single phase 400 V resistive heater of maximum 2.5 kW heats up the air. A PI-controller for the temperature after the throttle body uses a thyristor to regulate the power of the heater. Variable intake temperatures, T_{in} of up to 120°C can be achieved with this setup.

Measured Operating Points

The goal of the measurements is to gather a large set of data at different knock intensities for both fuels. Experiments are carried out for different speeds and intake temperatures. The spark timing is varied from late non-knocking conditions to early timing and strong knock. All measurements are conducted at steady state and wide open throttle. The air-to-fuel ratio is constant at stoichiometric conditions. At each operating point 288 consecutive cycles are recorded. An overview of all operating conditions is given in table 2. For ethanol at atmospheric conditions, even with very early spark timings, no knock is observed. Measurements at these conditions for ethanol are omitted. The engine speeds for the selected operating points are relatively high. The original design and purpose of the engine is aimed at operation at high speeds. Below 3000 r/min the maximum torque of the engine is low and the cycle-to-cycle variations are high, resulting in unstable operation.

Table 2. Operating conditions for the experiments.

Speed [r/min]	Intake temperature [°C]	Spark advance [°CAaTDCf]	
		Ethanol	Iso-octane
3000	35	-	41.25 - 54.75
	80	45.00 - 60.00	36.75 - 50.25
	100	35.25 - 60.00	35.25 - 48.00
	120	30.00 - 50.25	30.75 - 42.00
4000	35	-	39.00 - 52.50
	80	33.75 - 44.25	25.50 - 39.00
	100	30.00 - 43.50	26.25 - 37.50
	120	33.75 - 47.25	26.25 - 37.50
4500	35	-	33.00 - 46.50
	80	35.25 - 53.25	32.25 - 47.25
	100	33.75 - 44.25	29.25 - 42.75
	120	31.00 - 41.25	24.00 - 37.50
5000	35	-	38.25 - 51.75
	80	33.75 - 41.25	24.75 - 38.25
	100	33.75 - 43.50	26.25 - 37.50
	120	33.75 - 41.25	27.00 - 36.00
5500	35	-	46.50 - 60.00
	80	30.00 - 47.25	32.25 - 45.75
	100	30.00 - 41.25	27.00 - 40.50
	120	28.50 - 37.75	24.75 - 36.00

Fuel Selection

Ethanol and iso-octane are selected as the fuels for this study of knock. The main properties of the selected fuels are listed in [table 3](#). The energy density of ethanol is much lower than that of iso-octane. To obtain the same amount of energy per cycle, 66% more ethanol has to be injected compared to iso-octane. The latent heat of vaporization of ethanol is much higher than that of iso-octane. As a consequence the heat required to evaporate the same amount of fuel is three times higher for ethanol compared to iso-octane. The heat required to evaporate a stoichiometric mixture with the same amount of air is 4.89 times higher for ethanol compared to iso-octane. When all heat, required to evaporate the fuel, is taken from the incoming air, the reduction in temperature is 101.7°C for ethanol and 20.78°C for iso-octane. Cooling down a stoichiometric mixture of 150°C with evaporation of the fuel increases the charge density by 31% for ethanol and by only 5% for iso-octane. An accurate design of the injection system can thus work out as an effective supercharging for ethanol. Ethanol has a significantly higher laminar flame speed compared to iso-octane [39, 40, 41, 42, 43]. The laminar flame speed is however very sensitive to the temperature. Additionally, the flame speed that determines the combustion rate is the turbulent flame speed. Differences in temperature and turbulence can offset the nominally higher flame speed for ethanol. The research octane number (RON) of ethanol is higher than that of iso-octane, due to the better knock resistance of ethanol.

Table 3. Fuel properties [44].

Property	Unit	Ethanol	Iso-octane
Chemical formula	-	C ₂ H ₅ OH	C ₈ H ₁₈
Molecular weight	g/mol	46.07	114.23
Density	kg/l	0.794	0.692
Latent heat of vaporization	kJ/kg	920	307
Lower heating value	MJ/kg	26.75	44.42
Stoichiometric air to fuel ratio	-	9	14.7
RON	-	107-109	100

Measurement Results

[Figure 2](#) shows the measured torque for the measurements. The dashed lines represent the results for ethanol and the full lines for iso-octane. The curves show the typical parabolic shape as expected from spark timing variations. There is a clear difference in the torque output between the two fuels for the same operating point. For the same speed, intake temperature and spark timing, ethanol has a much higher torque than iso-octane. This observation can be attributed mainly to the difference in latent heat of vaporization. The cooling effect for ethanol is much higher than for iso-octane. Additionally, the injection of iso-octane was PFI. The PFI injector is located well before the intake valves so significant

wall wetting can be expected and thus the effect of charge cooling is lower. Whether the cooling effect associated with the fuel evaporation has an influence on the volumetric efficiency also depends on the operating point. For some speeds (e.g. 5500 r/min) the effect is strong, for others (e.g. 3000 r/min) the cooling effect on the volumetric efficiency is lower.

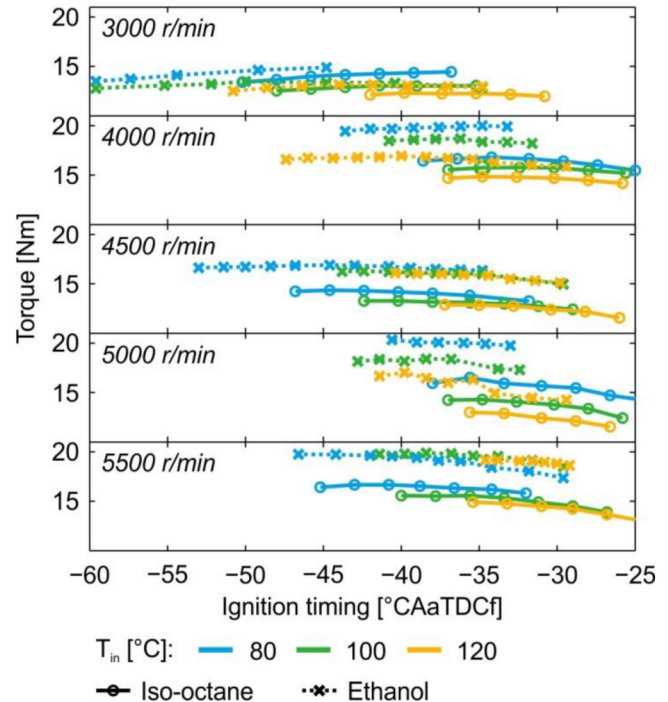


Figure 2. Torque as a function of spark advance for iso-octane (PFI) and ethanol (DI) at different speeds and intake temperatures.

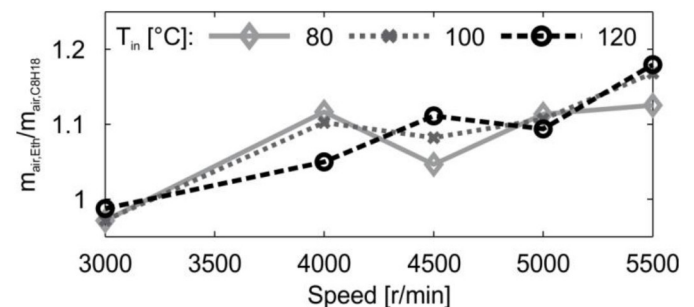


Figure 3. Ratios of air mass flow between ethanol and iso-octane due to evaporative cooling for different speeds and intake temperatures.

[Figure 3](#) shows the ratio of the air mass flow for ethanol to that of iso-octane. The air mass flow does not change significantly when changing the spark timing, so one data point per speed and intake temperature can be displayed. The figure shows the increase in volumetric efficiency due to charge cooling for ethanol over iso-octane. The effect is amplified for higher speeds. The intake temperature does not dominate. Increasing speed suggests higher turbulence levels. With increasing turbulence, the mixing between fuel and air improves and the wall wetting is reduced. The amount of heat taken from the incoming air, in this case, is increased and the volumetric efficiency rises. The charge cooling effects discussed in [figures 2 and 3](#) include both fuel characteristics and engine.

configuration (DI versus PFI) and are difficult to separate for the given measurements and also out of the scope of this work. For further analysis of the results it is important to know the difference in charge density between the two fuels.

THERMODYNAMIC ANALYSIS OF THE COMBUSTION PROCESS

Heat Release Rate and Two-Zone Temperature Calculation

For the thermodynamic analysis of the combustion process the software GT-Power, a part of the GT-Suite package developed by Gamma Technologies, is used. For the benefit of this work the so-called three pressure analysis is used. The advantages of this approach over conventional calculation methods include the better estimate of the residual gases and the conditions at IVC. In the three pressure analysis a 1D model of the engine ranging from the position of the intake pressure sensor to the exhaust pressure sensor is used. The measured pressures at these locations are imposed as boundary conditions. The geometry of the piping and the combustion chamber as well as the valve lift and flow coefficient profiles are input into the model. Two different calculations are performed for each iteration. A backward calculation only considers the high pressure phase between IVC and EVO and uses the measured cylinder pressure as an input. As a result of this calculation the heat release rate is obtained. A forward calculation considers the complete cycle (incl. gas exchange phase) and uses the heat release rate of the backward calculation as an input. The results of this calculation are the in-cylinder pressure trace and the composition at IVC. The conditions at IVC are then used as boundary conditions for the next backward run. Convergence is reached when the pressure profile, mass flow and temperatures of the forward run match the measured values. A summary of the algorithm can be seen in [figure 4](#). When the convergence criteria are met, the forward and backward run are equal and the results from either run can be used as the final result of the calculation.

The geometry of the piping, cylinder head and combustion chamber as well as the valve timing are known parameters for the calculations. The heat transfer and component temperatures in the engine however are unknown and must be modeled. For the heat transfer from the combustion chamber gases to the walls (piston, liner, valves and cylinder head) the heat transfer coefficients according to Woschni [45] are used. Instead of estimating the wall temperatures directly, the heat transfer coefficients on the outside of the components are modeled (interfaces cylinder head - coolant, liner - coolant, piston - oil, liner - oil). The cooling water pump is driven externally. The speed of the pump is not related to the engine speed. The heat transfer coefficient for the coolant can be kept constant for all operating points. Similar to the convection and conduction model for the combustion chamber, the heat transfer in the intake and exhaust paths to the coolant and the surrounding air is modeled using heat transfer coefficients on the outside of the piping.

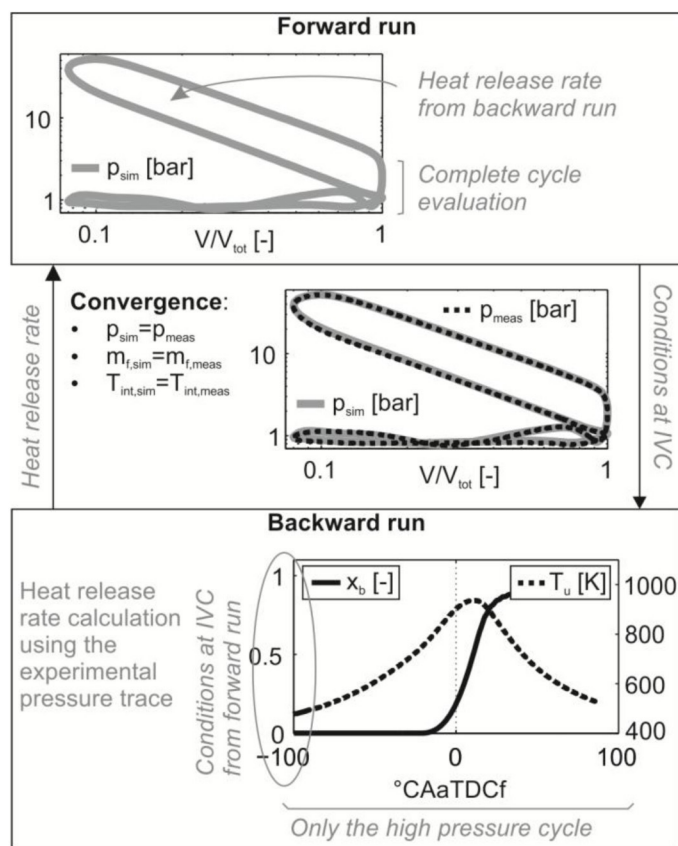


Figure 4. Summary of the algorithm for the calculation of the heat release rate and the two zone temperatures in GT-Power.

Each of the measured operating points consists of 288 single cycles. The data can be averaged over all of those cycles to obtain the mean engine cycle for this operating point. The parameters are calibrated such that the calculations for all mean cycles converge. The values for the heat transfer coefficients and the evaporation profile are listed in [table 4](#). These parameter data are then used for the calculation of the single cycles. In order to exclude the pressure sensor noise, vibrations from the valves and pressure oscillations due to knock, the in-cylinder pressure traces are filtered using a low-pass filter at 5 kHz before the heat release rate calculation.

Table 4. Parameter values for the thermodynamic analysis of the combustion process.

Parameter		Unit	Value
Heat transfer coefficient	Coolant – Head	W/m ² K	3500
	Coolant – Liner		3500
	Oil – Piston		3000
	Oil – Liner		3000
	Intake port		1000
	Exhaust port		1000
CA50		°CA	10

Sensitivity Analysis

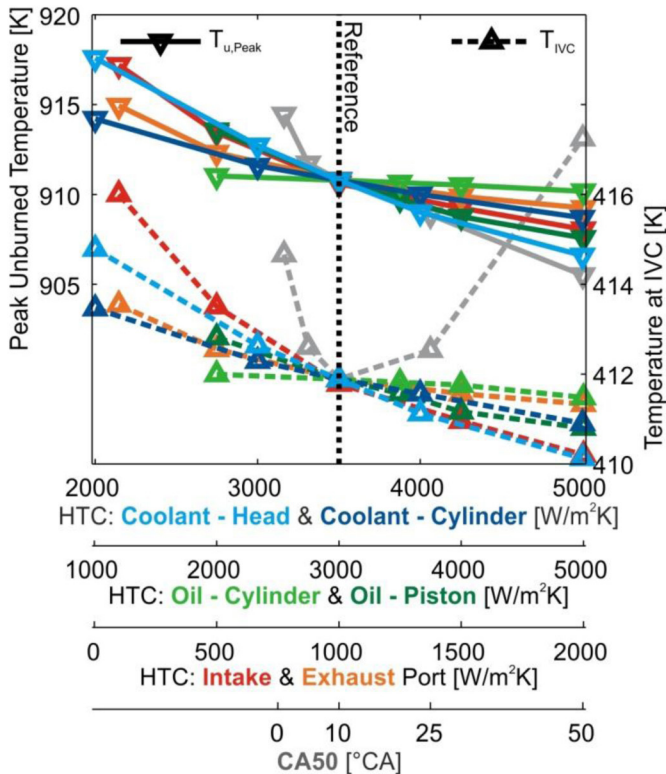


Figure 5. Sensitivity of the GT-Power model parameters (different x-axes) - heat transfer coefficients (HTC) and evaporation profile (CA50) - on the peak unburned temperature (left y-axis, full line) and the temperature at IVC (right y-axis, dashed line) for a representative operating point: 4000 r/min, $T_{in}=100^{\circ}\text{C}$, 36.75°CA SA, ethanol.

The sensitivity of the estimated parameter values on the results of the calculation is important, especially on the unburned temperature for the study of knock. The sensitivity of the estimated parameters in table 4 on the peak unburned temperature, $T_{u,peak}$, and the temperature at IVC, T_{IVC} , is shown in figure 5 for a representative operating point. The parameters are varied over a wide range of values deviating from the reference case as in table 4. A decrease in the heat transfer coefficients results in overall higher temperatures, as can be expected. The effect of the oil cooling on the liner has little effect. The influence of the oil cooling on the piston is valued at maximum 0.9°C on T_{IVC} and 3.2°C on $T_{u,peak}$ for the considered range. A stronger sensitivity is seen for the coolant with a higher effect for the cylinder head than the liner. The variation of the heat transfer coefficient coolant - head leads to a maximum deviation from the reference case of 6.8°C for $T_{u,peak}$ and 2.9°C for T_{IVC} . The heat transfer coefficient in the intake port has a stronger influence on the studied temperatures than the heat transfer coefficient in the exhaust port, because the incoming air passes here first. Strongly reducing the effect of the heat transfer in the intake port, thus neglecting the cooling circuit, increases the maximum unburned temperature by 6.4°C and the temperature at IVC by 4.1°C . The effect of the evaporation profile on the temperature in the cylinder is not uniform. A decrease of the parameter CA50 from the reference case results in an increase of both $T_{u,peak}$ and T_{IVC} . The

evaporation of the fuel requires energy and cools down the incoming air. Very fast evaporation cools down the charge quickly and the evaporation is finished much before the valves close thus increasing the temperature at IVC. The same effect can be seen for $T_{u,peak}$. Very slow evaporation however increases as well T_{IVC} . Because much of the fuel evaporates during the compression, the cooling effect of the evaporation is not yet seen at IVC. $T_{u,peak}$ however keeps reducing with slower evaporation.

Results of the Calculations

The measurement data are used to complete the thermodynamic analysis as described here, yielding e.g. the two-zone combustion temperatures and the heat release rates for both the mean and single cycles. Figure 6 shows the combustion duration between 10 and 50% of the mass fraction burned. Later spark timings result in slower combustion and longer durations. For the same operating point, the combustion duration for ethanol is always longer than for iso-octane.

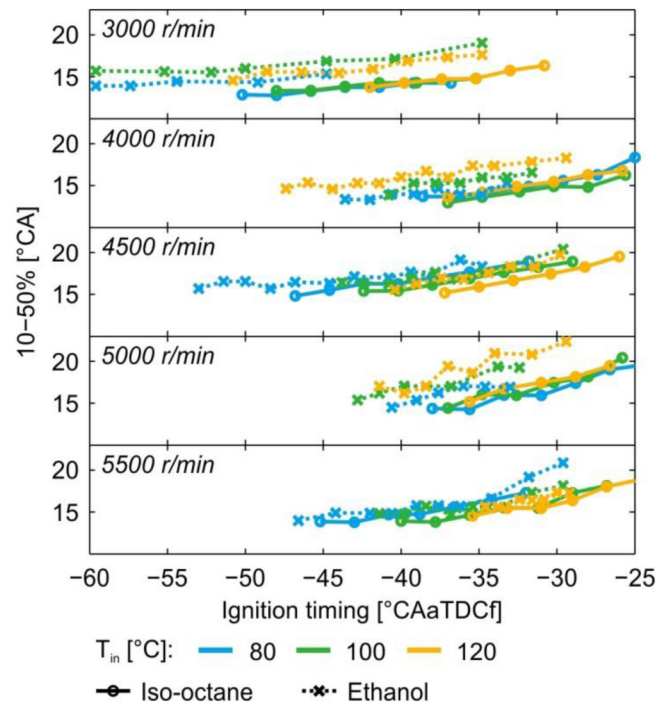


Figure 6. Combustion duration of the mean cycle between 10 and 50% burned mass fraction as a function of spark advance for iso-octane (PFI) and ethanol (DI) at different speeds and intake temperatures.

KNOCK DETECTION

Knock Detection Algorithm

When the conditions in the unburned zone in front of the flame are favorable, the unburned mixture can auto-ignite and cause knock. Knock is often related to early spark timings and high compression ratios leading to high pressures and temperatures and thus low auto-ignition delay times. Knock typically occurs at so called hot-spots in the combustion chamber: inhomogeneities in the temperature and mixture distribution in front of the flame

cause these auto-ignition locations. The amplitude of the resulting pressure oscillations is related to the chemical reaction rate of the auto-ignition [46] and thus also with the initial boundary conditions at the moment of auto-ignition. The waves propagate through the combustion chamber with the speed of sound. It has also been reported that this pressure wave reduces the auto-ignition delay time in the surroundings and can cause additional auto-ignitions, which can lead to a detonation. The pressure waves are reflected at the walls and the frequency of the pressure oscillations are therefore closely related to the cylinder geometry [47].

In order to detect knocking combustion in a SI engine, different methods have been proposed. Most of them try to quantify the severity of knock using a certain index, which increases from no knock over onset of knock to strong and severe knock. Usually a threshold value is defined as the knock limit, which determines whether a cycle is considered as a knocking cycle, since the transition between the different knock intensities is continuous. In absence of optical access to the combustion chamber, knock detection using the in-cylinder pressure is the most reliable method. Different knock indices for the detection of knock have been proposed. An overview of these indices is given in a recent publication [48]. The same knock detection method is used in this work. The in-cylinder pressure, p_{cyl} , trace is filtered using a high pass filter to obtain only the pressure oscillations, p_{filt} , as illustrated in figure 7. The maximum amplitude of the pressure oscillations (MAPO) is used as the knock index.

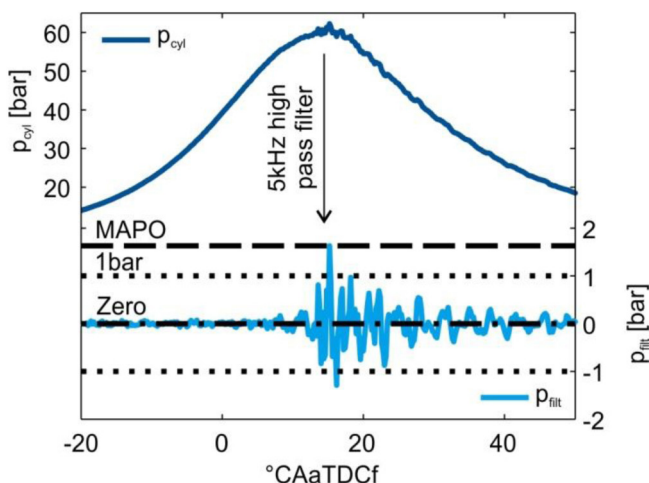


Figure 7. In-cylinder pressure trace and high pass filtered signal for an example cycle at operating point 4000 r/min, $T_{in} = 120^{\circ}\text{C}$, 33.00°CA SA, iso-octane.

The intensity of knock differs from cycle to cycle. Figure 8 shows the MAPO value for every cycle in three operating points: a late, an intermediate and an early spark timing. The operating point with early spark timing has high MAPO values for the single cycles, with peaks of almost 20 bar. In this operating point almost all the cycles knock and this operating point should not be kept for a long time. The operating point with intermediate spark timing has some peaks for the MAPO

values. Some of the cycles knock and the intensity is much lower than for the early spark timing. For the late spark timing all the MAPO values are low.

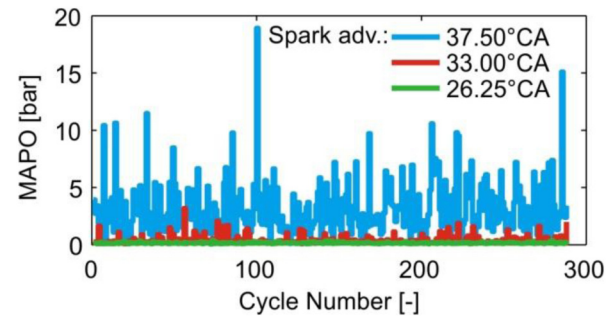


Figure 8. MAPO values for all cycles for three different spark timings: late, medium and early at 4000 r/min, $T_{in} = 120^{\circ}\text{C}$, iso-octane.

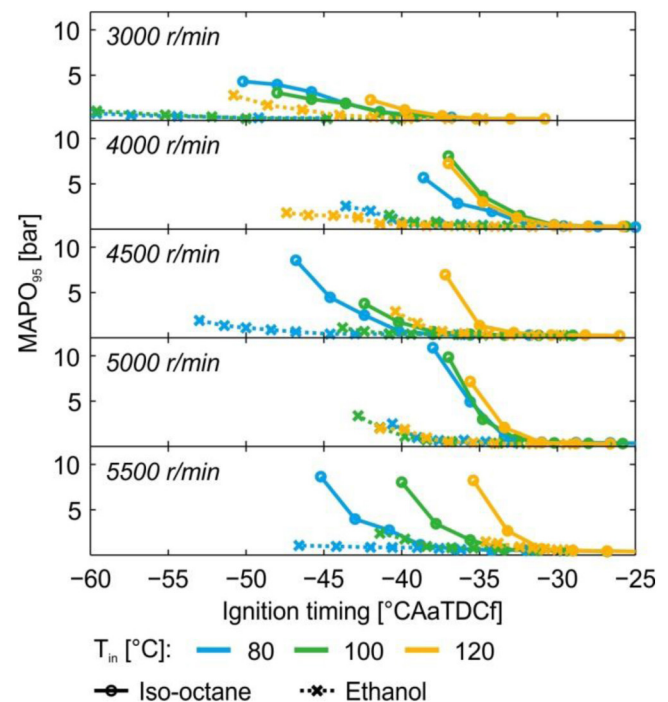


Figure 9. Knock intensity index MAPO_{95} as a function of spark advance for iso-octane (PFI) and ethanol (DI) at different speeds and intake temperatures.

The MAPO value changes for every cycle. To identify the knock intensity for the operating point the 95th percentile of the MAPO distribution can be used, MAPO_{95} . Figure 9 shows the MAPO_{95} values for both ethanol and iso-octane as a function of speed, intake temperature and spark timing. Earlier spark timings increase the knock intensity. Due to the overall lower temperatures and the much better knock resistance, the MAPO_{95} values for ethanol are consistently lower compared to those for iso-octane. The increase of MAPO_{95} occurs at later spark timings for higher intake temperatures. Regarding the speed no distinct influence can be observed. Auto-ignition of the unburned mixture in front of the flame is initiated when the ignition delay time for this mixture is reached. Both high temperatures and pressures (yielding a strong gamma-ray flux) and the availability of time promote the occurrence of knock. A