

Evaluation of Knock Behavior for Natural Gas - Gasoline Blends in a Light Duty Spark Ignited Engine

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ABSTRACT

The compression ratio is a strong lever to increase the efficiency of an internal combustion engine. However, among others, it is limited by the knock resistance of the fuel used. Natural gas shows a higher knock resistance compared to gasoline, which makes it very attractive for use in internal combustion engines. The current paper describes the knock behavior of two gasoline fuels, and specific incylinder blend ratios with one of the gasoline fuels and natural gas. The engine used for these investigations is a single cylinder research engine for light duty application which is equipped with two separate fuel systems. Both fuels can be used simultaneously which allows for gasoline to be injected into the intake port and natural gas to be injected directly into the cylinder to overcome the power density loss usually connected with port fuel injection of natural gas. Adding natural gas at wide open throttle helps to reduce knock mitigating measures and increases the efficiency and power density compared to the other gasoline type fuels with lower knock resistance. The used methods, knock intensity and number of pressure waves, do not show significant differences in knock behavior for the natural gas - gasoline blends compared to the gasoline type fuels. A knock integral was used to describe the knock onset location of the fuels tested. Two different approaches were used to determine the experimental knock onset and were compared to the knock onset delivered by the knock integral (chemical knock onset). The gasoline type fuels show good agreement between chemical and experimental knock onset. However, the natural gas -gasoline blends show higher discrepancies comparing chemical and experimental knock onset.

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INTRODUCTION

As generally known, the compression ratio (CR) and the gas properties are strong levers to increase the theoretical efficiency of an internal combustion engine. Stoichiometric operation is one of the boundary conditions for spark ignited engines allowing the usage of a comparably simple after-treatment system such as three-way-catalyst; therefore the CR is the remaining lever in order to increase the efficiency. However, the CR of spark ignited engines is limited by the knock resistance of the fuel. Natural gas (NG), which contains mostly methane, is an attractive fuel for internal combustion engines. Some of the intrinsic properties of methane -such as the higher knock resistance compared to gasoline - make it very attractive for the usage in internal combustion engines. The increasing availability and stable price of NG - compared to other oil-bases fuels - in the US, are the main reasons for the increasing interest in this fuel. As depicted in Figure 1, it is predicted that the production of NG will rise continuously within the next few decades [1].

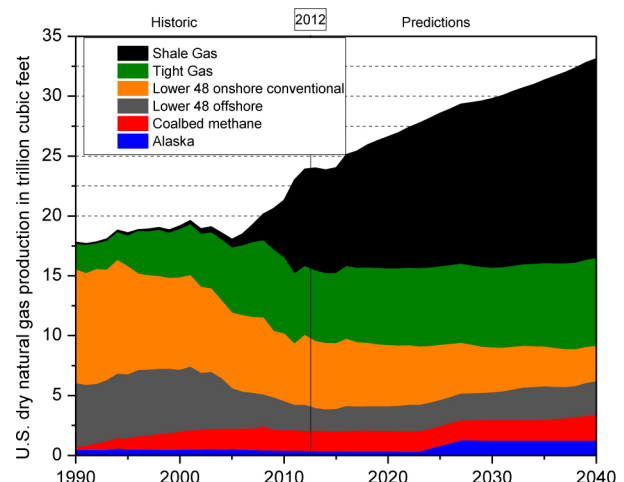


Figure 1. Historic and Projected Production of Natural Gas



Fueling a spark ignited engine with NG allows an increase in efficiency and decrease of toxic emissions, such as nitrogen oxides (NO_x) and particles at the same time. By transforming the vehicle fleet to NG fueled engines, the operating costs and the dependency on crude oil can be reduced [2]. Furthermore, increasing concerns regarding the emissions of internal combustion engines resulted in the implementation of stricter emission regulations in the United States, Europe and many other countries. However, the usage of NG can be beneficial for meeting current and future emissions legislations [3]. The molecular hydrogen-to-carbon ratio increased from 1.8 to 3.7-4.0 when NG is used instead of gasoline [3]. This in fact means that carbon dioxide (CO_2) emissions can be reduced by about 24% [4] when an engine is operated with NG compared to gasoline at the same efficiency. The high stability of the methane molecule, which leads to a very high knock resistance, is very advantageous for high load operation [5], but is harder to convert in the catalyst for the same reason. The higher knock resistance allows higher efficiencies at wide open throttle (WOT) compared to gasoline where knock mitigating measures such as spark retard and fuel enrichment need to be applied. Those measures reduce the efficiency and increase the fuel consumption. NG is gaseous at ambient conditions, which means the evaporation process does not need to take place, when compared to gasoline. Injecting NG directly is also advantageous in terms of particle emissions and total hydrocarbon (THC) emissions compared to gasoline direct injection (DI), especially during cold start conditions, since wall wetting does not occur [3]. A disadvantage by using NG is the lower power density when port fuel injected (PFI), due to the air displacement [5]. In order to overcome the power density loss, the engine for this study was equipped with a NG DI system. DI for NG, which has been a challenge so far because of the non-lubricating fuel and high required flow area [6], can increase the volumetric efficiency when fuel is injected after intake valve closing (IVC). The aforementioned displacement of air during the intake stroke does not take place anymore and the engine aspirates air only, therefore the power density can be increased [5]. This paper aims to investigate the knock behavior of different fuels; two gasoline type fuels and blends of one of the gasolines and NG. Furthermore, gasoline has known values to describe the knock resistance, such as RON, MON and AKI. However, when NG and gasoline are being blended, the knock resistance remains unknown. A second gasoline type fuel with known RON, MON and AKI values was also used to assess the knock resistance of blend ratios behaving similar to the reference fuel.

The Livengood-Wu knock integral (LWKI) method was used in order to mathematically describe the knock onset of different fueling strategies [7]. Correlations published by Kalghatgi et al. were used, in order to describe the LWKI for gasolines knowing RON and sensitivity (S) [8]. Furthermore, calculations in a constant volume reactor (CVR) in CHEMKIN were performed in order to describe the ignition delays for the gasoline type fuels and different blend ratios of E10 and NG [9]. An Arrhenius type equation was used in order to describe the ignition delay depending on in-cylinder pressure and temperature. A comparison was made between the ignition delay correlations published by Kalghatgi et al. and the gained results by CHEMKIN for the gasoline type fuels [8]. The knock onset delivered by LWKI, which will be named chemical knock onset in subsequent paragraphs, was validated against the knock onset found in the

experiments. Furthermore, two approaches were used to define the experimental knock onset and validated against the chemical knock onset. The approach, which was also used by Kalghatgi et al., showed earlier knock onset locations in the experiments and is thus closer to the chemical knock onset for most of the fueling strategies. However, 50% NG blended operation showed the highest difference between experimental and chemical knock onset. A difference up to 4.7 crank angle degree (CAD) was shown on an average for all knocking cycles considered.

EXPERIMENTAL BACKGROUND

Experimental Setup

Steady state experiments were conducted on a single cylinder research engine. The engine was equipped with gasoline PFI and side mounted DI for NG. The arrangement of injectors and the setup as mentioned above is shown in Figure 2. For DI, an outward opening injector is used. The injector is located between the intake valves and has a 60° angle relative to the cylinder axis as shown in Figure 2. The injection pressure of NG is 15 bar (230 psi). The port fuel injector was delivered by Ford and was operated with an injection pressure of 4.1 bar (60 psi). The paper comprises comparisons of neat E10 and EPA TIER II EEE fuel (EEE) and different energy based blend ratios of 25, 50, and 75% NG. The balance is covered by E10, injected into the intake port at -300 degree crank angle after top dead center ($^\circ\text{aTDC}$). A Motec M800 was used to control the ignition timing as well as the injection timing for both fuel systems. Specifications of the engine are shown in Table 1. Combustion air was supplied by an Atlas Copco air compressor; throttling was achieved by using a Parker Pilot regulator in the intake stream. Crank angle (CA) resolved data were recorded using an AVL 365X crank angle encoder. The physical resolution is 0.5 CAD; internal calculation delivered 0.1 CAD resolution for the crank angle based data [10]. In-cylinder pressure data were measured with an AVL GU21C. The spark timing (ST) was adjusted in order to maintain 50% of the rate of heat release (ROHR), (AI50), at 8°aTDC which corresponds to maximum break torque (MBT) timing. A conventional ignition coil was used, with a nominal energy level of 75 mJ.

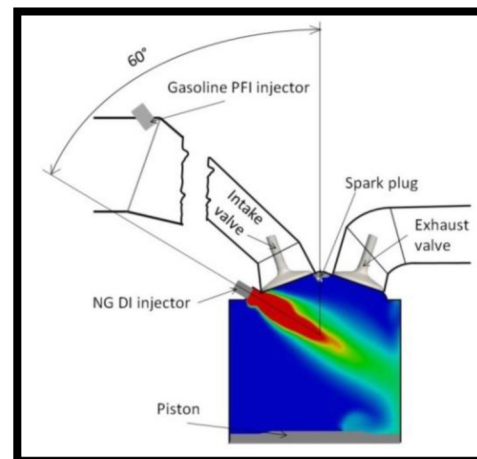


Figure 2. Schematic of the Engine Setup

Table 1. Engine Specific Parameters

Displacement	626.4 cm ³
Stroke	100.6 mm
Bore	89.04 mm
Geometric CR	10.5:1
Number of Intake/Exhaust Valves	2/2
Spark Plug	NGK, 0.7 mm gap
Tumble Ratio	0.6
Exhaust valve open/close [°aTDC fired]	150/-350
Intake valve open/close [°aTDC fired]	350/-140

For WOT operation, retarded ignition was preferred over enrichment as knock mitigating measure. Exhaust back pressure was held at ambient conditions. The properties of E10, EEE and NG are listed in Table 2.

Table 2. Fuel Specifications for E10, EEE and NG

	E10	EEE	NG	
C,H,O [wt%]	82.08, 14.13,3.79	86.58,1 3.41,0	CH ₄ , C ₂ H ₆ , CO ₂ , N ₂ [mol%]	94,3,2,1
LHV [MJ/kg]	42.05	42.61	LHV [MJ/kg]	46.93
AFR _{STOICH} [-]	14.1	14.5	AFR _{STOICH} [-]	16.2
AKI	87.0	92.7		
RON	90.7	97.4		
Sensitivity	7.0	8.3		

The properties of NG represent the typical composition available from the NG pipeline. However, in order to ensure consistent NG composition, NG was delivered in cylinders from a regional distributor.

Experimental Approach

Steady state engine testing was performed for part-load conditions as well as WOT at 1500 RPM. Stoichiometric operation was maintained for all operating points. Every data point plotted in subsequent figures represents three averaged data points collected over 375 cycles each (unless otherwise noted). WOT tests were performed adjusting the intake pressure to 1 bar (14.5 psia).

Figure 3 shows the pressure trace and the respective injection timing and injection pressure for neat fuel operation of both fuel systems at 1500 RPM and WOT. Figure 3 represents the pressure traces for E10 PFI as well as NG DI. The height of the rectangles represents the injection pressure and the width represents the injection duration. E10 PFI is injected into the intake port starting at -300°aTDC with injection duration of 165 CAD. Start of injection (SOI) for NG DI was chosen to be -150°aTDC since the highest indicated mean effective pressure (IMEP) values were achieved at WOT [11]. The injection duration for 1500 RPM and WOT is about 42 CAD for NG DI. The critical pressure ratio p_0 of 0.54 delivers a critical pressure of 8.1 bar by using a γ of 1.3 and a fuel injection pressure of 15 bar for NG DI. The red dotted line at -40°aTDC shows the crank angle location where the choked mass flow would be achieved at WOT.

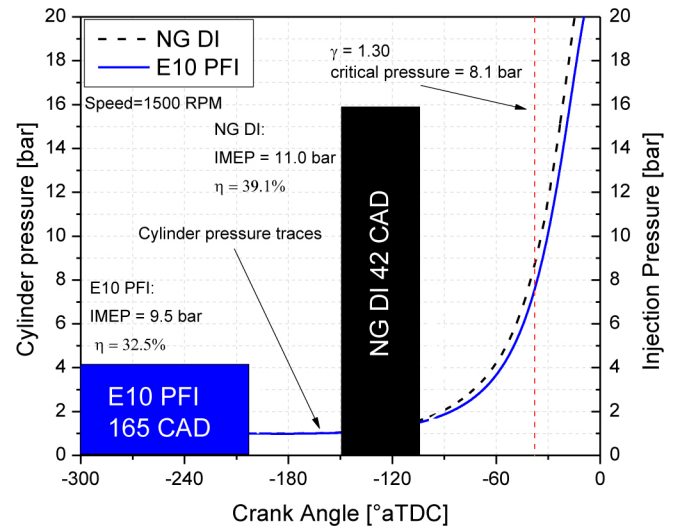


Figure 3. Cylinder Pressure and Injection Timing/Pressure for E10 PFI SOI -300°aTDC and NG DI SOI -120°aTDC, WOT and 1500 RPM

Combustion Metrics

Equation (1) represents the first law of thermodynamics applied on a closed system. By neglecting blow-by gases, the time frame between IVC and exhaust valve opening (EVO) can be seen as a closed system. In order to quantify the combustion, equation (2) is used, which integrates the ROHR. The equation is implemented within the data acquisition system and calculates the added fuel energy subtracted by the wall heat versus crank angle. Subsequent sections evaluate the progress of the combustion in more detail, expressed by different fractions of combustion energy released. AI10, AI50 and AI90 for example refer to 10, 50 and 90% combustion energy released.

$$\frac{dQ}{d\phi} = \frac{1}{\gamma - 1} \cdot \left[\frac{\gamma \cdot p \cdot dV}{d\phi} + \frac{V \cdot dp}{d\phi} \right] \quad (1)$$

$$AI = \int_0^{100} \frac{dQ}{d\phi} \cdot d\phi \quad (2)$$

The early flame development process is described by the flame development angle (FDA) which is defined in equation 3. It considers the CAD from ST to AI10.

$$FDA = AI10 - ST \quad (3)$$

The combustion duration (CD) is used in order to describe the actual combustion process, which is defined in equation 4.

$$CD = AI90 - AI10 \quad (4)$$

The variance of the IMEP (COV_{IMEP}) was used in order to quantify the stability of the engine operation. A 3% limit was imposed for stable engine operation which was derived over the collected cycles per condition.

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{\mu_{IMEP}} \quad (5)$$

Definitions

Figure 4 shows a knocking and a non-knocking cycle for E10 PFI with similar boundary conditions. The measured in-cylinder pressure trace is shown in the top chart. The band pass filtered incylinder pressure trace (BPF CPT) is shown in the bottom chart.

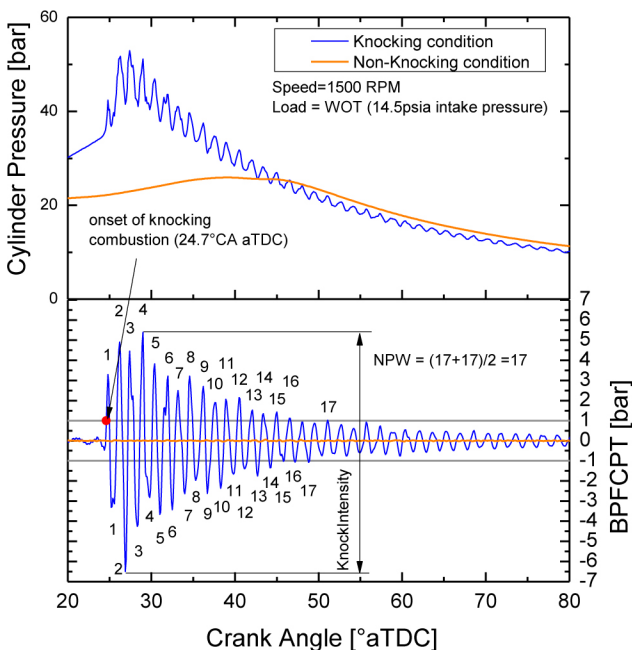


Figure 4. Characterization of Knocking Combustion

The in-cylinder pressure trace was filtered between 3kHz and 25kHz as suggested by the literature [12]. As can be seen in Figure 4, knocking combustion conditions contain high frequency pressure amplitudes which are not seen during non-knocking combustion conditions. Combustion was considered as knocking when the BPF CPT signal exceeded the threshold of ± 1 bar (another suggestion found in the literature [12] and [13]) shown by the grey lines in the bottom chart. Another reason for the ± 1 bar threshold is that knocking combustion conditions became clearly audible. The BPF CPT, in Figure 4, exceeds the 1 bar threshold at 24.7°aTDC the first time, indicated by the red dot, which is also considered as the onset of knock. The number of pressure waves traveling through the combustion chamber for a knocking combustion condition is calculated by adding up all spikes exceeding 1 bar on the positive side and adding up the spikes on the negative side as well. The mean value for both sides shows the number of pressure waves (NPW). Figure 4 shows an example of how NPW is calculated. Another measure to describe knocking combustion conditions is to take the pressure difference from the highest pressure spike to the lowest pressure spike, called peak-to-peak pressure, which is expressed by

knock intensity (KI). In Figure 4 the KI value shown is 11.5 bar using the difference of the fourth spike of the positive side with a pressure value of 5.5 bar and the second spike of the negative side with -6 bar. Adding up the absolute values of these two, results in KI of 11.5 bar. Subsequent plots compare experimental data close to 10% knocking cycles for all fueling strategies.

EXPERIMENTAL RESULTS

In order to investigate the knock behavior of the tested fuels, two different approaches were used. The first approach was increasing the IMEP by increasing the intake pressure and maintaining MBT timing until the engine started knocking. Thereafter, spark sweeps to achieve knocking and non-knocking conditions were performed. The second approach was to keep a constant intake pressure of 1 bar (14.5 psia), which corresponds to WOT. Tests were started with delayed ST, corresponding to a non-knocking condition. The spark timing was then advanced until the knock limited spark advance (KLSA) was reached. However, boosting the engine was necessary beyond 50% NG in order to achieve knocking combustion conditions.

Knock Investigations at MBT Timing

Figure 5 shows relevant combustion parameter for all five fueling strategies. E10 PFI shows the lowest knock resistance and knock occurs already at an IMEP of 6.3 bar. As shown in Table 1, E10 PFI features an AKI of 87. Operating the engine with EEE, which has an AKI of 92.7, allows maintaining optimal combustion phasing up to an IMEP of 8.1 bar. Fueling the engine with 25% NG allows extending the IMEP even further until the onset of knock occurs at 8.4 bar IMEP. Operating the engine with 50% NG, WOT operation was required to achieve knocking combustion. The onset of knock started around 10°aTDC which is close to what is considered MBT timing. The achievable IMEP for the 50% NG operation is about 11 bar. Boosted operation was necessary to get knocking combustion conditions with 75% NG. The achieved IMEP as shown in Figure 5 is about 17.9 bar. Despite similar IMEP values for EEE compared to 25% NG, the net indicated thermal efficiency (ITE) values are significantly higher for EEE operation. The difference in molar expansion ratio between NG and E10 and a possibly higher wall heat transfer because of higher in-cylinder turbulence for NG DI could be conceivable scenarios for the lower efficiency. Pamminger et al. show a loss analysis comparing E10 PFI and NG DI discussing that fact [11]. E10 PFI shows lower efficiency numbers because of throttle losses at part-load. The facts mentioned above, decreasing molar expansion ratio and higher wall heat losses by increasing the blend ratio might also explain the difference in efficiency between 50 and 75% NG. As can be seen in Figure 5, advancing the ST leads to shorter CD. FDA for all fuels is different but stays rather constant by changing the ST. However, the difference in ST as well as CD between pure gasoline PFI and injecting some NG directly into the cylinder is significant. Blended operation with NG DI allows sparking about 10 CAD later compared to gasoline PFI by maintaining the same AI50. The difference in CD between fueling the engine with PFI compared to DI is about 5 CAD, as shown in Figure 5.