

Ethanol Detection in Flex-Fuel Direct Injection Engines Using In-Cylinder Pressure Measurements

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ABSTRACT

A method for detection of ethanol content in fuel for an engine equipped with direct injection (DI) is presented. The methodology is based on in-cylinder pressure measurements during the compression stroke and exploits the different charge cooling properties of ethanol and gasoline. The concept was validated using dynamometer data of a 2.0L DI turbocharged engine with variable valve timing (VVT). An algorithm was developed to process the experimental data and generate a residue from the complex cycle-to-cycle in-cylinder pressure evolution which captures the charge cooling effect. The experimental results show that there is a monotonic correlation between the residues and the fuel ethanol percentage in the majority of the cases. However, the correlation varies for different engine operating parameters; such as, speed, load, valve timing, fuel rail pressure, intake and exhaust temperature and pressure.

INTRODUCTION

In recent years, there has been an increasing interest in renewable sources of energy to meet a growing demand and reduce emissions. Biofuels appear as a good option because they are compatible with current gasoline engine designs. Among biofuels, ethanol has been blended in varying percentages with gasoline, for many years. Typical blends generally range from 0% to 85% of ethanol in volume. Current flex-fuel vehicles (FFV) need to reach the same emission levels as conventional vehicles. In order to achieve the required emission level,

air-fuel ratio (AFR) control is essential and is currently performed using the Exhaust Gas Oxygen (EGO) sensor independently of the fuel composition.

However, ethanol and gasoline have different physical and thermodynamical properties [1] which require different engine control parameters depending on the ethanol percentage in the fuel (etoh%) to optimize performance and emissions. For instance, ethanol has a lower vapor pressure than gasoline which causes cold-start problems.

	Gasoline	Ethanol
RON	92	111
Density (g/cm ³)	0.74	0.79
Heat of Combustion (MJ/kg)	42.4	26.8
Stoichiometric A/F ratio	14.3	9.0
Boiling Point(°C)	20-300	78.5
Enthalpy of vaporization(kJ/kg)	420	845

Table 1: Ethanol and Gasoline properties

Hence, the fuel ethanol content needs to be detected to adjust the cold start-up procedure. Ethanol detection could also facilitate the adjustment of ignition timing and compression ratio to improve the engine efficiency by exploiting the higher RON and flame propagation speed of ethanol. Knowledge of the ethanol percentage could also allow the optimization of the feed-forward control strategies for the different fuel blends; such as, transient AFR control and torque control.

Although ethanol sensors [2, 3] have been developed to detect fuel ethanol concentration by placing them in the tank or in the fuel line, they are not widely used in production vehicles mainly due to the extra cost. The ethanol content in the fuel can also be indirectly detected by means of the closed-loop Air Fuel Ratio (AFR) correction signal based on the EGO sensor [4, 5]. Although this method is able to estimate the fuel composition with acceptable accuracy, it has a slow detection speed and can not be used at engine startup [6] due to the unavailability of the EGO sensor. Furthermore, since the closed-loop AFR correction signal is also used for on-board diagnosis, this method might misinterpret other faults, such as mass air flow (MAF) sensor or injector drifts, for a change in ethanol concentration [7, 8, 9]. Methodologies have also been developed to determine the fuel composition by exploiting the effects of ethanol concentration on the combustion behavior via in-cylinder pressure sensors [10, 11, 12]. However, to some extent, these approaches overlap with the EGO-based method because they all rely on combustion related properties.

The works mentioned above and others that also use in-cylinder pressure sensor to estimate AFR or other fuel/air properties [13, 14, 15, 16, 17] are all based on the pressure trace during combustion. The present work focuses on the cylinder pressure information during the compression stroke and in particular the charge cooling effect caused by direct injection of ethanol-gasoline blends. The additional cooling effect due to ethanol injection during the compression period is used in [18] to improve the engine efficiency and performance. This effect and its impact on the in-cylinder pressure is exploited here to detect the fuel ethanol content in an engine equipped with direct injection.

First, the ethanol detection principle and two algorithms to extract the charge cooling effect from the cylinder pressure trace during the compression stroke are presented. Both algorithms process the cylinder pressure during the compression stroke under two different fuel injection patterns to generate a detection residue which captures the charge cooling effect and consequently the ethanol content in the fuel. The algorithms differ only in the way the compression pressure traces are processed to produce the residue. Further, the experimental validation of the detection principle and algorithms is described. Dynamometer tests were conducted on a 4 cylinder 2.0L DI turbocharged engine at various speeds and loads, as well as, different valve timings and fuel injection pressures.

The experimental results show a monotonic correlation between the detection residues and the fuel ethanol content in the majority of the cases. However, the correlation varies for different engine operating parameters; such as, speed, load, valve timing, fuel rail pressure, intake and exhaust temperature and pressure.

DETECTION PRINCIPLE

The ethanol detection method presented in this work is based on in-cylinder pressure measurements and exploits the large difference in enthalpy of vaporization and consequently, charge cooling properties of fuels with different ethanol percentages. In order for the charge cooling to be observed on the cylinder pressure evolution, fuel must be injected directly in the cylinder while all valves are closed, i.e. during the compression stroke. If the fuel injection occurs during the intake stroke, while the intake valve is open, the cooling effect will increase the flow of air into the cylinder, improving the engine volumetric efficiency but having little or no effect on the in-cylinder pressure [18].

Under some simplifications, the compression stroke with a mixture of air and completely vaporized fuel, and without additional fuel injection can be modeled as a polytropic process with coefficient n_c [19, 20]; whose temperature and pressure are characterized by Eq. (1) and (2).

$$T = T_{ivc} \cdot \left(\frac{V}{V_{ivc}} \right)^{1-n_c} \quad (1)$$

$$P = P_{ivc} \cdot \left(\frac{V}{V_{ivc}} \right)^{-n_c} \quad (2)$$

Where, P , T and V are cylinder pressure, temperature and volume respectively; and the subscript ivc denotes the intake valve closure instant.

When fuel is injected during the compression stroke, its vaporization cools down the cylinder charge causing a deviation from the ideal polytropic model. Ethanol and gasoline has different enthalpy of vaporization, 845kJ/kg for ethanol and 420kJ/kg for gasoline. As a result, the charge cooling depends on the fuel ethanol percentage (etoh%).

Fig. 1 shows simulated compression pressure traces and how they can deviate from the ideal process when fuels with different ethanol percentages are injected and instantly vaporized. Note that, this work concentrates on the cylinder pressure evolution during the compression stroke and therefore, the 0 deg. crank angle reference used in all the plots corresponds to the compression Bottom Dead Center (BDC).

The charge cooling may alternatively be seen on the polytropic compression coefficient, n_c , as illustrated in Fig. 2. In an ideal polytropic process, the relationship between $\log(P)$ and $\log(V)$ is affine with a constant slope equal to $-n_c$. The fuel vaporization causes a deviation in the process which, in the over-simplistic case of instantaneous fuel vaporization, can be seen as a drop in the n_c curve. Clearly, etoh% correlates with the

magnitude of the deviation of the n_c -curves from the ideal constant value.

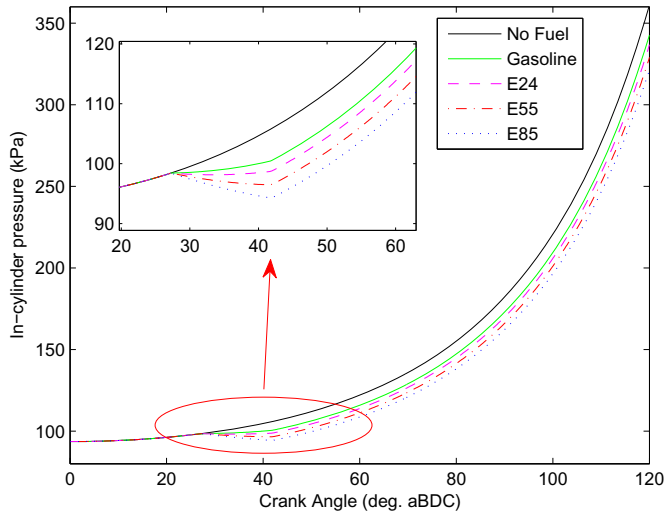


Fig. 1: Simulated in-cylinder pressure during the compression stroke. The area showed in detail corresponds to the injection event.

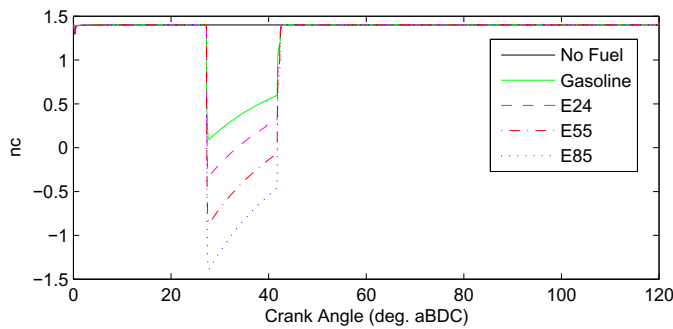


Fig. 2: Simulated polytropic compression coefficient (n_c) vs crank angle.

However, due to the complex interaction among the two-phase fuel-air mixture, fuel injection and vaporization characteristics, and heat transfer with the cylinder walls, the real process might not exhibit such a clear difference in behavior for different etoh%. This paper presents a study to verify the theoretical trends in Fig. 1 and 2 and the feasibility of the proposed ethanol detection methodology.

DETECTION RESIDUE GENERATION

Two approaches are presented for the generation of a detection residue from in-cylinder pressure measurements that captures the charge cooling effect due to fuel injection during the compression stroke. The first one is based on the deviation of the polytropic compression coefficient (n_c) during the fuel vaporization, n_c method. While, the second one focuses on the final deviation in the compression pressure trace after the fuel vaporization, gap method.

Both proposed methodologies exploit the difference in cylinder pressure during the compression stroke under two different injection modes in order to generate a detection residue which hopefully depends monotonically on the fuel ethanol content (etoh%):

1. **Single injection (Si) mode:** all the fuel is injected early in the intake stroke.
2. **Split injection (Sp) mode:** a fraction of the fuel is injected early in the intake stroke and the rest at the beginning of compression (when all the valves are closed).

Note that in both modes the total injected fuel is determined to achieve stoichiometric Air Fuel Ratio (AFR).

When fuel injection occurs during the intake stroke, most of the fuel vaporizes before the intake valve closes and consequently, the charge cooling associated with the fuel vaporization may improve the engine volumetric efficiency but has little or no impact on the cylinder pressure evolution. On the other hand, when fuel is injected during the compression stroke, there's no possibility of extra air induction and the charge cooling translates in a cylinder pressure drop. By subtracting the in-cylinder pressure traces for Single and Split injection modes, the pressure drop due to the charge cooling caused by the 2nd fuel injection in Split mode can be extracted from the complex cylinder pressure evolution. As explained above, this charge cooling effect depends on the fuel ethanol concentration through the enthalpy of vaporization.

Fig. 3 shows experimental in-cylinder pressure traces during the compression stroke for Split and Single injection modes and their difference. Clearly, the pressure difference in Fig. 3.b extracts the effect of the 2nd fuel injection in Split mode from the exponential pressure evolution characteristic of the compression process.

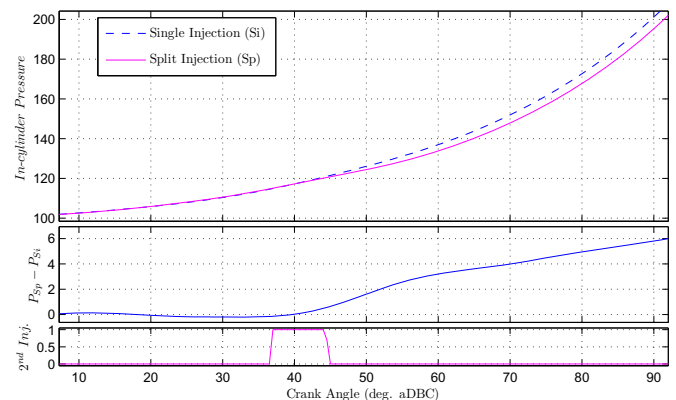


Fig. 3: a) Compression pressure traces for Single and Split injection modes. b) Pressure difference between injection modes. c) Split-mode 2nd injection pulse.

As shown in Fig. 4, the residue generation algorithm consists of a signal conditioning block and the actual detection residue generation (n_c or gap method).

algorithm can be combined in the future with an injection mode controller. The injection mode controller could determine when and how often Split injection mode is applied in order to achieve the desired estimation performance without compromising emissions or engine efficiency. This block was not implemented in this stage because the algorithm was executed off-line.

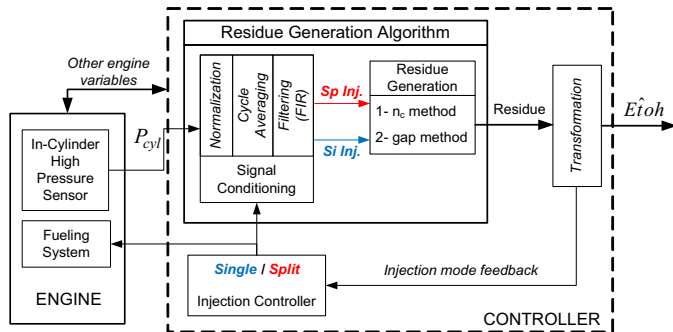


Fig. 4: High-level block diagram of the residue generation algorithm.

SIGNAL CONDITIONING - Regardless of the detection method, it is necessary to perform a pre-conditioning of the pressure measurements in order to eliminate several sources of error, as follows:

1. Engine conditions variations.
2. Cycle to cycle variations in the engine behavior.
3. Pressure sensor accuracy, noise, and quantization.
4. Disturbances and fast unmodeled dynamics.

The signal conditioning consists of 3 stages: normalization, cycle-average and filtering.

Normalization - Even small differences in the testing conditions can affect the process behavior and consequently, the algorithm results. In order to compare detection residues computed with data from different tests, a normalization of the measurements is required. For the current experimental data, a normalization only with respect to the load is being used which is computed using the intake manifold pressure, MAP, as shown in Eq. (3)

$$P_{norm}(k) = \frac{P_{cyl}(k) \cdot (k_{SoCp} - k_{SoIt}) \cdot 100}{\sum_{i=k_{SoIt}}^{k_{SoCp}} MAP(i)} \quad (3)$$

Where, P_{cyl} and MAP are vectors containing crank resolved cylinder and intake manifold pressure measurements respectively; k_{SoIt} and k_{SoCp} are the vector indexes corresponding to the start of the intake and compression strokes respectively; and 100 is a unit adjustment constant.

Cycle average - The cylinder pressure exhibits important cycle to cycle variations which are directly related to the stochastic nature of the combustion process. In order to

reduce this variability to an acceptable level and capture the charge cooling characteristics, several consecutive cycles need to be averaged.

$$\bar{P}(k) = \sum_{i=1}^n P_{norm}^i(k). \quad (4)$$

For the results presented in this work an averaging of ten cycles ($n=10$) was used. The choice responds mainly to the available number of cycles at each tested condition.

Filtering - A filter is applied to the measurements to minimize two sources of error. The first one is measurement noise and quantization error introduced by sensors and the acquisition system. The second one, which for our purposes will be also considered noise, corresponds to disturbances inherent to the engine operation and unmodeled fast dynamics. One of the most important disturbances of this second type is seen in the pressure trace during the intake valve closure.

The filtering is realized by a 128th order finite impulse response (FIR) filter with a normalized cut of frequency ranging from 0.01 to 0.1

$$P_f(k-64) = h_{FIR}(k) * \bar{P}(k) \quad (5)$$

where: h_{FIR} is the impulse response of the filter.

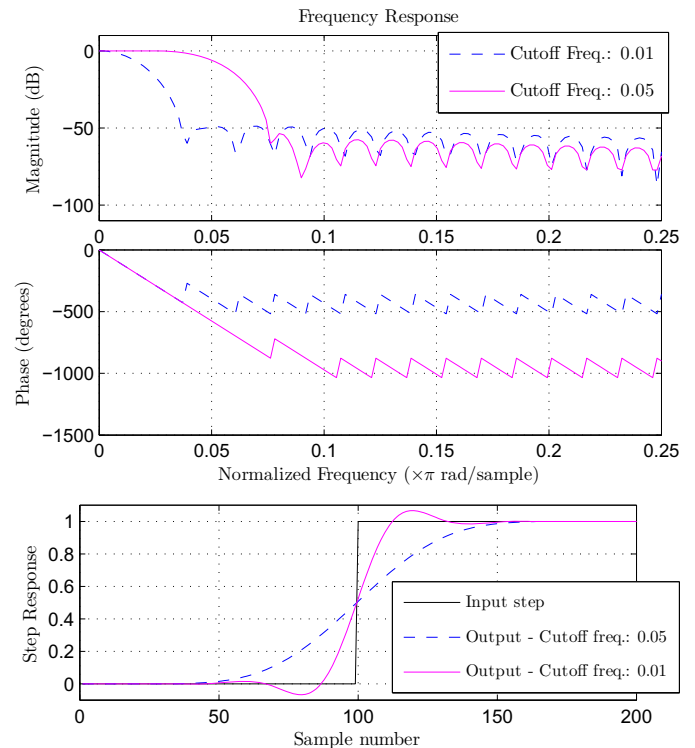


Fig. 5: Filter frequency and step responses

The cut of frequency selection responds to a trade-off between noise filtering capability and cylinder pressure