

Sensitivity Analysis Study on Ethanol Partially Premixed Combustion

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ABSTRACT

Partially Premixed Combustion (PPC) is a combustion concept which aims to provide combustion with low smoke and NO_x with high thermal efficiency. Extending the ignition delay to enhance the premixing, avoiding spray-driven combustion and controlling the combustion temperature at an optimum level through use of suitable lambda and EGR levels have been recognized as key factors to achieve such a combustion. Fuels with high ignitability resistance have been proven to be a useful to extend the ignition delay. In this work pure ethanol has been used as a PPC fuel.

The objective of this research was initially to investigate the required operating conditions for PPC with ethanol. Additionally, a sensitivity analysis was performed to understand how the required parameters for ethanol PPC such as lambda, EGR rate, injection pressure and inlet temperature influence the combustion in terms of controllability, stability, emissions (i.e. HC, CO, NO_x and Soot) and combustion and thermodynamic efficiency.

Investigations have been performed experimentally on a single cylinder heavy duty engine where each of the parameters swept one at time. The results showed that the combustion is not sensitive to changes in injection pressure but the adjustments in lambda, EGR and inlet temperature should be controlled very carefully as there are clear limitations for each of these parameters. These limitations are highlighted and presented in this study. Engine-out soot emissions showed to be detectable only once the global lambda is very low (1.25 or lower) with very high EGR rates.

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INTRODUCTION

Due to increasing global concern regarding pollution, this is much focus on the need to develop more efficient and environmental friendlier propulsion systems. The main systems for transportation have been the Otto and Diesel engine for a long time, but both systems have difficulties to meet the more stringent legislations without extensive after-treatment systems. This has driven the evolution to combustion concepts such as homogeneous charge compression ignition (HCCI) and later partially premixed combustion (PPC). The HCCI concept has been shown to have ultra-low engine-out emissions in soot and NO_x however it has shown to be unreliable in terms of combustion control and load limitations at higher loads. As a result, research activities were focused towards PPC.

PPC combustion has attracted high levels of interest during the past seven years due to the good controllability of

the combustion as well as having a high potential to reduce NO_x and soot levels to very low levels without penalties in engine efficiency. Different techniques have been reported to obtain PPC. The early version of PPC focused on a combination of low compression ratio, high EGR rate and operating conditions close to stoichiometric operation using diesel fuel [1,2]. The limited operating range and low combustion efficiency were the drawbacks of this concept. Later Kalghatgi introduced a PPC concept to promote air-fuel mixing by injecting gasoline fuel in a compression ignition engine [3,4]. Gasoline which has higher ignition resistance than diesel was used to separate the end of fuel injection from the start of combustion. Using a gasoline type fuel (i.e. fuel with high auto-ignition resistance) significantly extended the load operating range of PPC. Using dual fuel (gasoline-diesel) injection has been also significantly investigated in order to extend the PPC operating range limits and boosts the efficiency [5,6].

Division of Internal Combustion Engines at Lund University has been exploring the possibility of achieving PPC through the appropriate combination of EGR and lambda in combination with fuels with high auto-ignition resistance property [7,8,9]. The key parameters are correct in-cylinder temperature that should be low enough to result in low NO_x and in the same time it should be high enough for good oxidation of unburned hydrocarbon (HC) and carbon monoxide (CO). The correct combination of EGR rate and lambda can control the in-cylinder temperature. The in-cylinder temperature, a suitable injection strategy to control the combustion and a fuel with high resistance to auto-ignition all contribute to minimize the exhaust emissions without a penalty in efficiency.

In order to have simultaneously low CO, HC and NO_x, the maximum temperature inside the combustion chamber during the cycle should be between 1500 and 1800 K [10,11]. Adiabatic flame temperature can be used as a good indication for the maximum in-cylinder temperature. Figure 1 shows the adiabatic flame temperature as a function of lambda and EGR for a homogeneous mixture used ethanol as fuel. Adiabatic flame temperatures have been computed using CECIL, a chemical equilibrium solver developed by Per Tunestål at Lund University. The principle of the solver is to minimize Gibbs free energy using an active set method. NASA polynomials have been used to compute thermodynamic properties. If the EGR rate is between the ranges of [45%-55%] and lambda is in the ranges of [1.2-1.8] it is reasonable to assume that the temperature will be in the range of [1800K-1500K], meaning low enough NO_x and still good combustion efficiency.

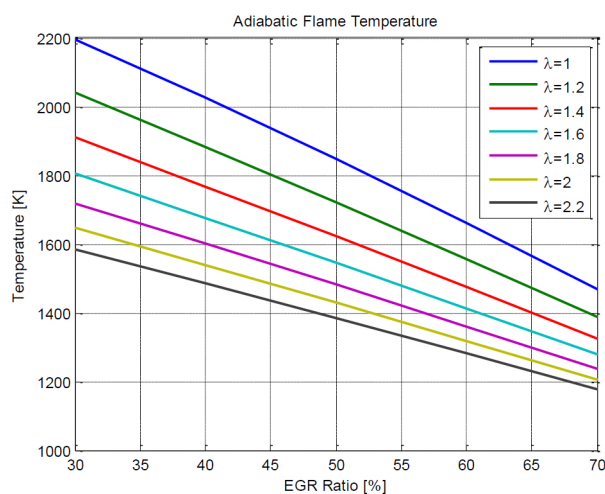


Figure 1. Adiabatic flame temperature with different EGR and Lambda combinations

Due to the unique properties of ethanol it was considered as a very good candidate for PPC. High octane number, the oxygen content, high volatility, low heating value and high evaporation enthalpy are the properties which make ethanol different to regular hydrocarbons and attractive as PPC fuel.

Manente in [7,8,9] proposed that EGR=50% and lambda=1.5 provided optimal operating requirements for PPC. To date, no complete study has been published using ethanol as PPC fuel.

The objective of this study was to perform a sensitivity analysis on the main control parameters for PPC to quantitatively assess the operating limitations for PPC fueled with ethanol. Investigations have been performed experimentally on a single cylinder heavy duty engine and effects from changes in EGR rate, global lambda, injection pressure and inlet temperature were quantitatively observed.

EXPERIMENTAL SETUP

The engine used for the experiments was a modified Scania D13. The engine specifications can be found in Table 1. Only one of six cylinders was operational and the remaining cylinders were motored. For inlet boost two superchargers were externally installed in a compressor package to reach desired pressure, independent of engine speed. To simulate and control the back pressure of a turbo charger, a valve was installed. Another back pressure valve was installed downstream to force EGR to the inlet. An inlet air heater was installed to facilitate ignition of the fuel and air mixture, since ethanol is highly resistant to auto ignition. The heater was PID controlled as combustion is sensitive to temperature variations. A scheme of the experimental setup is presented in Figure 2.

A Scania XPI common-rail fuel injection system was used. With only one cylinder in use, the fuel rail volume was increased to even out pulsations in the system. This was done by installing two fuel rails in parallel. A controlled fuel leakage was also created to simulate the missing injectors, providing normal operating conditions for the fuel pump.

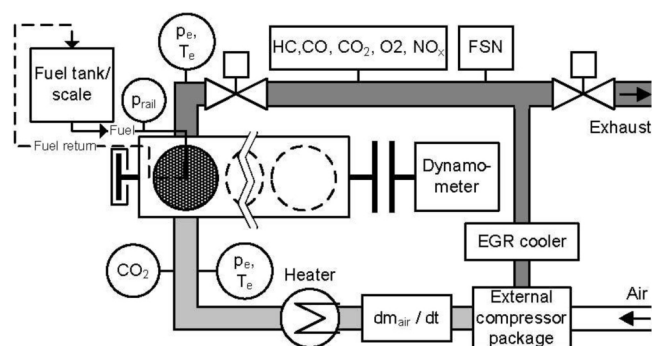


Figure 2. Scania D13 schematic experimental setup

Table 1. Specifications of the single cylinder engine and injector

Displacement Volume	2123 cm ³
Bore	130 mm
Stroke	160 mm
Connecting rod	255 mm
IVC	40 ABDC
EVO	50 BBDC
CR	17.3:1
Fuel system	Scania XPI
Orifices	8
Orifice diameter	0.19 mm
Umbrella angle	148 deg

All experiments were run with ethanol with specifications presented in Table 2. In order to insure that no damage was caused to the injection system, lubricity additive was added to the ethanol.

Table 2. Ethanol fuel specifications

Fuel	Ethanol 99.5%
RON	107
C	2
H/C	3
O/C	0.5
LHV [MJ/kg]	26.9
A/F stoich.	9

Pressure sensors were located in the inlet, exhaust, and in cylinder. The in-cylinder pressure signal was amplified with a Kistler converter, and together with the crank angle degree (CAD) incremental encoder, real time readings at a resolution of 0.2 crank angle degrees were recorded over 400 cycles. With this data the heat release was calculated and some parameters such as start of combustion (SOC), phasing where 50% of the fuel is burned (CA50) and combustion duration are extracted. SOC was given in CAD where 5% of the cumulative heat release occurred. Temperatures were measured with thermocouples in the inlet, exhaust and engine oil systems. Emissions were measured with a Horiba Mexa-9100 emission unit and with an AVL 415 smoke meter. The soot readings were presented with a smoke number [FSN], in a scale from 0-10. Other emissions were measured in [ppm] and presented with Index Emissions [g/kg (emission/fuel)]. The load is calculated as gross Indicated Mean Efficiency Pressure (IMEP), which is the total work done during the compression and combustion stroke, normalized with the displacement volume. Some part of data was evaluated in terms of combustion efficiency, thermodynamic efficiency and heat losses. A Sankey diagram and the used definitions in this paper are presented in the [appendix](#).

EXPERIMENT

This work is a part of PPC research work at Lund University. Based the previous research suggest in [7,8,9] the general operating requirement for PPC combustion was determined to EGR =50% and Lambda=1.5.

Double injection strategy was used in this experiment. For the experiments which are performed at 12 bar IMEP¹ the injection ratio was set almost equally for the both injection pulses (i.e. 52% of the total injection duration was set on the first injection pulse and the rest was set on the second injection) however for the experiment performed at 5 bar only 38% of the fuel was injected from the first injection pulse and the rest was injected from the second injection pulse. Injection timing setting for each experiment is described in the respective result section. Injection pressure and inlet temperature was set depending on the desired load. More detailed study on injection parameters can be found in a paper by the authors of this work [12].

The experiments were performed at 1200 rpm and 12 bar IMEP however since the inlet temperature is a very crucial parameter at low loads, an extra inlet temperature sweep was performed at 5 bar IMEP and the results were compared with the 12 bar case. The inlet temperature was swept from 180 to 125 C while the injection pressure was set to 700 bar, lambda=1,5 and EGR rate was equal to 50%. The general operating parameters for the investigated loads are presented in Table 3.

Table 3. Operating parameters for the investigated loads

IMEP(gross) [bar]	12	5
Lambda [-]	1.5	1.5
EGR [%]	50	50
Injection pressure [bar]	1500	700
Inlet temperature [C]	100	140

Totally 5 different parameter variations were performed and the operating conditions for four of them which were performed at medium load (i.e. 12 bar IMEP) are presented in Table 4. The engine speed was set to 1200 rpm for all experiments.

Table 4. operating parameters for each sweep

	Ref.	Sweeps			
		Lambda	EGR	Inj. P	Inlet.T
Lambda [-]	1.5	1.1-1.6	1.5	1.5	1.5
EGR [%]	50	50	40-70	50	50
Inj. pre [bar]	1500	1500	1500	1500-700	1500
Inlet temp	100	100	100	100	70-120

¹Since all the experiments have been performed on a single cylinder engine, all reported IMEP are gross calculated

RESULTS

The results are divided into four different sections as four parameters (i.e. Lambda, EGR, injection pressure and inlet temperature) were studied and presented in the same order as the experiments were performed. For each result section the inlet conditions for the experiments are presented followed by the evaluated data in terms of emissions index, heat release data, combustion and thermodynamic efficiencies and the heat losses.

Effects of Changes in Lambda with Constant EGR Level

The reference lambda value was determined to 1.5 and with keeping all other parameter constant it was desired to analyze the sensitivity of the combustion parameters to changes in lambda. Lambda can be changed in two ways, keeping either air or fuel mass constant. In this study, fuel mass is kept constant thus changing the air mass and hence the inlet pressure. The lambda change was performed by increasing the inlet pressure from 1.2 bar (abs.) up to 2.8 bar which corresponded to 1.1 to 1.6 in lambda. The inlet conditions for this experiment are presented in Figure 3.

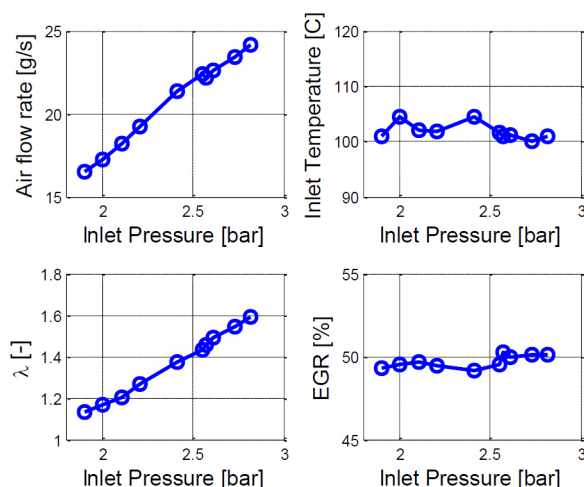


Figure 3. Inlet conditions as the inlet pressure was changed from 2.8 to 1.2 bar at 12 bar IMEP

As the fuel mixture gets richer the start of combustion is delayed and the ignition delay increases. This means that the homogeneity in the mixture increases which results in faster combustion, and as a consequence the maximum pressure rise will be higher. With a leaner mixture, the fuel from the first injection ignites separately followed by the second injection which ignites partially kinetically and partially in a diffusion flame. The spray-driven part of the combustion increases the combustion duration and moderates the combustion acoustic noise. The heat release data for the different lambda levels are presented in Figure 4 which clearly shows the transition of the combustion from partly spray-driven for the leaner mixture to more premixed combustion as lambda decreases.

The maximum pressure rise, maximum in-cylinder pressure as well as the cyclic variation of the IMEP for different lambda levels is presented in Figure 5.

NO_x, HC, CO and soot emissions are presented in Figure 6. As lambda increases the spray-driven part of the combustion becomes more distinguishable which can partly describe why NO_x increase as lambda increases in Figure 6. Another explanation for the lower NO_x emissions at lower lambda is the lower oxygen fraction.

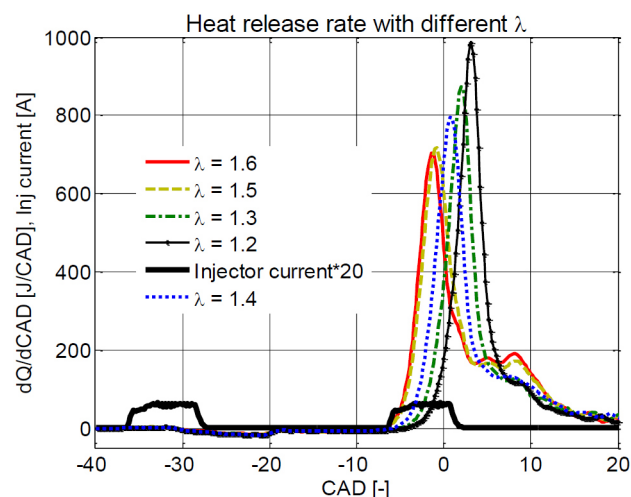


Figure 4. Heat release rate for different lambda at 12 bar IMEP

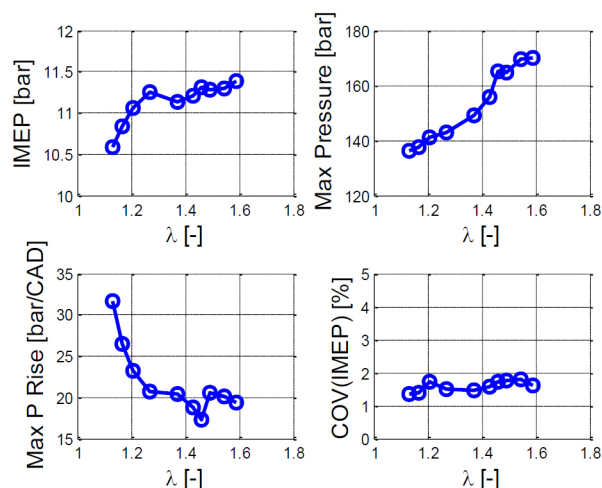


Figure 5. Pressure data such as Maximum pressure rise, Max pressure, IMEP(gross) and COV(IMEP) as a function of lambda

When lambda is lower than 1.3 the first detectable soot emissions can be detected. Soot formation is mainly due to the mixing of the fuel and air, and also the combustion temperature however the rate of the soot oxidation is determined by the oxygen concentration of the mixture, combustion temperature and combustion duration. As lambda decreases the mixture gets richer and the available oxygen