

Investigation of Performance and Emission Characteristics of a Heavy Duty Natural Gas Engine Operated with Pre-Chamber Spark Plug and Dilution with Excess Air and EGR

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ABSTRACT

This article deals with application of turbulent jet ignition technique to heavy duty multi-cylinder natural gas engine for mobile application. Pre-chamber spark plugs are identified as a promising means of achieving turbulent jet ignition as they require minimal engine modification with respect to component packaging in cylinder head and the ignition system. Detailed experiments were performed with a 6 cylinder 9.4 liter turbo-charged engine equipped with multi-point gas injection system to compare performance and emissions characteristics of operation with pre-chamber and conventional spark plug. The results indicate that ignition capability is significantly enhanced as flame development angle and combustion duration are reduced by upto 30 % compared to those with conventional spark plugs at certain operating points. Maximum possible dilution (limited by combustion stability index, Coefficient of Variation (COV) of Gross Indicated Mean Effective Pressure (IMEPg)) with excess air and EGR were investigated experimentally at engine speed of 1500 rpm and 5, 12 and 18 bar IMEPg operating load and results indicate that the lean limit is extended by 0.8-1 Lambda unit and 5-8% EGR rate units. It was also observed that pre-chamber spark plugs cause charge pre-ignition at loads exceeding 10-12 bar IMEPg. To avoid this, the minimum amount of dilution, with excess air and then EGR, which is required to operate above 10 bar IMEPg was estimated experimentally. Finally, to compare performance and emission characteristics of operation with these two ignition techniques, the engine was tested on an ESC-like (European Stationary Cycle) 12 mode cycle. Results indicated marginal reduction in cycle averaged NOx emissions with maximum excess air dilution and about 50% reduction with maximum EGR dilution, whereas CO and HC emissions increased. Other operating characteristics like the flame development angle, combustion duration, brake efficiency etc. are also compared for the tested operating range of the test engine.

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INTRODUCTION

Natural gas engines are an attractive alternative to the present day diesel engine technology for heavy duty application for reasons such as cheaper fuel, less expensive after treatment devices and increasing network of gas fueling stations worldwide. However, they lag behind the diesel counterparts in operating efficiency due to throttling losses and lower compression ratio which is limited by engine knock, high NOx emissions and excessive exhaust gas temperature when operated at stoichiometry. Fuel-lean combustion yields lower combustion temperatures which lead to lower NOx, and higher thermal efficiency due to lower heat losses but again, the extent of dilution is limited by the capability of the ignition system to reliably ignite the lean

mixture. To overcome this limitation, alternate ignition strategies [1] like laser induced spark ignition[2,3], diesel pilot injection [4,5] and turbulent jet ignition [6] have been proposed and evaluated by many researchers to a reasonable extent till now but they require much more detailed investigation before they can be commercialized.

Application of laser induced spark ignition in combustion engines is still in research phase and also difficult to implement in mobile application due to complexity of the laser ignition system. Diesel pilot injection in natural gas engine has been used to reasonable extent but mostly for stationary application due to fewer restrictions on space and ease in handling two fueling sub-systems. Recently, however, pilot ignited natural gas engines have been used in automotive applications [7]. As compared to these alternate

methods, turbulent jet ignition offers a more simplified solution since it requires minimum or no engine modifications and is comparatively less complex. Toulson et. al [6] have compiled a good review of development and state of the art of turbulent jet ignition systems. To the best of authors' knowledge, most of the studies till now have been done at low to intermediate engine load of upto 10 bar IMEP and mostly on single cylinder setup.

It is well known that premixed lean combustion is limited by decreasing flame velocity and unreliable ignition behavior leading to irregular mixture inflammation. Flame velocity can be increased both by increasing the degree of chemical reactivity with higher charge temperature (higher compression ratio) and increasing transport velocity with intense in-cylinder charge motion which can be achieved by appropriate intake manifold and combustion chamber design. Since the compression ratio is limited by knock, intensifying in-cylinder charge motion is an effective solution to enhance flame velocity; however, excessive turbulence near the spark plug electrode can cause high cycle to cycle variation in ignition and even misfire as the flame kernel may be blown off before it develops into a full-fledged flame. Hence, it is necessary to create a zone in the chamber, particularly in the area of the electrodes, with only slight charge motion and this is accomplished by a pre-chamber around the electrodes in a pre-chamber spark plug. Charge motion in the pre-chamber is mainly governed by the location and orientation of nozzle between the pre and main chamber; hence the ignition inside pre-chamber is less affected by cyclic variation of charge motion in the main chamber.

Another distinctive feature of pre-chamber spark plug is that after the ignition of charge, the pressure rise generates strong flame jets flowing out of the pre-chamber which offer two benefits. First, unlike a single point spark, these jets offer a spatially distributed ignition source which results in shorter combustion duration hence compensating for natural gas' inherent slow burning property. Secondly, the flame jets are believed to be formed of complete and partial combustion products with active species which further enhance ignition of charge in main chamber. The turbulent mixing of high velocity jet with main chamber charge also helps increase flame speed.

In this article, a pre-chamber spark plug has been extensively studied as a means to achieve effective turbulent jet ignition by experiments on a 6 cylinder 9.4 liter engine at operating loads up to 18 bar IMEPg. A 12 mode cycle averaged emission and performance comparison between pre-chamber and conventional spark plug operation are reported. Questions like the maximum operating load achievable with pre-chamber spark plugs and benefits in terms of extension of lean limits are answered.

DEFINITIONS

Some derived parameters used during this study will be defined in this section.

- Excess Air Ratio, Lambda (λ), dimensionless

$$\lambda = \frac{AFR_{actual}}{AFR_{stoichiometric}} \quad (1)$$

- Gross Indicated Mean Effective Pressure (IMEPg), bar

$$IMEPg = \frac{\int_{-180}^{180} p \cdot dv}{V_d} \quad (2)$$

- p = Cylinder pressure, Pa
- v = Cylinder volume, m^3
- V_d = Cylinder Displacement volume, m^3

- Exhaust Gas Recirculation (EGR) Rate, percentage (%)

$$EGR = \frac{C_{CO2,Inlet}}{C_{CO2,Exhaust}} \quad (3)$$

- $C_{CO2,Inlet}$ = Concentration of CO₂ in inlet, % vol/vol
- $C_{CO2,Exhaust}$ = Concentration of CO₂ in exhaust, % vol/vol

- Flame Development Angle, in milliseconds or degrees crank angle (deg. CA), is defined as time between spark timing and 10% heat release.
- Combustion Duration, in milliseconds or degrees crank angle (deg. CA), is defined as time between 10% and 90% heat release.

- Coefficient of Variation (COV) of IMEP, dimensionless

$$COV_{IMEP} = \frac{\sqrt{\frac{\sum (IMEP_g - \overline{IMEP})^2}{N}}}{\overline{IMEP}} \quad (4)$$

N = Number of cycle

METHODOLOGY

The experimental campaign was designed to study the dilution limit, maximum operating load and emission characteristics of engine operation with conventional spark plugs (C.SP) and pre-chamber spark plug (PC.SP), hereafter referred to as two cases. Experiments were performed in the following four sets.

Set 1 - Heat Release Characteristics

The intention was to compare heat release characteristics for the two cases at the same operating points with respect to engine speed, load and spark timing, so as to have a meaningful comparison of flame development angle and combustion duration which are representative of the ignition quality. The operating points were 1200, 1500 and 1800 rpm engine speed and 3, 6 and 9 bar IMEPg load at each engine speed with spark timing held constant at 12 degree crank

angle before top dead center (bTDC). This resulted in a basic 9 point test matrix and then dilution from excess air ratio (λ) of 1 to the lean limit in steps of 0.1 was studied. This load range was selected to avoid the region of operation where pre-chamber spark plug cause charge pre-ignition.

Set 2 - Comparison of Dilution Limit

The maximum dilution with excess air and EGR were separately determined for both cases. Experiments were performed at 1500 rpm engine speed and 5, 12 and 18 bar IMEPg engine load with spark timing set to knock limited maximum brake torque (MBT) timing for each operating point.

Set 3 - Minimum Dilution Required at High Operating Load

Operating a natural gas engine at stoichiometric air to fuel ratio above certain operating load causes problems like excessive exhaust gas temperature and charge pre-ignition when operating with pre-chamber spark plug. This means a minimum amount of dilution, with either excess air or EGR, is always required when operating above a critical engine load, which was found to be in the range of 10 to 12 bar gross IMEP for the experimental engine. This set of experiments were to determine this minimum dilution, while observing three main restrictions, namely exhaust temperature less than 700 deg. C, avoid charge pre-ignition and combustion stability (COV of IMEP less than 5%). It will be discussed in a later section how this minimum dilution imposes a restriction on the maximum achievable operating load for the two cases.

Set 4 - Cycle Averaged Emissions

With an intention to determine the benefits in terms of emission reduction due to higher diluted operation potential of pre-chamber spark plug, a 12 mode test cycle was employed for cycle averaged emissions and performance maps at stoichiometric, maximum Excess Air and maximum EGR diluted operating conditions. The test cycle was derived from the European Stationary Cycle (ESC) excluding the first mode when the engine is on idle. The weighing factors for the remaining 12 modes were increased proportionate with their original weighing factors to accommodate the weighing factor of the excluded first mode. The test cycle modes and weighing factors are tabulated in [Table 1](#). Engine speed of 1250, 1500 and 1750 rpm were selected as the speeds A, B and C with 15, 17 and 13 bar gross IMEP as the respective full (100%) load. These selections were made as they offer fair resemblance to the actual engine operating conditions for automotive application.

Table 1. 12 mode test cycle details.

Mode	Speed	Load (%)	13 Mode Weighing Factors	12 Mode Weighing Factors	Mode (new)
1	Idle	0	15		
2	A	100	8	9.42	1
3	B	50	10	11.76	2
4	B	75	10	11.76	3
5	A	50	5	5.88	4
6	A	75	5	5.88	5
7	A	25	5	5.88	6
8	B	100	9	10.6	7
9	B	25	10	11.76	8
10	C	100	8	9.42	9
11	C	25	5	5.88	10
12	C	75	5	5.88	11
13	C	50	5	5.88	12

Experimental Setup and Conditions

The experimental setup consists of a 6 cylinder, 9.4 liter engine equipped with a variable geometry turbocharger and a long route cooled EGR system. Main engine specifications are tabulated in [Table 2](#). Engine was equipped with a multi-point gas injection system and an inductive type ignition system with ignition energy of about 30 mJ. Both spark plug types were operated by same ignition system. Piston with “Quartette” type combustion chamber design ([Figure 3\(d\)](#)) that offers the highest turbulence level to enhance lean combustion is used, details of which can be found in [8]. The conventional spark plug used is ‘J’ type with iridium electrode and 0.3 mm electrode gap. All the experiments were carried out with inlet air temperature maintained at 40 ± 5 °C, coolant temperature at 82 °C and lubricating oil temperature at 100 °C. [Figure 1](#) briefly depicts the layout of experimental setup.

Table 2. Main specifications of experimental engine

Number of cylinders	6
Displacement	~ 9.36 liters
Bore	120 mm
Stroke	138 mm
Compression Ratio	12
Fuel	Natural Gas
Methane content	~ 89 mol-%
Methane number	~ 73
Stoichiometric air to fuel ratio	16.9
H/C ratio	3.84

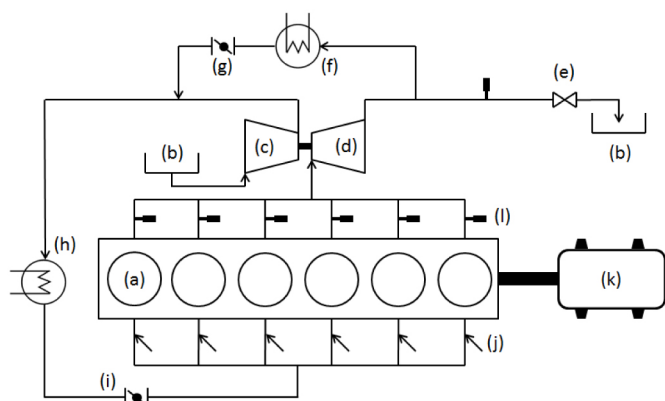


Figure 1. Layout of experimental setup. a) Cylinder, b) Reservoir, atmospheric temperature and pressure, c) Turbocharger compressor, d) Turbocharger Turbine, e) Back pressure valve, f) EGR cooler, g) EGR valve, h) Charge air cooler, i) Throttle, j) Gas injector, k) Engine brake and l) Lambda sensor

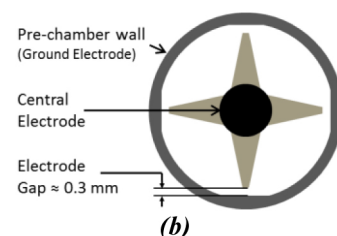
Each cylinder was equipped with a piezo-electric pressure transducer and pressure data was sampled with a resolution of 0.2 degree crank angle. All parameters were averaged over 200 consecutive cycles at each operating point. Emissions such as HC, CO, NO_x, CO₂ and O₂ are measured. EGR was calculated by measuring CO₂ at inlet and exhaust. Engine was also equipped with 7 wide-band lambda sensors, one for each cylinder and one after the turbocharger measuring engine's overall dilution with excess air.

Pre-Chamber Spark Plug

The pre-chamber spark plug was essentially a conventional spark plug with a hemispherical pre-chamber welded to its tip, as shown in [Figure 2 \(a\)](#). However, an important difference is that 4 pin-like arms extend out of the central electrode with their tips placed adjacent to the pre-chamber wall (ground electrode) with a spark gap of approximately 0.3 mm, as depicted in [Figure 2 \(b\)](#). The material of central electrode is platinum. Pre-chamber volume is approximately 0.2 ml, which equals to 0.141 % of the clearance volume for the experimental engine. Pre-chamber has 5 nozzles of 1.1 mm diameter arranged and oriented as depicted in [Figure 3\(a-b\)](#). Pre-chamber spark plug nozzle's orientation with respect to the combustion chamber cavity is kept identical during all experiments and is as depicted in [Figure 3\(c\)](#).

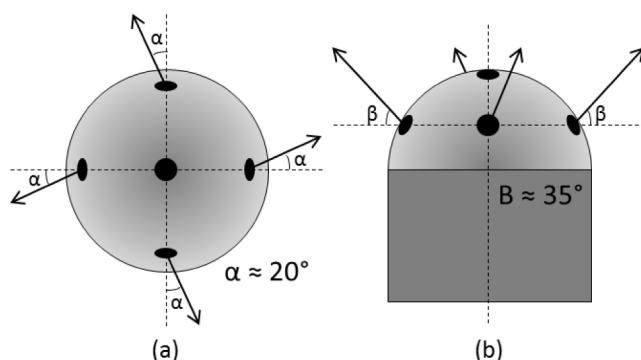


(a)



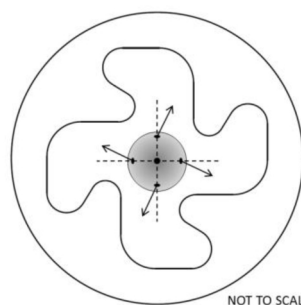
(b)

Figure 2. (a) Pre-chamber spark plug tip, (b) Central and ground electrode arrangement



(a)

(b)



(c)



(d)

Figure 3. Arrangement and Orientation of pre-chamber nozzles a) Top View, b) Front view; c) Spark plug location and orientation, d) Combustion chamber design