Fuel 153 (2015) 483–491

Contents lists available at ScienceDirect

Fuel

journal homepage: www.elsevier.com/locate/fuel

Impact of n-butanol and hydrous ethanol fumigation on the performance and pollutant emissions of an automotive diesel engine

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ARTICLE INFO

Article history:
Received 4 February 2015
Received in revised form 5 March 2015
Accepted 10 March 2015
Available online 24 March 2015

Keywords:
n-Butanol
Hydrous ethanol
Fumigation
Diesel engine
Particle number concentration

ABSTRACT

This work studied the impact of hydrous ethanol and n-butanol fumigation on the combustion characteristics, performance, pollutant emissions, particle number concentration and size distribution of an automotive diesel engine. Independently of engine load, both alcohols exhibited higher premixed combustion peaks, faster combustion process, and higher coefficient of variation of indicated mean effective pressure (imep) and reduced maximum in-cylinder temperature, in comparison with ultra low sulfur diesel (ULSD). Neither n-butanol nor hydrous ethanol presented better brake thermal efficiency (*bte*) and brake specific fuel consumption (*bsfc*) than ULSD. Engine performance with alcohol fumigation is highly susceptible to operating mode. Therefore, it is necessary to optimize the specific thermal conditions for implementing a fumigation strategy. Both alcohols increased carbon monoxide (CO) and total hydrocarbons (THC) and reduced nitrogen oxides (NOx) and particulate matter (PM), in comparison with ULSD fuel. However, the magnitude of this reduction was markedly affected by engine operating mode. The n-butanol showed the best trade-off (PM vs NOx + THC) among all fuels. In comparison with ULSD, hydrous ethanol fumigation decreased the total number concentration of particles while maintaining or increasing the geometric mean diameter, depending on the engine load. In comparison with ULSD, n-butanol maintained or reduced the total number concentration of particles and exhibited the opposite trend for the geometric mean diameter. The particle number concentration (PNC) and size distribution were not affected by engine load for n-butanol.

1. Introduction

Biomass-derived fuels such as biodiesel, alcohols, biomass-to-liquid, and hydrotreated vegetable oils, are being used as partial substitutes of conventional diesel fuel. In particular, alcohols are attractive because: they can be easily blended or injected into the engine, they are produced through fermentation processes from a high variety of non-edible vegetable and organic waste sources, and finally, they contain a high share of oxygen, which has the potential to reduce particulate matter (PM) and NOx emissions [1–3]. The most popular techniques for using alcohols in diesel engines are alcohol–diesel blending and alcohol fumigation in the intake port [4].

Although alcohol blending is the easier method to implement, there are some challenges to be faced: (i) some alcohols have poor solubility in diesel fuel (e.g., methanol [5]); (ii) although diesel/alcohols or biodiesel/alcohols reduce NOx, they may increase carbon monoxide (CO) and total hydrocarbons (THC) emissions [6]; (iii) alcohols might cause poor stability depending on blend temperature [7]; (iv) the hygroscopic capacity of some alcohols promotes corrosion in the injection system and the fuel tank [1,8]; (v) blends have less lubricity than diesel [1,9]; (vi) alcohols may reduce the cetane number of the blend (e.g., approx. 7 points for every 10% of ethanol in the blend [10]) and (vii) high alcohol losses due to evaporation from the fuel tank [11].

A good alternative to avoid these problems is alcohol fumigation, which has the following advantages: (i) high flexibility on diesel fuel substitution [12]; (ii) the amount of alcohol injected can be adjusted to match the actual engine requirement [13], (iii) hydrous alcohols can be used [14–16], (iv) simultaneous reduction of NOx and PM emissions [5,17], (v) in most cases, while the particle number concentration (PNC) decreases, the geometrical mean diameter is not affected [18] and (vi) evaporation losses are lower in

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http://dx.doi.org/10.1016/j.fuel.2015.03.022
0016-2361/

comparison with alcohol–diesel blends. However, this technique faces some disadvantages: (a) possibility of severe knock under high-load conditions due to the low cetane numbers of the alcohols [19] which limits the quantity of diesel substitution [15,20–24]; (b) the high heat of evaporation of alcohols may lead to ignition difficulties and high aldehyde emissions at cold start, warm up and low load operations [25–27]; (c) although it has proven to reduce PM and NOx, increases CO, THC [5,17] and NO_2 emissions [18,28,29] and (d) requires an additional fuel injection system and fuel tank adaptation.

From the literature review it can be concluded that alcohol fumigation: (i) has been studied mainly with ethanol (anhydrous and hydrous) and methanol [17], and only few recent works employ n-butanol [24,30,31]; (ii) increases fuel consumption due to the lower energy content of alcohols [12,30,32,33]; (iii) increases CO and THC emissions due to the combination of the decrease of in-cylinder temperature and the adsorption of alcohols in the lubricating oil layers [12,24,32,34]; (iv) reduces NOx emissions due to lower in-cylinder average bulk temperature induced by alcohol evaporation [12,20,24,30–34] and the water content in alcohol promotes additional NOx reduction [14–16]; (v) reduces PM due to the additional oxygen and to the decrease of aromatic and sulfur share of the diesel fuel [12,24,30–34], (vi) increases the biological activity of the soluble organic fraction [14,20,35,36]; (vii) needs more research on PNC and size distribution, especially for n-butanol, since data have not been reported to date; and finally, (viii) comprehensive engine mapping has started recently with methanol [37] but there is not reported research with other alcohols.

Considering the low price and availability of hydrous ethanol, the prospective of n-butanol as renewable fuel, its emerging research as a fumigation alternative, the lack of characterization on its particle number concentration (PNC) and size distribution, and finally, the absence of comparing research between both alcohols, this work aims to analyze the impact of hydrous ethanol and n-butanol fumigation on performance, combustion, pollutant emissions, and PNC and size distribution of an automotive diesel engine.

2. Methodology

2.1. Engine test rig

The schematic of the engine setup is presented in Fig. 1. Experiments were carried out in a 4-cylinder, 2.5 l, turbocharged, DI automotive diesel engine (Table 1) which was modified with a built-in house intake multipoint port injection system to substitute 10% of diesel fuel in energy basis by hydrous ethanol (H-Et10) and n-butanol (n-Bu10). In preliminary tests, it was found that the alcohol fumigation system, allowed replacing from 10% up to 20% in energy basis under the selected operating modes commented below. The lower reliable bound of 10% was given by the minimum time for injector opening and the upper limit of 20% was limited by intake valve closure. In this research 20% was not considered due to driveability issues. A substitution of 10% was selected because it is commonly used in governmental policies, it is also easily attainable for the international fuels market and it might be attractive for engine manufacturers in order to keep their engine warranties. In order to guarantee 10% alcohol substitution at a specific engine load (M2/M4), the engine was started with ULSD fuel and maintained at desired load until reaching stationary conditions [38]. Afterwards, the brake power was reduced to 90% by decreasing engine speed and maintaining engine torque. Finally, the alcohol fuel was injected until reaching back 100% of the initial brake power.

The engine was coupled to a Schenck W230 eddy current dynamometer. The air flow rate was measured with a Magnetrol TA2 hotwire sensor, and diesel and alcohol flow rates were measured with two separate Shimadzu electronic weight scales (±0.01 g). The instantaneous in-cylinder pressure was recorded with a Kistler 6056A piezoelectric pressure transducer coupled to a Kistler 5011B charge amplifier. The instantaneous piston position was measured with a Heidenhain ROD 426 angular encoder of 1024 pulses/rev. High speed data were acquired using the Labview™ software and a National Instruments™ data acquisition system (NI PCI 6024E and NI PCI MIO-16E-4).

A zero-dimensional, one-zone thermodynamic combustion diagnosis model [39], based on in-cylinder pressure signal was used. A total of 100 pressure curves were registered at each operation mode to ensure reliability in the combustion diagnosis results. CO and NO_x emissions were measured with an AVL Dicom 4000 NDIR sensor and total hydrocarbons (THC) emissions were recorded with a ThermoFID 2000e flame ionization detector through a heated line (190 °C). The PNC and distribution size were measured at two meters downstream from the exhaust manifold utilizing a Dekati Electrical Low Pressure Impactor (ELPI+) provided with a double diluter, using a 30% constant dilution. The temperature was set to 120 °C for both heaters of the Fine Particle Sampler (FPS), applying a vacuum pressure of 4000 Pa and the impactor charger voltage was set to 3900 V and the impactor charger current to 1.02 μA.

Specific PM was obtained with a dilution rate of 10 through a Ricardo partial dilution tunnel. Whatman microfiber glass filters of 47 mm diameters were conditioned to 22 °C and 45% humidity in a climatic chamber for 48 h before and after PM collecting procedure. A Shimadzu high precision weight scale ($\pm 1 \times 10^{-5}$ g) was used to determine the collected mass of PM.

2.2. Electronic fumigation system

Each alcohol was injected at a pressure of 300 kPa. The needle lift of each injector was controlled with a built-in house electronic control unit which was programmed in a *Freescale*™ microcontroller using *Labview*™ software. To ensure the synchronization of the alcohol injection timing, a proximity sensor was installed in the intake valve of cylinder #1. An engine speed sensor of 60 pulses per revolution was also implemented. Both sensors were connected to the microcontroller, which was programmed to configure and manage the alcohol injection process. The algorithm first configures the initial parameters for the correct operation of the microcontroller and then sets the injector opening time from the Labview-based software. Afterwards, the algorithm locates the intake top dead center (TDC) of cylinder #1 in order to synchronize the injection process of all alcohol injectors, which followed the order *2-3-4-1*, and calculates the injection duration (ms) according to the set point established by the user (open loop).

The duration of the intake stroke in a complete thermodynamic cycle (two engine revolutions) at 2410 min^{-1} (or 40.16 s^{-1}) is 24.89 ms (or 1000/40.16 ms). This means that the available time for alcohol injection is 12.45 ms (or 24.89/2 ms). Calculations established that alcohol droplets move from the injector nozzle to the intake valve in 4 ms. Considering this, the alcohol injector can be opened for up to 8.45 ms, to assure that the alcohol enters into the cylinder during intake stroke. Injection duration was set to around 2 ms for mode M4 and about 4 ms for mode M2, which are described below.

2.3. Design of experiments and repeatability tests

The following engine operating modes were selected: M2 (95 N m or 0.478 MPa of bmep at 2410 min^{-1}) and M4 (43 N m