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A numerical study on the effects of boot injection rate-shapes on the combustion and emissions of a kerosene-diesel fueled direct injection compression ignition engine

Kun Lin Tay, Wenming Yang *, Feiyang Zhao, Wenbin Yu, Balaji Mohan

Department of Mechanical Engineering, National University of Singapore, 9 Engineering Drive 1, Singapore 117576, Singapore

HIGHLIGHTS

- Boot injection rate-shapes give lower nitrogen oxide emissions in general.
- Soot mass, particle number and size for kerosene combustion are lower than diesel.
- Lower boot injection velocity gives larger soot particles during initial combustion.
- High main injection velocity gives narrow soot mass distribution during late combustion.

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ABSTRACT

In this work, the effects of boot injection rate-shapes on the combustion process and emissions formation of a direct injection compression ignition engine fueled with kerosene and diesel are investigated through numerical simulations. Boot injection rate-shapes with varying boot injection velocity and boot injection duration are used. The KIVA4-CHEMKIN code is used in conjunction with a phenomenological soot model and an improved kerosene-diesel reaction mechanism to study the combustion process and emissions formation. The phenomenological soot model consists of a number of sub-models from literature that accounts for soot particle inception, soot coagulation, soot surface growth via the hydrogen-abstraction-carbon-addition (HACA) mechanism and soot surface oxidation by oxygen (O_2) and hydroxyl radical (OH). It should be noted that the improved kerosene-diesel reaction mechanism is robust enough to predict the combustion and emissions trends of kerosene with respect to diesel. From this study, boot injection rate-shapes are seen to cause combustion phasing and cause lower nitrogen oxide (NO) emissions in general. Furthermore, it is observed that when kerosene replaces diesel, engine efficiency and NO emissions increase while carbon monoxide (CO) and soot emissions decrease. Soot mass quantity, soot particle number and soot particle size are the lowest for pure kerosene combustion. Finally, detailed analyses of the effects of boot injection rate-shapes on soot particle dynamics are also presented.

1. Introduction

In light of stricter engine performance and emissions standards [1], diesel engine technology has improved tremendously over the past years to keep up with the need for a more efficient and cleaner form of combustion. Nowadays, engine researchers are looking for ways to decrease carbon dioxide (CO_2), carbon monoxide (CO), unburnt hydrocarbon (UHC), soot and oxides of nitrogen (NO_x) emissions which are detrimental to human health [2] and the environment [3]. Moreover, enhancement of engine combustion efficiency is also high on the agenda of many engine researchers.

For a conventional direct injection compression ignition (DICI) engine, the entire combustion process and emissions formation are very much dictated by the fuel spray itself as well as the ambient in-cylinder conditions [4]. Therefore, achieving a more efficient combustion or a cleaner combustion requires the optimization of fuel injection as well as in-cylinder ambient conditions. For instance, introducing exhaust gas recirculation (EGR) is able to reduce thermal NO_x emissions [5] while increasing fuel injection pressure is able to reduce soot emissions [6] and increase efficiency [7] in a DICI engine. Of the many technological improvements to a conventional DICI engine, the fuel supply system is one aspect that

* Corresponding author.
E-mail address: mpeywm@nus.edu.sg (W. Yang).

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has advanced greatly over the years. Presently, the delivery of fuel in a DICI engine is achieved through a high pressure common-rail together with solenoid injectors [8] which are all electronically controlled. This allows the precise control of fuel delivery into the combustion chamber and more specifically the number of injection pulses, the injection quantity, the injection duration and the crank angle of fuel injection can all be controlled. Lately, the introduction of piezo-electric injectors [9,10] has enhanced the flexibility of fuel delivery as the response time of the piezo-stack is more rapid and needle-lift can be better controlled. Due to the recent introduction of this technology, highly defined injection rate-shapes are now practically achievable as seen in the recent works of Payri et al. [11] and Macian et al. [12].

Currently in literature, studies on the effects of injection rate-shapes on engine combustion and emissions have been limited. Furthermore, from the review paper of Mohan et al. [6], it was seen that of the many injection strategies employed by researchers for their works, injection rate-shape studies on engine combustion and emissions are few. From literature, Desantes et al. [13,14] investigated the effects of boot injection rate-shapes in a heavy-duty diesel engine by making alterations to the fuel injection system. From their studies, they concluded that boot injection rate-shapes were able to reduce NO_x emissions. With longer boot lengths or low boot pressures, NOx emissions decreased with inevitable increase of the break specific fuel consumption (BSFC) and soot emissions. Moreover, the heat-release profile was seen to change with injection rate-shaping as the combustion was dictated by the fuel spray. Also, d'Ambrosio and Ferrari [8] experimentally studied the effects of injection rate-shaping in a low temperature combustion (LTC) diesel engine using solenoid injectors. This was done by placing the injectors' electrical signals very close to one another such that two separate injections fused together to give a rate-shaped injection. They concluded from their study that injection rate-shaping was able to mitigate engine noise and simultaneously affect NO_x emissions albeit to a smaller extent. Apart from engine experiments, simulations were also carried out to study the effects of injection rate-shapes on engine combustion characteristics. Juneja et al. [15] from the University of Wisconsin-Madison numerically investigated the effects of rate-shaping in a diesel engine. They used the KIVA3V-CHEMKIN code for their simulations. From their study, it was found that using the appropriate injection rate-shape, NO_x as well as soot precursors could be decreased with an inevitable loss of engine power. In addition, Ghaffarpour et al. [16] computationally investigated the integrated effects of intercooler with injection rate-shaping and found that under certain diesel engine operating conditions, NO_x emissions could be cut by as much as 50%. Furthermore, Shuai et al. [17] studied a variety of injection rate-shapes in a diesel engine using the KIVA3V-CHEMKIN code. The rate-shapes included conventional rectangular rate-shape, triangular, trapezoidal as well as boot rate-shapes. From their simulations, it was found that the rectangular as well as the boot rate-shapes were able to reduce UHC, CO and soot emissions. More recently, Mohan et al. [18], through numerical simulations, investigated the effects of boot injection rate-shapes on combustion in a biodiesel fueled diesel engine. They found that a high boot pressure and a long boot length were able to give NO_x-soot trade-off.

From the above, it can be seen that investigations on the effects of injection rate-shapes on engine combustion and emissions are still quite limited. Moreover, the fuels used in the studies were high viscosity fuels like diesel and biodiesel. No low viscosity fuels were used. In addition, to the best of the authors' knowledge, no detailed analysis was done regarding the effects of injection rate-shapes on the trends of engine in-cylinder soot particle dynamics such as soot mass density, soot particle size and number distribution. Therefore, it is highly desirable to address the above issues. Hence, the objective of this work is to investigate through numerical simulations the effects of injection rate-shapes on the combustion process and emissions formation of a DICI engine fueled with a low viscosity fuel and its blends with diesel. The trends of engine in-cylinder soot particle dynamics will also be looked into in this work. The low viscosity fuel chosen for this work is kerosene because of two major reasons. First, the North Atlantic Treaty Organization (NATO) military has the intention of using JP-8 for all military ground vehicles and power generators in order to overcome the challenge of logistical fuel supply during war and peacetime operations [19,20]. Second, fuel adulteration is a big problem in certain parts of the world where kerosene is illegal mixed with diesel and used in diesel engines [21,22]. Thus, under these circumstances, studying the effects of kerosene-diesel blends on the combustion process and emissions formation in a DICI engine becomes essential.

2. Methodology

2.1. Chemical reaction mechanism

Numerical simulations will be done to investigate the effects of injection rate-shapes on the combustion characteristics of a DICI engine fueled with kerosene and diesel. Therefore, a recently developed kerosene-diesel reaction mechanism by the authors' group [23] will be used for the simulations. Concisely, the kerosene-diesel reaction mechanism had been thoroughly validated for DICI engine simulations. The kerosene sub-mechanism is capable of giving good ignition delay time agreements with that of experimental shock tube data from literature for initial shock tube conditions of 20 atm and equivalence ratios of 0.5, 1.0, 1.5 and 2.0. Furthermore, the kerosene sub-mechanism is able to predict the flame lift-off lengths (FLOLs), apparent heat-release rates (AHRRs) as well as spray combustion ignition delays in a constant volume combustion chamber (CVCC) reasonably well. As the decoupling methodology, which was proposed by Chang et al. [24], was used in the construction of the reaction mechanism, the reaction mechanism is kept very compact with only 48 species amongst 152 reactions, inclusive of a detailed set of $H_2/C_1/CO$ reactions as well as NO_x reactions. On the other hand, the diesel sub-mechanism, which is represented by n-decane, had been thoroughly validated by Chang et al. [24] in their work. For more details, refer to [23,24].

Due to the fact that soot particle dynamics trends such as soot mass density, soot particle size and number distribution are to be investigated in this work, a phenomenological soot model [25] is included for the engine simulations. More details about this model can be found in the following section. This soot model uses acetylene (C_2H_2) as the soot precursor specie and for soot surface growth. From diesel engine and CVCC experiments in literature, it can be clearly seen that diesel fuel generates more soot as compared to kerosene fuel under the same set of engine operating conditions [26,27]. This is due to the fact that diesel has a higher aromatic content than kerosene [26] and more importantly diesel's higher viscosity and surface tension [20,28] contribute to poorer

Table 1
Kerosene and diesel properties [20,28].

Properties	Kerosene	Diesel
Lower heating value (MJ/kg)	43.228	42.975
Cetane number; CN	38	46
Density (kg/m^3) @ 15 °C	812	843
Kinematic viscosity (mm^2/s) @ 40 °C	≈1.4	2.35
Surface tension (dyn/cm)	27.7	29.8
Volume fraction of aromatics	0.11	0.27