The effect of water injection on the combustion of diesel fueled and n-heptane fueled DI diesel engine

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(Manuscript Received: 17 July 2015; Revised: 29 October 2015; Accepted: 26 February, 2016)

Abstract

Diesel engines feature the most efficient type of internal combustion engines that suffer high emission content at the same time. This investigation proposes numerical water injection strategy from separate nozzle holes into combustion chamber in an effort to reduce NOx and soot. The water injection is practiced on both n-heptane fueled and diesel fueled engine in order to evaluate the capacity of this approach on different fuels combustion. It was found that the effect of water injection on n-heptane fueled engine imposes a moderate penalty on the engine efficiency; indicated efficiency of diesel decreases from 34.2% to 27.9% due to water injection whereas for n-heptane it falls from 33.84% to 28.7%. The results also show a significant NOx reduction and a slight reduction in soot emission. The numerical simulation with finite volume framework shows that for trade-off between emission control and maintaining engine efficiency, water injection approach must be applied to n-heptane fueled engine.

Keywords: CFD simulation; Diesel; Emissions; n-heptane fueled engine; Water injection

1. Introduction

The renewed attention to the global warming as the phenomenon of gradual increase in the average temperature of the Earth's atmosphere is being paid from policy makers, engineers, governments, and environmentalists. Some impacts of climate change includes glacier retreat, changes in the timing of seasonal events, generation of destructive tornadoes, reduction of agricultural production, etc. Internal Combustion (IC) based vehicles play a significant role in the total air pollution. For example, transportation produces almost 30 percent of all U.S. global warming emissions and collectively, cars and trucks account for nearly one-fifth of all U.S. emissions, emitting around 24 pounds of carbon dioxide and other global-warming gases for every gallon of gas [1]. Most ozone pollution is caused by motor vehicles, which account for 72% of nitrogen oxides and 52% of reactive hydrocarbons (principal components of smog) [2].

One of the main concerns of power generation in different segments of industry is to maintain the level of emitted hazardous gases within a prescribed limit. The attempts have also been accompanied by producing the different clean sources of energy such as biodiesel and electric vehicles. Although these trends may have been to reduce emissions or remove it (electric vehicles); however, each of the solutions are afflicted to some serious drawbacks and concerns. The electric vehicles are incomparably more expensive than IC engine vehicles even some governments provide subsidies to buy electric vehicles. Another major reason for the more popularity of the IC engine automobiles is the greater power that can be captured from the combustion of fossil fuels in the engine. Diesel engines demonstrate an acceptable rate of power delivery, although it exposes such as dramatic emission in exhaust gas that requires professional handling. Taking measures like after-treatment operation has been considered as a costly method, which involves expensive equipment.

Application of water due to its cooling effect is an efficient strategy to decrease the temperature and as a result to curb the engine out emissions. Application of water in internal combustion engines is implemented in one of the following ways: injection of water into the inlet manifold [3-5], direct injection of water into the combustion chamber [6-10], injection by another system of injection designed for water injection [11], involvement of water in combustion interaction thorough water emulsion such as diesel in water emulsion [12-16]. In the field of inlet water injection, Brusca et al. [17] reported the effect of water injection on diesel combustion by varying the ratio of water-to-fuel mass in the span of 0 to 15. The results indicated that this methodology is successful in controlling the level of NOx formation in diesel engine. In direct water injection, the quantity of water is limited compared to that of inlet water injection. A study is dedicated to examine the effect of direct water injection on DI diesel engine by a CFD KIVA-3v program [18]. It is concluded that water injection led to 35% increase in spray penetration, while the water injection is once again proved a pragmatic approach for lowering the amount of NOx and soot pollutants. While water injection can reduce the emissions, it was argued that it may inflict the engine efficiency and indicated power. However, Gonca [19] observed sim-
ultaneous increase in engine power and NOx emission by steam injection on a diesel engine running with ethanol-diesel blend. The results are substantial since when the engine mode is shifted from the diesel-ethanol to a pure diesel, NO is increased and torque is reduced. The typical fuel-water emulsion only contains around 12% water that is not sufficient for optimum PM and NOx decrement. Nonetheless, in some cases the researchers pronounced a satisfactory performance of water-fuel emulsion performance in terms of emission reduction in diesel engine. For example Fahd et al. [20] used 10% water emulsion diesel (ED10) in experimental study to evaluate performance and emission of a 2.5L turbo-charged Toyota diesel engine. The results showed that applying ED10 in engine produces lower engine out power and also a slight increase in fuel consumption is inevitable. On the other hand, this strategy led to a significant reduction in cylinder temperature and NOx emission.

In this study, the potential of applying direct water injection from a separate nozzle hole in combustion chamber of a diesel engine is assessed and afterwards the efficiency and emission indexes are compared with that of n-heptane-fueled diesel engine. In light of the provide literature survey, the numerical investigation of water injection on diesel engine fueled with alternative fuel in numerical method is scarce and this study tries to address the challenging issues and present a suitable solution for application of water injection in industrial scale.

2. Numerical Simulation

The used engine for simulation of the effect of water injection is coupled with an alternating current (AC) alternator for engine loading. The numerical model was established for a 2.0 liter water-cooled HSDI diesel engine with the specifications listed in Table 1. The engine was operated at medium load for every operational condition of the engine speed of 3500 rpm. The diesel engine is equipped with a VGT turbocharger and an intercooler. The numerical grid of combustion chamber is illustrated in Fig. 1 at 20˚ CA ATDC position with 35327 cells.

The numerical computations of the governed equations are conducted for a closed system from IVC (52˚CA BTDC) to EVO (110˚CA ATDC). The turbulent flow within the combustion chamber is so important especially after initiation of combustion and is modeled by using the RNG k-ε turbulence approach [21]. Meanwhile, the standard WAVE model was considered for the primary and secondary breakup process of the injected spray. In order to handle the heating and evaporation of the fuel spray, the Dukowicz model [22] was incorporated in the developed code. The interconnection between the water/fuel spray particles and the turbulent eddies as a result of fluctuating velocity has to be included in the mean gas velocity by a stochastic dispersion model [23]. This is assumed since the fluctuating velocity has a random Gaussian distribution. The Extend Coherent Flame Model, 3-Zone (ECFM-3Z Model) is based on the turbulent mixing methodology for the combustion in the combustion chamber [24, 25]. The model is implemented on the basis of a flame surface density transport equation and a mixing model that explains the inhomogeneous turbulent premixed and diffusion combustion phases. In this model, the fuel is assumed as a combination of more than one chemical species, such that the rate of reaction is divided for each fuel species. In this sense, it is possible to calculate the consumption rate of each component individually. The combustion model is based on Coherent Flame Model coupled with the spray simulation and has the capability of stratified combustion modeling and NO formation [26]. The extended combustion flame model is according to the ECFM-3Z [27] that is consisted of three zones of unmixed air plus EGR (if any exists), the mixed air and fuel zone, and unmixed fuel. Hence, the flame propagation moves from the burned gas to unburned gas area. This model recognizes three main combustion modes, namely: Auto Ignition, Premixed Flame (oxidation), and Diffusion Flame. For a diesel spray, there is a close distance of fuel droplets to each other and the fuel droplets can be classified at unmixed fuel part of the computational cell. After evaporation of droplet fuels, a certain time is required to transport from the pure fuel to mixed fuel and air region. As a result, the mixing of fuel with air is modeled by initially placing the fuel into the ‘unmixed fuel’ zone of the ECFM-3Z model [27]. The transport equation from the unmixed to the mixed zone is solved and presented in detail in Ref. [27].

The water and fuels (n-heptane and diesel) are co-sprayed at 3˚CA BTDC by two separate injector holes. The characteristics of each individual fuels have deep impact on combustion phase and when the water is injected, it will have different cooling effect on combustion of different fuels. The specifications of different fuels are listed in Table 2.
3. Results and Discussion

The combustion phenomenon is the result of fluid flow and the subsequent thermodynamic processes occurring in post-injection period. As a result, the quality of spray injection and mixture formation is pivotal factor for engine performance and the concentration of emissions. Two flow parameters that describe the preparation of mixture in the combustion chamber are quantified as following:

\[
\omega_{sx} = \frac{\sum_{i=1}^{n} m_i [(x_i - y_i)w_i - (z_i - z_0)w_i]}{\sum_{i=1}^{n} m_i [(z_i - z_0)^2 - (y_i - y_0)^2]}
\]

(1)

\[
SR_x = \frac{\omega_{sx}}{N}
\]

(2)

where \(\omega_{sx}\) [26] and \(SR_x\) [27] are the angular swirl velocity and swirl ratio, respectively to measure the swirl motion as
affected by water injection to diesel and n-heptane fuels. Here $n$ is the total number of cells in the meshed combustion chamber, $m_i$ denotes the mass within the cell, $(x_0, y_0)$ coordination is for the cylinder axis, while the Cartesian coordinates for the local cell centroid is $(x_i, y_i, z_i)$. The velocity components in direction of $x$, $y$, and $z$ directions are represented by $u_i$, $v_i$, and $w_i$, respectively. $SR_i$ is the swirl ratio about the $x$-axis and $N$ is the engine's angular velocity.

The fuel/air mixing quality is measured by “equivalence ratio uniformity index”, which is called “uniformity index”. The uniformity index, $\gamma$, is defined:

$$\gamma = 1 - \frac{1}{n} \sum_{i=1}^{n} \sqrt{(w_i - w)^2}$$  \hspace{1cm} (3)

where $w_i$ and $w$ are the local and average equivalence ratio, respectively. According to Eq. (3), is the ranged between 0-1 where $\gamma = 1$ implies the complete mixture uniformity and $\gamma = 0$ is when no mixing happens between fuel and air.

The variation of uniformity index with CA for different fuels on n-heptane and diesel with/without water injection is depicted in Fig. 2. It is observed that in the case of pure fuel injection, the uniformity of mixture is increasing consistently, while with water injection a local maximum happens at 728˚ CA. When water is injected, the mixing is more non-homogenous that shows the water injection is not desirable for mixing of fuel with accessible air. In this manner, the cooling effect of water droplets slugs the fuel vaporization and its mixing with the air. Moreover, n-heptane fuel has lower dynamic viscosity making smaller spray droplet and higher uniformity. The swirl number as a function of CA under the influence of water injection for different fuels is presented in Fig. 3. It is noted that without involvement of water in combustion chamber, there is no sensible change in swirling of air and mixture within combustion chamber for n-heptane and diesel. The effect of water injection however is noteworthy such that it increases the swirl for n-heptane. For the case of diesel, water injection increases the swirl only in early combustion period. The swirl of mixture in diesel-fueled engine with water injection at post combustion period is significantly lower than that of the case without water injection.

The effect of application of different fuels and water injection on heat release rate (HRR) can be seen in Fig. 4. It is evident from the figure that applying the water injection (WI) strategy is conducive to increase in ignition delay (this is in agreement with the results of previous studies [28,29]). By taking WI into account for diesel fuel combustion, an ignition delay of 0.247 ms is resulted whereas the ignition delay by water injection for the case of n-heptane fuel is 0.181 ms that is shorter than that of diesel. When the water droplets are introduced to the combustion chamber, the cylinder temperature falls and this gives enough time for fuel-air diffusion, therefore the ignition is retarded with water injection. The longer ignition delay provides more homogenous mixture that has higher HRR peak. The premixed combustion peak increases from 22.83 J/deg (at 725˚ CA) to 34.82 (at 731˚ CA) corresponding to diesel fuel ignition of pure diesel and “diesel+WI”, respectively. Water injection strategy deteriorates the combustion and as seen in Fig. 5, the pressure trace decreases for both n-heptane and diesel fuels. Based on the obtained results, the water injection in n-heptane fueled engine has lower adverse influence in comparison with diesel fuel combustion. This is in agreement with results of indicated engine power for different operational modes in Fig. 6, where n-heptane gives higher power than diesel and water injection in diesel-fueled engine comes with more power reduction penalty. The higher power of n-heptane is attributed to its inherent specifications such as higher cetane number compared to diesel. In order to achieve a broad insight into the performance of diesel-fueled and n-heptane fueled engines performance, Fig. 7 is presented for investigation of WI on indicated specific fuel consumption (ISFC). Firstly, diesel fuel has higher HRR peak and wall heat flux due to rapid heat transfer and lower stoichiometric AFR associated with diesel fuel. Secondly, it is observed that with WI, ISFC tends to increase since water droplets can decrease the LHV of both fuels hence increase the fuel consumption.

With regard to emission control measures by WI, temperature is graphed with respect to CA in Fig. 8 to justify the trends of NOx emission. Water injection due to its cooling effect caused temperature drop as seen in Fig. 8, subsequently in Fig. 9 NOx has been reduced significantly by adoption of WI strategy. This approach shows to be very effective in reducing NOx that a 83.4% decrease in diesel fuel and a 62.4% decrease in n-heptane fuel is resulted with water injection. The contour plots of soot emission in two positions of 10˚ CA ATDC and 20˚ CA ATDC are illustrated in Fig. 9 for n-heptane and diesel with/without WI. The soot mass fraction distribution clearly shows the effect of water injection on soot decrease for both n-heptane and diesel fuels. It can be concealed that WI is the most efficient approach to reduce the soot concentration of diesel-fueled engine rather than n-heptane fuel combustion. This is mainly due to lower temperature of chamber and higher swirl motion with water injection. This approach is capable of a remarkable soot accumulation zone especially in bowl area at 740˚ CA.
Figure 2. The variation of uniformity index in the range of 0-1 versus CA

Figure 3. Variation of swirl number for different fuels with/without WI versus CA

Figure 4. Variation of heat release rate for different fuels combustion with/without WI versus CA
Figure 5. In-cylinder pressure history for different fuels combustion with/without WI versus CA

Figure 6. Comparison of indicated power for n-heptane fueled and diesel fueled engine with/without WI

Figure 7. Comparison of ISFC for n-heptane fueled and diesel fueled engine with/without WI
Figure 8. in-cylinder temperature history for different fuels combustion with/without WI versus CA

Figure 9. Variation of NOx emission for different fuels combustion with/without WI versus CA

Figure 10. Spatial distribution of soot emission at 730 and 740°CA positions for different operational modes
4. Conclusions

The 3D CFD approach is conducted on a baseline DI diesel engine that uses both diesel and n-heptane fuels. Fuel and water is directly injected from two separate nozzle holes into combustion chamber. The engine is run on high-speed 3500 rpm and the results were obtained for fluid flow pattern, combustion and emission, and engine efficiency criteria. It was found that water injection may affect the mixing process adversely and it causes a serious combustion malfunctioning. Meanwhile, it is seen that water injection increases the ignition delay due to water vaporization cooling effect that subsequently reduces in-cylinder temperature and NOx concentration. The main advantage of water injection is reducing the main hazardous emissions from engine out manifold. It is also noted that n-heptane is more suitable for water injection scheme since the requirement for significant efficiency sacrifice is obviated.

References

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