INJECTION CHARACTERISTICS OF AN IN-LINE FUEL INJECTION SYSTEM USING THE ALTERNATIVE FUELS

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Abstract
The ever increasing ecological requirements encourage the use of alternative fuels. Some alternative fuels have advantages in production, some in transport and some in excellent injection as well as engine characteristics. Till today several alternative fuels for compression ignition engines have been investigated. This paper deals with the bio-diesel and with the wasted cooking oil. Biodiesel is produced from different plants (rape, Canola, Soya beans, palm tree...), while the wasted cooking oil is received from the catering industry. In this paper attention is focused on the fuel injection as well as fuel spray characteristics using diesel, bio-diesel and wasted cooking oil. The analysis of the fuel injection characteristics is obtained using the one-dimensional mathematical model for numerical simulation of injection process. In this model the influence of the snubber valve, the cavitation phenomena as well as variable fuel properties are taken into account. The influence of alternative fuels on spray characteristics is investigated using some empirical models. Furthermore, a computational fluid dynamics package is used for some analyses of the nozzle flow characteristics. Finally, on the basis of the numerically and experimentally obtained results, the harmful emissions using alternative fuels are predicted to some extent.

1 INTRODUCTION

Ecological and economical requirements set new limits in using the alternative fuels in Diesel engines. A look back to the Rudolf Diesel's invention origin shows, already his first ideas were using peanut oil as a fuel for his compression ignition engine. Latter on he decided for the petroleum oil based diesel fuel, which was also used through the decades.

Energy source independence demands, caused by oil crisis, lead to the idea of using the vegetable and animal fat based oils instead of petroleum oils. At the end of 80's several countries decided to turn partly to the alternative fuel produced from the vegetable sources, generally named Biodiesel.

Biodiesel has similar characteristics as petroleum based diesel fuel. The main advantage is its agricultural source, so Biodiesel produced from plants doesn't contribute to net rising of the CO₂ emissions in the atmosphere and consequently to the greenhouse effect (if the CO₂ emitted from fertilizer production and the process of esterification is not taken into account)[1][2][3]. So the use of the Biodiesel should help countries to lower the net CO₂ emissions.

Biodiesel can be produced not only from the vegetable source and animal fats, but also from the waste cooking oil (WCO) from the catering industry. Waste cooking oil can be cleaned and either mixed together with other fuels (petroleum diesel, Biodiesel) or used alone.

Many analyses assembled by the American Environmental Protection Agency (EPA)[4] and the results in [5] showed that the emissions of the particulate matter (PM), unburned hydrocarbons (HC) and carbon monoxide (CO) when using Biodiesel are less or at least equal to those of the petroleum diesel fuel. At the other side the nitrogen oxides emissions can be
lower as well as higher. According to [6] the lubrication characteristics of the Biodiesel are better in comparison to those of the low sulphur diesel fuel.

To date known disadvantages are slightly higher fuel consumption, negative influence on the rubber materials and bad low-temperature operation conditions.

Analysis on operation with different filtered and transesterified waste cooking oils [7] showed the increase of fuel consumption and all emissions except HC.

The analysis of the fuel injection characteristics is obtained using the one-dimensional mathematical model for numerical simulation of injection process [8]. The influence of alternative fuels on spray characteristics (Sauter mean diameter, penetration length and spray cone angle) is investigated using some empirical models. Furthermore, a computational fluid dynamics package (FIRE ver.62b) is used for some fast analyses of the nozzle flow characteristics. At the end, on the basis of the numerically and experimentally obtained results, the emissions using alternative fuels are predicted to some extent.

2 THEORETICAL BACKGROUNDS

2.1 Fuel characteristics

Even though vegetable oil and animal fats could be burned as pure vegetable oil, they are rarely used in this way. Pure oil exists in the form of triglycerides (C12-C18), consisting of three hydrocarbon chains connected together by glycerol. The main problem considering the vegetable oil is its high viscosity, causing the problems concerning the fuel flow from the tank to the engine. For example: the viscosity of Canola, which is a special kind of rape seed, is 12 times higher as diesel fuel at 20°C and 8 times even at 80°C [3].

Those problems can be mitigated by preheating the oil and using wider fuel lines, by blending the vegetable oil with diesel fuel or by chemical modification, i.e. producing the Biodiesel. Biodiesel is a generic name for fuels obtained by transesterification of the vegetable or animal oils. At this process the ester bonds in triglyceride are hydrolysed to form free fatty acids, which react with methanol or ethanol to form methyl or ethyl esters. This produces thinner, less viscous (for example 39 mm²/s (oil), ca. 5 mm²/s (ester) [9]) and more volatile fuel. Because of those esters Biodiesel fuels are commonly named Rape Methyl (or Ethyl) Ester (RME or REE).

The characteristics of Biodiesel may vary in dependence of the source plant/oil. The properties of some methyl esters, assembled from different sources, and diesel fuel are compared in Table 1.

Table 1: Comparison of typical properties of different fuels [3][10][11]

<table>
<thead>
<tr>
<th></th>
<th>D2 Diesel fuel</th>
<th>Biodiesel (DIN)</th>
<th>Canola</th>
<th>Palm oil methyl ester</th>
<th>Sunflower oil methyl ester</th>
<th>Soy methyl ester</th>
<th>Tallow methyl ester</th>
<th>Waste cooking oil</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\rho$ [kg/m³]</td>
<td>820-845</td>
<td>875-900</td>
<td>922</td>
<td>880</td>
<td>880</td>
<td>884</td>
<td>877</td>
<td>915</td>
</tr>
<tr>
<td>$\nu$ [mm²/s]</td>
<td>2-4.5</td>
<td>3.5-5.0</td>
<td>37</td>
<td>5.7</td>
<td>4.6</td>
<td>4.08</td>
<td>4.1</td>
<td>36.7</td>
</tr>
<tr>
<td>$H$ [MJ/kg]</td>
<td>42.6</td>
<td>37.3</td>
<td>36.9</td>
<td>39.8</td>
<td>38.1</td>
<td>40.3</td>
<td>39.9</td>
<td>n.d.</td>
</tr>
<tr>
<td>Cetane number</td>
<td>46</td>
<td>&gt;49</td>
<td>n.d.</td>
<td>62</td>
<td>49</td>
<td>46</td>
<td>58</td>
<td>n.d.</td>
</tr>
</tbody>
</table>
2.2 Injection system

The calculations were made for the conventional fuel injection system (CFIE) with the characteristics presented in Table 2 and Figure 1. The values presented in the table are in mm or mm².

Table 2: CFIE characteristics

<table>
<thead>
<tr>
<th>PUMP: Bosch PES4P120A72</th>
<th>plunger lift</th>
<th>11</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>plunger diameter</td>
<td>12</td>
</tr>
<tr>
<td></td>
<td>plunger stroke</td>
<td>11</td>
</tr>
<tr>
<td></td>
<td>plunger prelift</td>
<td>3,47</td>
</tr>
<tr>
<td></td>
<td>retraction volume</td>
<td>90</td>
</tr>
<tr>
<td></td>
<td>snubber valve hole diameter</td>
<td>0,5</td>
</tr>
<tr>
<td>HP tube</td>
<td>inner diameter</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td>length</td>
<td>616</td>
</tr>
<tr>
<td>INJECTOR: Bosch KBAL65S NOZZLE: Bosch DLLA 148S</td>
<td>no. nozzle holes</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td>hole diameter</td>
<td>0,375</td>
</tr>
<tr>
<td></td>
<td>seat diameter</td>
<td>2,5</td>
</tr>
<tr>
<td></td>
<td>needle guide diameter</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>max. needle lift</td>
<td>0,35</td>
</tr>
</tbody>
</table>

Figure 1: Schematic figure of the CFIE

2.3 Mathematical model for fuel injection

The calculations were performed by using our one-dimensional mathematical model for calculation of the CFIE injection processes, presented in [8]. In this model the influence of the snubber valve, the cavitation phenomena as well as variable fuel properties are taken into account. Using the mathematical model, the injection pressure \( (p_m) \), pressures along the injection system, injection rate \( (q') \), fuelling \( (q \text{ or } q_c) \), delivery valve lift \( (h_{dv}) \), snubber valve lift \( (h_{sn}) \) and the needle lift \( (h_n) \) in dependence of the camshaft \( (\varphi) \) angle are calculated. Some other important injection parameters are also given. Among them the following are considered here: mean injection rate (MIR), fuelling during the first 0,5 ms of injection \( (q_{0,5}) \) and squarness \( (s_q) \). In addition, the empirical models for calculation of the spray characteristics are included in the mathematical model.

2.4 Empirical models for spray characteristics calculations

By using the below printed equations the following spray characteristics were analysed: Sauter mean droplet diameter, spray penetration length and the spray cone angle.

SAUTER MEAN DIAMETER \( (d_{32}) \) represents the ratio between the sums of the droplets' volumes to the sum of the droplets' surfaces and is commonly used for the definition of the fuel spray atomisation. In the literature several empirical models for the definition of \( d_{32} \) can be found [12]-[17]. In the present research the models presented by Filipović [12] (eq.1) and Hiroyasu [14] (eq.2) were employed.

\[
d_{32} [\mu m] = 324.6 \left( \frac{\mu_a \cdot u_0^2 \cdot d_h}{\sigma_f} \right)^{-0.231} \left( \frac{\rho_f \cdot d_h \cdot \sigma_f}{\mu_f^2} \right)^{-0.082}
\]  
(eq.1)
\[ d_{32} \ [\mu m] = 2.39 \cdot 10^{-3} \cdot \Delta p^{-0.135} \cdot \rho_a^{0.121} \cdot \mu_c^{0.31} \]  
(eq.2)

Where \( \rho_a \) present air and \( \rho_f \) fuel density, \( \sigma_f \) is the fuel surface tension and \( \mu_f \) is the dynamic viscosity of the fuel. The outflow velocity is defined as \( u_0 \), \( \Delta p \) is the pressure difference, while \( d_h \) is the nozzle hole diameter.

The first one was chosen because the fuel properties are taken into the consideration, while the other one (developed for the calculation of the diesel fuel sprays) predicts the real size of the droplets at the diesel fuel injection process very good.

SPRAY PENETRATION LENGTH \( (L_p) \) is mainly dependent on injection duration and the pressure in the injection/combustion chamber. Again several empirical models were presented, for example [14],[18]-[21]. The models presented by Yule and Filipović [18] (eq.3) and Jimenez et al.[19] (eq.4) were employed to calculate the penetration length in this paper.

\[
L_p \ [mm] = 2.65 \cdot 10^3 \cdot d_h \cdot \left( \frac{\rho_a \cdot u_0^2 \cdot d_h}{\sigma_f} \right)^{0.1} \cdot \left( \frac{\rho_f \cdot u_0 \cdot d_h}{\mu_f} \right)^{0.3} \cdot \left( \frac{\rho_f}{\rho_a} \right)^{0.08} 
\]  
(eq.3)

\[
L_p \ [mm] = 0.6 \cdot 10^{-3} \cdot u_0 \cdot t^{0.9} \cdot \left( \frac{\rho_a}{\rho_f} \right)^{-0.163} 
\]  
(eq.4)

Where, beside of already mentioned parameters, \( t \) is the injection time.

SPRAY CONE ANGLE \( (\theta) \) was calculated from the empirical equations derived by Hiroyasu [14] (eq. 5) and Siebers [22] (eq.6).

\[
\theta = 0.05 \cdot \left( \frac{d_h}{\mu_a} \right)^{0.5} \cdot (\rho_a \cdot \Delta p)^{0.25} 
\]  
(eq.5)

\[
\tan \frac{\theta}{2} = 0.0043 \cdot \sqrt{\frac{\rho_f}{\rho_a}} 
\]  
(eq.6)

Here \( \mu_a \) is the dynamic viscosity of the air and \( c \) is a constant dependent on the nozzle hole diameter. From the results of Siebers [22] and by taking into account the dimensions of the nozzle, the constant \( c \) is 0.257.

3 NUMERICAL EXAMPLES

The analyses were performed at three different engine operating conditions (Idle, Rated and Peak torque) for three different fuels: Biodiesel, Waste Cooking Oil and D2 fuel.

To perform the calculation, the pump and the nozzle dimensions must be specified. The dimensions needed for the program run are presented in Table 2. The operating conditions are defined by setting the pump rotational speed and the geometrical duration of delivery (IDLE: 250 min\(^{-1}\), 1.05 mm, TORQUE: 600 min\(^{-1}\), 1,976 mm, RATED: 1000 min\(^{-1}\), 1,976 mm). The parameters were set according to the values of idle, rated power and peak torque at the pump (i.e. engine) operation using the D2 diesel fuel. Those settings are probably not the optimal conditions for the alternative fuels, but they enable a direct comparison. For the optimal operating conditions the geometrical duration of delivery, as well as the injection timing for the engine operating conditions should be changed. The characteristics of the used fuels are presented in Table 1. For the calculation of the spray characteristics, the above mentioned empirical models were employed.
4 RESULTS

4.1 Fuel injection characteristics

As can be seen from Figure 2 and Table 3, the differences between the results by using different fuels are quite small. Actually, the highest differences occur for the injection pressure and the fuelling during the needle closing at the rated conditions.

In all three operating conditions the maximum pressure and fuelling are always the highest in case of using the WCO. The other important parameter connected with the injection pressures is squarness, which is defined as the ratio of mean injection pressure to maximum injection pressure. To avoid unnecessary pump load, caused by a narrow peak of maximum injection pressure, the ratio should be high enough. The differences between the squarnesses at the rated and torque operating conditions are not significant, since the maximal difference is less than 4%.

![Figure 2: Injection pressure and fuelling for RATED (left) and TORQUE (right) operating conditions](image)

Table 3: Comparison of some calculated values

<table>
<thead>
<tr>
<th></th>
<th>RATED</th>
<th>TORQUE</th>
<th>IDLE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Percentage of fuel injected during the first 0.5 ms of injection</td>
<td>( \frac{q_{0.5}}{q_c} ) [%]</td>
<td>D2 16,40</td>
<td>9,16</td>
</tr>
<tr>
<td></td>
<td>Bio</td>
<td>16,01</td>
<td>9,83</td>
</tr>
<tr>
<td></td>
<td>WCO</td>
<td>16,56</td>
<td>9,39</td>
</tr>
<tr>
<td>Percentage of fuel injected during the needle closing</td>
<td>( \frac{q_{close}}{q_c} ) [%]</td>
<td>D2 12,96</td>
<td>8,60</td>
</tr>
<tr>
<td></td>
<td>Bio</td>
<td>7,88</td>
<td>8,90</td>
</tr>
<tr>
<td></td>
<td>WCO</td>
<td>7,91</td>
<td>9,17</td>
</tr>
<tr>
<td>Mean injection rate</td>
<td>MIR [mm³/CAM]</td>
<td>D2 87,5</td>
<td>75,3</td>
</tr>
<tr>
<td></td>
<td>Bio</td>
<td>87,8</td>
<td>76,3</td>
</tr>
<tr>
<td></td>
<td>WCO</td>
<td>88,8</td>
<td>75,4</td>
</tr>
<tr>
<td>Max. injection pressure</td>
<td>( p_{inj,\text{max}} ) [MPa]</td>
<td>D2 73,4</td>
<td>62,5</td>
</tr>
<tr>
<td></td>
<td>Bio</td>
<td>85,9</td>
<td>63,2</td>
</tr>
<tr>
<td></td>
<td>WCO</td>
<td>92,2</td>
<td>65,1</td>
</tr>
<tr>
<td>Squarness ( p_{\text{mean}}/p_{\text{max}} )</td>
<td>( s_q )</td>
<td>D2 0,609</td>
<td>0,595</td>
</tr>
<tr>
<td></td>
<td>Bio</td>
<td>0,587</td>
<td>0,598</td>
</tr>
<tr>
<td></td>
<td>WCO</td>
<td>0,587</td>
<td>0,597</td>
</tr>
</tbody>
</table>
To predict the emissions of the combustion process, two very important parameters are the fuelling during the first part of the injection and during the needle closing. A larger amount of fuel injected during the so-called period of the ignition delay yields higher emissions of the NOx, while the fuel injected during the needle closing or even later yields higher particulate mater (PM) emissions. Considering the differences in fuelling, the percentages of the fuel injected during the observed period were compared. The differences comparing the first parameter are not significant. A larger difference occurs only in case of Biodiesel at the idle operating conditions. Since the quantity of the fuel injected during the needle closing should be less than 8% of the total fuelling, the use of alternative fuels yields good results at the rated operating conditions, were the percentage was lowered from 13 to 8%. At the torque operating conditions, the percentage of fuel injected during the needle closing remains at the almost same level for all three fuels.

4.2 Spray characteristics

The results of the SAUTER MEAN DIAMETER calculations using the empirical model of Filipović are presented on the following figure. The differences by using different fuels are significant. On the other side, the results by using the Hiroyasu model doesn’t differ much. That is the effect of the model, where the fuel characteristics are not considered.

![Figure 3: d_{32} results using the Filipović model](image)

Considering the SPRAY PENETRATION LENGTH, the results differences between the two employed models are significant. The first model, derived for the alternative fuels by Jimenez, predicts quite high values. These do not differ significant from one to another fuel. The highest differences are at the idle conditions were the largest difference is about 5%. The predicted values for maximal penetration length are about 230 mm at idle, 520 mm at torque and 490 mm at rated conditions. The calculated values of the spray penetration length using the Yule - Filipović model are presented on the figure below.

![Figure 4: Spray penetration length results using the Yule - Filipović model](image)
The SPRAY CONE ANGLE calculated using either the model of Hiroyasu or Siebers doesn’t differ significant between the different fuels. In the first case, the spray cone angle is dependent only on the air properties, nozzle hole diameter and pressure difference, while the other model takes into account also the fluid properties. The results of the Hiroyasu model are around 60° at rated and torque and 55° at the idle conditions, while for the model presented by Siebers the values are around 5° at all conditions. These large differences between both models and the fuel independence of the results indicate that the employed models are probably not suitable to be used on the present problem.

The above presented results by using the Filipović and Yule-Filipović model shows that the diesel fuel spray is shorter and better atomised compared to other (alternative) fuels. Those results show that the fuel density and viscosity influence the spray formation significantly. The higher the fuel density and viscosity, the larger droplets occur. The result of worse fuel atomisation is also a higher spray tip penetration. The presented facts were also verified by some fuel spray observations at the pintle nozzle, where the spray formed at the WCO injection was narrower and longer than the spray in other cases.

4.3 Numerical analysis

Some fast analysis using the CFD package FIRE (ver.62b) considering the in-nozzle flow and the flow coefficient calculations were made by some previous researches [23]. The results (figure 5) showed that the use of different alternative fuels has no significant influence on the in-nozzle flow characteristics (velocity flow field and pressure distribution). Slightly higher flow coefficients (i.e. mass flow at the outlet) were calculated for the WCO, Biodiesel and some other plant oil or animal fat based methyl esters.

Figure 5: Numerical analysis results
5 CONCLUSIONS

From the above mentioned results the following conclusions and guidelines for the future work can be made:

- The results of the existing one-dimensional mathematical model show no significant differences between the injection characteristics using either diesel or alternative (Biodiesel, WCO) fuel. WCO and Biodiesel predict slightly higher injection pressures and fuellings, which match with the results of the numerical analysis [23]. To verify the results of the injection process calculation with alternative fuels, some experimental work considering the injection pump testing should be done.

- Since the mathematical model was programmed for the calculation of the diesel fuel injection, probably some modification, to assure higher accuracy at the injection process calculation using the alternative fuels, should be done.

- Considering the spray analysis, the alternative fuels gave worse atomisation (larger \(d_{32}\), longer \(L_p\)), which could cause problems in the combustion process. For optimal injection and combustion process, the pump and the engine should be optimised to run either on Biodiesel or WCO.

6 REFERENCES


[17] Hakki O.Z., Calculation of Spray Penetration in Diesel Engines, SAE 690254
[22] Siebers D.L., Scaling Liquid-Phase Fuel Penetration in Diesel Sprays Based on Mixing-Limited Vaporization, SAE 1999-01-0528