Flow simulation of petroleum diesel fuel and rapeseed oil in the nozzle of a diesel injector

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Abstract. The flow of petroleum diesel fuel (DF) and rapeseed oil (RO) in the nozzle of a diesel injector under operating conditions was simulated numerically. Investigated the general flow characteristics, such as mass flow rate, discharge coefficient, injection rate, averaged turbulent kinetic energy at the nozzle outlet and volume concentration of fuel vapor in the nozzle, and local flow parameters. The saturation pressure of RO is determined by summing the saturation pressure of constituting RO triacylglycerides. At a fuel temperature of 40 °C, the mass flow rate, discharge coefficient, injection velocity and turbulent intensity at the outlet of the nozzle hole for RO are significantly less than petroleum DF for all injection pressures, and the RO cavitation level (fuel vapor) in the nozzle hole is also lower than petroleum DF due to extremely slow saturation pressure of RO. However, this difference between RO and petroleum DF decreases with increasing injection pressure due to the different cavitation levels of DF at different injection pressures.

1. Introduction
In recent years, alternative vehicle fuels have been widely considered in light of energy security and environmental degradation. Vegetable oils are considered as a promising energy source to replace fossil fuels in compression ignition engines (CIEs), because vegetable oils are renewable and their direct use in CIEs reduces smoke, SOx, CO2 and aromatic compounds-carcinogens emissions [1].

The wide use of vegetable oil in diesel engines is hindered by the difference between the physicochemical properties of vegetable oils and those of petroleum diesel fuel (DF), especially the high viscosity of vegetable oils is 10-20 times higher than that of petroleum DF [2]. Markov et al. [3,4] studied the viscosity characteristics of two-component and multicomponent mixed biofuels based on vegetable oils and developed a method for determining their viscosity. It is noted that vegetable oils are characterized by increased viscosity, which has a negative effect on the processes of mixture formation and combustion. Increased emissions of carbon monoxide (CO) and unburned hydrocarbons (CHx) in exhaust gases from diesel engines fuelled with pure vegetable oils have been reported in experimental studies [5-9]. This may be due to the fact that the higher viscosity of vegetable oil makes fuel difficult to atomize and evaporate, which leads to locally rich mixtures contributing to incomplete combustion and formation of CO and CHx due to a lack of oxygen [8, 10-12]. However, the opposite results of the influence of vegetable oils on the emissions of these harmful substances have also been obtained [13-
[15], which is related to the presence of oxygen in the vegetable oil molecules. Therefore, in order to clarify the effect of vegetable oils on the combustion process in the engine combustion chamber, first of all, it is necessary to study the influence of vegetable oil properties on fuel injection, atomization, evaporation and mixture formation processes, in particular, the fuel flow in the nozzle, which has a great influence on the fuel atomization, fuel evaporation and formation of a mixture of fuel and air, which may further affect the performance of diesel engines [16].

Dernotte et al. [17] experimentally studied injection rates in a non-cavitation diesel nozzle for different fuels with kinematic viscosity from 0.6 to 7 mm²/s and density from 683 to 876 kg/m³. At low injection pressures, an increase in fuel viscosity leads to a decrease in fuel discharge coefficient to 10%, and at high pressures, only fuel density affects mass fuel rate. However, the kinematic viscosity of most vegetable oils at a temperature of 20 °C is in the range of 50 to 130 mm²/s [3,4,18-20]. Numerical simulation of the cavitation flow of Jatropha curcas oil [21], performed for a diesel nozzle, showed that the cavitation level of Jatropha curcas oil is much less than that of diesel fuel. However, the authors did not report the saturation pressure of Jatropha curcas oil, which is depended on oil composition and affects the occurrence and development of the cavitation process in nozzle [22].

A lot of research has been related to the flow of biodiesel fuels obtained from vegetable oils in injector nozzles [23-26], but the flow characteristics of vegetable oils have not been studied enough. The objective of this study is to investigate the steady flow characteristics of pure rapeseed oil (RO) compared to petroleum DF in the nozzle of a diesel injector. Analyzed the distribution of pressure, flow velocity, vapor volume concentration and turbulence inside the nozzle hole at various injection pressures, as well as general parameters of the fuel flow at the outlet of the nozzle hole, such as mass flow rate, injection rate and turbulent kinetic energy.

2. Nozzle geometry and fuel properties

Flow simulation of pure RO and petroleum DF was performed in an AZPI nozzle 171.07.00 form Altai precision products plant (AZPI Russia). This nozzle is assembled in a diesel injector FDM-22 for a 4-stroke turbocharged diesel engine D-245.12S (with cylinder bore-to-stroke ratio 11/12.5). The structural scheme of the studied nozzle is presented in figure 1. Some specifications of this nozzle are given in table 1.

![Figure 1. Structural scheme of The design diagram of the AZPI nozzle 171.07.00](image)

Table 1. Some specifications of the investigated nozzle.

| Manufacturer, Mark number | Diameter of nozzle holes \( d_n \), mm | Number of nozzle holes | Maximum needle lift \( h_{n\,\text{max}} \), mm | Length of the nozzle hole, \( L_{\text{hole}} \), mm |
|---------------------------|--------------------------------------|------------------------|----------------------------------|----------------------------------|
| AZPI, 171.07.00           | 0.35                                 | 5                      | 0.32                             | 1.1                              |

Flow simulation in the flow part of the diesel injector nozzle was carried out for petroleum DF and RO. Some physicochemical properties of these fuels are given in table 2 [2, 3, 28]. The saturation pressure of RO at low temperatures was determined by summing the saturation pressure of constituting
RO triacylglycerides. And the saturation pressure of triacylglycerides can be calculated by the Clausius-
Clapeyron equation [29]:

\[
\ln P(T) = \frac{-\Delta G_{\theta}^{vap}}{R \theta \ln 10} + \frac{\Delta H_{\theta}^{vap}}{R \theta} \left( \frac{1}{T} - \frac{1}{\theta} \right)
\]

where, \(\theta\) – the reference temperature (298.15 K), \(R\) – the universal gas constant, \(\Delta G_{\theta}^{vap}\) – Gibbs free
evaporation energy at reference temperature \(\theta\), \(\Delta H_{\theta}^{vap}\) – the enthalpy of evaporation at reference
temperature \(\theta\). Herewith \(\Delta G_{\theta}^{vap}\) and \(\Delta H_{\theta}^{vap}\) were determined by using Fragment based-approach as the
sum from each fragment of triacylglycerides - glycerol and three fatty acids [30]. Figure 2 presents the
calculated results of the saturation pressure of RO depending on the temperature.

**Table 2.** Physicochemical properties of petroleum DF and RO.

| Property                               | Petroleum DF | RO   |
|----------------------------------------|--------------|------|
| Conditional composition formula        | C_{16,2}H_{18,5} | C_{57,0}H_{101,6}O_{2} |
| Molecular mass, kg/m³:                 | 223.3        | 883.04 |
| - at 20 °C                             | 830.0        | 909.8 |
| - at 40 °C                             | 822.7        | 896.3 |
| Kinematic viscosity, mm²/s:            |              |      |
| - at 20 °C                             | 3.8          | 75.0  |
| - at 40 °C                             | 2.4          | 41.5  |
| Dynamic viscosity, mPa·s:              |              |      |
| - at 20 °C                             | 3.15         | 68.24 |
| - at 40 °C                             | 1.97         | 37.20 |
| Saturation pressure, kPa:              |              |      |
| - at 20 °C                             | 2.7          | -     |
| - at 40 °C                             | 4.8          | -     |

**Figure 2.** Saturation pressure of RO at different temperatures.

3. Numerical models

The steady flow inside the nozzle at the maximum needle lift position was simulated using the CFD
program Fluent v17.2. The pressure at the inlet to the computational domain (injection pressure \(p_{in}\)) was
selected to be 51.5, 40, 23 and 20 MPa corresponding to the pressure range before injector in the fuel
injection system of diesel engine D-245.12S [31]. The pressure at the outlet of the nozzle hole
(backpressure \(p_b\)) was 8.878 MPa, which corresponds to the working pressure in the cylinder at the start
of injection [31]. The fuel temperature was set to be constant and equaled to 40 °C, which is the fuel
temperature before entering the injector during high-pressure fuel pump speed of 1250 min⁻¹ [31]. In
order to limit the calculation time, the symmetric geometry element of the nozzle flow part with one
hole was applied. Hexahedral structured grids were meshed for the computational domain using CFD ICEM and are shown in figure 3.

3.1. Cavitation model

In order to simulate two-phase flow in nozzle a multiphase equilibrium model-mixture model was used. The cavitation process is described by the Schnerr-Sauer model [32], in which the mass transfer equation has the following form:

$$\frac{\partial}{\partial t}(\rho \rho_V) + \nabla \cdot (\alpha \rho_V \vec{V}_V) = \frac{\rho_l \rho}{\rho} \frac{d}{d t} \frac{d a}{d t}, \tag{2}$$

where $\alpha$ – the volume concentration of the vapors, $\rho_V$ – the density of the vapor phase, $\rho_l$ – the density of the liquid phase, $\rho$ – the density of the mixture phase ($\rho = \alpha \rho_V + (1 - \alpha) \rho_l$), $\vec{V}_V$ – the velocity of the vapor phase. The density ratio between the mixture phase and vapor phase is represented by the following equation

$$\frac{d \rho}{d t} = - (\rho_l - \rho_V) \frac{d a}{d t}. \tag{3}$$

Mass transfer rate is written by the formula

$$R = \frac{\rho_l \rho}{\rho} \frac{d a}{d t}. \tag{4}$$

Herewith, the volume concentration of the vapors can be written in the correlation to the number of bubbles per unit volume $(n_b)$, which is represented as:

$$\alpha = \frac{n_b 4 \pi R_b^3}{1 + n_b 4 \pi R_b^3}, \tag{5}$$

where $R_b$ – the bubble radius.

The formula for the bubble expansion velocity derived from the dynamic theory without taking into account the second-order terms has the form

$$\frac{d R_b}{d t} = \frac{2}{3} \left| \frac{p_e - p}{\rho_l} \right|, \tag{6}$$

where $p_e$ – the pressure in a bubble, which is usually taken equal to the pressure of the saturated vapors; $p$ – the pressure in the external liquid phase.

From the abovementioned formulas (4), (5), (6), we can get the formula for describing the mass transfer rate
\[ R = \frac{\rho \rho I}{\rho} \alpha (1 - \alpha) 3 \frac{2}{3} \left(\frac{p_v - p}{\rho l}\right). \]

And the bubble radius can be expressed as
\[ R_b = \left(\frac{\alpha 3 \frac{1}{1 - \alpha}}{4\pi n_{ll}}\right). \]

3.2. Turbulent model

For describing turbulence flow in the nozzle, we chose a popular turbulent model with two equations — the k-\( \varepsilon \) model with Enhanced Wall treatment for equation \( \varepsilon \) in the near-wall boundary layers [33, 34]. In this model, the flow region is separated into a completely turbulent region and a viscosity-affected region.

In the completely turbulent layer, the conservative equations for the turbulent kinetic energy \( k \) and turbulent dissipation \( \varepsilon \) have the form
\[ \frac{\partial}{\partial t} (\rho k) + \frac{\partial}{\partial x_i} (\rho k u_i) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \mu_t \right) \frac{\partial k}{\partial x_j} \right] + G_k - \rho \varepsilon \]
and
\[ \frac{\partial}{\partial t} (\rho \varepsilon) + \frac{\partial}{\partial x_i} (\rho \varepsilon u_i) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \mu_t \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_1 \varepsilon \frac{\varepsilon}{k} G_k - C_2 \varepsilon \rho \varepsilon, \]

where \( \mu \) – the molecular viscosity; \( \mu_t \) – the turbulent viscosity, which is calculated by the formula:
\[ \mu_t = \rho \mu_t \frac{k^2}{\varepsilon}; \]
\[ G_k = 2 \mu_t S_{ij} S_{ij}, \]
\[ S_{ij} = \frac{1}{2} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right). \]

\( C_{1 \varepsilon}, C_{2 \varepsilon}, C_{\mu}, \sigma_{k}, \sigma_{\varepsilon} \) are the constants and have the values: \( C_{1 \varepsilon}=1.44 \), \( C_{2 \varepsilon}=1.92 \), \( C_{\mu}=0.09 \), \( \sigma_{k}=1.0 \), \( \sigma_{\varepsilon}=1.3 \).

In the region subject to molecular viscosity, the equation for the turbulent kinetic energy \( k \) is the same as the equation for \( k \) in the completely turbulent region, but the turbulent viscosity is determined according to the method proposed by H.C. Chen [35],
\[ \mu_{t,2} = \rho \mu_l l_{\mu} \sqrt{\varepsilon}. \]

In this formula, \( l_{\mu} \) serves as a length scale and is determined by the formula
\[ l_{\mu} = y C_t \left( 1 - e^{\frac{Re_y}{A_{\mu}}} \right), \]

where \( y \) – the distance to the wall, \( Re_y \) – the turbulent Reynolds number (\( Re_y = \frac{\rho y \sqrt{\varepsilon}}{\mu} \)), \( C_t = k \mu - \frac{3}{2} \), \( A_{\mu}=70 \). Different from the equation for turbulent dissipation \( \varepsilon \) in a completely turbulent region, in this region \( \varepsilon \) is determined by the formula
\[ \varepsilon = \frac{k^2}{l_{\varepsilon}}, \]

where \( l_{\varepsilon} \) – the length scale calculated by the formula
\[ l_e = yC_l \left( 1 - a \frac{Re_y}{2C_l} \right). \] (17)

4. Results and discussion

Using the above-presented models, numerically simulated the stationary flow of petroleum DF and RO in the AZPI nozzle. To analyze the flow of fuel characteristics inside the nozzle hole, a uniaxial coordinate system was used, the starting point of which coincides with the cross-section center of the inlet of the nozzle hole, and the coordinate axis of which is along the axis direction of the nozzle hole and has the coordinate \( l_{\text{axis}} \). Thus, the value \( l_{\text{axis}} = 0 \) mm corresponds to the cross-section of the inlet of the nozzle hole, and the value \( l_{\text{axis}} = 1.1 \) mm - the outlet of the nozzle hole.

4.1. Pressure in the nozzle

Figure 4 presents the pressure distribution of DF and RO on the longitudinal section of the nozzle hole when the pressure at the inlet to the computational domain (injection pressure) \( p_{\text{in}} = 51.5, 40, \) and 20 MPa. We noted different structures of the pressure distribution between DF and RO. A zone with a small pressure in the upper inlet region of the nozzle hole, arising from a sharp change of flow direction and distortion of flow pattern, can lead to the appearance of fuel vapor (cavitation) under the condition when the fuel flow pressure is less than the fuel saturation pressure. As shown in figure 4, the zone with a small pressure decreases for RO compared with DF under the same \( p_{\text{in}} \). The larger zone with a small pressure for DF is due to the higher saturation pressure of DF and the lower viscosity of DF [36].

![Figure 4](image)

**Figure 4.** Pressure distribution on the longitudinal section of the nozzle hole for DF and RO at different injection pressures: a – DF at \( p_{\text{in}} = 51.5 \) MPa; b - DF at \( p_{\text{in}} = 40 \) MPa; c - DF at \( p_{\text{in}} = 20 \) MPa; d - RO at \( p_{\text{in}} = 51.5 \) MPa; e - RO at \( p_{\text{in}} = 40 \) MPa; f - RO at \( p_{\text{in}} = 20 \) MPa.

Figure 5 shows the curves of the variation of the pressure averaged over the cross-section of the nozzle hole for DF and RO along the flow direction. When there is no fuel vapor in the nozzle for both two fuels (see figure 7), i.e. at an injection pressure \( p_{\text{in}} \) of 20 MPa and a backpressure \( p_{\text{b}} \) of 8.878 MPa, the average pressure of RO in the cross-section of the nozzle hole (line 6) exceeds the average pressure of DF (line 3) in the entire nozzle hole region due to the larger pressure loss resulted from the higher viscosity of RO. And accordingly, the flow velocity of RO is slower than that of DF, as shown in figure...
When the pressure difference $\Delta p$ between the nozzle inlet and outlet ($\Delta p = p_\text{in} - p_b$) increases, cavitation starts to occur inside the nozzle hole, the average pressure of DF and RO drops in the middle region of the nozzle hole and increases in the inlet region of the nozzle hole. In addition, during the same pressure differences, the average pressure of DF in the middle region of the nozzle hole is lower than the average pressure of RO but higher in the inlet region of the nozzle hole. This is due to the higher cavitation level of DF (see figure 6). The pressure inside fuel vapors during flowing remains unchanged and equals to the fuel saturation pressure, which is much less than the pressure of liquid fuel in the zone without cavitation. Therefore, the expanded cavitation zone of DF leads to a decrease in the average pressure on the cross-section of the nozzle hole.

4.2. Cavitation in the nozzle

**Figure 5.** Curves of the average pressure of DF and RO on the cross-section of the nozzle hole along the flow direction: 1 – DF at $p_\text{in} = 51.5$ MPa; 2 – DF at $p_\text{in} = 40$ MPa; 3 – DF at $p_\text{in} = 20$ MPa; 4 – RO at $p_\text{in} = 51.5$ MPa; 5 – RO at $p_\text{in} = 40$ MPa; 6 – RO at $p_\text{in} = 20$ MPa.

**Figure 6.** Distribution of fuel vapor volume concentration on the longitudinal section of the nozzle hole for DF and RO at different injection pressures: a – DF at $p_\text{in} = 51.5$ MPa; b – DF at $p_\text{in} = 40$ MPa; c – DF at $p_\text{in} = 40$ MPa; d – RO at $p_\text{in} = 40$ MPa.
$p_{in}=40 \text{ MPa}$; c - DF at $p_{in}=20 \text{ MPa}$; d - RO at $p_{in}=51.5 \text{ MPa}$; e - RO at $p_{in}=40 \text{ MPa}$; f - RO at $p_{in}=20 \text{ MPa}$.

Figure 6 shows the distribution of the vapor volume concentration for DF and RO on the longitudinal section of the nozzle hole at injection pressures $p_{in}$ of 51.5 and 40 MPa. These results confirm that the cavitation level of liquid RO is in a very low degree compared to the cavitation process of liquid DF, the vapor zone for the case of DF is many times larger than the vapor zone for the case of RO and even at an injection pressures $p_{in}$ of 40 MPa the maximum vapor volume concentration of diesel fuel is approximately 12 times the maximum vapor volume concentration of RO. At the injection pressure lower 35 MPa and keeping the other conditions unchanged, cavitation does not occur inside the nozzle hole. From figure 7, which shows the dependence of the total vapor volume concentration of DF and RO in the nozzle hole on the pressure difference $\Delta p$, it is found that when the pressure difference $\Delta p$ is higher than 16.122 MPa, the total vapor volume concentration of DF in the nozzle hole rapidly increases and, accordingly, the critical cavitation number for the case of DF $N_{c,cr}=1.55$; when the pressure difference $\Delta p$ is higher than 26.122 MPa, the total vapor volume concentration of RO in the nozzle hole starts to increase slowly and is several times lower than the total vapor volume concentration of DF. For example, when the pressure difference $\Delta p$ is 42.622 MPa, the total vapor volume concentration of DF in the nozzle hole is 27.5 %, and the total vapor volume concentration of RO is 5.7 %. The critical cavitation number for the case of RO is 2.94. The cavitation difference in the nozzle hole between DT and RO is mainly due to the lower saturation pressure of RO compared with DF. At a temperature of 40 °C, the saturation pressure of DF is 4800 Pa but the saturation pressure of RO is $7.74 \times 10^{-10}$ Pa. Similar results about the cavitation difference for fuels with different saturation pressures were also obtained during injecting naphtha and n-dodecane through a diesel injector nozzle [22], during petroleum DF and biodiesel from the waste cooking oil flowing through an injector nozzle [25] and during the flow petroleum DF and light gasoline fuel [37].

![Figure 7](image-url)  
*Figure 7. Total vapor volume concentration of DT and RO in the nozzle hole at various pressure differences $\Delta p$ between $p_{in}$ and $p_b$.  

4.3. Velocity in the nozzle

Figure 8 presents the distribution of absolute flow velocity of DF and RO on the longitudinal section of the nozzle hole at a pressure at the inlet to the computational domain (injection pressure) $p_{in}=51.5, 40$ and 20 MPa. It was noted that at the same injection pressure, the structure of the velocity distribution for DF and RO is different and the maximum flow velocity of DF is always greater than RO. In figure 9, the curves of the average axial flow velocity on the cross-section of the nozzle hole for DF and RO along the flow direction were plotted. It is obvious that with the same pressure difference $\Delta p$ between $p_{in}$ and $p_b$, the average axial flow velocity of DF is always higher than RO. At $p_{in}=51.5, 40$ and 20 MPa the flow velocity of DF at the nozzle hole outlet (injection rate) is higher than the flow velocity of RO by 22.22%, 17.9% and 21%, respectively. This is due to two factors. One is that RO has an increased viscosity, which leads to an increase in the thickness of near-wall viscosity layers [22]. The other is that for the case of DF the cavitation process is stronger (more vapors generated - figure 6), which leads to
a reduction of effective flow cross-sectional area and an increase of flow velocity in the no vapor region [38].

Figure 8. Distribution of the absolute flow velocity of DF and RO on the longitudinal section of the nozzle hole at different injection pressures: a – DF at $p_{in} = 51.5$ MPa; b - DF at $p_{in} = 40$ MPa; c - DF at $p_{in} = 20$ MPa; d - RO at $p_{in} = 51.5$ MPa; e - RO at $p_{in} = 40$ MPa; f - RO at $p_{in} = 20$ MPa.

Figure 9. Average axial flow velocity on the cross-section of the nozzle hole for DF and RO along the flow direction: 1 - DF at $p_{in} = 51.5$ MPa; 2 - DF at $p_{in} = 40$ MPa; 3 - DF at $p_{in} = 20$ MPa; 4 - RO at $p_{in} = 51.5$ MPa; 5 - RO at $p_{in} = 40$ MPa; 6 - RO at $p_{in} = 20$ MPa.
4.4. Turbulence in the nozzle

Figure 10. Distribution of turbulent kinetic energy of DF and RO flow on the longitudinal section of the nozzle hole: a - DF at \( p_{in} = 51.5 \) MPa; b - DF at \( p_{in} = 40 \) MPa; c - DF at \( p_{in} = 20 \) MPa; d - RO at \( p_{in} = 51.5 \) MPa; e - RO at \( p_{in} = 40 \) MPa; f - RO at \( p_{in} = 20 \) MPa.

To analyze the effect of RO on fuel flow turbulence, it is very useful to introduce the turbulent kinetic energy (TKE), which is characterized by the squared effective fluctuating velocity:

\[ TKE = \frac{u'^2 + v'^2 + w'^2}{2}. \]  

Figure 10 shows the TKE distribution of DF and RO flow on the longitudinal section of the nozzle hole as \( p_{in} = 51.5, 40 \) and \( 20 \) MPa. In figure 11 the curves of the average TKE of DF and RO on the cross-section of the nozzle hole along the flow direction were plotted. It was found that during injecting DF the turbulent disturbance core (with the maximum turbulence level) is closer to the nozzle hole outlet and the turbulent disturbance core for the case of RO is closer to the nozzle hole inlet. Moreover, with a decrease of pressure \( p_{in} \), the turbulent disturbance core gradually moves away from the nozzle hole outlet. The maximum TKE of RO is larger than DF at high pressures \( p_{in} \) and less than DF at low pressures \( p_{in} \). In addition, at \( p_{in} = 51.5 \) and \( 40 \) MPa, the average TKE of RO on the cross-section with \( l_{axis} \) from 0 to 0.8 mm is also higher than this value for DF. But the TKE of RO flow at the outlet of the nozzle hole is always less than that of DF.
4.5. General flow parameters at the nozzle outlet

To analyze the general flow characteristics of different fuels, the fuel mass flow rate, discharge coefficient, average injection velocity and average TKE at the outlet of the nozzle hole are studied. These parameters are shown in figures 12-14.

Figure 12 shows the mass flow rate and discharge coefficient for the case of DF and RO at different injection pressures. It was shown that with the same pressure differences $\Delta p$ between $p_{in}$ and $p_b$, the mass flow rate and discharge coefficient of DF are higher than that of RO. Moreover, as the pressure difference $\Delta p$ decreases, the discharge coefficient of DF increases at first and then almost remains unchanged when the pressure difference $\Delta p$ is lower than 16.122 MPa ($p_{in} = 25$ MPa, $p_b = 8.878$ MPa). However, the discharge coefficient of RO at first almost does not change and then decreases. The reasons can be explained as follows. For the low-viscosity and easily evaporating DF, the main role affecting inner nozzle flow is cavitation, which prevents the fuel from flowing. RO is characterized by increased viscosity and low volatility so that its flow is mainly affected by viscosity.
Figure 13. Fuel injection velocity of DF and RO at various pressure differences $\Delta p$ between $p_{in}$ and $p_b$.

Figure 14. Average turbulent kinetic energy of DF and RO flow at the nozzle hole outlet at various pressure differences $\Delta p$ between $p_{in}$ and $p_b$.

Figure 13 and figure 14 show the dependence of injection velocity and the dependence of average TKE at the outlet of the nozzle hole on the pressure difference $\Delta p$ between $p_{in}$ and $p_b$. Obviously, with the same pressure difference $\Delta p$, both the injection velocity and average TKE of DF are higher than these of RO. The reduced injection velocity and TKE of RO might lead to a decrease of primary breakup rate and injection cone angle and an increase of spray penetration length [24-26, 37]. Based on this, using RO in a diesel engine, it is necessary to make an appropriate modification to combustion chamber geometry and the working parameters of the fuel injection system to ensure complete mixing of fuel with air and to prevent fuel spray from hitting the combustion chamber walls.

5. Conclusions

A computational study was performed to investigate the injection characteristics of petroleum diesel fuel and rapeseed oil. Important observations are as follows:

(1) The distribution structures of pressure, velocity and turbulent kinematic energy of DF and RO flow in the investigated nozzle hole are different;

(2) As the injection pressure increases, the turbulent disturbance core of RO flow tends to the nozzle hole inlet and the turbulent disturbance core of DF flow tends to the nozzle hole outlet;

(3) In the case of DF the flow velocity exceeds that for RO case. During the injection pressure $p_{in}=51.5$, 40 and 20 MPa, the DF flow velocity at the nozzle hole outlet (DF injection velocity) is 22.22%, 17.9% and 21% higher than the RO flow velocity, respectively;

(4) The cavitation level of DF is many times higher than RO and the critical numbers for DF and RO are 1.55 and 2.94, respectively;

(5) The DF discharge coefficient is higher than the RO discharge coefficient in all the range of injection pressure. With injection pressure decreasing, the discharge coefficient of DF grows at first and then almost stay unchanged after the injection pressure is lower than 25MPa, but the RO discharge coefficient does not change at first and then decreases.

Based on the investigation results it is necessary to point out that the direct use of pure RO in a diesel engine might lead to a reduction of injection rate and the injection turbulence level, thereby the quality of mixture and combustion process will be deteriorated. To solve this problem, an appropriate modification to combustion chamber geometry and the working parameters of the fuel injection system should be considered.

References

[1] Torres-García M, Garcia-Martín J F, Jiménez-Espadafor Aguilar F J, Barbin D F and Álvarez-
Mateos P 2020 Vegetable oils as renewable fuels for power plants based on low and medium speed diesel engines. *J. Energy Inst.* 93 953–61

[2] Markov V A, Devyanin N S, Semenov V G, Vagrov V V and Zykov S A 2019 *Engine Fuels from Vegetable Oils* ed V A Markov (Riga: Lambert Academic Publishing)

[3] Markov V A, Devyanin S N, Zykov S A and Sa B 2017 Viscosity Characteristics of Biofuels Based on Vegetable Oils *Truck* 3 40–6

[4] Markov V A, Zykov S A, Biryukov V V and Sa B 2017 Viscosity characteristics of multicomponent and emulsified fuels *AutoGas Fill. Complex + Altern. fuel* 16 105–21

[5] de Almeida S C A, Belchior C R, Nascimento M V G, Vieira L dos S R and Fleury G 2002 Performance of a diesel generator fuelled with palm oil *Fuel* 81 2097–102

[6] Wang Y D, Al-Shemmeri T, Eames P, McMullan J, Hewitt N, Huang Y and Rezvani S 2006 An experimental investigation of the performance and gaseous exhaust emissions of a diesel engine using blends of a vegetable oil *Appl. Therm. Eng.* 26 1684–91

[7] Shah P R and Ganesh A 2016 A comparative study on influence of fuel additives with edible and non-edible vegetable oil based on fuel characterization and engine characteristics of diesel engine *Appl. Therm. Eng.* 102 800–12

[8] Devan P K and Mahalakshmi N V 2009 Performance, emission and combustion characteristics of poon oil and its diesel blends in a DI diesel engine *Fuel* 88 861–7

[9] Agarwal A K and Rajamanoharan K 2009 Experimental investigations of performance and emissions of Karanja oil and its blends in a single cylinder agricultural diesel engine *Appl. Energy* 86 106–12

[10] Hebbal O D, Reddy K V and Rajagopal K 2006 Performance characteristics of a diesel engine with deccan hemp oil *Fuel* 85 2187–94

[11] Singh P J, Khurma J R and Singh A 2010 Coconut oil based hybrid fuels as alternative fuel for diesel engines *Am. J. Environ. Sci.* 6 69–75

[12] Yilmaz N and Morton B 2011 Effects of preheating vegetable oils on performance and emission characteristics of two diesel engines *Biomass and Bioenergy* 35 2928–33

[13] Nwafor O M I, Rice G and Ogbonna A I 2000 Effect of advanced injection timing on the performance of rapeseed oil in diesel engines *Renew. Energy* 21 433–44

[14] Soltic P, Edenhauser D, Thurnheer T, Schreiber D and Sankowski A 2009 Experimental investigation of mineral diesel fuel, GTL fuel, RME and neat soybean and rapeseed oil combustion in a heavy duty on-road engine with exhaust gas aftertreatment *Fuel* 88 1–8

[15] Sonar D, Soni S L, Sharma D, Srivastava A and Goyal R 2015 Performance and emission characteristics of a diesel engine with varying injection pressure and fuelled with raw mahua oil (preheated and blends) and mahua oil methyl ester *Clean Technol. Environ. Policy* 17 1499–511

[16] H. Q D, Q. D X, B. Z W and K. Y 2019 Spray Characteristics and Engine Performance of Vegetable Oil–Diesel–Ethanol Hybrid Fuel. *J. Energy Eng.* 145 4019011

[17] Dernotte J, Hespel C, Foucher F, Houillé S and Mounaïm-Rousselle C 2012 Influence of physical fuel properties on the injection rate in a Diesel injector *Fuel* 96 153–60

[18] Esteban B, Riba J-R, Baquero G, Rius A and Puig R 2012 Temperature dependence of density and viscosity of vegetable oils *Biomass and Bioenergy* 42 164–71

[19] Ramadhas A S, Jayaraj S and Muraleedharan C 2004 Use of vegetable oils as I.C. engine fuels—A review *Renew. Energy* 29 727–42

[20] Yilmaz N 2011 Temperature-dependent viscosity correlations of vegetable oils and biofuel–diesel mixtures *Biomass and Bioenergy* 35 2936–8

[21] Cui H, Luo F, Wang Z and Dong S 2013 Simulation analysis of flow characteristics of Jatropha curcas oil in diesel injector *Trans. Chinese Soc. Agric. Eng.* 29 63–71

[22] Torelli R, Som S, Pei Y, Zhang Y and Traver M 2017 Influence of fuel properties on internal nozzle flow development in a multi-hole diesel injector *Fuel* 204 171–84

[23] Wei M, Gao Y, Yan F, Chen L, Feng L, Li G and Zhang C 2017 Experimental study of cavitation
formation and primary breakup for a biodiesel surrogate fuel (methyl butanoate) using transparent nozzle [24] Hwang J, Bae C, Patel C, Agarwal A K and Gupta T 2017 Near Nozzle Flow and Atomization Characteristics of Biodiesel Fuels (SAE Technical Paper)

[25] Yu S, Yin B, Jia H, Wen S, Li X and Yu J 2017 Theoretical and experimental comparison of internal flow and spray characteristics between diesel and biodiesel Fuel 208 20–9

[26] Ishak M H H, Ismail F, Che Mat S, Abdullah M Z, Aziz A and Idroas M Y 2019 Numerical analysis of nozzle flow and spray characteristics from different nozzles using diesel and biofuel blends Energies 12 281

[27] Hoang A T and Pham V V 2019 A study of emission characteristic, deposits, and lubrication oil degradation of a diesel engine running on preheated vegetable oil and diesel oil Energy Sources, Part A Recover. Util. Environ. Eff. 41 611–25

[28] Markov V A, Kamaltdinov V G, Zykov S A and Sa B 2019 Optimization of the Composition of Blended Biodiesel Fuels with Additives of Vegetable Oils Int. J. Energy a Clean Environ. 20

[29] Cruz-Forero D-C, González-Ruiz O-A and López-Giraldo L-J 2012 Calculation of thermophysical properties of oils and triacylglycerols using an extended constituent fragments approach CT&F-Ciencia, Tecnol. y Futur. 5 67–82

[30] Zong L, Ramanathan S and Chen C-C 2010 Fragment-based approach for estimating thermophysical properties of fats and vegetable oils for modeling biodiesel production processes Ind. Eng. Chem. Res. 49 876–86

[31] Markov V A, Devyanin S N, Zykov S A and Gaidar S M 2016 Biofuel for Internal Combustion Engine (Moscow: Science Research Center Engineer)

[32] Schnerr G H and Sauer J 2001 Physical and numerical modeling of unsteady cavitation dynamics Fourth Int. Conf. on Multiphase Flow vol 1 (ICMF New Orleans)

[33] Wolfshtein M 1969 The velocity and temperature distribution in one-dimensional flow with turbulence augmentation and pressure gradient Int. J. Heat Mass Transf. 12 301–18

[34] Jongen T 1992 Simulation and Modeling of Turbulent Incompressible Flows (EPF Lausanne)

[35] Chen Hc and Patel V C 1988 Near-wall turbulence models for complex flows including separation AIAA J. 26 641–8

[36] Liu W and Zhao J 2016 3-D Numerical study on the effect of variant injection pressure in a diesel injector with cavitation formation J. Adv. Veh. Eng. 2 174–81

[37] Yu W, Yang W and Zhao F 2017 Investigation of internal nozzle flow, spray and combustion characteristics fueled with diesel, gasoline and wide distillation fuel (WDF) based on a piezoelectric injector and a direct injection compression ignition engine Appl. Therm. Eng. 114 905–20

[38] Zhao J, Liu W, Zhao J and Grekhov L 2020 Numerical investigation of gas/liquid two-phase flow in nozzle holes considering the fuel compressibility Int. J. Heat Mass Transf. 147 118991