Calculation of geometric parameters of diesel fuel ignition flares

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Abstract. The fuel flare in the gas-diesel cylinder is the main source of ignition of the methane-air mixture, so the effective combustion of fuel depends on the processes that occur in it when vaporized fuel particles penetrate into the air and form a fuel-air mixture. Thus, the consideration of the parameters and structure of the fuel phase allows us to approach the analysis of the interaction between the methane-air environment and fuel particles, as well as the problems associated with the formation and burning of soot particles in the form of gas diesel.

The sputtered fuel torch is usually characterized by a length $L_f$, a width $B_f$, and a solid angle $\gamma_f$ (figure 1). The volume of the fuel flare will be determined [1-3]

$$V_f = L_f \cdot \sin \frac{\gamma_f}{2}.$$  \hfill (1)

In gas diesel, the ignition dose of diesel fuel is constant and is 20% of the nominal supply, which is also divided into five torches, hence $V_f=3$ mm$^3$.

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{figure1.png}
\caption{Diagram of a sprayed diesel fuel torch: $\gamma_f$ - angle of the spray jet cone; $B_f$ - width of the fuel torch; $L_f$ - fuel flare length.}
\end{figure}

The angle of the cone $\gamma_f$ varies from $4^{\circ}$ to $40^{\circ}$ and higher depending on the design of the sprayer. Of practical interest are the data obtained at high back pressures in the atomization chamber, and it is recommended to determine the angle of the jet cone of the dispersed fuel [4-6]:

$$\gamma_f = 2 \cdot Arctg\left(F_\epsilon \cdot W_c^{0.32} \cdot M^{-0.07} \cdot \rho^{0.5}\right).$$  \hfill (2)
where $F_f$ - empirical coefficient for closed injectors in pulse injection for calculating the maximum speed criteria $U_o$, $F_f=0.008$;

$W_e$ - Weber’s criterion, which characterizes the ratio of surface tension and inertia forces;

$M$ - criterion that characterizes the ratio of surface tension, inertia, and viscosity forces;

$\rho$ - the ratio of air and fuel densities at the start of fuel injection.

Expression (2) is valid when the criteria are changed in the pre-cases $W_e=(140…725)10^3$, $M=(7.39…33.4)10^{-4}$, $\rho=0.0095…0.028$.

The Weber criterion is defined from the expression [7-9]

$$W_e = \frac{U_o^2 \cdot d_c \cdot \rho_f}{\sigma_f},$$

(3)

where $U_o$ - the average flow rate of the fuel jet from the nozzle, m/s;

$d_c$ - diameter of the nozzle hole of the sprayer, m;

$\rho_f$ - fuel density, $\rho_f=830 \text{ kg/m}^3$;

$\sigma_f$ - coefficient of surface tension of the fuel, $\sigma_f=28\cdot10^{-3} \text{ N/m}$;

According to [10-14] the average velocity of the fuel jet out of the nozzle

$$U_o = \sqrt{(p_f - p_n) \cdot 2 \cdot 10^6 / \rho_f},$$

(4)

where $p_f$ and $p_n$ - are, respectively, the average injection pressure of the fuel and the average gas pressure in the cylinder during the injection period, MPa.

The $p_n$ value during the injection period will be determined [15-17]

$$p_n \approx 0.95 \cdot p_c,$$

(5)

where $p_c$ - pressure at the end of compression, MPa.

Pressure at the end of polytropic compression

$$p_c = p_a \cdot e^{n_1},$$

(6)

where $n_1$ - the average value of the compression polytrope, determined by the equation [18-20]

$$n_1 = 1.41 - \frac{100}{n},$$

(7)

where $n$ - the engine crankshaft speed, min$^{-1}$.

Taking $p_f=17.5$ MPa, we get the average speed of fuel consumption through the spray holes $U_o=179 \text{ m/s}$. After substituting the calculated values in formula (3), the Weber criterion will be equal to $W_e=2.3\cdot10^5$.

The $M$ criterion is defined from the expression [21-23]

$$M = \frac{\mu_f^2}{\rho_f \cdot d_c \cdot \sigma_f},$$

(8)

where $\mu_f$ - the coefficient of dynamic fuel viscosity, $\mu_f=3\cdot10^{-3} \text{ Pa/m}^2$.

Substituting the found values of indicators in the formula (8) we get the criterion $M=1.3\cdot10^{-3}$.

The ratio of air and fuel densities at the time of fuel injection is determined from the formula [24-26]

$$\rho = \frac{\rho_a}{\rho_f},$$

(9)
where $\rho_a$ - the average air density during the injection period, kg/m$^3$.

The average air density during the fuel injection period will be determined using (5) of the equation [27-29]

$$\rho_a = \frac{0.95 \cdot p_e \cdot 10^6}{R_a \cdot T_e},$$

(10)

where $R_a$ - the universal gas constant for air, $R_a=287$ J/kg·deg; $T_e$ - temperature at the end of the compression process, K.

$$T_e = T_a \cdot e^{(m-1)},$$

(11)

where $T_a$ - the temperature at the end of the inlet, K.

Substituting the calculated and experimental data in the expression (11), we get $T_e=817$ K. At the same time, $\rho_a=15.6$ kg/m$^3$; $\rho=0.019$; $\gamma=11.1^\circ$.

Then the length of the torch is determined from expression (2) and will have the form [30-33]

$$L_f = \sqrt{\frac{V_f}{\sin^2 \gamma_f}} = 17.9 \text{ mm.}$$

(12)

Based on the found length of $L_f$, it is possible to construct a geometric model of the propagation of five ignition flares in the combustion chamber and determine the average coefficient of excess air $\alpha$ in the experimental zone 1 (figure 2) and in the zone where the methane-suffocating mixture prevails [34-37].

The hemispherical volume occupied by fuel flares in the combustion chamber is calculated by the formula [38-40]

![Figure 2. Model of distribution of fuel flares when injecting an initial dose of diesel fuel with a multi-hole nozzle into the combustion chamber of the CNIDI type: 1 - zone of distribution of fuel flares; 2 - zone with a predominance of a methane-air mixture.](image)
When the value $L_f = 17.9$ mm found in equation (12), the hemispherical ignition volume $V_i$ is $16.1\%$ of the volume of the diesel combustion chamber, equal to $V_c = 74.2$ cm$^3$ [41-43]. This indirectly confirms the correctness of the calculation of the torch length $L_f$ and the accepted hemispherical shape of the ignition volume (13).

The process of mixing the ignition dose of diesel fuel with air when the gas-diesel engine is running at idle should not differ from this process in a diesel engine that has a volume–film mixing at idle. At idle this length of fuel flares $L_f$ can provide a mostly volumetric method of mixing in the ignition volume [44-46]

$$V_i = 0.218 \cdot V_c.$$  \hspace{1cm} (14)

At the same time the average coefficient of excess air when the diesel engine is running at the rated mode

$$\alpha_n = \frac{G_{an}}{14.3 \cdot G_{an}},$$  \hspace{1cm} (15)

where $G_{an}$ – air consumption by diesel when operating at rated mode, kg/h.

The average coefficient of excess air in the hemispherical volume covering the fuel flares when the diesel engine is running at idle is determined by

$$\alpha_{ix} = \alpha_n \cdot \frac{G_{fk}}{G_{fn}} \cdot \frac{V_{sx}}{V_i}.$$  \hspace{1cm} (16)

where $G_{fn}$ and $G_{fk}$ - diesel fuel consumption, respectively, in nominal mode and at idle, kg/h.

Below are calculations of the average value of the excess air coefficient depending on the average effective pressure, which are shown in figure 3.

![Figure 3. The values of the average coefficient of excess air from the average effective pressure: 1 – in the zone of distribution of fuel flares (zone 1 in figure 2), 2-in the zone of predominance of the methane-air mixture (zone 2 in figure 2).](image-url)

The second zone is dominated by a methane-air mixture. Carbon is formed in areas at a relatively high temperature, in which the particles quickly burn without soot formation, while at low loads the growth of soot particles in the exhaust gases should be insignificant at modes close to the nominal and higher, the supply of gas-like fuel increases, with the constant supply of diesel fuel, and consequently decreases $\alpha$, which leads to a lack of oxidizer and intensive soot formation [45-47].
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