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SPECIAL ASPECTS OF PARAMETER MANAGEMENT OF LOW CETANE FUEL INJECTION

Statement of the problem in a general form. The problem of increasing of combustion efficiency for a heat engine (diesel, gas turbine, etc.) has been solved for about a century, because economic, power and environmental indicators depend on it. In addition, which is no less important in relation to the stated indicators, the efficiency of combustion and the rate of fuel burnout in the specified coordinates at the designated time interval in the combustion chamber determine strict requirements for the used fuels in terms of thermophysical parameters that affect atomization, evaporation, and mixing with oxidizer . Hence, there are restrictions on the use of alternative fuels for heat engines, such as gas condensate, etc.

The solution of this problem is important for the national economy of Ukraine, especially when solving the problems of energy independence of the state. The acute shortage of fuel has firmly taken its position and dictates prices for almost everything (water, bread, heat, etc.) and can become the main argument in special conditions. From this follows the important scientific and technical task of weakening this factor - to find opportunities and scientifically justify their implementation by preparing of burning of alternative fuels so that the power, economic and environmental indicators of the engines would be at the standard level.

Analysis of recent research publications.

At present, one of the most relevant global trends in the field of engine construction is a comprehensive solution to the problems of fuel efficiency and environmental friendliness of internal combustion engines, which is taking place against the background of dieselization of the world fleet of motor vehicles [1]. The majority of specialists in this field agree that the raw material resources for the production of motor fuel have a tendency to be exhausted. The two main ways to overcome the problem, which is being transformed into a global energy crisis, are the use of renewable energy sources, in particular, alternative fuels of biological origin (pure or mixed) and increasing the efficiency of the use of chemical energy of motor fuel by improving the design of the internal combustion engines, application of energy recovery systems, rationalization of models of their operation, etc. [2]. The first of these ways is currently implemented with insufficient speed, since each of the types of alternative motor fuels has disadvantages caused both by the peculiarities of the process of obtaining raw materials for its processing of raw materials into fuel, and by the need to make significant changes in the design of internal combustion engines [3, 4]. As for the second way, the main problem for its implementation is that the designs of modern internal combustion engines are approaching the limits of their capabilities, as well as the limits for its improvement [5-7].

Setting the task. We offer a solution to the tasks using an additional power supply system in the form of a vortex mixer-evaporator [8]. The vortex vaporizer mixer works on the energy of waste gases and allows the use of cheap, relatively standard, low-cetane fuels, for example, stable gas condensate.

But the use of high-viscosity fuels in piston engines is complicated by the fact that their thermophysical parameters are significantly different from traditional fuels, which requires a careful approach when evaluating the formation of a fuel mixture - atomization, evaporation and mixing with an oxidizer. In addition, the high sulfur content poses additional challenges, especially regarding the environmental cleanliness of exhaust gases [9].

Presentation of the main research material.

Low-cetane alternative fuels differ from standard diesel by the parameters that determine the geometric characteristics of the torch (spray angle, torch range) and by the thermal-physical parameters that determine the quality of the spray, that is, the Weber and Laplace criteria

$$We = \frac{\rho_f d_n \upsilon}{\sigma_n}; \tag{1}$$

$$Lp = \frac{d_c \rho_f \sigma_s}{\mu_n^2}, \tag{2}$$

which are of great importance in the organization of the fuel combustion process, and estimate the average droplet diameter is estimated according to the differential injection characteristic, such as an average volume diameter of the droplet [10]

$$d_{зо} = A_{зо} (We)^{a_{зо}} (L_f)^{b_{зо}}, \tag{3}$$

where A, a, b are constant coefficients;

ρ_f – fuel density;

d_n – nozzle diameter;

σ_s – surface tension;

μ_n – dynamic viscosity;

υ – fuel flow rate from the sprayer nozzle.

Analysis of equations (1 – 3) shows that the most controlled argument is the fuel flow rate, which is equal to

$$\upsilon = \mu_m \sqrt{\frac{2}{\rho_f}} \sqrt{p_s - p_c} \quad , \tag{4}$$

where μ_n is the flow coefficient of the atomizer nozzle;

p_s – fuel pressure in the sprayer;

p_c – gas pressure in the cylinder during the injection period.

The pressure change in the cylinder during the injection period can be neglected, and the rate of change of pressure in the sprayer can be given by differential equations

$$\frac{dp_s}{d\tau} = \frac{1}{\beta V_a}\left[2 f_p e^{-\alpha L}\left(\frac{p_k}{z} + e^{-\alpha L}\cdot W\right)_{(\tau-\Delta\tau)} - f_n \frac{dh_n}{d\tau} - f_p - \frac{dq}{d\tau}\right], \tag{5}$$

where β is fuel compressibility;

V_a – volume of fuel in the nozzle atomizer;

α – damping factor of fuel velocity waves;

L is the length of the high-pressure pipeline;

f_p , f_n , – cross-sections of the pipeline and needles along the closing cone, respectively;

h_n – movement of the needle;

$W_{\tau+\Delta\tau} = e^{-\alpha L}\left(\frac{p_k}{z} + e^{-\alpha L}\cdot W\right)_{\tau-\Delta\tau} - \frac{1}{z} p_p$ – velocity wave in the atomizer cross-section;

$z = W \cdot \rho_n$ – acoustic resistance of the atomizer;

$\Delta\tau$ – transport delay;

$\frac{dq}{d\tau} = \mu f_s(h_n)\sqrt{\frac{2}{\rho_n}}\sqrt{p_s - p_c}$ – the differential characteristic of up-risking;

p_k – fuel pressure in the volume of the nozzle of the injection valve, which is determined by integrating the equation of the rate of change of pressure in the indicated volume

$$\frac{dp_k}{d\tau} = \frac{1}{\beta V_k}\left[f_k \frac{dh_k}{d\tau}(1-\sigma) + \left(f_n \frac{dh}{d\tau} - \frac{dq_n}{d\tau} \right)\sigma - \frac{f_m}{z} p_k - 2 f_m e^{-\alpha L} W_\tau \right], \tag{6}$$

where Vk is the volume of the injection valve fitting;

f_k, f_n – cross-sectional area of the valve along the unloading belt and the plunger, respectively;

h_k, h – valve and plunger movement, respectively;

σ – single function;

$\frac{dq_n}{d\tau} = (\mu f)_o \sqrt{\frac{2}{\rho_n}}\sqrt{p_u - p_{cw}}$ – differential fuel consumption through the cut-off windows of the sleeve of the plunger pair;

$(\mu f)_o$ – effective cross-section of cut-off windows;

p_{cw} – fuel pressure in the above-plunger volume before closing the cut-off windows;

p_u is the fuel pressure in the underplunger cavity, which is determined by the differential equation

$$\frac{dp_н}{d\tau} = \frac{1}{\beta V_н}\left(f_n \frac{dh}{d\tau} - f_k \frac{dh_k}{d\tau} \right)(1-\sigma) - \frac{dq_n}{d\tau}(1-\sigma), \tag{7}$$

where $V_н$ is the volume of the underplunger cavity.

In equations (6, 7) and further:

$$\sigma = \begin{cases} 0 \text{ при } h_k < h_{k_0}; \\ 1 \text{ при } h_k \geq h_{k_0}; \end{cases} \quad V_k = \begin{cases} V_k & \text{при } h_k < h_{k_0}; \\ V_k + V_н & \text{при } h_k \geq h_{k_0}, \end{cases}$$

where h_{k_0} is the movement of the valve to the exit of the unloading belt from the saddle channel.

To close equations (3 – 7), it is necessary to introduce regulated and controlled functions, that is, kinematic and dynamic parameters of moving precision pairs of fuel equipment:

is the differential equation of motion of the nozzle needle

$$\frac{d^2 h_n}{d\tau^2} = \frac{1}{m_n}\left(p_f f_n - \delta_n h_n - F_{\text{n}}\right); \tag{8}$$

is the differential equation of motion of the discharge valve

$$\frac{d^2 h_k}{d\tau^2} = \frac{1}{m_k}\left[f_k\left(p_н - p_k\right) + z\left(f_n \frac{dh}{d\tau} - f_k \frac{dh_k}{d\tau} - \frac{dq_n}{d\tau}\right)\sigma - \delta_k h_k - F_v \right]; \tag{9}$$

is the differential equation of motion of the plunger

$$\frac{dh}{d\tau} = 6n \cdot k_\varphi \tau + \upsilon_o , \tag{10}$$

where $m_г, m_k$ is the mass of the needle and valve (together with the 1/3 mass of the spring), respectively;

δ_n, δ_v – stiffness of the springs of the injector needle and the injection valve, respectively;

F_v, F_{n} – pre-tightening forces of the valve and the injector needle, respectively, which determine the nature of the pressure waves at the entrance to the pipeline and the differential injection characteristics;

n – camshaft rotation frequency;

υ_o – the speed of the plunger at the time of closing the cut-off windows.

The value of the angular coefficient k_φ is determined by the profile of the cam, which provides movement to the plunger. For example, for a convex cam profile, the movement and speed of the plunger are described by the equations:

$$\left.\begin{aligned} h &= a_j\varphi^2 + b_j\varphi + c_j; \\ \frac{dh}{d\tau} &= 6n\left(2a_j\varphi + b_j\right), \end{aligned}\right\} \tag{11}$$

where $a_j, b_j, c_j \Rightarrow k_{\varphi_j}$ - for the linear sections of the cam profile.

Then the movement of the plunger by the angle of rotation of the cam will be shown in the form:

$$h = \begin{vmatrix} a_j \\ 0{,}55 \\ 0{,}66 \\ 0{,}24 \\ -1{,}67 \end{vmatrix} \cdot 10^3 \varphi^2 + \begin{vmatrix} b_j \\ 0{,}00 \\ -0{,}54 \\ 2{,}58 \\ 23{,}9 \end{vmatrix} \cdot 10^2 \varphi + \begin{vmatrix} c_j \\ 0{,}00 \\ 0{,}06 \\ -0{,}516 \\ -6{,}45 \end{vmatrix} \qquad \begin{matrix} -\varphi_1 = 0 - 23^0 \\ -\varphi_2 = 23 - 37^0 \\ -\varphi_3 = 37 - 55^0 \\ -\varphi_4 = 55 - 72^0 \end{matrix} \tag{12}$$

In this way, we obtained a mathematical model for calculating changes of parameters, spray quality and differential characteristics of fuel injection, from which it is clear that the controlled arguments are the stiffness of the springs of the injector needle $\delta_ф$ and injection valve δ_k, the force of pre-tightening the valve F_v and injector needle F_n, and moving the valve to output of the unloading-shaft belt from the channel of the saddle h_{k_0}. Along with these arguments, the controlled function should take into account the fuel pressure in the atomizer:

$$p_p = \frac{1}{\beta V_p} \int \left(\frac{dq_ф}{d\tau} - \frac{dq_p}{d\tau} \right) d\tau \ , \tag{13}$$

where τ_2 is the duration of injection;

$\frac{dq_ф}{d\tau}$ – differential fuel consumption through the injector.

Conclusions and prospects for further research.

The analysis of the system of equations (1 – 11) shows that with a decrease in the frequency of rotation of the camshaft n, the injection speed will be insufficient to achieve the required spray quality due to a decrease in speed υ (4), because the fuel pressure in the sprayer p_p will sharply decrease, and the increase in stiffness of the injector needle δ_n and injection valve springs δ_k, as well as the pre-tightening force of the valve F_v and injector needle F_n will act in the opposite direction (5). This analysis allows you to take into account the peculiarities of management of the injection parameters, which is confirmed by experimental studies and consists of the following:

– the start-up mode and small cyclic feeds are provided with the necessary parameters of injection and ignition by using the parallel operation of the vortex evaporator-mixer;

– electromagnetic and electrodynamic drives of the nozzle needle allow to obtain the specified injection speed of small cyclic feeds (up to 12-18 mm^3/cycle) and the quality of spraying in the range of the average volume diameter of the drops within 15-20 microns.

– in the case of using a separate double supply pump with a cam drive of the plunger with volumes of 80 mm^3/cycle and more, the first dose of 30-40 mm^3/cycle should be injected into the intake manifold at the beginning of the compression process in the cylinder.

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