Theoretical principles of formation of homogeneous fuel mixtures in diesels

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Abstract. The article discusses molecular changes in the working mixture based on the composition of the biofuel blend. Theoretical calculations on the variations in heat release rates and fuel particle velocities reveal differing increases in heat utilization efficiency across various flow sections during atomization at different time intervals. The movement conditions of particles within the combustion chamber vary, particularly in fuels with multicomponent mixtures. Notably, a significant number of droplets (up to 20 million) of varying sizes are observed. Under these circumstances, abrupt spraying speeds are not recommended as they lead to fuel accumulation around the burner, hindering the combustion process. Furthermore, the article introduces analytical relationships aimed at promoting turbulence in flows of diverse fuels within the combustion chamber through the intensive mixing and throttling of fuel mixtures with varying compositions.

1 Introduction

Over the past 20 years, the main sources of energy have been oil, coal, and natural gas, which account for more than 60 percent of the country's electricity generation.

There are two possible ways to help overcome these problems. The first is the development and application of new energy-efficient technologies. The second is the greater use of renewable alternatives to fossil hydrocarbon fuels. Replacing gasoline or diesel as an energy source with biofuel can reduce greenhouse gas emissions by an average of 30% to 70%, depending on the type of fuel. However, renewable sources make up only 3% of the world's primary energy production today. In this regard, the socio-economic development of our Republic in terms of ensuring fuel-energy and food independence envisages saving fuel-energy and natural resources, increasing production efficiency due to the use of renewable energy sources. According to the evaluations of international experts, it is based on the increase in the price of oil products in the world and the volume of the searched oil reserves for about 30 years. But each country has its own resources and needs [1-4].

In this regard, the Republic of Uzbekistan has adopted a number of programs for efficient use and saving of these energy resources. One of the ways to solve this problem is the use of

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alternative energy sources, in particular, biofuels. According to a number of experts, biomass for obtaining biofuel can be a significant source in the future, and first of all in countries located in warm climate regions. Foreign scientists dealing with oxygenated fuel types say that biofuel can be mixed with good quality combustible fuels or emulsified. The use of agricultural machinery in hot climates places serious demands on fuel as well as on their self-ignition system. Transition to multi-component fuels (diesel fuel, biogas, bioethanol) is also one of the most important problems today.

But this issue, in turn, requires many scientifically based technical, technological and production solutions. Today, internal combustion engines (IYoD) are considered the most common heat engines and are widely used in all sectors of the economy due to their compactness, high efficiency and durability. The successful application of IYoD, the development of experimental structures, the increase of power and economic indicators in many cases is the careful development of the theoretical basis of the engine's working process. The first prof. The engine heat calculation method developed by V. I. Grinevetsky was further developed and supplemented by scientists such as N. R. Briling, E. K. Moezing and V. S. Stechkin [4-7].

2 Problem statement

One of the main issues in the field of tractor engines is the problem of increasing the efficiency of their relative power index, the production of a multi-discipline flexible system, and the reduction of the amount of harmful gases in them due to the reduction of the cost of creating a multi-component mixture of fuel. As we know from the analysis, according to the methods of ignition of working mixtures, IYoD is classified as follows: engines that ignite the working fluid under the influence of an electric spark and engines that burn under the influence of compression (diesels).

The flammability of the fuel is evaluated by the cetane number (TsC), the higher the TsS, the shorter the ignition period. Because it is accepted to obtain the elemental composition of multi-component mixed fuels, which is one of our research issues, in a mass unit. We can express 1 kg of liquid fuel consisting of sulfur-free carbon C and oxygen O, shares as follows [8-10].

\[ C + H + O_m = I \] (1)

The calculation of heat is based on the equations of thermodynamics and the values of parameters obtained during testing of internal combustion engines in practice. It is known that the more the data obtained during the calculations are used during the calculations, which are close to the engines being designed in terms of a number of their main parameters, the closer the results of the heat calculation are to the truth [10-14].

3 Theoretical analysis of sources and data

A very important parameter characterizing the combustion process of biofuels in the engine is the heat release, among other things:
- pressure in the cylinder,
- temperature in the cylinder,
- average indicator pressure,
- heat load on engine parts,
- engine noise,
- combustion indicators,
- affects the change in the use of heat.
The coefficient of heat utilization of biofuel mixtures increases, which in turn leads to an increase in the rate of heat release. To determine the temperature at the end of combustion, we use the heat balance equation of the combustion process:

\[ \mu \cdot C_p \cdot T_z = (\mu_v + D \cdot \lambda)T_c + \frac{\xi Q_h}{\alpha L_o(1+y)} \]  \hspace{1cm} (2)

\[ C_v = 20.16 + 1.74 \frac{T_c}{10^3} \]  \hspace{1cm} (3)

\[ L_o = \frac{L_o}{2g} \]  \hspace{1cm} (4)

The amount of air required for complete combustion of 1 kg of fuel

\[ L'_o = \frac{1}{0.23} \left( \frac{8}{3} C + 8H - O \right) \]  \hspace{1cm} (5)

We calculate the amount of air required for complete combustion of the biofuel mixture using the following expression. Hydrogen H bio and oxygen O bio in biofuel are taken into account.

\[ L'_o = \frac{1}{0.23} \left( \frac{8}{3} C_{\text{dieselfuel}} + C_{\text{biofuel}} + 8H_{\text{dieselfuel}} + H_{\text{biofuel}} - O_{\text{dieselfuel}} + O_{\text{biofuel}} \right) \]  \hspace{1cm} (6)

Considering the formula applied to light petroleum products with a calorific value in the range of 10400-11500 cal/g and a hydrogen content of 10.85-16.00%.

\[ H = 0.005Q_B - 41.4 \]

where: \( Q_B \) – high combustion heat of fuel, cal/g;
\( H \) – hydrogen in the fuel, %.

\[ 8(0.005Q_B - 41.4) + 0.005Q_B - 41.4 \\
0.04Q_B - 331.2 + 0.005Q_B - 41.4 \\
0.045Q_B - 372.6 \]

The amount of air required for complete combustion of the biofuel mixture, taking into account the heat of combustion

\[ L'_o = \frac{1}{0.23} \left( \frac{8}{3} C_{\text{dieselfuel}} + C_{\text{biofuel}} + 0.45Q_{\text{common}} - 372.6 - O_{\text{dieselfuel}} + O_{\text{biofuel}} \right) \]  \hspace{1cm} (7)

If we consider Craig's formula from the number of empirical formulas, if its density is known, then it allows us to determine its heat of combustion [14-16]. To study the high heat of combustion of liquid fuels, this formula is presented:

\[ Q_B = 12400-2100\rho^2 \]  \hspace{1cm} (8)

Here: \( \rho \) – fuel density.

The following formula can be used to determine the heat of combustion of liquid fuels:

\[ Q_H = 11,088+757\rho - 2100 \rho^2 \]  \hspace{1cm} (9)

In order to ensure universality when comparing combustion processes in engines with different cylinder sizes and different rotation frequencies, it is convenient to compare the ratio of the amount of heat released at 10PKV to the air mass supplied to the cylinder or to the cylinder volume.

Taking into account the heat of combustion \( Q_B \) and the fuel density \( r \), we change the formula (7) as follows:

\[ L'_0 = \frac{1}{0.23} \left( \frac{8}{3} C_{\text{dieselfuel}} + C_{\text{biofuel}} + 0.45(12400 - 2100\rho^2) - 372.6 \\
- O_{\text{dieselfuel}} + O_{\text{biofuel}} \right) = \\
= \frac{1}{0.23} \left( \frac{8}{3} C_{\text{dieselfuel}} + C_{\text{biofuel}} + 558 - 945\rho^2 - 372.6 - O_{\text{dieselfuel}} \\
+ O_{\text{biofuel}} \right) = \\
= \frac{1}{0.23} \left( \frac{8}{3} C_{\text{dieselfuel}} + C_{\text{biofuel}} + 558 - 945\rho^2 - 372.6 - O_{\text{dieselfuel}} \\
+ O_{\text{biofuel}} \right) = \]
The rate of heat release is determined by the burning rate of fuel containing biofuel:

\[
\frac{d\theta}{d\alpha} = \frac{(W_d+b_c)dm_p}{\alpha}
\]  

where: \((W_d+b_c)\) – the lowest calorific value of fuel containing biofuel, kDj/kg; \(m_p\) - the mass of the mixture containing biofuel, kg, \(\alpha\) - the turning angle of the crankshaft, degree.

We can use this connection for a single case:

\[
\frac{1}{m_{po}} = \frac{dQ}{d\alpha}
\]

where: \(m_{po}\) - is the mass of air supplied to one cylinder, kg.

After the formation of the biofuel mixture, a number of changes occur in the combustion process, among which \(M_z\) is the total amount of combustion products \([17-19]\).

As we know, the total amount of combustion products in standard fuels is:

\[
M_z = \alpha L_0 + \frac{H}{4} + \frac{O}{32}
\]

Taking into account the heat of combustion \(Q_b\) and fuel density \(\rho\), we change the formula (11) as follows:

\[
M_z = \alpha L_0 + 216.3\rho^2 + \frac{O}{32} K_b
\]

where: \(K_b\) – determines the amount of excess O oxygen in the biofuel. Taking into account that \(M_c = \alpha L_0 (1 + \gamma)\), we present the formula for the molecular change coefficient \(\mu\) of the working mixture:

\[
\mu = \frac{M_z}{M_c}
\]

We express the coefficient of molecular change of the working mixture according to the composition of the biofuel mixture as follows.

\[
\mu = K_{b1} + 216.3\rho^2 + \frac{Q_{diesel\,fuel}}{32}
\]

Based on the above, the equation of the heat balance of the combustion process is expressed as follows:

\[
216.3\rho^2 + K_{b1} 1.25 \cdot 10^{-5} \cdot C'_p T_z = (C_{v1} + D\lambda) T_c + \frac{5Q_H}{a_t a_0 (1+\gamma)}
\]

Adjustment characteristics of the composition of the mixture serve to determine the dependence of power, fuel economy and other indicators on the relationship between fuel and air in the combustible mixture when operating in the state where the fuel contains a biofuel mixture. They make it possible to determine the standard use adjustments of the composition of the mixture, as well as to establish appropriate limits in terms of achieving the specified efficiency, thermal efficiency and toxicity.

Indicator of maximum combustion pressure unevenness:

\[
X_{pcb,\, max} = \frac{\sigma(P_{c,\, max})}{P_{cb,\, max}} = \sqrt{\frac{1}{k} \sum_{i=1}^{k} \left( \frac{P_{c,\, max} + P_{b,\, max}}{P_{c,\, max}} - \bar{P}_{c,\, max} \right)^2}
\]
where \((p_{c,\text{max}} + p_{b,\text{max}})\) – maximum pressure of biofuel on combustion; \(\sigma(p_{c,\text{max}})\) – maximum pressure deviation of mixture combustion; \(p_{c,\text{max}}\) – average value of the maximum pressure of the mixture combustion; \(k\) – number of cycles that come after each other. 1- in the picture \(p_{c,\text{max}}\) determination of registry invited.

In Figure 1, \(p_{c,\text{max}}\) registration (registration) for determining the maximum was offered.

Fig. 1. \(p_{c,\text{max}}\) is a graph of the average value of the maximum pressure of the combustion of the mixture.

Based on the graph presented in Figure 1, the indicator of the maximum combustion pressure unevenness is \(X_{p_{c,\text{max}}}\).

\[X = \frac{\sigma(F_c)_{\rho}}{(F_c)_{\rho}}\]  

\((F_c)_{\rho}\) - the standard deviation of the area of the indicator chart,  
\((\bar{F}_c)_{\rho}\) - the average value of the indicator indicator fields in one rotation cycle.  

\[
\bar{F}_c = \int_{\alpha_1}^{\alpha_2} \rho_c d\alpha
\]

The complete indicator is built on the coordinates of \(P\) and \(\alpha\), which allows determining the \(F_c\) area.

Fig. 2. Graph of change of pressure in the cylinder.

b) average pressure overlap index:

\[X_{pi} = \frac{\sigma(p_i)}{p_i}\]  

\(\sigma(p_i)\) - is the standard deviation of the mean reading pressure,
\[ p_i - \text{The average value of the indicator pressure of the engine operating cycles, } i = 1 ... \]

\[ .k \text{ – number of cyclic fuel transfers.} \]

s) Inequality indicator of unadjusted indicator chart:

\[ X_{(F_c)_0} = \frac{\sigma((F_c)_0)}{(F_c)_0} \]  \hspace{1cm} (21)

where \( (F_c)_0 = \int_{\alpha_3}^{\alpha_4} p d\alpha \); \( (F_c)_0 \) - the area of the incomplete indicator diagram in the segment \( \alpha_3 - \alpha_4 \); \( \sigma((F_c)_0) \) is the standard deviation of the index table at the limit; \( (F_c)_0 \) is the average value of the boundary line area. The incomplete indicator diagram represents the range from the closing of the inlet valve \( \alpha_3 \) to the closing of the exhaust valve \( \alpha_4 \).

4 Research results

According to the analysis, the existing gaseous fuels used in engines are composed of various gases, fuel and inert components. Gaseous fuels have a much higher detonation stability than gasoline. It is taken into account that the research is mainly focused on diesel engines. Diesels are divided into split-combustion and split-combustion types, of which the split-combustion diesel engine is efficient in terms of energy performance during startup. Volumetric and film self-ignition types are adopted in separate combustion chambers. The disadvantage of the volume ignition method is that it is impossible to form highly dispersed droplets from the mixture in the resulting fuel torch. The formed ash is formed due to the kinetic energy of the fuel ejected from the small holes in the formation of primary droplets, i.e.:

\[ E_{yo} = \frac{w_{yo}}{2} = \frac{\varphi_{yo} \Delta P}{\rho_{yo}} \]  \hspace{1cm} (22)

or

\[ E_{yo} = \frac{\varphi_{yo} \Delta P}{\rho_{yo}} \cdot 10^4 \]

where \( E_{yo} \) - is the kinetic energy of the flow; \( w_{yo} \) – speed coefficient \( (w_{yo}=0.7) \); \( \Delta P \) – variable pressure in the opening of the speaker of the exhaust, \( \text{N/m}^2 \); \( \rho_{yo} \) - fuel density, kg/\( \text{m}^2 \).

As we know, mixture formation in diesels is very complicated, and indicators of internal combustion engines: the complexity of heat utilization, noise emitted during engine operation, smoke and toxicity level of used gases, start-up qualities, thermal stress of combustion chamber elements depend on the quality of mixture formation. In order to achieve high performance of diesel fuel in the proposed biofuel mixture, the following basic requirements are imposed on the fuel injection process:

- at the end of the compression stroke of the biofuel mixture, the selected advance angle in the process of starting should correspond to the angle of rotation of the crankshaft by 10-30\(^\circ\);
- the duration of the fuel injection phase should not exceed 40-45\(^\circ\);
- supply of the biofuel mixture to the cylinders by volume should be changed according to the required turning angle of the crankshaft;
- during the cycle, the amount of biofuel mixture injected into the cylinder should correspond to the load, the speed regime, and when they change, this amount should also change accordingly;
- quality adjustment of the spraying parameters of the biofuel mixture and distribution to the combustion chamber in accordance with the adopted mixture method;
- the process of providing the biofuel mixture should be stable in all cycles of the engine.

Based on the above requirements, the duration of the mixture formation process is short, taking into account that the process of fuel refinement in diesels is found to be more complicated than in gasolines. When the fuel passes through the filter, the first turbulent
The processes of the flow appear, and the turbulent process becomes larger if the speed of the fuel is high and the channel walls are thick - the roughness is high. Under the influence of initial turbulence, aerodynamic resistance of charges compressed to 3-4 MPa in the chamber, a large part of the biofuel mixture breaks up near the nozzle opening. The speed of movement of fuel particles varies according to flow sections, at different times of spraying, and the conditions of movement of particles in the volume of the combustion chamber are also different. As a result, a large number of droplets (from 0.5 to 20 million) appear, and their sizes differ greatly from each other (from 5 to 100 μm). The concept of average droplet diameter is used to evaluate the parameters of the combustion process of fuel entering the combustion chamber, mixture formation.

The smaller the average diameter of the drops, the smaller the correction. The average diameter of the drops does not allow to correctly determine the homogeneity of the composition Ω, how different the drops are. But this indicator is one of the most important indicators for evaluating the quality of weaving. We can see this more fully from the correction characteristics in Figure 3.

This graph (line 1) divides the relative size increase of current diameter droplets from the smallest diameter droplets, and the steeper the curve, the smoother and smoother the fuel composition.

![Fig. 3. Correction characteristic. 1- increase in the relative volume of drops; the relative amount of droplet 2 in the stream.](image)

Considering the Bernoulli equation for incompressible fluid flow, we present the expression below.

\[
\frac{dV_{kr}}{dt} = f_n W = f_n \frac{2}{\rho_y} \sqrt{P_{nu}}.
\]  

Where: \( W \) - is the theoretical speed of fuel flow from the nozzle, \( f_n \) - is the effective cross-section of the injector nozzle, \( P_{nu} = P_f - P_{ts} \) - is the fuel injection pressure, \( P_f \) - is the fuel pressure in the nozzle in front of the nozzle hole, \( P_{ts} \) - is the gas pressure in the cylinder (antispray pressure).

According to the formula, the speed of the relative amount of droplets in the flow depends on the injection pressure and changes during the fueling process. Based on this, we will focus on the parameters of fuel adjustment and the change graph of the process of injecting it into the cylinder. Figure 2 shows a graph of the variation of the formed droplets at different times of spraying. It can be seen that at the beginning and at the end of fueling, the drops are the largest and are not homogeneous. Variation of trim parameters during spray characteristics is of great importance for mixture formation and combustion processes in diesels.

The physical properties of the fuel, primarily its viscosity, greatly affect the flow breakup and droplet size. With an increase in the viscosity of the fuel, the smoothness and homogeneity of the wear deteriorates, because the speed of the initial turbulence in the flow...
decreases. The aerodynamic resistance of the gas environment increases the counter-effect on the droplet breakup.

The main part of the amount of fuel should be given at a high speed, at a high pressure, because in this case the fuel particles will go to the edges of the combustion chamber, and the air in this zone will be fully used. It is not advisable to complete the injection of mixed fuel at a sudden speed, because in this case the fuel will accumulate around the injector, which will worsen the combustion process and cause the engine to stall.

In recent years, in order to increase the efficiency of the heat engine operating on different fuels, various designs of injectors have been created.

![Figure 4](image)

**Fig. 4.** Scheme of the combustion problem: I is the speed of drops in the cross sections of the torch; II - distribution of the mixture in the sections of the torch; 1 - the outer layer of the torch; 2 - the inner layer of the torch; $gf$ is the flame cone angle.

One of the main disadvantages of these injectors is the difficulty in obtaining the same homogeneous fuel mixture in the combustion chamber of the engine. In Fig. 4, the conical-shaped flame emitted from the nozzle, the intensity of homogeneous fuel with a large fuel capacity, $L_f$ and width $V_f$, and due to the reduction of the self-ignition process, it becomes difficult to create a uniform fuel mixture in intensity.

According to the second law of thermodynamics, the thermal efficiency for a theoretical cycle performed by 1 kg of working fluid:

$$\eta_t = 1 - \frac{|q_2|}{q_1} = \frac{q_1 - |q_2|}{q_1} = \frac{L_r}{q_1}, \quad (24)$$

where:
- $q_2$ - the absolute value of the amount of conversion of the cold source into heat, J/kg;
- $q_1$ is the amount of heat supplied per second, J/kg;
- $L_r$ - is the work done by liquid per 1 kg in the cycle, J/kg;

$$\eta_t = \frac{q_1 - |q_2|}{q_1} = \frac{\Delta L}{q_1}, \quad (25)$$

where:
- $L$ – volume, kg/m$^3$;
- $A$ - is the thermal equivalent in kcal/(kg·m).

**5 Proposal and discussion**

Analyzing the above data, we can say that if a high-pressure ("Common rail") modern system is installed in the supply system of a diesel engine, a higher level of fuel refinement will be achieved. This condition ensures good fuel distribution and good combustion. The work done during the cycle is not completely used by the heat engine, part of it is spent on friction and other losses in engine parts [20-24]. All these losses are accounted for by the efficiency of individual engine elements. In order to increase the efficiency of the mixed fuel that can be used in diesel engines, we suggest to equip the injector with a mixing mesh turbulator as shown in Figure 5.
Fig. 5. Mesh Turbulizer Injector: 1- corpus; 2-nozzle holes; 3- needle (pin); 4-way turbilizer.

It consists of a body 1, a nozzle 2, a needle 3, a replaceable turbulizer 4. The fuel mixture is adjusted to the mesh turbulator installed at a distance of 20-25 mm through the nozzle hole 2 of the nozzle, compared to the available nozzles, and exits at a high speed.

The high-performance cylinders of the mesh turbulator provide turbulence in the fuel flow, resulting in intensive crushing and coagulation of the fuel mixture.

6 Conclusion

The expansion cycle of a mixed fuel diesel engine takes place under conditions of long distance and wide fuel flow, so the auto-ignition process occurs with a large emulsion.

The work done during the cycle is not completely used by the heat engine, part of it is spent on friction and other losses in engine parts. All these losses are accounted for by the efficiency of individual engine elements.

At a short distance (20-25 mm), due to the turbulence of the fuel flow, it is fully supplied with the recommended fuel mixture and creates optimal conditions for creating a uniform mixture without waste. The use of multi-component biofuels is recommended for use in engines equipped with the common rail supply system.

The main advantage of this system is to increase the fuel injection pressure by three to four times, i.e. to 150...200 mPa. At this pressure, the fuel particles injected into the cylinder are very small and are better oxidized by air oxygen, and the combustion process is complete. As a result, the efficiency of diesel increases.

References