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## Optimization of Mixing Energy in Two-Chamber Engines.

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**ABSTRACT**

The article concerns the analysis of the effect of the interaction between air charge and fuel flame factors on mixing energy. Based on the available statistics on the parameters of mixing in various types of diesel engines, the dependence of the operating parameters change on the components of the total mixing rate has been revealed. The optimum total mixing rate, inter alia through the reallocation of components in favor of a pneumatic component, has been determined. A perspective diagram for the two-chamber engine operating cycle, as well as the calculation methods for the main geometrical parameters of its pre-combustion chamber, has been suggested. Particular focus has been made on the selection of the optimum smallest throttling cross-section of the pre-combustion chamber. The experimental data about the performance of the suggested workflow diagram for the retrofit engine have been set forth. Besides, the immunity of the engine to various fuel grades within a wide range of octane numbers and their viscosity has been revealed.

**Keywords:** pre-combustion chamber, throttling section, multifuel engine, internal combustion engine, mixing energy.

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## INTRODUCTION

Nowadays, internal combustion engines occupy a highly important place in world's power engineering. Statistics proves that reciprocating internal combustion engines, possessing higher economic performances, as compared to engines of other types, account for a greater part of all energy generated within the recent years [1].

The comparative analysis of the characteristics of diesel engines with different combustion chambers reveals some trends in the change of mean values of the fuel injection pressure and the air charge motion velocity. The threshold limits, the values of which change within the limits of 150…8 MPa … of injection pressure in diesel engines with undivided chamber and volumetric mixing and pre-combustion … respectively. The change of the air charge velocity falls within similarly comparable limits, from 5 to 320 m/s. Therefore, it can be assumed that the increase in the air charge motion, and consequently, in its energy is compensated by the decrease in the fuel injection energy, and vice versa.

It would be reasonable to ask how far such compensation can go, if it is possible to discharge high pressure fuel injection equipment completely by endowing it with dosing functions only and leaving mixing to the moving (migrating) air charge. One of the main tasks of the investigation will include studying the processes of mixture migration between the divided chambers of pre-combustion engines.

The studies have shown [2] that the wall temperature affects the droplet evaporation rate.

However, at the same time, the extended surfaces of combustion chambers deteriorate starting characteristics of the diesel engine [3-6].

The effect of temperature rise in cold diesel conditions cannot be used, that is why, in order to enhance the starting characteristics, as well as to additionally mitigate the ignition delay, and therefore, to smoothen combustion at operating modes, multifuel engines are provided with the function of significant increase in the compression degree to $\varepsilon$ = 24…26, which results in the increase of the combustion pressure and mechanical losses [7].

At the same time, it has been proved [8] that the range of compression degrees, optimum from the thermodynamic point of view, falls within the limits of $\varepsilon$ = 12…14 (and is characterized by a slight heat utilization and mean pressure gain), i.e. the overcompression by 10 units takes place. Until today, the operation at such compression degrees has hit stiff headwinds related to the insufficiency of octane numbers of petrol engines and a higher operating roughness of diesel engines, especially of smaller size engines.

Based on the above:

1. In order to decrease the injection energy, either the air charge energy shall be increased, or the available air charge energy shall be used more rationally (taking a pre-combustion version as a basis).
2. In order to enhance the cycle reproducibility indicators and to mitigate the impact of fuel physical and chemical properties, a forced electric-spark ignition should be applied.

The diagram of a smoothly running and actively controlled combustion process [9], shown in Figure 1, comprises prerequisites for the real solution of the vast variety of issues related to both, the use of multifuels, and the enhancement of economic and environmental characteristics of reciprocating internal combustion engines (ICE). Due to its novelty and evident potential, the latter has been chosen as an object of research [9].

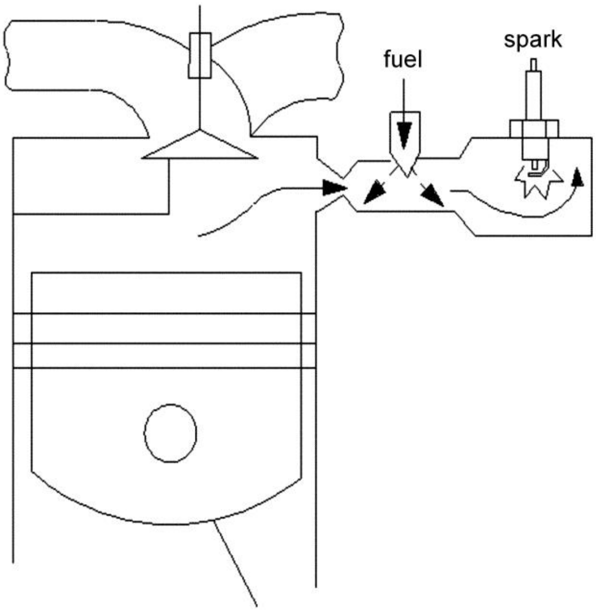


**Figure 1. Workflow diagram**

According to the above diagram, fuel is injected into the channel connecting the main and the auxiliary chambers. The cyclic dose distribution as per the chamber volume takes place in the course of injection.

Mixing in the main chamber and in the cylinder is effected on account of intensive swirls created by the combustion products displaced in the pre-combustion chamber at a high rate and of the displacer located opposite the connecting channels.

## METHODS

### Optimization of mixing energy in two-chamber engines

The mixing energy of conventional diagrams is composed of several components. They are: injection pressure, air flow velocity, cylinder temperature and pressure:

$$E_{\mathrm{mix}} = E_{af} + E_a + E_t \ . \tag{1}$$

where $E_{af}$ – air flow energy;
$E_a$ – atomization energy:
$E_t$ – thermal energy.
The value of kinetic energy related to 1 kg of the atomized fuel (specific atomization energy)

$$E_{\mathrm{a}} = \frac{\omega_d^2}{2} = \frac{\varphi_d^2 \cdot P_d}{\rho_f} \ , \tag{2}$$

where $\omega_d$ – fuel discharge rate in m/s;
$\varphi_d$ – discharge rate coefficient;
$P_d$ – pressure drop at nozzle holes in MPa;
$\rho_f$ – fuel density in kg/m$^3$.

For practical purpose, it is more convenient to relate the atomizing energy to 1 g of fuel. Table 1 sets forth the values of rate values and the specific atomization energy at $\varphi_c = 0.7$ and $\rho_d = 830$ kg/m$^3$.

As is stated above, a pre-combustion diesel, characterized by coarse atomization of fuel by a pintle or single-hole nozzle, has been taken as an alternative for mixing. It has also been taken into account that the increase of the flow rate from 12 to 100 m/s reduces the duration of the evaporation and combustion of similar size fuel droplets three-fold.

**Table 1 – Atomization parameters**

| Fuel parameters | Mechanical atomization under pressure $P_d$ (MPa) | | | | | |
|---|---|---|---|---|---|---|
| | 10 | 13 | 25 | 32 | 45 | 130 |
| Discharge rate, m/s | 108 | 123 | 170 | 192 | 229 | 386 |
| Specific kinetic energy $E_t$ J/g | 5.9 | 7.66 | 14.8 | 18.8 | 28.5 | 76.6 |

Disregarding the atomizing fineness and assuming that there exists a certain optimum mixing energy, the optimum mixing energy can be expressed through velocity, by replacing the atomization energy and the air charge energy of Formula 1 with the discharge rate and air charge motion velocity:

$$\omega_{mix} = \omega_d + \omega_m \cdot \tag{3}$$

Where the fuel discharge rate

$$\omega_d = \mu_c \cdot \sqrt{\frac{2g}{\gamma}\left(P_{cr} - P_i\right)}, \tag{4}$$

where $P_i$ – chamber pressure at injection;
$\gamma$ – fuel specific weight.

Based on the calculations by the formula (3), the analysis of 55 models of diesels has been summarized in Table 2.

**Table 2 – Parameters of diesel engines**

| Parameter | Combustion chamber | | | | |
|---|---|---|---|---|---|
| | Non-divided | Semi-divided | M-process | Swirl chamber | Pre-combustion chamber |
| $S_p$(m/s)* | 8.4…11 | 6.5…7.2 | 11 | 5.5…8.1 | 8.5…10.2 |
| $\omega_{mix}$(m/s)** | 300…200 | 250…200 | 250…210 | 360…300 | 450…370 |
| $\omega_m/S_p$*** | 0.25…2.3 | 3…8 | 8…9 | 10…40 | 20…60 |
| $g_e$(g/kW·h) | 208…232 | 224…258 | 230…258 | 258…313 | 278…340 |

* Piston speed.
** Mixing rate equivalent calculated by the formula (3).
*** Ratio of air charge motion velocity to piston speed.

The analysis of the above data shows that the total missing rates and specific fuel consumptions of the first three types of diesels are comparable in terms of absolute values. At the same time, pre-combustion and swirl diesels are characterized by the mixing rate which is by 100 m/s higher, and thus, by a higher specific fuel consumption (Figure 2).

Taking into account that the mean injection pressure of swirl engines is 12.5 … 15.0 MPa, which corresponds to 115 … 140 m/s, and that of pre-combustion engines – 8 … 13 MPa, which corresponds to 90 … 120 m/s, then, taking 230 … 250 m/s as an optimum mixing rate, we will get the following: swirl chamber engines at the lowest limit fall within this area, and approximately 20% of their injection energy is consumed by mixing, while in pre-combustion chamber engines all injection energy is consumed inefficiently and is parasitic. In pre-combustion chamber engines the energy is consumed by a throttled cross flow, leading to the mixture overichment in an auxiliary chamber. As a result, fuel underburning, smoking and, consequently, decrease in the power-to-weight ratio and economical operation take place.

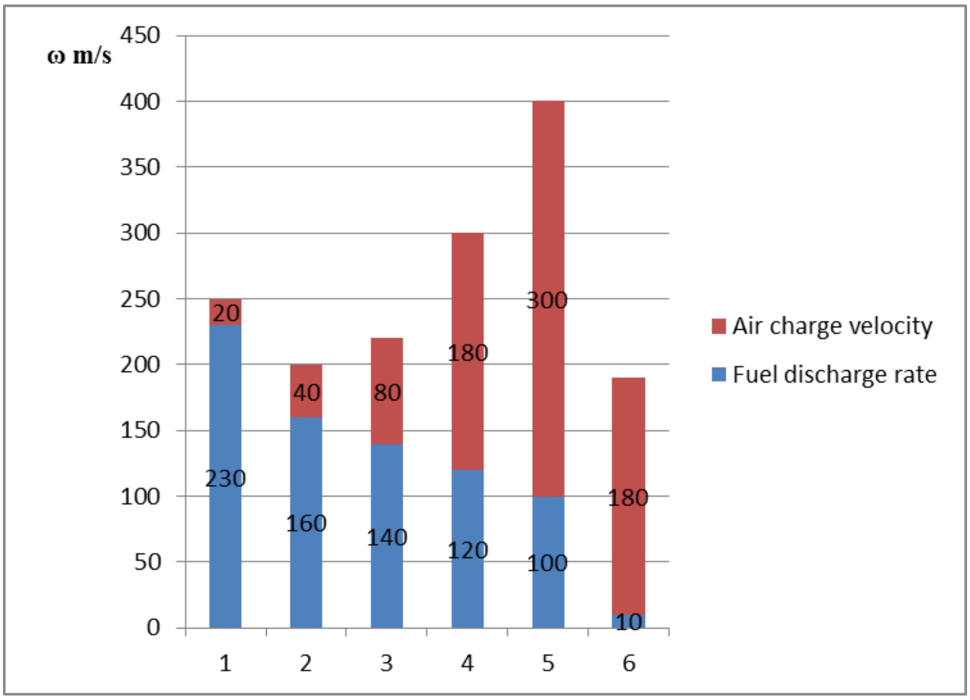


**Figure 2. Total mixing rates:**
**1 – diesel with undivided combustion chamber; 2 – diesel with semi-divided combustion chamber;**
**3 – with M-process; 4 – swirl chamber; 5 – pre-combustion chamber; 6 – perspective workflow diagram**

Based on the analysis results, it would be fair to say that the perspective diagram for the pre-combustion mixing shall be characterized by the minimum atomization energy at the cross flow rate of from 200 to 300 m/s, depending on the change of the nominal speed rate. In practice, this can be achieved by the application of open type fuel injectors with reduced hydraulic resistance of the nozzle hole.

**Determining the optimum throttling cross section for the connecting channel of the engine pre-combustion chamber**

One of the problems concerning the implementation of the burning process stipulated by this diagram is in maintaining the optimum level of mixing energy, as well as the increase of pre-combustion chamber products pressure, i.e. the assurance of heat release dynamics within specified limits.

All-mode adjustment of this parameter is one of the main problems, solving which would make it possible to forecast engine operating parameters at the design stage.

It is evident that the simplest solution lies in the creation of certain pressure ratios at the pre-combustion chamber outlet which would not change significantly following the change of the engine rotation speed and load, and would ensure constant specified conditions in the pre-combustion chamber cavity for the development of the combustion process, as well as the use of the accompanying effect of combustion products discharge rate with the aim to enhance mixing in the main volume.

In actual practice, this can be achieved by designing a connecting channel between the pre-combustion chamber and the main combustion chamber with the outlet into the main cavity performed in the form of a nozzle of specific section. Taking into account the influence of the throttling channel (its section) between the pre-combustion chamber and the main volume on the cross flow energy during compression (mixing), the issue of substantiating the selection of the throttling section must be considered thoroughly.

The flowchart diagram for the add-on pre-combustion chamber is shown in Figure 3.

The basic parts of the pre-combustion chamber are: pre-combustion cavity 2 – (between the sections v-v and k-k); 1 – segment nozzle (area between k-k and a-a); by its smallest section or throat kr-kr, the nozzle is divided into two parts – a convergent part (connecting channel limited by the area from k-k to kr-kr) and a divergent part (area between kr-kr and a-a).

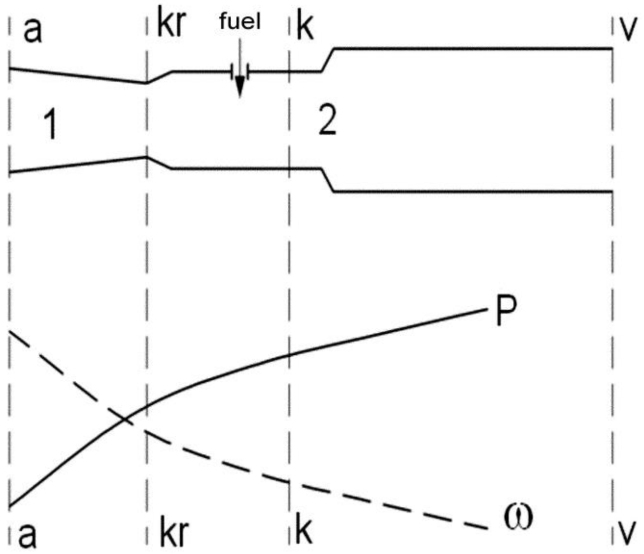


**Figure 3. Diagram for the flow path of the pre-combustion chamber**

The ratio of the nozzle outlet section area $F_a$ to its smallest section area $F_{kr}$ is called nozzle divergence. The ratio of the pre-nozzle pressure $P_k$, i.e. at the k-k chamber section, to the pressure $P_a^c$ at its outlet section is called the ratio of gas expansion in the nozzle and is expressed as follows:

$$\delta_c = \frac{P_k}{P_a^c}. \qquad (5)$$

The connecting channel, into which the fuel is injected, performs the function of a mixing apparatus, both at the compression stroke, and at the subsequent stages. In the course of combustionб it will be also considered as a part of the common combustion chamber. The process of combustion products expansion takes place in the nozzle. As a result of this process, the pressure of combustion products and temperature T fall, and the rate ω [Omega] rises, as is schematically shown in Figure 3.

The absolute value of the combustion products velocity $\omega_a$ at the nozzle outlet section shall be determined by their initial pre-nozzle temperature, i.e. by the temperature at the combustion chamber outlet $T_k$, by the degree of expansion in the nozzle, as well as by physical and chemical properties of combustion products and energy losses in the nozzle.

The expansion of combustion products and their discharge from the pre-combustion chamber is accompanied by fuel under-burning. This is determined by the above mentioned reasons; that is why not only combustion products enter the nozzle, but also unburned part of the fuel in a vapor and fine state. This fuel burns out while being expanded in the nozzle and in the main cavity of the combustion chamber. Therefore, due to this reason, only approximate calculations for the process of combustion products expansion and discharge in the nozzle are possible.

In order to determine the rate of combustion products discharge from the pre-combustion chamber, one should know the temperature $T_a$, pressure $P_a^c$ and the composition of the combustion products at the nozzle outlet section. These parameters can be determined when performing thermal calculations for the engine. Besides, they will be determining when performing further calculations.

The theoretical rate of combustion products discharge from the pre-combustion chamber is determined based on the obtained value k' – the adiabatic exponent calculated using the known values of temperature, pressure and gas constants at the initial, final or intermediate section of the nozzle. For this purpose, the following adiabatic equation is used:

$$\left(\frac{P_a^c}{P_k}\right)^{\frac{k'-1}{k}} = \frac{R_a^c \cdot T_a^c}{R_k \cdot T_k}. \tag{6}$$

After taking logarithms, we can find

$$k' = \frac{\lg \dfrac{P_k}{P_a^c}}{\lg \dfrac{R_a^c \cdot T_a^c \cdot P_k}{R_k \cdot T_k \cdot P_a^c}}. \tag{7}$$

The theoretical rate of combustion products discharge from the nozzle shall be determined using the following formula:

$$\omega_a' = \sqrt{2g \cdot \frac{k'}{k'-1} \cdot R_k \cdot T_k \cdot \left[1 - \left(\frac{P_a^c}{P_k}\right)^{\frac{k'-1}{k'}}\right]}, \tag{8}$$

where $T_k$ – temperature at combustion chamber outlet;
$P_a$ – pressure at nozzle outlet section;
$R_k$ – gas constant for combustion products;
g – gravitation acceleration;
$P_k$ – in-chamber pre-nozzle pressure.

The theoretical discharge rate at the smallest nozzle section can be determined in a similar way, i.e.

$$\omega_{kr}' = \sqrt{2g \cdot \frac{k'}{k'+1} \cdot R_k \cdot T_k}. \tag{9}$$

Using the values of $T_k$ and $R_k$, we can determine the specific volume of combustion products

$$V_k = \frac{R_k \cdot T_k}{P_k}. \quad (10)$$

The actual specific volume of combustion products at the outlet section of the pre-combustion chamber, with the combustion products gas constant $R_a$ and actual temperature $T_a$, is

$$V_a = \frac{R_a^c \cdot T_a^c}{P_a^c}, \quad (11)$$

where $P_a^c$ – rated compression pressure at nozzle outlet.

The basic nozzle dimensions determine the throat and the outlet section $F_{kr}$ and $F_a$, as well as the ratio of these areas $F_a$ / $F_{kr}$ (i.e. nozzle divergence). The areas of these sections and the divergence are calculated in accordance with theoretical parameters of the state and the combustion products velocities in the nozzle as per the following formulae:

The theoretical area of pre-combustion chamber outlet section

$$F_a = \frac{V_a}{\omega_a'}. \quad (12)$$

Theoretical area of the nozzle throat

$$F_{kr} = \frac{V_k}{\omega_{kr}'} = \frac{R_{kr} \cdot T_{kr}}{P_{kr} \cdot \omega_{kr}'} \quad (13)$$

Theoretical ratio of nozzle divergence

$$\frac{F_a}{F_{kr}} = \frac{V_a \cdot \omega_{kr}'}{V_{kr} \cdot \omega_a'}. \quad (14)$$

Another factor influencing the area of the smallest section is the relative area:

$$f_k = \frac{F_k}{F_{kr}}. \quad (15)$$

where $F_k$ – area of the flow path of the combustion chamber (connecting channel).

The calculation of the relative area is determined by a number of reasons:

1. Possibility to use cross flow energy in the course of compression-combustion for endowing injectors with dosing function only.
2. Economically optimum combustion products consumption mode and, at the same time, maintaining dynamic combustion process.
3. Minimal throttling.

4. Limitation of structural components dimensions.

It should be noted that the experimental use of chambers $f_k < 2$ hasn't given any positive results. Stable operation was observed in idle mode or under partial loads. In the chambers where $f_k = 1$ even the effect of flame quenching was observed.

According to the calculations carried out for UD-15 engine (УД-15), converted into a multifuel version, and based on the selection of the optimum cross flow rate and the criteria of combining the area of the connecting channel and its smallest section, the below values have been obtained: the area of the connecting channel $F_k$ = 95 mm$^2$ ($d_k$ = 11 mm); the smallest area of the connecting channel $F_{kr}$ = 11.6 mm$^2$ ($d_{kr}$ = 3.85 mm).

In actual practice, the diameter of the throttling channel connecting the main and the auxiliary chambers, impacts pressure rise in the main combustion chamber in the process of compression, which is evidenced by experiments [9].

Determining nozzle divergence and substantiating the selection of a dimensionless area of combustion chamber, as well as defining the optimum mixing rate as 230…250 m/s allows to solve the issues of uncertainty of the geometric characteristics of the pre-combustion chamber under the studied workflow diagram.

The determination of the cross flow energy is based on the calculation of the smallest throttling section, the accurate calculation of which makes it possible to calculate the degree of pressure increase in the pre-combustion chamber, if any.

## RESULTS

As it was mentioned above, when fuel is injected into the connection channel in the engine, using the cross flow energy during mixing – combustion and forced ignition, one should first of all identify the primary operating trends at various combinations of the connecting channel and outlet nozzle parameters.

Single-cylinder four-stroke engine UD-15 (УД-15) was taken as an object for carrying out the whole research.

**Table 3 – Technical characteristics of the engine**

| Parameter | Characteristic value |
|---|---|
| Number of cylinders | 1 |
| Number of strokes | 4 |
| Crankshaft speed | 3,000 min$^{-1}$ |
| Cylinder diameter | 72 mm |
| Piston travel | 60 mm |
| Initial degree of compression in single chamber modification | 6 |
| Capacity | 245 cm$^3$ |
| Cooling | air |

1. The cylinder head of a series-production engine has been subject to modification with the aim to increase the compression degree. Its volume was reduced from 50 cm$^3$ to 35 cm$^3$, which allowed to increase the compression degree to $\varepsilon = 8$.
2. A number of experiments has been carried out on the modified head with the attached add-on chamber, the volume of which varied from $a = 0.3$ to $a = 0.15$ of the cylinder head volume. As this took place, the compression degree changed from 6.3 to 6.8 respectively.
3. The engine is equipped with a high-pressure fuel pump driven through a magneto shaft gear drive. The fuel injection equipment has a high-pressure single-plunger pump with a plunger diameter of 7 mm.

In the workflow diagram under study, the combination of the advantages of fuel injection into an auxiliary chamber with the advantages of the pre-combustion ignition system is assumed. The improvement of economic performances in this regard is supposed to be achieved due to the intensification of the air-and-fuel mixture combustion as a result of charge ignition in the main chamber not by a point source of ignition, but by a flame jet exiting the auxiliary chamber at a high rate.

Such ignition creates additional turbulent swirls in the air-and-fuel mixture during the second phase of combustion, which contributes to nondetonating combustion at higher compression degrees and levels the difference between the physical and chemical properties of various kinds of fuel.

Theoretical substantiation for the selection of the smallest section of the connecting channel and the channel zone where fuel injection takes places was set forth above. According to the calculations performed, based on the selection of the optimum cross-flow rate and the criteria of combining the connection channel area and its smallest section, we obtain the following values: connecting channel area: $f_k$ = 100 mm$^2$, the smallest area of the connecting channel $f_{kr}$ = 13.6 mm$^2$.

The verification of the rated parameters was carried out on three chambers of similar volume, $a$ = 0.3, but of different outlet nozzle parameters. The areas of the smallest sections: 7 mm$^2$; 12.5 mm$^2$; 19.6 mm$^2$, and of the outlet sections, with the accepted expansion coefficient 2.5-18 mm$^2$; 30 mm$^2$; 50 mm$^2$ respectively. The areas of the main connecting channel were successively changing upwards, from 28 mm$^2$ to 113 mm$^2$.

The summary of the experimental data, as well as the mentioned comparisons, allow us to state that during fuel injection into the channel connecting the main and the auxiliary combustion chambers, and by dividing a cycle dose in the course of mixing-combustion, the engine acquires a tendency for automated operation with the mixture composition optimum for the specific mode [10].

It is worth noting that engine operation with the optimum mixture composition can be achieved by the use of the advantages of flame ignition in combination with high-quality methods of engine power adjustment [11].

The availability of a sufficient air volume within the main chamber, as well as charge turbulence in the course of combustion, contributes to good mixing therein and complete combustion [12]. In this case, both power and economic performances of the process are determined by the total thermal effect of mixture combustion in the main and auxiliary chambers with a tendency for being operated with the lean mixture.

As is evident from the foregoing, carrying out the process as per the diagram described enables to implement the principle of qualitative power adjustment. In doing so, changing of the mixture quality is achieved by changing the fuelling value.

## CONCLUSION

1. The reserve for the gain in the pre-combustion engine performance lies in the rational use of the mixing energy, which is expressed in atomization energy liquefaction and cross-flow energy rational use.
2. The determination of the cross-flow energy is based on the calculation of the smallest throttling section. The accurate calculation of the throttling section makes it possible to find the degree of pressure rise in the chamber.
3. The determination of nozzle divergence and substantiating the selection of a dimensionless area of combustion chamber, as well as the finding the optimum mixing rate, which falls within the range of 230…250 m/s, allows to remove the uncertainty in selecting geometric characteristics of the pre-combustion chamber under the studied workflow diagram.
4. The reviewed diagram of carrying out the process by injecting into the connecting channel and using cross-flow energy for dividing a cycle dose during mixing-combustion makes it possible to adjust engine power by changing the quality of the mixture, from idle to full power, and to increase the compression degree to the level which is considered to be optimum from the thermodynamic point of view.
5. By virtue of the above factors, engines acquire tendency for automated operation with the mixtures ensuring optimum efficiency at the given load with no mixture quality automation required. This circumstance facilitates the fuel injection control system significantly.

6. The studied structural diagram reveals potential opportunities for the use of different grades of fuel, both commercial and non-conforming.
7. The selected design parameters of pre-combustion chamber, injectors and connecting channel section ensure stable operation of the engine across the whole range of its speed rates.

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