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Method for calculating thermodynamic processes in the organization of a gas-steam cycle in an internal combustion engine

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ABSTRACT

The article considers the organization of the internal combustion engines' working process aimed at increasing environmental safety and the degree of use of the internal energy of the working fluid in order to increase the energy efficiency and environmental safety of ship, stationary and transport engines and the possibility of increasing the efficiency of the working cycle of internal combustion engines. The method of calculating the expected indicators of the proposed technology of workflow organization, calculation of the required amount of water is proposed. The pressure in the cylinder after the end of vaporization is determined using partial pressures, the presence of water vapor in the exhaust gases can help simplify their cleaning in the fight against harmful emissions. The results of calculations carried out according to the proposed method allow us to expect an increase in the average effective pressure by an amount from 15% to 35%, depending on the moment of water injection.

1 Introduction

Taking into account the constant expansion of the number and aggregate power of internal combustion engines, the danger of full production of natural hydrocarbon reserves becomes more and more real. At the same time, the level of pollution of the human habitat by spent combustion products is becoming increasingly large and approaching critical [1], which, objectively, inspires great concern.

Heat engines make it possible to convert the chemical energy of fuels into thermal and, further, into mechanical energy, which is used directly or transformed into other types of energy. In our country, about 80% of all electric energy is produced by thermal power plants and only 20% by hydroelectric. Thermal energy, converted into mechanical energy and used directly to drive cars and vehicles, is 4-5 times higher than the energy generated by all electric stations.

A piston engine with a crank mechanism, from the point of view of converting the chemical energy of fuel into mechanical energy, is a product far from perfection: on average, its effective efficiency does not exceed 0.35 for spark-ignition engines and 0.45 for diesels.

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There are many reasons for this: incompleteness of fuel combustion caused by imperfection of the working process, heat loss with exhaust gases and coolant, mechanical losses in the engine, etc. Researchers and designers, of course, try to eliminate them or at least to some extent weaken their effect. In particular, while reducing mechanical losses in the engine, undoubted successes have been achieved, for example, through the use of new structural materials, improving the quality of fuels and lubricants, improving the technology of machining parts and assembling engines. However, there is a reason that, in principle, it is impossible to influence – the crank mechanism, more precisely, the transformation of the reciprocating movement of the piston into the rotational movement of the crankshaft. Due to the relatively small expansion stroke, it is not possible to completely convert the thermal energy of the working body formed as a result of fuel combustion into mechanical work.

Thus, the problem of maximizing the use of fuel energy is of great importance, not only because it gives a great economic effect, but also due to the need for economical use of natural fuel reserves. Plus, more efficient use of fuel energy increases the environmental safety of engines, since reducing fuel consumption leads to a reduction in the total amount of emissions.

Thermal energy cannot be completely converted into another type of energy. Therefore, the most important task of power engineering is to find the most efficient ways to convert thermal energy into its other types and, in particular, into mechanical energy.

The main scientific idea of the study is to increase the degree of use of the thermal energy of the working fluid in order to increase the energy efficiency and environmental safety of ship, stationary and transport engines.

2 Analysis of the possibilities of increasing the degree of use of internal energy of an expanding working fluid

Due to the sufficiently high degree of knowledge of the combustion processes and the significant successes achieved in improving the efficiency of their flow, it is advisable to analyze the possibilities of improving the efficiency of the thermodynamic cycle of marine internal combustion engines in order to make the fullest possible use of heat in the cycle, that is, to increase the degree of use of the heat of the working fluid.

When analyzing work cycles, their images are used in the coordinates p–v and T–s. These diagrams are convenient for analysis because in p-v coordinates, the area under the process curve characterizes the mechanical work of this process, and in T–s coordinates, the area under the process curve characterizes heat. In addition, the diagram in T–s coordinates allows for a convenient comparison with the Carnot cycle, which gives us the possible maximum of the thermal efficiency to which we need to strive. It is irrational to create an engine running on the Carnot cycle, since the work received for a single cycle is too small and a real engine is largely spent on overcoming the friction of moving parts. Therefore, in the thermodynamic cycle of existing engines, heat is supplied to the working fluid not at a constant temperature (by isotherm), but at a constant pressure (by isobar), and even better at a constant volume (by isochore) or by some intermediate process between isobar and isochore [2].

In the p – v diagram, the cycle mechanical work is characterized by the area inside the diagram. In order to increase the mechanical work of the cycle and, accordingly, its efficiency, it is necessary to increase the completeness of the diagram. With the fuel supply unchanged (that is, without changing the heat supply), this can be done by changing the nature of the compression and expansion processes. The work of the expansion process can be increased if the polytrope of the expansion process is made to flow more steeply. This could help to use the exhaust gas energy in the cylinder more fully, that is, to expand to lower pressure values at the end of the process. But the crank mechanism does not allow this without additional measures (for example, you will have to increase the stroke of the piston or other characteristics of the mechanism).

Considering the process of expansion of z' – b, we see the same losses in the cooling fluid. And this suggests the idea of creating an engine with a heat-insulated cylinder and, possibly, a cylinder cover. Indeed, cooling in modern internal combustion engines is only necessary in order to prevent the growth of temperature stresses in parts to values that threaten their strength and possible jamming of engine parts. In the presence of materials capable of working normally in such conditions without cooling and with special lubrication conditions, it would be possible to approximate the compression and expansion processes to adiabatic ones, as in the Carnot cycle, and thus increase the efficiency of the working cycle [2].

However, once again everything rests on the impossibility of carrying out a complete expansion in existing crank mechanisms. As a result, an additional increase in the internal energy of the working fluid, obtained by reducing losses to the cooling system due to the thermal insulation of the cylinder, will turn into increased losses with exhaust gases.

On the one hand, this can be recognized as a positive phenomenon, since the exhaust gases, having more energy, can increase the production of mechanical energy in the turbocharger and, possibly, feed part of it to the crankshaft. With such a decrease in the heat sink into the cooling system, most of the heat is removed with exhaust gases and then used for turbocompounding. On the other hand, this leads to an additional complication of the design, and an increase in the efficiency of the thermodynamic cycle will not be achieved. Thus, in fact, the heat insulation of the cylinder leads to a redistribution of the heat balance items - with the use of thermal insulation, the degree of use of thermal energy in the cycle decreases and attempts to use it are transferred to the turbocharger, that is, to another thermodynamic cycle. At the same time, losses appear when the working fluid is moved into this other cycle (for example, pressure losses, and hence the energy of combustion products, heat losses in the exhaust pipeline and in the turbocharger itself, which also needs to be cooled). In addition, such measures lead to an increase in the emission of nitrogen oxides, as they increase the temperature in the cylinder [2].

That is, it is necessary to somehow solve the problem of using this additional heat, which can be obtained due to the thermal insulation of the cylinder. It would be useful to increase the expansion stroke while leaving the compression stroke unchanged. This would make it possible, without increasing energy costs during the compression process, to trigger a large temperature drop during the expansion process, that is, to use this additional thermal energy "returned" to the cylinder in the form of an increase in the internal energy of the combustion products.

Atkinson did this at the time, proposing his own design of a mechanism for transforming motion, where the course of the expansion process is greater than the course of the compression process [4]. However, he pursued slightly different goals. His proposal was not widely distributed due to the noticeably greater complexity in comparison with the traditional crank mechanism.

Engines with continued expansion are also offered, which were mentioned above, when the exhaust gases are sent to a cylinder with a larger piston area, where additional work is performed. However, such a solution again leads to a significant complication of the design. A compromise solution may be the organization of continued expansion of the working fluid in serial engines of traditional design with engine boost by supercharging while reducing the number of working cylinders. For this, a part of the cylinders is used for continued expansion. When the cylinders are turned off, the implementation of the traditional workflow stops, and they are transferred to the mode of continued expansion. In working cylinders using supercharging, the compression ratio should be reduced to maintain an acceptable level of loads on the engine parts. This leads to the fact that a decrease in the compression ratio in the working cylinders causes a decrease in the indicator efficiency, and there are also increased mechanical losses due to the switched-off cylinders [6].

An increase in the stroke of the piston would also allow engines with non-piston mechanisms of motion conversion, which were also mentioned above, but their application involves a significant change in engine production technology, which, unfortunately, is too early to talk about, since well-established production technologies of traditional piston internal combustion engines, allowing for a high resource they leave engines with spinless motion conversion mechanisms out of competition.

The tasks of this part of the study also do not include a significant change in the technologically proven design of the engine, therefore it is necessary to use this additional heat in some other way.

In [2, 3], the technology of organizing the workflow with water injection into the cylinder at the end of the combustion process is described. It is proposed to introduce water into the cylinder after reaching the maximum cycle temperature, which uses part of the internal energy of the combustion products for its evaporation. The resulting steam will increase the pressure in the cylinder, which will increase the cycle mechanical work. Due to the introduction of an additional working fluid, for the formation of which heat is spent, the temperature in the cylinder will decrease. A decrease in the temperature in the cylinder will lead to a decrease in the heat flow from the combustion products into the cylinder walls [3]. It is expected that water injection at the end of the combustion process will increase the energy efficiency of the engine without significantly complicating the design of its main elements. The water vapor in the exhaust gases can facilitate their purification in the fight against harmful emissions [5]. Thus, the proposed workflow of an internal combustion engine can be called a gas-steam cycle.

It is worth noting that water injection after completion of combustion has been offered before. For example, in a Crower engine, water is injected into the combustion chamber at a pressure of 15 MPa after the completion of the exhaust stroke [6]. Taking heat from the heated surfaces of the combustion chamber, the water evaporates. When the piston moves from the upper dead center to the lower, the water vapor, expanding, performs useful work (the fifth stroke is the steam stroke). When the piston moves from the lower dead center to the upper (sixth stroke), the exhaust steam is released. The spent steam enters the condenser, where it cools and turns back into water. The Crower cycle differs from the traditional Otto cycle not only in the number of cycles, but also in the ratio of the number of working cycles to their total number. So, in the Otto cycle, this ratio is 1: 4, and in the Crower – 1 : 3, an additional 40% of the useful work is performed on a constant amount of fuel [6].

The advantages of the Crower engine are high fuel efficiency due to the utilization of heat from the walls of the combustion chamber and the ability to reduce the dimensions of the elements of the cooling system, or to abandon it altogether. In addition, internal cooling can significantly increase the compression ratio of the spark-ignition engine, which will also have a positive effect on fuel efficiency [6].

Among the disadvantages of the Crower engine is the need to install additional equipment for storing and condensing water on the car. A serious problem is the need to prevent freezing of water in winter operating conditions. An increase in tact is also associated with the appearance of certain problems that need to be solved, or at least taken into account. The addition of additional clock cycles, all other things being equal, inevitably leads to a decrease in the liter power of the engine. Additional cycles require changes to the gas exchange system, complicating the design of the engine [6].

In addition, this design does not solve the problems with the loss of heat to the coolant in the combustion and expansion processes, which this part of the study is aimed at reducing.

3 Calculation of the required amount of water

Since such a cycle is proposed for the first time, there is no methodology for calculating it.

In order to determine the amount of water that can be supplied at the end of the combustion process, it is necessary to determine the amount of heat that we have for this. As mentioned above, this is the loss of heat to the coolant. Figure 1 shows a diagram of the Trinkler cycle in T–s coordinates. These losses are highlighted by hatching with thickened red lines.

However, we can use not all this heat, but only the heat up to the temperature at the water injection point.

The expansion in the proposed cycle takes place up to some pressure at point b, which is not known. To solve this problem, the method of successive approximations is applied. The pressure Pb is assumed to be the same as in the Trinkler cycle, the available heat is determined, the amount of water is calculated, the parameters of the end of the expansion process are determined, after which Pb is corrected and the computational and analytical study is repeated [5].

Water injection must be carried out a little later than the moment when the maximum cycle temperature is reached. The calculation was carried out for a four-stroke high-speed (1500 rpm) engine. During the calculation, several water injection moments were selected: 380, 390 and 400 degrees of rotation of the crankshaft.

To construct a cycle diagram, a thermal calculation of the working cycle indicators was carried out, after which the obtained theoretical indicator diagram was displayed in T–s coordinates. The method of thermal calculation used in this study is classical and is described in many sources [7, 8, 9].

In the course of the study, a methodology was proposed that allows theoretically determining the expected indicators of the proposed cycle, which is based on the Grinevetsky-Mazing method, while changes were made to the calculation of the expansion process in accordance with the idealization described above.

The amount of available heat q_{avail} is determined by the area of the diagram in the coordinates T–s. At the selected water injection points, the amount of available heat will be different $Q_w = q_w \cdot \mu_{gas}$. The letter «w» in the designation hereafter symbolizes water – "water".

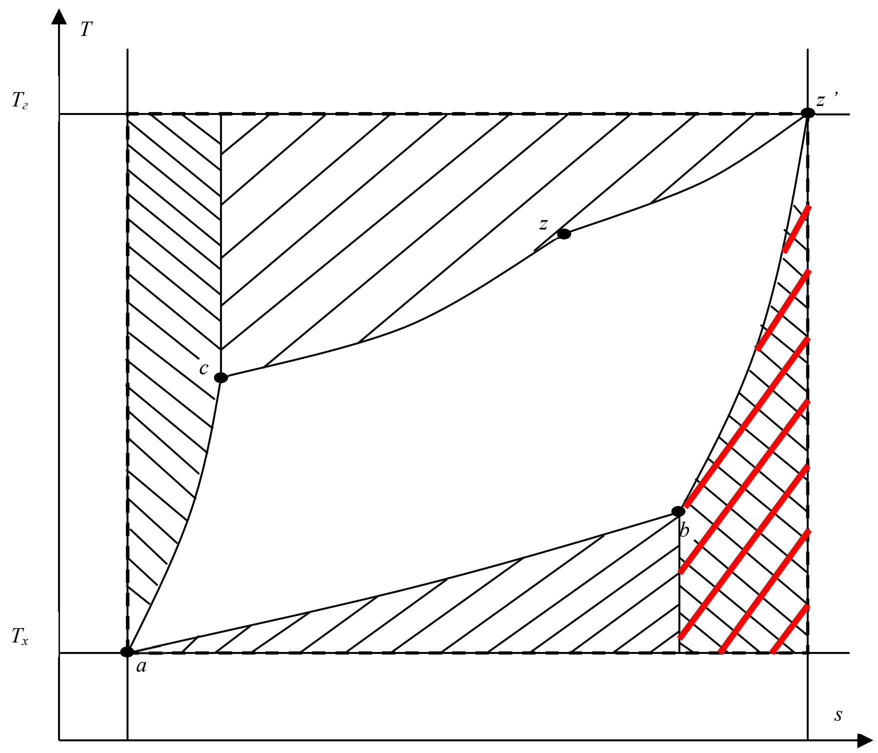

Fig.1 – Heat used for vaporization

Heat for heating water to boiling point $Q_{boil} = c \cdot m \cdot (T_{boil} - T_{water})$, kJ/ kg. The heat for evaporation of water after heating it to the boiling point $Q_{eva} = r \cdot m$, kJ.

Thus, the amount of heat

$$Q_w = Q_{boil} + Q_{eva} = c \cdot m \cdot (T_{boil} - T_{water}) + r \cdot m = m \cdot (c \cdot (T_2 - T_1) + r), \text{kJ} \tag{1}$$

Hence the amount of water

$$m = \frac{Qw}{(c \cdot (Tboil - Twater) + r)}\ kg \tag{2}$$

The difference between the proposed cycle and the Trinkler cycle lies in the course of the expansion process. At the moment of water injection, almost instantaneous vaporization will occur, the temperature will decrease, the amount of the working fluid will increase with a change in the parameters of its state. How the curve of the vaporization process will take place, we cannot assume, therefore, some idealization of the process was adopted during the calculations: it was believed that water injection and vaporization occur instantly. With this idealization, the thermodynamic system after the end of vaporization will jump into a new state with new parameters p, V, T, M, etc.

In order to determine how the pressure in the cylinder will change at the time of water injection, it is necessary to determine the volume of steam produced. As already mentioned above, steam formation at elevated pressure occurs with lower specific volumes of steam generated than at atmospheric pressure.

According to the first principle of thermodynamics, the change in internal energy during the transition to a new state is equal to the sum of the work of external forces and the amount of heat transferred to the system. Since we assumed that vaporization occurs instantly, there is no piston displacement, therefore, the work of external forces is zero. Thus, vaporization leads to: a decrease in the internal energy of the combustion products in the cylinder and an increase by the same amount of steam energy, that is, vaporization should lead to a change in internal energy: a decrease in the internal energy of the combustion products in the cylinder and an increase by the same amount of steam energy. Since no energy was supplied from the outside, the entropy did not change and the change in internal energy should lead to a decrease in the temperature of the combustion products.

The heat Qw that went to change the internal energy was determined earlier. The change in internal energy $\Delta U = Q_w = c_v \cdot \Delta T_w$, from where $\Delta T_w = Q_w / c_v$.

The temperature of the working fluid in the new state $T_w' = T_w - \Delta T_w$.

Thus, the parameters known to us in the new state are: PV [MPa], V_w [m^3], T_w [K], μ_{steam} [mole].

According to the law of partial pressures [10], the total pressure of a mixture of gases is equal to the sum of the partial pressures of the gases composing the mixture. The partial pressure of the combustion products at the selected injection points is known from the theoretical diagram constructed based on the calculation results. The partial pressure of steam can be determined by the Mendeleev-Clapeyron equation

$$Psteam = \mu \text{steam} \cdot R \cdot Tw \tag{3}$$

To determine the nature of the expansion polytrope after the end of vaporization, it is necessary to determine a new polytrope index.

The total heat capacity of a mixture of gases is the sum of the heat capacities of the gases that make up the mixture. Therefore, to the expression for determining the average molar heat capacity of the combustion products used in calculating the cycle indicators, it is necessary to add the average molar heat capacity of water vapor, multiplying both heat capacities, respectively, by the volume fractions of combustion products and water vapor.

An exploratory study was undertaken to determine the amount of water that would bring the temperature down as far as possible at the end of the expansion. In order to avoid condensation in the exhaust tract, the flue gas temperature is usually assumed to be 200 °C (473 K). The calculation was carried out in relation to the engine type Sh9.5/11, which has a cylinder volume of 0.78 liters. The search calculation showed that in a given volume of the cylinder, when water is injected at 400 degrees of rotation of the crankshaft the internal energy of the combustion products is sufficient to evaporate about 5 kg of water. The pressure in the cylinder will reach 18 MPa when the temperature of the exhaust gases at the end of the expansion process drops to 473 K. The maximum combustion pressure in a serial engine is 7.4 MPa.

That is, water injection will significantly increase the use of thermal energy of combustion products, but it will not be possible to convert it completely into mechanical within the existing crank mechanism. In addition, it will be technically very difficult to inject such an amount of water at an acceptable time. The volume of injected water in this case significantly exceeds the volume of the cylinder. Therefore, the injection of such an amount of water is possible only after the completed vaporization of the already injected portions of water, that is, the injection stretches over time. As the amount of steam increases, the pressure in the cylinder increases significantly. In order to continue injecting the remaining portion, it is necessary to exceed the pressure in the cylinder in order to obtain a satisfactory atomization of water, otherwise the vaporization process will be delayed, since large drops will warm up longer before turning into steam. It is technically difficult to provide such injection pressure. Such a value of the pressure of the working body in the cylinder will lead to an excessive increase in mechanical loads. In addition, at such pressures, it is possible for steam to break through the piston rings, wash off the oil film from the cylinder walls and then water the engine oil.

Therefore, it is advisable to determine the amount of injected water at which the pressure in the cylinder, based on the conditions of long-term strength of the parts of the crank mechanism, would not rise above the maximum combustion pressure.

According to the calculations carried out, an increase in the power of the engine operating according to the proposed cycle can be expected: when water is injected by 380 degrees of rotation of the crankshaft - an increase of 15.76%; by 390 degrees of rotation of the crankshaft – an increase of 26.23%; by 400 degrees of rotation of the crankshaft – an increase of 35.06% [11, 12].

4 Analysis of the results obtained

The results obtained show that the greatest effect is given by the later injection of water from the three proposed injection moments. This is probably caused by the following: after reaching the maximum combustion temperature, the combustion products in the cylinder have the greatest efficiency due to high temperature. As the expansion and completion of the work, the combustion products lose their efficiency, but still have significant internal energy. However, the increased volume of the cylinder does not allow it to be effectively converted into a mechanical one, since the pressure in the cylinder decreases. By supplying water at this moment, we convert the internal energy of the combustion products into the potential energy of water vapor and thereby increase the pressure in the cylinder, which leads to an increase in work. When water is supplied immediately after reaching the maximum combustion temperature, we will receive less work from the combustion products, converting it immediately into water vapor energy.

Also, the analysis of the calculation results showed that it makes no sense to strive to expand the obtained vapor-gas mixture to the lowest possible values, since, while we gain in increasing the completeness of the diagram at the beginning of expansion, we lose with a decrease in pressure at the end of expansion. Therefore, it is advisable to expand to the pressure of the end of the expansion process as in a serial diesel engine, maybe even stop at a higher pressure. After all, an increase in the degree of heat utilization has been achieved – the temperature of the combustion products has been reduced during the expansion process, that is, this heat is used in the cycle, and is not lost in the cooling system. Also, an increase in mechanical work is achieved. In the Trinkler cycle, this heat would not be used. Therefore, it is possible to expand to higher pressures in order to convert the thermal energy of the combustion products into the potential energy of steam. At the same time, the increased pressure in the cylinder will produce more mechanical work. Losses with exhaust gases even at elevated pressure at the end of the expansion process in the proposed cycle will be less than in the Trinkler cycle, since the heat expended to increase the completeness of the diagram in the Trinkler cycle is extracted with exhaust gases, and in the proposed cycle it is used to increase the cycle mechanical work [11, 12].

5 Conclusion

It should be noted that with an increase in the pressure of the end of the expansion process, the work on cleaning the cylinder from the spent working fluid will increase. Also, an increase in the pressure of the end of the expansion process will lead to an increase in the coefficient of residual gases, which means a deterioration in filling and subsequent combustion [5]. Here, perhaps, it is worth solving an optimization problem so that an increase in the cost of cleaning the cylinder does not lead to eating up the increase in the work of the expansion process. However, at this stage we will not be able to carry it out correctly, since there is no method for calculating the expansion process with vaporization embedded in it, which gives reliable results.

Therefore, it is advisable to postpone optimization for the period of the experiment. That is, it is possible to solve the optimization problem in the course of conducting experimental studies, after which it is possible to propose a calculation methodology and carry out computational optimization.

The increased pressure of the spent working fluid can be useful for a turbo-pound. At the same time, the turbine will already be closer to the steam one. The pulses in the exhaust tract will no longer be so important, so the turbine should not be radial-axial, but axial. That is, the system will not be pulsed, but isobaric.

According to the calculations carried out, an increase in the power of the engine operating according to the proposed cycle can be expected: when water is injected by 380 degrees of rotation of the crankshaft - an increase of 15.76%; by 390 degrees of rotation of the crankshaft – an increase of 26.23%; by 400 degrees of rotation of the crankshaft – an increase of 35.06% [11, 12].

Nomenclature

°RoC	degrees of rotation of the crankshaft
rpm	revolutions per minute
q_{avail}	the amount of available heat
w	water
p–v	diagram in "pressure – volume" coordinates
T–s	diagram in "temperature – entropy" coordinates
Q_w	the amount of heat for vaporization [kJ]
q_w	specific amount of heat for vaporization [kJ / mol]
μ_{gas}	number of combustion products in the cylinder [mol]
Q_{boil}	the amount of heat to heat water to boiling [kJ]
c	heat capacity of water [kJ / kg*K]
m	water weight [kg

T_{boil}	the boiling point of water at a given pressure [K]
T_{water}	the temperature of the injected water [K]
Q_{eva}	the amount of heat to evaporate water after heating it to boiling point
r	latent heat of vaporization [kJ / kg]
T_2	the boiling point of water at a given pressure [K]
T_1	the temperature of the injected water [K]
p	pressure
V	volume
T	temperature
M	molar mass
ΔU	internal energy
c_v	average molar isochoric heat capacity of combustion products
ΔT_w	temperature change of combustion products at point w1
T_w'	the temperature of the combustion products at the same point after vaporization
P_{steam}	partial steam pressure
μ_{steam}	the amount of steam formed

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