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Analysis of Regulatory Parameters of Fuel Consumption by Gasoline Engines

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Abstract: All known automotive concerns and institutes specialized in ICE problems have worked to identify the relationship between the compression ratio of ICE and its efficiency and to investigate the nature of thermodynamic processes taking place in ICE.

Numerous experiments have also been carried out to increase the compression ratio of ICE. But these works had a negative result. Building on this negative result, ICE theory adopted, as axioms, claims that the compression ratio of a gasoline engine cannot be higher than 14. That the most effective compression ratio of the diesel internal combustion engine is in the region 17-23, and at the compression ratio 40 it becomes zero. Experts and theorists were so established in the correctness of these provisions that at this stage the slightest attempt to question them caused a sharp reaction.

Keywords: the internal combustion engine (ICE), engine, system, Indicator diagram, ultra-high compression, inversely proportional, qualitative.

1. Introduction

An interesting feature of the interpretation of open thermodynamic cycles by the theory of internal combustion engines concerns the question that:

"The cycle proceeds with a constant amount of the same working fluid (gas), as a result of which both the loss of the working fluid due to its leaks through leaks and the loss of energy that occurs when a fresh charge enters the engine and exhaust gases are removed from it are excluded from consideration. In this case, the exhaust gas removal process is replaced by a fictitious process of heat removal from the working fluid to a cold source." (D.N.Vyrubov. p. 7). Here is the classic definition of a theoretical cycle with heat supply by *V=const*: the theoretical cycle of a heat engine with heat supply by isochore and heat removal by isochore and with adiabatic compression and expansion processes.

This provision of the internal combustion engine theory is fundamentally erroneous, because the removal of exhaust gases has nothing to do with the process of heat removal from the working fluid to a cold source. This is confirmed by the following:

1. According to the second law of thermodynamics, heat is allocated to a cold source, which was used by the thermodynamic system as a "payment" or "compensation" for converting heat into work. I.e., heat Q_2 allocated to a cold source cannot be used in principle to turn it into work in this system. At the same time, in an internal combustion engine, the energy of the gases being

removed can be used to get work done. This is done by continuing the expansion in the gas turbine, or the blade part of the combined engine.

This means that at least part of the energy contained in the exhaust gases being removed is not thermodynamic heat compensating for the operation of the cycle.

2. The more work is done per cycle, the more heat is spent on compensation. If the exhaust gases removed in the internal combustion engine contain the heat removed from the cold source, then as the amount of work performed by the cycle increases, the exhaust gas temperature should increase, not decrease.

The amount of work performed by the cycle can be increased by increasing the compression ratio. But as the compression ratio increases, the temperature of the exhaust gases being removed decreases rather than increases. So, in a gasoline internal combustion engine (T_a =350^{0}K, k =1.35) with a compression ratio of ε=5, the temperature T_2 of the completion of the expansion process would be T_b = 1769^{0}K. In a similar engine with a compression ratio of ε.=50, the temperature Tb of the completion of the expansion process at the same values Ta and k will be T_b = 986^{0}K. It follows from the above that the value of the thermal efficiency of the system can be increased by reducing the amount of internal energy of the exhaust gases.

3. In a hypothetical internal combustion engine with a compression ratio of ε=5, operating according to the Carnot cycle, the temperature of completion of the adiabatic expansion process (at a = 1) would be T_2 = 17690K. The process of heat removal Q_2 to a cold source begins and ends at the specified temperature (*T=const*). That is, the internal energy of the working medium at temperature T_2 has nothing to do with heat removal in the amount of Q_2, even its microscopic part has nothing to do with. In the form of compensation heat, the system gives off only the energy that is communicated to the working body by the work of isothermal compression. i.e., in a hypothetical engine operating on a circular closed Carnot cycle, the heat of compensation Q_2 is the work of isothermal compression (minus the energy that is spent on increasing pressure).

Thermodynamics explains this situation as follows:

In isothermal processes, work is performed not at the expense of a decrease in internal energy U (as is the case with adiabatic processes), but at the expense of free energy *F*. The associated energy TS (in this case, the internal energy of the working fluid at a temperature of T_2) has nothing to do with either the performance of work or the removal of heat to a cold source.

If I may say so, the bound energy T_C is a "thermodynamic platform" on which the free energy F is converted into work and heat compensation.

But this question has another feature:

"The value of Tc increases with the expansion of temperature limits, and it is easy to show that this increase is more significant with a decrease in T_2 than with an increase in T_1".

This means the following:

1. The thermodynamic potential of any working fluid used in thermal machines is limited, i.e. it is not possible to transfer to the working fluid an amount of heat greater than Q.

2. The greater the amount of bound energy T_C the working fluid contains, the less free energy F (respectively, heat Q) can be communicated to it.

3. Ideal closed cycles are not efficient, because high values of the amount of bound energy make their thermodynamic potential low.

The removal of a working fluid with bound energy T_C from the thermodynamic system turns the cycle into an open one. This allows, due to the small amount of bound energy contained in the fresh (updated) working body, to significantly increase the thermodynamic potential of the system and, accordingly, increase the amount of cycle work. Moreover, an increase in the thermodynamic potential of an ideal cycle in no way affects its efficiency.

4. According to the ratio 1/ek-1, with an infinitely large compression ratio $\varepsilon \rightarrow \infty$, the temperature of the exhaust gases will be equal to the temperature of the beginning of compression, i.e. Tb→≈Ta. This means that the gases removed from the cylinder do not contain even a negligible part of the heat that goes to compensate for the cold source.

Thus, the removal of the working fluid from the thermodynamic system theoretically has nothing to do with the removal of heat to a cold source.

This means that in theoretical cycles, the internal energy of the removed working fluid in the temperature difference T_b-T_a refers to free energy and is a source of increasing the efficiency of the internal combustion engine.

The ratio $\frac{1}{e^{k-1}}$ shows that as the compression ratio increases, the temperature Tb of the end of the adiabatic expansion decreases. The expression e^{k-1} shows that as the compression ratio increases, the temperature of the Tc end of the adiabatic compression increases. The expression e_k shows that as the compression ratio increases, the pressure of the compression end Pc also increases. This allows us to draw the following conclusions:

1. In actual cycles, the concepts of "adiabatic compression" and "adiabatic expansion" are conditional, since the working fluid exchanges heat with the inner surface of the cylinder. The intensity of this heat transfer will increase as the compression ratio increases according to the expression ek-1. Therefore, in actual cycles, heat losses in the cylinder walls occurring during the period of temperature change from the value of Ta to the value of Tc and from the value of Tc to the value of Tb (Ta→Tc→Tb) are heat losses to compensate. Heat losses in the walls, which are caused by the difference in the temperature range Tc→Tz→Ts are not heat losses to compensate for and can be turned into work.

2. An increase in the compression ratio entails an increase in compression pressure, which leads to an increase in mechanical losses. Mechanical (thermal) losses, which are caused by a change in the pressure of the working fluid in the range Ra→Pc→Pb, are also a loss of heat for compensation. The mechanical losses caused by a change in the pressure of the working fluid in the pressure range Pc→Pz→Pc are not heat losses to compensate and they can also be turned into work.

Thus, the above allows us to conclude:

In actual cycles, the amount of heat loss (compensation heat) will have an ideal value if the maximum cycle temperature is equal to the temperature of the end of compression (T_z=T_c), and the maximum cycle pressure is equal to the pressure of the end of compression (P_z=P_c).

Now let's try from the standpoint of workflow theory to solve the question of how the imaginary operation of an ideal engine (and, accordingly, the calculation formula) differs from the operation of a real engine.

"...the method of energy transfer associated with changing external parameters is called work."(I.P. Bazarov, p. 23).

The method of energy transfer, without changing external parameters, is called heat input or removal, and the process itself is called heat exchange. I.e., heat transfer not associated with a change in the external parameters of the working fluid is the process of changing its internal energy without performing work.

In the Carnot cycle, all the work of a hypothetical internal combustion engine transferred to the consumer is performed according to the isotherm according to the rule *T=const*. In the internal combustion engine cycle with *P=const*, part of the work is performed using an isobar when heat is supplied to the working fluid at constant pressure. In the internal combustion engine cycle with *V=const*, when heat is supplied to the working body via an isochore, i.e., at a constant volume, no work is performed!

When considering ideal cycles, this factor does not matter at all. But in actual cycles, it becomes the main factor determining both efficiency and efficiency. Since theoretical cycles are prototypes of actual cycles, the influence of this factor should also be analyzed at the level of theoretical cycles.

In the theoretical cycle with $V=const$, the heat supply process is completely separated from the work completion process. I.e., heat input and work completion are two consecutive stages following each other. At first, only heat is introduced. Then only the work is done. The question of the efficiency of converting the internal energy of the working medium into work can arise only when the thermodynamic system proceeds to change the internal energy of the working medium through a change in its volume. If the volume has not changed, then the work has not been completed. If it has changed, then it is perfect.

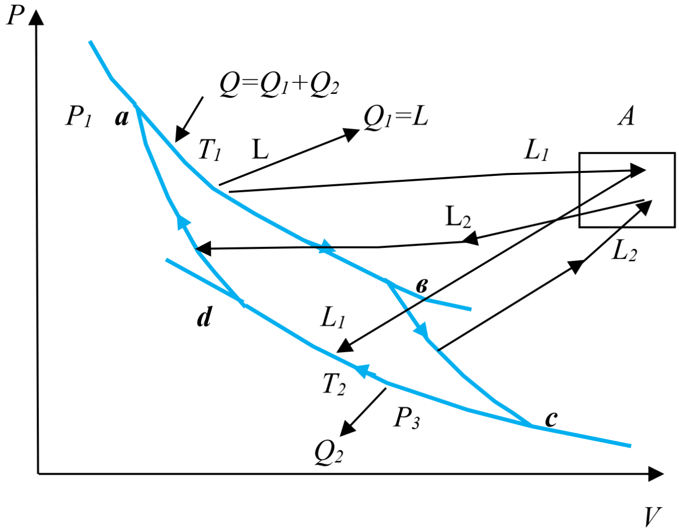

Fig.1

1. Operating conditions of the hypothetical Carnot engine: In order to activate, the thermodynamic Carnot system must be supplied with energy in the form of compression work, which creates pressure P_1 and temperature T_1. Next, the system performs work due to pressure. When performing work, the pressure P_1 (due to expansion) and the temperature T_1 (due to cooling) will tend to decrease. In a closed expanding vessel, pressure is created by temperature. The introduced heat completely compensates for the decrease in temperature and thus partially compensates for the decrease in pressure. Therefore, if we consider the process of introducing heat and performing work not as a whole, but a specific mechanism for using heat at each point of the process, it turns out that the work performed by the system on the isothermal section ***ab*** consists of two components: 1. The compression work L_1 introduced into the system when compressing the working fluid on the ***cd*** sections. 2. The work L obtained as a result of converting the input heat in the amount of Q_1 into pressure (through temperature). In total, the two types of these works are equivalent to the input heat of Q. As for the input heat Q, then: 1. Part of the input heat Q_1 is spent on performing the work. 2. The second part of the heat Q_2 is irrevocably (which is fundamentally unavoidable) consumed to compensate for the temperature of the working fluid. If the isothermal expansion continues until the pressures $P_b = P_a$ equalizes, at the end of the process, the internal energy of the working medium will consist only of temperature T_1 enclosed in volume V_{60} at pressure $P_{60} = P_a$ **and the working medium is no longer capable** of performing work. In order to repeat the cycle, it is necessary to update the state of the working fluid, i.e. remove heat from it until the temperature T_2 is reached. The heat withdrawn Q_2 will be equal to the amount of heat consumed to compensate for the temperature of the working fluid when performing work.

Ideal open loops are prototypes of theoretical loops, and theoretical loops are prototypes of real cycles. The transition from an ideal sample to a prototype always means a deterioration in efficiency and effectiveness. In order to correctly reflect the result of the transition from an ideal cycle to a theoretical one, as shown by the formula for calculating the thermal efficiency of a cycle with a mixed heat supply, the calculation should be based on 2 indicators characterizing the effect of changes in the specific area of the heat sink on efficiency and efficiency. But the theory of internal combustion engines was limited only to indicators of an increase in volume and pressure, ignoring the influence of temperature on an increase in the area of heat dissipation. For example, in an ideal cycle with heat supply at *V=const*, the pre-expansion index is *p=1*. But this does not mean that the specific area of the heat sink of the cycle does not change. An increase in temperature from the value *Tc* to the value *Tz* at *V=const* means that the specific area of heat removal due to temperature changes has increased to the same extent as it would have occurred with an increase in the index p or λ. Accordingly, when switching from an ideal to a theoretical cycle with *V=const*, the thermal efficiency calculation formula should contain an indicator of a preliminary temperature increase during the heat supply period *Tz /Ts*.).

If cycles with mixed heat supply and with heat supply at P = const are represented as ideal cycles, then from the formulas for calculating thermal efficiency The indicators of the preliminary increase in pressure and volume must be removed. In this case, the formulas for calculating the thermal efficiency of these ideal cycles will look like:

$t = 1 - 1/e^k - 1$.

In theoretical cycles of internal combustion engines (with external and internal mixing) with a low and medium compression ratio with a mixed heat supply (i.e., if there is a section with signs of an isobaric process P =const in the cycle diagram), the formulas for calculating thermal efficiency should contain indicators of a preliminary increase in volume and pressure. At the same time: a) the formula of the theoretical calculation cycle of a diesel engine should look like:

$t = 1 - (\frac{\lambda\rho^{k-1}}{\varepsilon^{k-1}}[(\lambda - 1) + k\lambda(p - 1)],$

(Sabate-Trinkler formula); b) the formula of the theoretical calculation cycle of Ibadullaev for a gasoline engine should look like:

$t = 1 - \frac{\lambda\rho^{k-1}}{\varepsilon^{k-1}}[k(p - 1) + \rho(\lambda - 1)].$

In theoretical cycles of internal combustion engines (with external and internal mixing) with high and ultrahigh compression ratios and with heat supply through the isothermal process T =const, the factors influencing the value of thermal efficiency will be the indicators of volume increase (indicator p) and temperature increase (indicator t). The indicator of the preliminary increase in pressure in such cycles is $\lambda=1$. Therefore, in the second part of the formula for calculating the thermal efficiency of such a cycle, along with the indicator of a preliminary increase in volume, an indicator of a preliminary increase in temperature should be present. Provided that the temperature of the isothermal section of the diagram of the actual cycle is absolutely equal to the temperature of the Vehicle, the indicator of the preliminary temperature increase should be taken *t=1*. For this cycle, the indicator should be output by the ratio *t= Tc/Tc*.

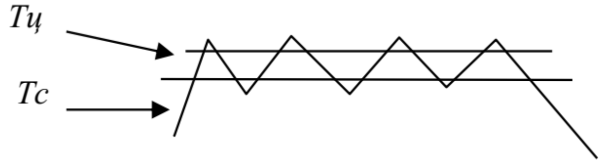

Fig. 3

To base the cycle, you need to take a reference engine. Hardware states of the reference engine: ε=10, Pa - 1 kg/sm^2, k = 1.36, Ta = 50^{0}C, predicted gasoline AI-98, operating mode - full load (100% throttle), generator frequency - 800 rpm.

When the reference engine operates at the specified frequency Pc=25 kg/sm^2, Tc=580^{0}C, UOS θ=2-3^0 before TDC. The starting point of heat release is 00 TDC. Pz=56 kg/sm^2, which corresponds to approximately 7-8^0 PKV after TDC, Tz=2400^{0}C, 16-180 PKV after TDC. When operating at this frequency, the processes of formation of a flame source, development of preliminary flame mixtures, propagation of the flame front, ignition, increase in the volume of the combustion chamber, and pressure increase will be completely balanced and brought to the equilibrium point of the processes. Reducing the speed without throttling will cause detonation, increasing the speed without adjusting the speed limit will cause a sharp drop in efficiency. To prevent this from happening, it is necessary to increase the SOP. At a permissible signal frequency of 3000 rpm, the SPD indicator will be approximately θ = 35^0 BTDC, and the engine diagram will look like the one in the above figure. 58. That is processes will occur under equilibrium conditions. A deviation of the SOP by several degrees, or a change in the octane number by several units, is practically not observed.

2. Calculation engine. Hardware conditions: ε=20, P_a- from 0.425 to 1 kg/sm^2, k=1.36, Ta=50^{0}C, AI-98 gasoline used, operating mode-full load (42.5-100% throttle), minimum rotation speed 800 rpm, maximum 6000 rpm.

When the engine is running at 800 rpm with full load, we must withstand the conditions of flame formation, front propagation, etc. of the reference engine. Therefore, we limit the filling by throttling to 42.5%, Pa =0.425, (in VAZ-2110 with ε = 20 actually reaches 0.6) Pc=25 kg/sm^2, Tc = 5800C, UOZ θ =2-30 to TDC. The starting point of heat release is 0^0 TDC. Pz= 56 kg/sm^2, reached approximately 8-90 PCV after TDC, Tz =2200^{0}C, reached approximately 18-19^0 PCV after TDC. When operating at this frequency, the processes of flame formation, the development of pre-flame reactions in the mixture, the propagation of the flame front, combustion, an increase in the volume of the combustion chamber, pressure build-up, as in the reference engine, will be fully balanced and brought to the point of equilibrium of the processes. Reducing the RPM without reducing the Pa will cause detonation, increasing the RPM without increasing the Pa or UO will cause a sharp drop in efficiency. When the speed is increased to 1600, the passage time of these processes is halved. Accordingly, we have twice as long a delay time for self-ignition at our disposal. Therefore, it is possible, as the speed increases, to proportionally increase the Ra to a value of 1 kg/sm^2. Up to this number of revolutions, the processes of flame formation, the development of pre-flame reactions in the mixture, the propagation of the flame front, combustion, an increase in the volume of the combustion chamber, and pressure build-up will be fully balanced and brought to a point of equilibrium of the processes. At the same time, P_c=59 kg/sm^2, Tc=670^{0}C, UOZ θ= 5-60 to TDC. The starting point of heat release is 00 TDC. Pz>80 kg/sm^2, is reached at about 200 PCV after TDC, Tz=2600^{0}C, is reached at 300 PCV after TDC. At a rotation speed of 1800 rpm and above, the delay time of self-ignition begins to lag behind the speed of the processes, therefore, we increase the efficiency. At, say, a rotational speed of 3000 rpm, the optimal speed will be approximately θ= 150 to TDC and the engine diagram will look like in the above Figure 58-A. I.e., the processes will occur in a strongly narrowed equilibrium zone. A deviation of 2-3 degrees, or a change in the octane number by several units, will become strongly noticeable either in the form of detonation or in the form of a drop in efficiency.

CONCLUSION

What has been said in this work can be summarized as follows:

1. The author's ideas about the possibility of increasing the compression ratio of internal combustion engines to ultra-high values at this stage, as one of the respected professors put it, terrify theorists and practical designers. The basis of this approach and inadequate perception of facts is not the fantastic nature of ideas, but the conservatism of thinking. An idea after its implementation cannot be considered fantastic. The history of the development of science, including the theory of internal combustion engines, is in fact the history of the struggle and overcoming of such "horrors". For those who have doubts, the author recommends reading the

life story of R. Diesel. His ideas caused no less “horror” among famous theorists and practitioners of that time.

2. In recent decades, it has become abundantly clear that the strength potential of our environment is not infinite. Energy reserves are limited. Their thoughtless extraction and ineffective use, poisoning of the environment with greenhouse and toxic gases is leading humanity to energy and environmental disasters. This will affect everyone: not only those who extract and consume petroleum products, but also those who walk and believe that they are not participating in the poisoning of the environment. There is no alternative to reducing the consumption of petroleum products through their more efficient use.

3. Many theorists whom the author asked to read his works and express their opinions refused to do so, not because they lacked healthy curiosity or the ability to think, but because they were preoccupied with solving current problems. The main question for each of us is how to feed our family today, and not think about the global problems of tomorrow. The consumer society has turned theorists into simple artisans concerned with daily earnings.

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