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Compression Pressure Measurements, Starting and Stopping of an Ultra-High Compression Ratio Gasoline Engine

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Abstract: As the As the automobile industry develops and the population increases, the demand for it increases. Therefore, reducing the fuel problem and environmental pollution remains one of the pressing issues. All known automobile concerns and institutions specializing in ICE problems are trying to solve them by means of a turbo device to improve the ICE. This makes the design structure of the car engine complex. It is necessary to simplify the car engine for long-term high-quality operation and increased fuel efficiency. The solution to these problems can be found by increasing the compression ratio of the engine. Currently, some research is being conducted to increase the compression ratio of internal combustion engines.

Keywords: the compression, the plugs, nozzles, Indicator diagram, or fuel-air mixture, cycle, engines.

1. Introduction

One of the important components of checking the serviceability of the engine is the measurement of compression pressure. The following rules must be observed: the engine must be warmed up, the throttle valve must be fully open, the plugs must be unscrewed. The scroll speed of the starter must be at least 300-min. At the same time, the values of the compression values for a serviceable engine are indicated in the operating manuals. For gasoline engines with a compression ratio of 8-10.5, the compression should be 12-15 kg/sm^2, with a compression ratio of 11-12.5, the compression pressure should be 15-17 kg/sm^2. For diesel engines with a compression ratio of up to 20, the compression pressure should be 22-26 kg/sm^2, for engines with a compression ratio of 20-23 it should be 26-32 kg/sm^2.

In a serviceable VAZ-2110 engine with a compression ratio of 9.9, the compression should be 14-15 kg/sm^2. The theoretical calculation of the compression end pressure is performed according to the formula Pc = Pa*ε 1.35. According to this formula, the compression end pressure for engines with compression ratios from 10 to 25 at Pa = 0.9 bar (1 kg/sm^2) should be:

Table 1

adia=1.35 pa= 0.9 bar		
eps	Pc,bar	Tc,K.
10.00	20.148	738.778
12.50	27.231	798.790
15.00	34.831	851.424
17.50	42.889	898.622

20.00	51.361	941.617
22.50	60.213	981.246
25.00	69.416	1018.106

(compiled by N.A. Ivashchenko.)

That is, as can be seen from the table, in the VAZ-2110 engine with a compression ratio of 9.9, the compression should not be 14-15 kg/sm^2 (12.6-13.5 bar), but 22 kg/sm^2, or 20 bar. In a diesel engine with a compression ratio of 23, the compression end pressure should not be 26-32 kg/sm^2 (23.4-28.8 bar), but 77 kg/sm^2, or 69.4 bar.

Based on the above table, in the author's engine, which was demonstrated to the participants of the International Conference and showed a compression end pressure of 40.5 kg /cm2 (36.45 bar), the compression ratio is not 22, but 15.5.

That is, the discrepancies between tabular and calculated data are very large. The reason is as follows:

The pressure of the compression end depends on how much air is compressed in the cylinder. The amount of air in the cylinder when the starter is scrolled depends on the moment when the intake valve is closed. During the measurement, it turned out that the intake valve of the VAZ-2110 engine (serial camshaft) closes at the moment when the piston has moved 23 mm from the LDC (lower dead center) towards the TDC. The full stroke of the piston is 74.8 mm. Accordingly, 69% of the amount of air that the cylinder would hold if the piston were located in the LDC remains in the engine cylinder when the starter is scrolled.

But this is theoretical. In fact, there are gaps and gaps between the walls of the cylinder and the rings, through which part of the air leaks. According to the training data, in a serviceable gasoline engine, the leakage rate during compression measurement is 2%. If the scroll speed is equal, the higher the pressure of the compression end, the greater the leakage rate. Therefore, in an engine with a compression ratio of 22, the leakage rate is approximately 5%.

Accordingly, in order for the calculated data to coincide with the results of instrumental measurements, the following table must be used for this engine:

Table 2.

adia=1.35 pa= 0.6 bar		
eps	Pc,bar	Tc,K
10.00	13.432	492,52
12.50	18.154	532,47
15.00	23.221	567,56
17.50	28.593	599,02
20.00	34.241	627,68
22.50	40.142	654,1
25.00	46.278	678,67

P_a is the value of atmospheric pressure. If the compression pressure is measured in kg/cm2, Ra has a value of 1 (units), and if in bars, 0.9. If you use the table below, it turns out that the engine cylinder is completely filled with air, but it is in a dilution state equal to 69% of the ambient atmospheric pressure. Taking into account the magnitude of the leaks, the vacuum will be 64% of the atmospheric one. I.e., reality is actually being replaced by a fictional circumstance.

In fact, the cylinder is filled to 64% of its volume and at the same time the air pressure corresponds to the ambient pressure. Such a situation in the works of the author is designated as the actual compression ratio.

The actual compression ratio of the VAZ-2110 engine with a compression ratio of 22 when measuring compression is 15.5. The actual compression ratio of the serial VAZ-2110 with a compression ratio of 9.9 is 6.8.

It turned out that the results of compression pressure measurements do not depend on the position of the throttle valve. Both with the throttle fully closed and with the throttle open, 40 kg/sm^2 was obtained.

If measurements were taken with the throttle closed, the result had little effect on whether the spark plugs were unscrewed or not. With the candles screwed in we got 38-40 kg/sm^2, with the candles unscrewed 40 kg/sm^2.

Reason: When air is sucked into the first cylinder, the pressure in the intake manifold drops and a vacuum is created in it. In the next cylinder, at this time, the exhaust stroke ends and the valves enter the overlap sector. Because of this, for a certain period of time the intake manifold communicates freely with the exhaust manifold through this cylinder. An intensive suction of combustion products from the exhaust pipe into the intake pipe occurs and the pressure in the intake manifold is restored to ambient pressure. In this case, the amount of air sucked in from the exhaust manifold depends on the crankshaft speed.

This factor had a negative impact when starting a warm engine. The first experiments showed that due to the suction of exhaust gases from the exhaust manifold, the pressure Pa in the cylinders exceeds the permissible limits and, upon compression, a flash of the combustible mixture occurs due to self-ignition. The first method used to combat this phenomenon was a small car vacuum cleaner with a one-way valve. The vacuum cleaner was connected to the intake manifold through a valve and worked like a vacuum pump. Before starting the engine, the vacuum cleaner was turned on for 1-2 seconds. It sucked air from the intake manifold and created a slight vacuum there. Then the starter turned on. When stopping the engine, if you do not turn off the fuel supply after turning off the ignition, the following picture will be observed:

“Increasing the compression ratio in modern car engines has led to the fact that it is often difficult to stop a normally warmed-up engine by turning off the ignition; it continues to idle for quite a long time with interruptions and shaking.

Specially conducted studies have shown that after turning off the ignition, the flashes immediately stop, as a result of which the crankshaft rotation speed quickly decreases, but not to zero, but only to a value at which the duration of the ignition delay is shorter than the residence time of the mixture heated by compression in the cylinder. As a result, spontaneous flashes occur, which leads to a slight increase in the rotation speed to a value at which the flashes stop again, the rotation speed decreases again, flashes occur again, etc. (A.N. Voinov, p. 179).

But if, in the case given from the textbook, without turning off the fuel supply, you limit the flow of air into the cylinder, the engine will stop. Those. The above example is another confirmation that a certain balance must be maintained between the volume of the combustion chamber, the compression time and the amount of working fluid, which will ensure the reliability of both starting and stopping the engine when the ignition is turned off.

Subsequently, as the compression ratio increased, it was necessary to increase the scroll speed at startup. To do this, it was necessary to reduce the diameter of the flywheel and increase the diameter of the starter gear, while simultaneously increasing the starter power. If earlier, with Pc = 26-27 kg/sm^2, the starter had a power of 0.9 kW, and the ratio of the teeth of the flywheel and the starter gear was 129/11=11.73, then with Pc = 38-40 kg/sm^2, a starter with a power of 2.2 kW with a tooth ratio of 124/15 = 8.27. Those. The rotation speed at startup was increased by 1.42 times, which was quite enough to start the engine in any condition without using a vacuum cleaner. But even in this case, when starting a hot engine, it was necessary to start the starter 1-2 seconds before fuel injection.

Starting a gasoline engine with ε=20 in cold weather at temperatures down to –25^{0}C (there were no lower temperatures during the experiments in Moscow) was carried out in the usual way. Those. The starter turns on simultaneously with the supply of fuel and ignition. The engine starts immediately after turning the ignition key, i.e. almost instantly.

Installing separate throttle valves for each cylinder changed the picture dramatically. With the throttle valves closed, 20-22 kg/sm^2 was obtained, and with the throttle valves open, 40 kg/sm^2.

With a further increase in the compression ratio, the suction of exhaust gases from the exhaust tract will begin to affect the engine's idle speed. Therefore, at compression ratios above 22, each cylinder must be equipped with a separate throttle valve.

The issue of eliminating the suction of combustion products and reducing pump stroke losses will be fundamentally resolved by switching to regulating air flow by the intake and exhaust valves by removing its excess into the intake and exhaust manifolds. In this case, when starting and warming up a cold engine, the amount of air in the cylinder is regulated by a second intake valve with a closing angle adjustable on the compression stroke. The excess air charge will be pushed into the intake manifold, and fuel will be injected directly into the cylinder after the valve closes. After the engine warms up, the amount of air in the cylinder will be controlled by the second exhaust valve, pushing excess air into the exhaust manifold.

2. Ideal, theoretical and actual cycles.

The thermodynamic cycle with a mixed supply of heat (Sabate-Trinkler), known to the theory, has, according to A.N. Voinov, the following disadvantages and advantages:

1. The engine compression ratio is limited to $\varepsilon<25$ due to an increase in the maximum permissible operating severity.

“Another significant drawback of diesel engines is the harshness and noise of their operation, associated with high rates of pressure increase at the beginning of the main combustion phase. In diesel engines with open chambers and jet mixture formation, the maximum values of $\frac{dp}{d\varphi}$ reach 1.2–1.5 MPa/ versus 0.15–0.2 MPa/ in gasoline engines with spark ignition.

Although diesel engines of this type achieve the lowest specific fuel consumption [up to $165g/(h_p - h)$], and in marine diesel engines with large cylinders - up to 150g/(hp-h), but the noise they create exceeds permissible standards. In this regard, the efforts of a large number of researchers and designers have long been aimed at finding ways to organize the processes of mixture formation and combustion that would reduce the values of p_z and $dp/d\varphi$.”

“To achieve high combustion completeness in a high-speed diesel engine at low values of the excess air coefficient, it is necessary to organize the injection in such a way that would ensure, perhaps, a more uniform distribution of fuel throughout the entire volume of the air charge, for example, when injecting through nozzles with a large number nozzle holes. But at the same time, the simultaneous occurrence of a significant number of initial sources of ignition and its rapid development are inevitable, which leads to a rapid increase in pressure and its high maximum values. If we strive to obtain moderate rates of pressure increase, it is difficult to ensure rapid completion of combustion and avoid prolonged afterburning during the expansion period.”

2. When a diesel engine operates with a compression ratio of up to 25 "at idle speed at low speed, the atomization fineness usually deteriorates significantly and at the same time the temperature of the walls of the combustion chamber decreases, which is accompanied by an increase in the duration of the delays. This leads to the fact that a significant proportion of the injected fuel droplets manage to completely evaporate by the time of ignition. With a uniform distribution of fuel vapor in the combustion chamber, a homogeneous mixture of a composition is obtained that already exceeds the limits of flammability, which can lead to the emission of products of incomplete oxidation of fuel with an unpleasant odor from the engine. Some of these products are toxic.

An effective way to combat this drawback is to reduce the amount of air entering the cylinders at low loads and at idle by using throttling.” (Emphasis added);

3. “One of the main disadvantages of diesel engines associated with the combustion process is the appearance of black smoke at the exhaust under heavy loads in the case of an increase in the cyclic fuel supply or, what is the same, a decrease in the total excess air coefficient below a certain certain limits. This is explained by the fact that during the diffusion combustion of inhomogeneous mixtures in zones of local over-enrichment, the formation of particles of solid carbon (soot) occurs at high temperatures of burnt gases in adjacent zones, where local values of α are close to unity.”

4. “Although the pi values in diesel engines can be significantly increased (up to 2.0 MPa or more) by using supercharging, the engine design becomes heavier due to very high maximum pi values.”

Advantages:

1. “The effect on such explosive ignition of an increase in the compression ratio, intake pressure and thermal state of the engine is directly opposite to the influence of the same factors on detonation in engines with pre-mixing. “Everything that contributes to the occurrence of detonation in light fuel engines with spark ignition, in diesel engines, on the contrary, eliminates the occurrence of shock waves due to a reduction in ignition delays and, accordingly, a reduction in the amount of fuel supplied to the cylinder before it ignites.”

2. “The fact that as the amount of injected fuel decreases, an increasing proportion of it burns in the chamber volume away from the walls, contributes to a decrease in heat transfer. Equally important is the reduction in the average heat capacity of combustion products, which increases the efficiency of using the released heat to perform useful work.

All this leads to the fact that, unlike light fuel engines, the indicator efficiency of which decreases at low load conditions, in diesel engines the values of ή i increase with decreasing load. Accordingly, fuel economy in the operating conditions of cars with diesel engines compared to the fuel consumption of cars with gasoline engines reaches an average of 40%, while the differences in the minimum specific fuel consumption according to load characteristics are much smaller - only 20-25%.”.

Thus, the disadvantages in organizing the operation of diesel engines boil down to the following:

1. To ensure high combustion completeness at low values of the excess air coefficient, it is necessary to ensure fine atomization and early injection. But this leads to a rapid increase in pressure.

2. When operating at low speeds, a homogeneous mixture of a composition is formed that goes beyond the limits of flammability.

According to A.N. Voinov, an effective means of combating these shortcomings is to reduce the amount of air entering the cylinders at low load and idle modes using throttling.

2. An increase in the average pressure of the cycle leads to an increase in the maximum pressure and a heavier engine structure.

The listed disadvantages indicate that the causes of detonation in gasoline engines and high rates of pressure rise in diesel engines have the same root: this is an insufficient compression ratio, due to which the main phase of heat release has to be organized in the zone of small changes in the volume of the working fluid. Experiments with gasoline engines with high compression ratios show that the identified possibilities for regulating the rate of pressure increase by increasing the compression ratio and shifting the main period of heat release in phase can also be used when organizing the combustion process in diesel engines, which will eliminate the listed disadvantages. What is important here is that increasing the compression ratio of diesel engines to ultra-high values (up to 51) will lead to a reduction in weight and size indicators and an increase in the service life of such engines compared to engines with conventional compression ratios.

3. ICE with external mixture formation.

When the compression ratio increases to high and ultra-high values, one fundamental difference remains between internal combustion engines - the method of preparing the mixture for combustion. Preliminary preparation (external mixture formation) or direct preparation (internal mixture formation). For engines with ultra-high compression ratios (up to 51) with internal mixture formation, the ignition method will not matter. In engines with ultra-high compression ratios, it is necessary to use multi-stage direct injection.

In engines with external mixture formation, from the moment the spark is supplied, the possibility of influencing the combustion process is eliminated. For this reason, the compression ratio in them can be high (up to 30), but not extremely high. Such an engine will operate according to the rules set out in the article "Gasoline engine with an ultra-high compression ratio" with the following clarification: at low speeds with throttling at a flow rate of approximately up to 35% of the combustible mixture - according to the Beau De Roche cycle. At low and high speeds at a flow rate of approximately 35% to 100% of the combustible mixture with throttling (when the filling is limited, the maximum permissible amount of mixture for a given speed is taken as 100% of the flow rate) and when operating at an external speed characteristic (without throttling) - according to Ibadullaev's cycle.

A gasoline engine with internal mixture formation with an ultra-high compression ratio (up to 51) at low and high speeds with air throttling will operate according to the Ibadullaev cycle. At medium and high speeds on the external speed characteristic without air throttling according to the Imam cycle.

Any of the cycles proposed for consideration can be presented in the form of an ideal, theoretical and actual.

4. Ibadullaev cycle.

Based on the thermodynamic processes of an ideal gas, an open cycle with heat supplied to the working fluid first along an isobar (at constant pressure $P=const$), and then along an isochore (at constant volume $V=const$), with heat removal through compression and expansion and renewal of the working fluid according isochore.

The cycle is completed as follows: in the section ab of the diagram, polytropic compression of the working fluid occurs with heat removal. The pressure of the working fluid rises to value P1. At point b, part of the heat is supplied to the working fluid. In section bb', the pressure of the working fluid remains equal to the value P_1. In section b'c of the diagram, at a constant volume, the second part of the heat is supplied. The pressure of the working fluid rises to the value P_2. In the SD section, polytropic expansion of the working fluid occurs with heat removal. In the section of the diagram, the working fluid is updated.

Ibadullaev's theoretical cycle is characterized by the indicators of a preliminary increase in volume in the isobaric section and a preliminary increase in pressure in the isochoric section, which affect the value of thermal efficiency.

Diagram of Ibadullaev's thermodynamic cycle

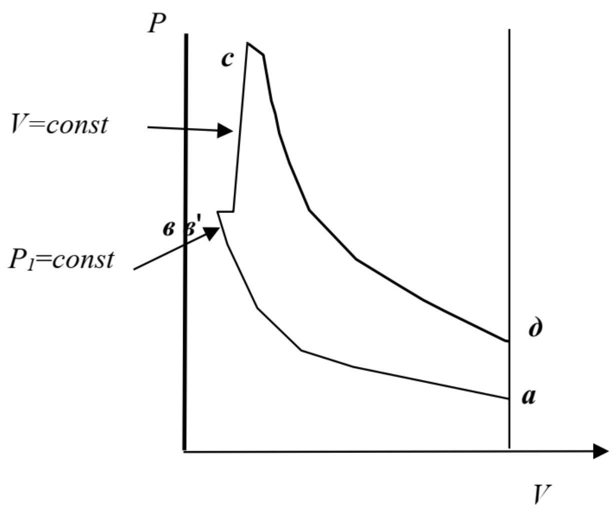

Fig. 1

Ibadullaev's actual cycle formula: Mixed cycle of operation of a gasoline internal combustion engine with a high (up to 30) compression ratio, in which on the first stroke a fresh charge of air (or fuel-air mixture) is injected, on the second stroke this charge is compressed to pressure P1, on the third On the fourth stroke, combustion products expand; on the fourth stroke, combustion products are removed from the cylinder, characterized in that when operating at the external speed characteristic, depending on the rotation speed, on the intake stroke, the filling of the cylinder with the combustible mixture (air) is limited to ensure for the period of flame propagation along the front of constant pressure P_1 by synchronizing the process of increasing the volume of the combustion chamber and the pressure of the mixture. In this case, the starting point of heat release is at TDC (top dead center), during expansion, part of the heat is released at a constant pressure P_1, and the rest of the heat is released at a pressure increasing to the value P_z. In this case, the compression process is conditionally adiabatic, the process of preliminary expansion is isobaric, and the process of preliminary increase in pressure is conditionally isochoric. Calculation of the thermal efficiency of Ibadullaev's theoretical cycle should be made according to the formula:

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

Diagram of Ibadullaev's theoretical cycle.

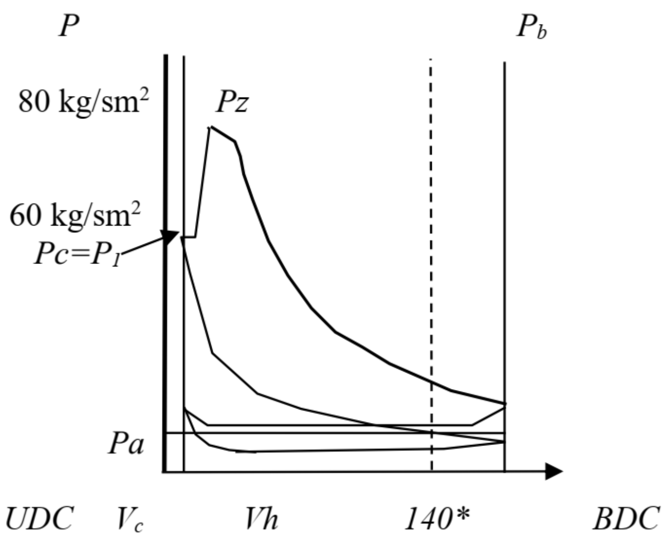

Fig. 2

CONCLUSION

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

1. 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.

2. In modern high-speed diesel engines, the maximum cycle pressures are 220-260 kg/sm^2. Increasing the compression ratio to ultra-high values will allow reducing the maximum values of pressure (up to 200-210 kg/sm^2) and temperatures (up to 1500-1700^{0}K, instead of 2200-2600^{0}K). Those. For mass production of such engines, neither special materials nor special technologies are needed. Modern industry has all the necessary conditions for this.

3. The basis for the construction of modern gasoline and diesel engines is the outdated dogma of the internal combustion engine theory that in order to ensure maximum efficiency and efficiency, it is necessary to ensure the flow of the main phase of heat generation in the TDC zone. This leads to the fact that the limits of increasing the compression ratio in gasoline engines are limited by detonation, and in diesel engines by the dynamism factor. At the same time, in the last decade, there has been a tendency in the practice of engine building to abandon this dogma. The use of direct and multistage injection in engines shows that the main phase of heat generation should occur during expansion.

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