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ANALYSIS OF EXISTING GAS EXCHANGE PROCESSES IN INTERNAL COMBUSTION ENGINES

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Abstract
This article presents the development of methodological recommendations and the application of their results in production in order to improve the environmental safety of motor vehicle traffic .

Keywords : Transport, transport problems, environmental mathematical problems, harmful substances.

Introduction

To complete the working cycle in a piston internal combustion engine, the combustion products formed during the previous cycle must be removed from the cylinder and a fresh charge of air or a fuel-air mixture must be introduced. It should be noted that the amount of fresh air introduced depends on the quality of the engine cylinder cleaning. Therefore, the intake process is analyzed in conjunction with the parameters characterizing the exhaust process, considering the entire complex of phenomena related to the gas exchange process as a whole.

The process of intake or charging in engines is intended to fill fuel cylinder a mixture of air and fuel or air alone.

The amount of air or combustible mixture entering the cylinder during its filling depends on a number of factors, the main ones of which are [17]:

- hydraulic resistance of the intake and exhaust systems;
- heating of the fresh charge from contact with hot engine parts;
- the presence in the cylinder at the beginning of filling with a fresh charge of residual exhaust gases from the previous cycle.

In a four-stroke engine, gas exchange occurs within two crankshaft revolutions. Thus, gas exchange in the engine cylinder performs two main functions:

- removal of exhaust gases from the engine cylinder during the previous cycle;
- ensuring the supply of a fresh portion of the working fluid.

Valve timing, and thus gas exchange, is regulated by the camshaft. The camshaft rotates at half the speed of the engine's crankshaft, from which it is driven.

Most modern internal combustion engines use overhead valves, which can be driven either from the overhead camshaft using a lever mechanism, or directly from the cams of this shaft.

The camshaft is designed to transmit movement to the valves from the crankshaft. It is usually made integral with the cams (eccentrics) and drive components of certain engine components.

The upper suspension valves are located in the cylinder head, the lower and side valves in the cylinder block.

In carbureted engines with a single-row valve arrangement, to ensure intake manifold heating and improved fuel evaporation, the intake and exhaust manifolds are usually located on the same side of the cylinder head. Placing the manifolds on both sides of the cylinder head is most often used in diesel engines, which reduces intake manifold heating and thereby improves engine filling efficiency.

To improve cylinder filling, reduce exhaust valve temperatures and reduce the mass of moving parts of the valve timing mechanism per valve, in some cases it is advisable to install three or four valves per cylinder.

V.M. Arkhangelsky et al. [16] propose the following dependence for determining fuel consumption, kg/s:

$$G_T = 7200\frac{iV_n\rho_k\eta_V}{\tau\alpha l_0};(1)$$

It follows that the weight of the combustible mixture sucked into the internal combustion engine per unit of time depends on the cylinder displacement, engine speed, and mixture density ρ_k. The mixture pressure is most significantly affected by the resistance in the intake manifolds and the flow cross-sections of the distribution components, as well as the temperature at the exhaust end. Increasing the temperature at the exhaust end reduces the filling efficiency η_V, and therefore engine power.

When the mixture entering the cylinder burns, it releases a quantity of heat proportional to the weight of the charge. Part of this heat, the amount of which depends on the perfection of the combustion process and thermal and mechanical efficiency, is converted into useful work by the crank mechanism. Despite extensive research, thermal efficiency has not been significantly improved. Increasing the power per liter, necessary to reduce engine weight, leads to increased heat loss through the cylinder walls during combustion and with the exhaust gases. This limits the lower limit of achievable specific weight of the engine.

Heat transfer from the valve to the cooling medium occurs primarily through the valve seat and partially through the valve stem. Insufficient valve movement and untimely seating of the valve head can lead to valve timing failure due to overheating. The use of sodium-filled valves, multiple exhaust valves, and coating the valve heads and seats with heat-resistant materials improve the situation somewhat, but are not considered radical measures.

Since the acceleration of the valve movement increases proportionally to the square of the number of revolutions, a further increase in the number of revolutions above the already reached limits is limited by the magnitude of the accelerations in the distribution parts.

Increasing the liter capacity can be achieved by the following measures:

-improving the quality of cylinder cleaning and reducing the amount of residual gases as a result of reducing the resistance to gas passage;

- increasing the permissible compression ratio due to the favorable shape of the combustion chamber and the elimination of highly heated areas, the temperature of which significantly exceeds the average temperature of the walls;

- increasing the working speed of the crankshaft to the limits limited by the strength of the crank mechanism, by eliminating the distribution parts subject to the action of significant alternating accelerations.

Valves are not suitable for forced valve timing, as their movement when closing is limited by valve seats. The position of these stops depends on the engine's temperature, so valve springs must be used to close the valves. Research conducted to determine the power expended in valve timing has shown that with valve timing, almost all the energy absorbed by the spring when opening the valve is returned when it closes. Power is expended only to overcome the excess pressure when opening the exhaust valve and the vacuum when opening the intake valve.

The distributions of engines with different piston diameters and strokes, but the same speed, can be compared using the specific flow area.

When designing engine valve timing mechanisms, it is convenient to use the parameter proposed by I.M. Lenin [3] – the specific area of the middle inlet opening of the valve:

$$Q_{уд} = \frac{f_{\alpha cp}}{V_h^{/}(n_{eN}/1000)} = \frac{f_{\alpha cp}}{V_h^{/}(n_{ерег}/1000)}, \qquad см^2/(л \cdot 1000\ об/мин) \quad (2)$$

where is $V_h^/$ – the working volume of one engine cylinder; f_{acp} – the opening area of the intake valve;

$n_{epez} = 1{,}35\ n_{eN}$

The discrepancy in the number of revolutions n_{epez} is n_{eN} explained by the fact that the value n_{eN}, in addition to filling, is also affected by the continuous increase in power spent on overcoming mechanical resistance and pumping losses in the engine.

The specific intake port area of 2.45 cm2 / (l 1000 rpm) has been verified many times on various engines.

When evaluating the distribution type, it's important to remember that the comparative value $Q_{y\partial}$ doesn't account for the nature of the flow area changes as they open and close. According to hydrodynamics, it's desirable to have slow-opening and fast-closing valves on the intake side, while, conversely, fast-opening and slow-closing valves are desirable on the exhaust side.

It should be noted that with an increase in valve lift, the flow coefficient μ, related to the cross-sectional area in the valve seat, decreases significantly, and at maximum lift . $\mu = 0{,}45....0{,}50$.As a result, an excessive increase in the maximum valve lift has little effect on the effective flow area $\mu f_{\kappa\pi}$ and is not accompanied by a noticeable increase in the filling coefficient.

Calculating the average intake velocity based on the geometric time-section $\int_{t_1}^{t_2} f_{\kappa\pi}\, dt$ does not take into account the hydraulic properties of the intake tract as a whole or its individual components. The conducted research allows for a more accurate assessment of the intake tract based on the effective time-section value $\int_{t_1}^{t_2} \mu f_{\kappa\pi}\, dt$.

When calculating based on the effective time-section, which takes into account gas-dynamic losses in all elements of the intake tract, the average actual intake velocities are higher.

Pressure loss due to resistance in the intake system [4]:

$$\Delta p_a = k_1 \frac{n^2}{f_{\kappa\pi}^2}; \quad (3)$$

where is k_1 – a constant value

$$k_1 = \frac{\pi^2 D^4 R^2 \left(1 + \lambda^2\right) \rho_k \left(\beta^2 + \xi_{bn}\right)}{8}$$

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$D-$cylinder diameter; $R-$crank radius; connecting $\lambda=\frac{R}{L}$; $L-$rod length; ρ_k-charge density at the intake; $\beta-$charge velocity attenuation coefficient at the intake; $\xi_{bn}-$drag coefficient of the intake system.

From expression (4), it is clear that is Δp_a proportional to the square of the rotational speed and inversely proportional to the square of the area. Increasing the area f_{bn} is a way to reduce pressure losses. In modern four-stroke automobile engines with overhead valves, the possibilities for increasing the area f_{bn} are limited by the valve placement in the cylinder head.

The total cross-sectional area of the intake valves can be increased by using four valves per cylinder. A four-valve configuration is advisable for turbocharging and at higher engine speeds, as the reduced mass and, consequently, inertial forces of each valve ensure more reliable valve timing at high engine speeds.

Due to the hydraulic resistance of the intake and exhaust systems and the presence of residual gases in the cylinder, the actual amount of fresh charge entering the cylinder during the intake period will always be less than the amount that could fill the working volume of the cylinder under ambient conditions. The degree of perfection of cylinder filling with fresh charge is assessed using the filling coefficient:

$$\eta_V=\varphi_1\frac{\varepsilon}{(\varepsilon-1)}\frac{P_a}{P_0}\frac{T_0}{T_0+\Delta T+\varphi\cdot\gamma_{OCT}\cdot T_r},(3)$$

where is $\varphi_1-$the cylinder recharging coefficient; $\varepsilon-$compression ratio; P_a-pressure at the end of the intake; P_0 and T_0-pressure and ambient temperature; $\gamma_{OCT}-$residual gas coefficient; T_r-residual gas temperature.

From expression (58), it is evident that the filling coefficient is affected by pressure P_a and temperature T_0, charge preheating temperature , ΔT, residual gas coefficient γ_{OCT}, temperature $T_r-$and pressure P_r, compression ratio ε, and recharging and cleaning coefficient. These quantities, in turn, depend on a number of factors and are, moreover, interrelated. Therefore, in addition to analyzing the impact of individual factors on the coefficient, η_V it is advisable to consider the totality of their influence on it, depending on the engine operating mode.

Among other factors, pressure P_a has the greatest influence on the coefficient η_V. It has been determined that the reduction in pressure loss ΔP depends on the resistance in the intake system and is proportional to the square of the average charge velocity in the smallest cross-section of the intake system. η_V The design of the intake tract, primarily the valve clearance, also influences the value $f_{кл}$. Pressure loss and the associated deterioration in filling, according to the laws of hydrodynamics, are proportional to the square of the mixture velocity. To reduce the mixture velocity, high-speed engines have short, large-diameter intake pipes with increased lift and valve timing designed for the duration of valve opening.

To increase the filling efficiency, the number of exhaust valves is increased to two in each cylinder. To reduce friction of the mixture flow against the walls and to mitigate vortex formation, which negatively impacts the filling, the internal surfaces of the intake manifold, diffuser, intake pipe, and intake port in the head are polished.

The engine's thermal state also affects the filling factor. The thermal conductivity of the material, combined with intensive cooling, reduces the average temperature of components subject to high temperatures. M.M. Aripdzhanov [6] is dedicated to solving this problem, theoretically exploring the complex influence of the cyclical nature of the heat transfer process, the temperature boundary layer, and the thermal insulation elements of the cylinder-piston assembly components on reducing heat loss, specific fuel consumption, and the emission of toxic components with exhaust gases.

As the engine speed increases, the filling efficiency begins to drop due to the increased mixture flow velocity. Even if all measures are taken to reduce flow resistance, a significant drop in filling efficiency only occurs at high speeds. Improved cylinder filling was achieved by adjusting the length and cross-section of the intake tract to induce oscillations in the mixture flow.

The most effective way to increase the charge ratio is to force-feed the engine with the mixture from the supercharger. In this case, at high boost pressures, the charge ratio can be significantly greater.

Mechanical efficiency is increased by reducing friction losses. Engine testing revealed that the valve timing drive accounts for 11.2% of total mechanical losses.

It is known that the effective engine power per unit of cylinder displacement is estimated by the engine power per liter:

$$N_U = \frac{H_U n \eta_i \eta_M \rho_k \eta_V}{30 \tau l_0 \alpha}, \quad (59)$$

where is η_U – the mechanical efficiency, which characterizes friction losses and pumping losses; η_i – the indicated efficiency of the engine; H_U – the calorific value of the working mixture at normal pressure and atmospheric temperature.

It should be emphasized that a naturally aspirated engine operates with higher mechanical efficiency and a high compression ratio; therefore, it ensures good heat recovery and low specific fuel consumption; engine power is limited by a lower filling factor.

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Forced engines, i.e., engines with supercharging via a supercharger, operate with lower mechanical efficiency and a low compression ratio, resulting in poor heat recovery and increased specific fuel consumption. With turbocharging, mechanical losses are lower, and a reduction in compression ratio is also necessary. However, any supercharging method dramatically increases the filling factor, resulting in significant overall fuel consumption per unit of time. High energy consumption in this case ensures high power per liter, despite poor heat recovery.

From the above analysis, it follows that increasing crankshaft speed cannot be considered an independent factor for increasing power per liter. Speed increases power only if it increases the product $\eta_M \eta_V n$, despite the decrease in the first two factors. The drop in mechanical efficiency and filling factor at a certain speed is no longer compensated for by increasing speed, and power begins to decline.

All design measures that increase the filling factor and mechanical efficiency simultaneously result in an increase in the rotation speed corresponding to the maximum power, since in this case the product $\eta_M \eta_V$ decreases more slowly with an increase in rotation speed.

Based on the conducted research, it can be concluded that increasing the global power of spark-ignition engines is facilitated by increasing ε and approaching α values of 0.85...0.95. Improved fuel efficiency can be achieved by increasing ε and approaching α 1.0...1.10, which simultaneously improves the engine's environmental performance due to more complete combustion of the mixture.

Therefore, to increase the power and efficiency of engines, it is necessary to strive to lean out the combustible mixture in parallel with increasing the compression ratio; this makes it possible to most effectively use the advantages of high compression ratios.

For all types of spool valve timing, the most important feature is the nature of the mechanism's movement:

- progressively moving spool valves that perform an oscillatory motion, which, to a first approximation, has a sinusoidal character;
- rotating spool valves moving at a constant angular velocity and therefore not subject to the action of inertial forces at a constant crankshaft speed.

A reciprocating spool valve control for a four-stroke process consists of two tubular spool valves nested within the cylinder, enclosing the working piston. The spool valves are moved by auxiliary connecting rods. The cranks are offset by approximately 70 ° relative to each other .

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Changing the exhaust cross-sectional area is achieved by shifting the intake port on the outer valve upward. Further changes in cross-sectional area can be achieved by altering the phase angles between the inner and outer valve cranks, as well as by changing the height of the cylinder ports. Figure 1 shows the intake and exhaust cross-sectional areas as a function of crankshaft rotation angle.

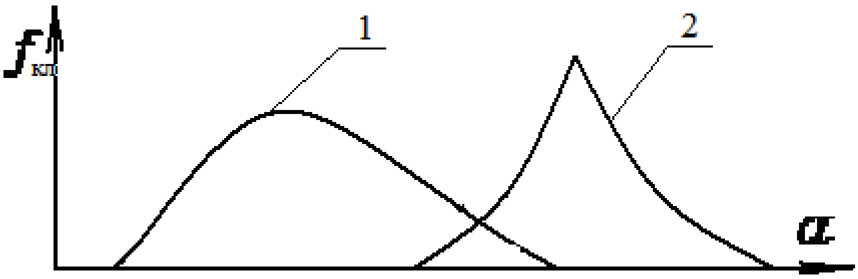

Fig. 1. Time-section of two spool valve timing. 1-outlet; 2-inlet

The significant advantages of the distribution with two progressively moving spools are:
- favorable shape of the combustion chamber;
- fully forced movement of the valves;
- silent operation;
- the ability to change individual distribution parameters without significantly affecting other parameters.

Disadvantages of the mechanism:
- complex drive;
- the spool valves are driven by offset four-link crank mechanisms and are therefore subject to significant inertial forces;
- poor heat dissipation from the combustion chamber through three oil films;
-increased friction of large surfaces of valve spools, especially when the engine is cold;
- skewing of the spools due to non-central connection with the drive connecting rod and one-sided pressure of the working piston;
- low overlap of the distribution windows, which worsens the tightness during the compression stroke:
- difficulty in lubricating the rubbing surfaces of both sleeves:
- deterioration of the mixture due to oil vapor; difficult heat dissipation to the cooling water in the cylinder jacket, caused by the presence of valves and three oil films and leading to overheating.

The power consumed by the two-spool valve timing drive is 0.4...0.9% of the useful power, which is 2-3 times higher than in engines with valve timing.

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In some air-cooled aircraft engines, the most successful types of spool valves were those that combined reciprocating movement of the spool parallel to the cylinder axis with rotation around the same axis.

The main disadvantages of this type of distribution are as follows.

- significant sliding surfaces between the valve, cylinder head, piston and finned cylinder, which cause high friction forces, especially in a cold engine.
- the distribution drive forces are applied at a significant distance from the center of the spool;
- the spool valve experiences one-sided pressure from the piston and is simultaneously subject to the action of large inertial forces: as a result, the increase in the number of revolutions that can be achieved with this type of distribution is insignificant and amounts to approximately 10% compared to the highest values achieved in an engine with valve timing;
- the implementation of a hermetically sealed connection of a thin-walled valve with a finned cylinder, piston and cylinder head, which would not depend on the temperature and load conditions of the engine, requires a lot of experience.

Of greatest interest is the design of an engine with a flat spool valve with α 1, 2, or 5 valve groups. In this engine, valve timing is accomplished using articulated spool valves, i.e., valves that are not affected by inertial forces at a constant crankshaft speed. Maximum permissible engine speeds are limited only by the crank mechanism and the sliding speed of the valve distribution surfaces.

Rotating valve distribution, suitable for high speeds, allows for significantly larger flow areas of the distribution elements, even compared to the use of multiple valves per cylinder. Due to this, and the short gas flow paths, these types of distribution, when properly implemented, offer high filling capacity and create minimal exhaust gas resistance.

With rotary valve distribution, no components are in the combustion chamber, which is hot from the previous cycle, during filling and compression. This allows for a higher compression ratio, allowing for other advantages of this distribution to be realized.

The disadvantages of this design are:

- unfavorable shape of the combustion chamber;
- small distribution cross-section; due to the eccentric arrangement of the sealing bushings, the distribution cross-section at full opening can only be approximately 18.9% of the piston area.
- insufficient cooling of the spool; the cooling medium is supplied to the spool through a diaphragm and rings with ground ends; but since only one common annular hole is provided for the supply and discharge of the cooling medium to the spool, intensive

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cooling of the spool cannot be achieved; supply of cooling water without an appropriate seal entails a violation of operational reliability due to water ingress into the lubrication system;

- uneven wear of the spool; the sealing surfaces are subject to wear, which, among other things, depends on their sliding speed, which increases with the spool radius; therefore, these surfaces wear more on the outside than on the inside.

A group of developments of cylindrical spool valves assumes placement of the spool valve in the cylinder head with the axis directed perpendicular to the cylinder axis.

The rotating valve has one channel each for fresh charge and exhaust gases, located diametrically. For four-stroke engines, the gear ratio between the crankshaft and the rotating valve is 4:1. Therefore, the peripheral speeds on the outer surface of the valve are low.

The nature of the change in flow sections during intake in a cylindrical valve and valve timing is shown in Fig....2.

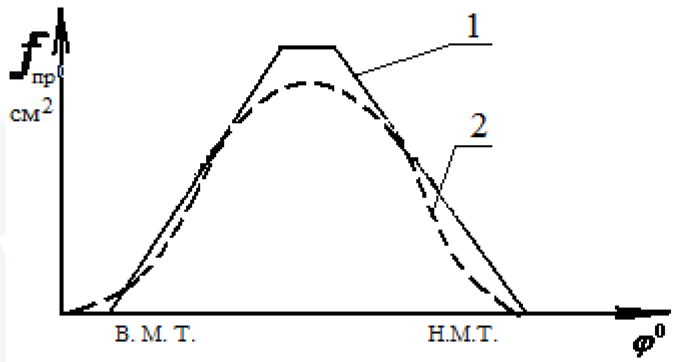

Fig. 2. The nature of the change in the flow sections of the intake with distribution with a rotating spool (1) and with valve distribution (2).

Cylinder efficiency characterizes the liter-per-revolution power of an engine. A comparative analysis of engines with spool valve distribution and valve timing shows that the cylinder efficiency is 1000 and 1000, respectively $C_1 = 0{,}00966$. $C_2 = 0{,}00722$The cylinder efficiency of an engine with spool valve distribution is approximately 80%.

Very large inlet and outlet cross-sections are difficult to achieve, as accommodating two gas channels in the spool valve shaft, diametrically opposed to the cooling medium channels, requires significant dimensions. Filling and cleaning are limited, as the gas flow is distributed twice by the spool valve.

Thanks to good cooling of the valve stem, fuels with a lower octane number can be used, and despite this $\varepsilon = 7{,}2$, the engine operates without detonation at high efficiency.

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