Oleksandr Kakun, Master, Pridneprovsky State Academy of Civil Engineering and Architecture, aleksandr060501@gmail.com
Tatyana Kolesnikova, Candidate of Technical Sciences, Associate Professor, Pridneprovsk State Academy of Civil Engineering and Architecture, tnk1403@ukr.net
Artem Didyk, Master, Pridneprovsky State Academy of Civil Engineering and Architecture, Artem.yotube2000@gmail.com
Danylo Podolskiy, Master, Pridneprovsky State Academy of Civil Engineering and Architecture, danil.podolskiy.00@gmail.com
Mikhailo Kobilyatsky, Master, Pridneprovsky State Academy of Civil Engineering and Architecture, gamegorobec@gmail.com
Sergei Zavgorodnii, Master, Pridneprovsky State Academy of Civil Engineering and Architecture, sergeu32@icloud.com
Yehor Yemets, Master, Pridneprovsky State Academy of Civil Engineering and Architecture,
Egoremec345@gmail.com

THEORETICAL STUDIES OF TWO-STROKE ENGINE WITH ADJUSTABLE COMPRESSION RATIO

Introduction. Environmental issues and economic aspects of internal combustion engines have a great influence on the development of their design. Competition with other energy sources is also accelerating development. However, the closest competitor, electric engines, must be tied to the electric grid or have a reliable low-cost battery as part of the vehicle. The problem is still far from solved. But work in this direction is a stimulus for development, particularly of promising internal combustion engine designs. Two-stroke engines with spark ignition compared to four-stroke engines have high liter power and good mass-size indicators, have almost an order of magnitude less NOx emission, are cheaper to manufacture and maintain. However, these engines are characterized by increased fuel consumption and significant emissions of CO and especially CH, which is associated with the peculiarities of gas exchange and imperfect combustion process, especially at idle and low loads, when the filling of the cylinder with fresh charge decreases, and the amount of residual gases increases. It is possible to improve economic and at the same time toxic indicators of a two-stroke gasoline engine by improving the combustion process in partial modes by adjusting the compression **ratio.**

Problem Statement. The task of theoretical study of a two-stroke engine with crank-chamber blowdown is to reveal the influence of compression ratio control εx on the indicator and effective engine performance at its operation in partial modes.

To solve this problem it is necessary to develop a method of calculation of the engine, allowing to determine the parameters of the working cycle, taking into account the mode of operation of the internal combustion engine and the compression ratio in the cylinder.

Basic material. Existing methods with a certain reliability allow to carry out the calculation of the engine working process, but for this purpose the initial data of such calculation should be correctly determined [1; 2]. In a two-stroke engine with crank-chamber blowdown such initial data can be the excess blowdown air ratio ϕ_0 and the blowdown pressure pk. The value of these values can be determined from the design of the blowdown compressor. The influence of εx on the variation of ϕ_0 and p_k must be taken into account.

These methods of calculation of the blowdown compressor do not allow to study the regularities of influence of various design factors on its parameters (performance, power consumption, etc.) and to study the features of the engine operating process depending on the parameters of the blowdown compressor, so they can not be sufficiently used in the design and development work.

Thus, for the possibility of research and fulfillment of appropriate design calculations of two-stroke engine with variable compression ratio it is necessary to develop a methodology for calculating the purge compressor and obtained equations that allow to carry out qualitative and quantitative assessment of the influence of various factors on the indicators of the working process of the engine.

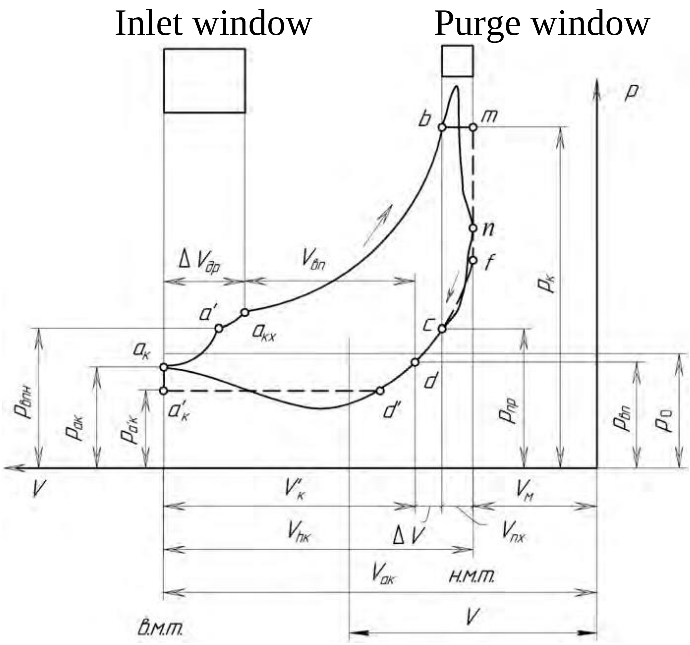
The calculation of the blowdown compressor of a two-stroke engine with a variable compression ratio required a separate very labor-intensive calculation and experimental study of the parameters of the gas exchange system. In our case, we proceeded from the experimental data of existing two-stroke internal combustion engines.

The main parameters characterizing the working process of the purge compressor are its capacity, indicator work and average indicator pressure.

To determine these parameters, we replaced the indicator diagram of the real blowdown compressor with the diagram of the so-called design cycle, observing the condition of equality of the areas of these diagrams.

In the calculated indicator diagram shown in Fig. 1 with dashed lines, the inlet process (solid line $dd'a$k) is replaced by two lines - the expansion polytropy dd' with the exponent n2k and the dashed line $d'a'$k of constant mean inlet pressure pa'k.

The crank chamber recharging process at the aka' section is caused by the pressure difference $p$vp in the inlet pipe and in the crank chamber p (p < $p$vp). At the end of the recharging process, the mixture ejection process takes place. This process begins when the pressure p = $p$vpn reaches equality (point a') and ends when the inlet window closes (point $a$kx). The mixture pressure in the crank chamber during the ejection process (point $a'a$kx) due to hydraulic losses in the inlet window is higher than in the inlet pipe and is variable. The variable nature of hydraulic losses during back ejection is explained by the fact that during this period the piston velocity and the cross-sectional area of the window change.



- - - calculated; --- actual

Figure 1 - Indicator diagram of the blowdown compressor

The gas exchange process considered above can have three special cases depending on the choice of gas distribution phases. In the case of optimum timing, the inlet window closes at the moment of pressure equality p = $p$vpn (point $a$kx coincides with point a') and the filling of the crank chamber with fresh charge reaches its maximum value. If the window height is less than the optimum height (point $a$kx lies on the curve aka'), the crank chamber is undercharged because $p$aq x < $p$vn. If the height of the inlet window is greater than the optimum value as shown in Fig. 3.1, the mixture will be pushed out of the crank chamber into the inlet pipe at the section $a'a$kx.

The compression $a'b$ and reverse expansion cd in the calculated indicator diagram are assumed to be exactly the same as in the actual indicator diagram of the blowdown compressor.

Purge compressor capacity. Since in the described engine the piston acts as both a normal piston and a blowdown compressor piston, it is more convenient to use not the second capacity of the blowdown compressor, which is usually used in compressor technology, but its cycle capacity, i.e. per one revolution of the crankshaft.

$$V_k = \varphi_0 V_{hk} \tag{1}$$

where V_k - is the volume of mixture delivered by the purge compressor to the cylinder per cycle (or one crankshaft revolution); V_{hk} - is the total volume described by the piston per stroke (purge compressor displacement); ϕ_0 - purge air excess ratio (or crank chamber fill factor).

For the crank chamber, the described volume

$$V_{hk} = \frac{\pi D^2}{4} S \tag{2}$$

where D - is the cylinder diameter of the blowdown compressor; S - is the total stroke of the piston.

Indicator work. Let us determine the indicator work of the purge compressor *Liq* for one crankshaft revolution.

$$L_{d'ak} = V_{hk} p_{ak} \left[1 + a_M - (a_M + a_{\Pi x}) \left(\frac{p_{np}}{p_{ak}} \right)^{\frac{1}{n_{2k}}} \right] \tag{3}$$

Average indicator pressure

$$p_{ik} = \varphi_{ik} \left[\begin{array}{c} \frac{\int_{V_{ak}}^{V_{akx}} pdV}{V_{hk}} + p_{ak} \left\{ \lambda_{\text{доз}} \frac{1 + a_M - a_{\text{ВП}} + a_{\Pi x} - a_{\Pi}}{n_{1k} - 1} \right\} \text{x} \\ \text{x} \left[1 - \left(\varepsilon_{\text{Кх}} - \frac{a_{\text{ВП}}}{a_{\text{М}} + a_{\Pi x}} \right)^{n_{1k} - 1} \right] - \\ -\lambda_{\text{ДОЗ}} a_{\Pi x} \left(\varepsilon_{\text{Кх}} - \frac{a_{\text{ВП}}}{a_{\text{М}} + a_{\Pi x}} \right)^{n_{1k}} + \\ + \frac{a_{\text{М}} + a_{\Pi x}}{n_{2k} - 1} \left[\delta \left(1 + \frac{a_{\Pi x}}{a_{\text{М}}} \right)^{n_{2k} - 1} - \delta^{\frac{1}{n_{2k}}} \right] + \\ + \left[1 + a_{\text{М}} - (a_{\text{М}} + a_{\Pi x}) \delta^{\frac{1}{n_{2k}}} \right] \end{array} \right] \tag{4}$$

Indicator power

$$N_{ik} = \frac{p_{ik} V_{hk}}{30\tau} \tag{5}$$

Conclusion. The obtained analytical dependences allow to estimate the influence of compression ratio control on power and economic indicators of the engine in conditions of variable modes, and also can be used for practical calculations of internal combustion engines.

Literature

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