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EXPERIMENTAL ANALYSIS OF LOAD CHARACTERISTICS OF A TWO-CYLINDER PISTON COMPRESSOR

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Annotation. *The article presents the results of a theoretical justification and experimental analysis of the load characteristics of a two-cylinder piston compressor. The data have been summarized and a mathematical apparatus has been developed to determine the moment of resistance of a piston compressor based on SA-7A when its rotation frequency changes.*

Key words: *compressor unit, rotor, piston, cylinder, moment of resistance, rotation speed.*

Introduction. Compressor units are widely used in industry, in chemical production, in the production and transportation of oil, natural gas, in transport, in refrigeration equipment, etc. [2,3].

There are various design solutions for compressors, which have their own characteristics. Compressors, according to their method of operation, are usually divided into volumetric (piston and rotary), vane (centrifugal and axial) and jet [1,2]. Piston compressors are the most numerous and widespread due to their advantages: high efficiency at low and medium capacities, the ability to develop high pressures and the accumulation of compressed gas energy. Since consumers of compressed gas in most cases have a time-varying nature, there is a need to regulate the performance of the compressor unit.

The performance of the compressor can be changed by influencing the elements of the gas path, the electric motor or its connection to the compressor, suction or discharge valves, the ratio between the volume of the compression cavity and the volume described by the compressor piston, elements of the movement mechanism, as well as various combinations of these influences.

Let's look at some ways to regulate productivity. The method of bypassing gas from discharge to suction does not require any special changes in the design of the compressor, is simple to manufacture and operate, allows for smooth changes in performance over a wide range, but increases the cost of specific work for the production of compressed air. Bypass should be used mainly to reduce the

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compressor resistance torque during startup. Throttle bypass of gas after the first stage into the compressor suction line reduces the cost of specific work compared to bypassing; It is simple to manufacture and operate, but due to an increase in the pressure ratio in the last stage of the compressor and the possibility of dangerous gas temperatures, it is limited in the permissible range of capacity control.

Changing the performance of the compressor by temporarily switching its operation to idle by pressing the suction line valve plates or bypassing gas from the cylinder into the suction cavity using special valves can be used in systems with a limited range of pressure changes [2,3]

However, switching the compressor to idle does not eliminate the energy consumption of the engine at zero compressor performance. Regulating the pressure in the network by attaching cavities of additional dead spaces does not greatly complicate the design of the machine, is more economical than other methods, and maintains the pressure quite accurately. Reducing compressor performance by throttling the suction gas is easy to manufacture, allows for a smooth change in performance, but increases specific operating costs. The degree of performance reduction is limited by increasing the pressure ratio in the last stage of the compressor. This method should be used only for small performance control ranges. A smooth change in drive speed directly proportionally changes the performance of the compressor unit. This method is the most economical.

When periodically starting or stopping the drive, the required performance is achieved by choosing the ratio of the time of operation or stop of the engine. This method is quite economical, but has a number of disadvantages: current surges when starting an electric motor, which reduces the life of the engine, pressure drops during regulation with a long on or off time. This method is used for asynchronous motors with a squirrel-cage rotor up to 100 kW, and for induction motors with a wound rotor up to 250 kW. Step change in drive speed allows you to change productivity in steps of 50%, 33%, etc. The disadvantages of this method are the use of multi-speed IMs with lower efficiency at rated load. In the process of cavitation, the most effective method of crushing [4,5] liquid pesticides or liquid biologically active polymers is the use of an aerodynamic flow created by a compressor.

When regulating the performance of a compressor, a method is proposed in which the rotation speed changes, and it is necessary to take into account the change in the moment of resistance created by the compressor when the rotation speed of the compressor shaft changes.

In addition to understanding the generally recognized need for the development of resource-saving technologies, certain regulatory documents - in particular the European Union directive on greening (EU Ecodesign Directive 2009/125/EC [6]) stipulate the requirement for constant optimization of the energy efficiency of each product. To meet the high energy efficiency requirements in the field of reciprocating compressors, which will only become more stringent in the future, it is necessary to find the most optimal use of the existing potential. As in many other industries, the topic of heat loss recovery is leading, because In a reciprocating compressor, particularly a multi-stage reciprocating compressor, in systems operating in the kilowatt and megawatt range, the compressed gas is typically cooled to ambient temperature after each stage. The amount of heat energy removed is very significant and is lost in almost all cases. The most important task is considered to be to propose a method for quantifying the available heat generated and its potential use.

For reciprocating compressors, the maximum discharge temperature is typically 200 °C, and in some cases slightly higher [7]. For many applications, this temperature is defined by standards: for oil-lubricated multistage air compressors above 1.0 MPa (abs), it is 160 °C according to [8]. API 618 [9], the leading international standard for the oil and gas industry, recommends a general maximum temperature of 150 °C and 135 °C for high hydrogen content. A general limitation of up to 135 °C in the oil and gas industry is given in [10]. Sometimes, with special precautions, higher discharge temperatures are possible. For most compression scenarios, the minimum discharge temperature is at least 80 °C. Thus, it can be assumed that for most reciprocating compressors, the discharge temperature is between 80 °C and 200 °C.

Material and methods. The first analysis of the piston pump-compressor operation was performed by the authors [11,12], where one can also get acquainted in detail with the operation of this unit. The studies showed the operability of the design and the possibility of increasing the efficiency of the compressor cavity by reducing the cylinder temperature. In these studies, an obvious design approach was used, based on the crosshead design of the piston drive, which divides the cylinder into two cavities - compressor and pump. That is, in essence, a vertical design of a double-acting compressor was used [5], the above-piston space of which is connected through self-acting valves to the source and consumer of gas, and the under-piston space, in which the rod is located, is connected to the source and consumer of liquid.

For the work, the definitions of the torque created by the compressor were studied; it is necessary to take into account the geometric parameters of the piston machine, the dependence of the torque on the angle of rotation of the shaft at a unit radius and pressure on the piston, as well as when changing the pressure in the compression cavity [2].

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The torque is determined in the following order:

1. Determining the torque created by one piston on the compressor shaft per shaft revolution at a unit crank radius and force on the piston.
2. Determining the position of the piston in the cylinder at different values of the shaft rotation angle.
3. Determining the theoretical indicator diagram for given parameters, using data.
4. Determining the torque on the crank shaft created by one piston using the corresponding data.
5. Determining the torque on the crank shaft for the compressor.
6. Determining the torque on the compressor shaft taking into account the flywheel masses at different compressor shaft rotation frequencies.

The study used methods of scientific analysis, synthesis, Bernoulli's equation and Fisher's variance statistical data processing.

Result and their discussion. For a real piston machine, the moment of resistance must take into account the pressure created:

$$M_k = \frac{P\, r\, M_1\, \pi D^2}{4} \qquad (1)$$

where P- is the pressure in the cylinder from the indicator diagram; r -is the crank radius; D -is the diameter of the compressor cylinder.

The pressure in the cylinder is determined by the theoretical indicator diagram. Indicator diagrams are used to describe the processes occurring in the cylinder during one revolution of the shaft. The construction of the theoretical indicator diagram is made with some assumptions: the compressed gas is an ideal gas, where there is no resistance to gas movement and there are no leaks.

Thus, with a force on the piston in the III and a crank radius in the 1st, the compressor will create a moment:

$$M_1 = \frac{\sin(\omega\,\theta + \beta)}{\cos(\beta)} \qquad (2)$$

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where ω θ - is the angle of rotation of the crank mechanism, rad/s; β- is the angle between the connecting rod and the compressor movement.

The general form of the unit moment of resistance for a single-cylinder compressor is shown in Fig. 1.

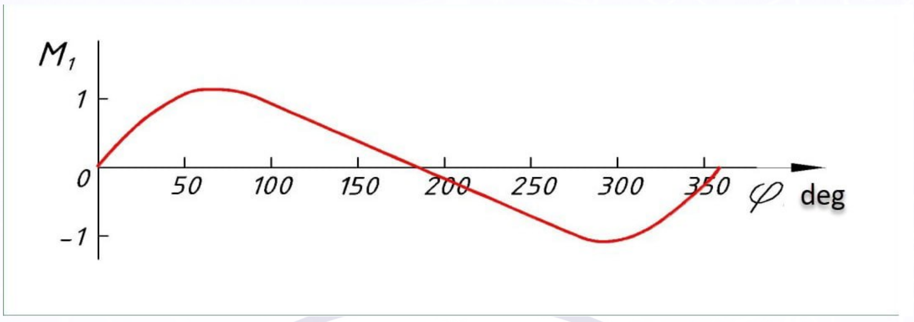

Fig. 1. Dependence of the unit moment on the angle of rotation of the crank shaft.

When constructing a diagram, it is necessary to know how the volume in the cylinder changes when the compressor shaft position changes. Since the volume depends only on the piston position in the cylinder, it is sufficient to determine the piston position at different shaft position angles.

Piston position relative to the bottom dead center:

$$S=\frac{r\,(l-\cos(\omega t))+l}{4\,(l-\cos(2\omega t))} \qquad (3)$$

where r- is the crank radius, m; l= t/1 is the ratio of the crank radius to the connecting rod length; 1- is the connecting rod length, m.

The piston positions relative to the bottom dead center when the compressor shaft rotates are shown in Fig. 2.

The indicator diagram is determined for the CO-7A compressor at an excess discharge pressure of 0.2 MPa. (Fig. 3).

The gas compression curve 1-2 is calculated based on the polytropic equation:

$$pV\kappa = const, \qquad (4)$$

where k- is the polytropic exponent, for air k= 1.4; p is the pressure, Pa; V- is the volume, m^3.

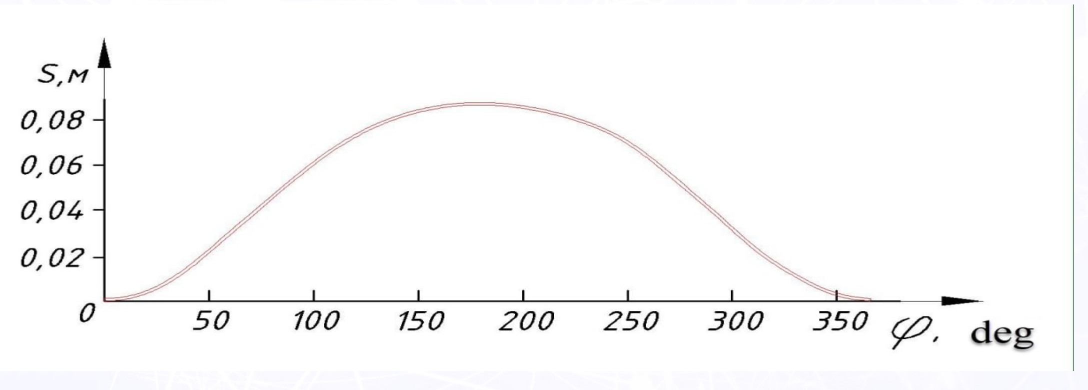

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Fig. 2. Position of the piston relative to BDC depending on the angle of rotation of the compressor shaft.

The discharge curve 2-3 is taken to be equal to the discharge pressure. The curve (expansion from dead space 3-4 is determined similarly to the curve 1-2. The suction curve 4-1 is taken to be equal to the suction pressure

The indicator diagram is shown in Fig. 3.

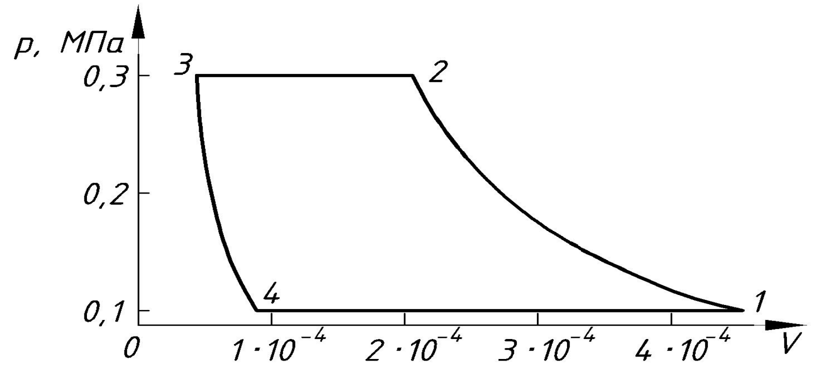

Fig. 3. Theoretical indicator diagram of the compressor CO-7A at excess discharge pressure of 0.2 MPa.

Based on the theoretical data obtained earlier, the torque on the compressor shaft is determined using formula (2). The calculation result is shown in Fig. 4.

The dependence of the torque on the shaft rotation angle, taking into account that the compressor CO-7A is a two-cylinder compressor, is shown in Fig. 5, curve

Flywheels are used to smooth out the torque pulsations created by the compressor and, as a consequence, reduce the current pulsations in the drive circuit.

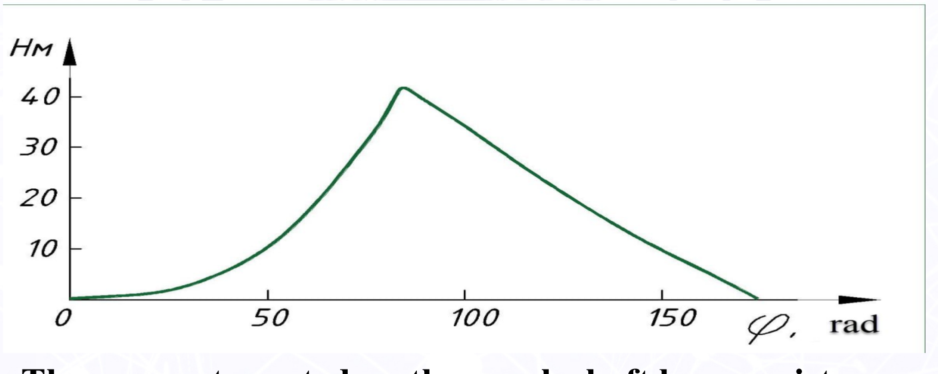

Fig.4. The moment created on the crank shaft by one piston.

With an increase in the pulsation rotation frequency, the torque decreases (Fig. 5, curve 3), and with a decrease in the pulsation rotation frequency, the torque increases (Fig. 5, curve 2).

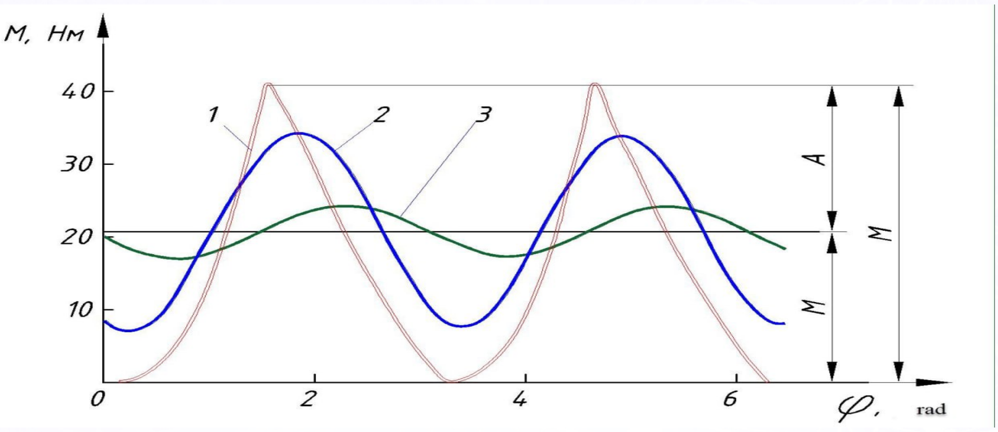

Fig. 5. The moment created on the crank shaft (curve 1) and on the compressor shaft, taking into account the moment of inertia at a rotation speed of 10 rpm (curve 2) and 100 rpm (curve 3) depending on the shaft rotation angle.

The dependence of the moment of resistance on the rotation speed, according to formula (1), will be equal to:

$$M_{дв} = M_{ср} + \frac{A}{\sqrt{2+(\frac{I\omega}{c})}} \sin(\omega t - arctqj\omega c \qquad (5)$$

where Mcp - is the average torque during one revolution (from the load torque graph), nm; J - is the moment of inertia reduced to the engine shaft, kg m2; A is the amplitude of the torque oscillations on the crank shaft; c - is the rigidity of the mechanical characteristic of the engine, $c = M_{hom}/\omega_{nom}$.

The dependence of the torque on the rotation speed is shown in Fig. 6. Analysis of the load characteristic of a two-cylinder compressor showed that the torque pulsations increase smoothly with a decrease in the rotation speed to 15 rad/s, and with a further decrease in the rotation speed the pulsations increase sharply.

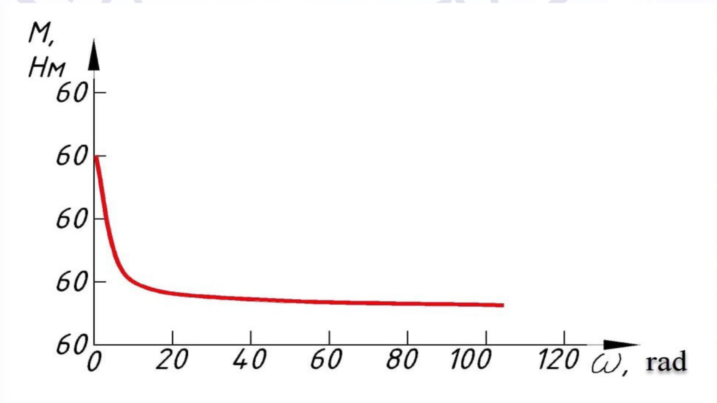

Fig. 6. Dependence of the maximum torque on the compressor shaft on the shaft rotation speed.

Conclusions: 1. Mathematical descriptions of the load moment were carried out using the example of a two-cylinder piston compressor SO-7A, analytical expressions for its determination were obtained.

2. Digital data and graphical analyses of the effective load moment of piston compressors, which can be used to determine the rational law of frequency regulation in the system of piston machines.

3. When regulating by changing the rotation frequency of the compressor drive, it is necessary to perform regulation in the frequency range exceeding the critical level of 15 rad/s.

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