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Determination of kinematic parameters of the compressor unit applied to low-pressure petroleum gas

Wyznaczanie parametrów kinematycznych agregatu sprężarkowego w zastosowaniach z LPG

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ABSTRACT: The collection and transportation of low-pressure gas released into the atmosphere from the oil fields is important in increasing gas production. This issue, which differs in its actuality, has both economic and pollution preveniton benefits. One such project, led by the UN Development Fund, was successfully implemented in the city of Siyazan, Azerbaijan. An electrically driven compressor unit of NQK-7/1-5 type with two oscillating cylinders is commonly used for such work. On visual inspection of the compressors undergoing repair, it was found that the seals in the pneumatic part and the elements of the cylinder-piston group are completely out of order, the gear teeth in the transmission part are broken, worn, the bearings are destroyed, the belts are torn, etc. Compressors sent for repair are almost always reassembled. A structural-kinematic analysis of the compressor installation was carried out to find out the causes of the existing malfunctions. For this purpose, a kinematic diagram of the conversion mechanism of the compressor – a slider-crank linkage mechanism was drawn up using the method of closed contours from projection equations relative to the X and Y axes, and based on the condition that the rotation angles of the piston and cylinder axes are equal, the instantaneous rotation angle, angular velocity and angular acceleration of the cylinder rotation, as well as instantaneous values of displacement, speed and acceleration of the piston inside the cylinder were found. It was shown that the designer's decision on the selection of the design scheme and determination of the product dimensions, which are one of the main issues in the design of the mechanism, should be made based on the results of the kinematic and dynamic analysis.

Key words: compressor, slider-crank linkage mechanism, displacement of the piston, angle of rotation, angular velocity, angular acceleration, pressure angle.

STRESZCZENIE: Gromadzenie i transport gazu LPG uwalnianego do atmosfery z pól naftowych są istotnymi kwestiami jeśli chodzi o zwiększenie wydobycia gazu. Zagadnienie to, zróżnicowane pod względem realiów, niesie za sobą korzyści z punktu widzenia ekonomicznego jak i zapobiegania zanieczyszczeniu środowiska. Jeden z takich projektów, prowadzony przez Fundusz Rozwoju ONZ, został zrealizowany, z pozytywnym wynikiem, w mieście Siyazan w Azerbejdżanie. Podczas wykonywania takich prac najczęściej stosowany jest agregat sprężarkowy z napędem elektrycznym typu NQK-7/1-5 z dwoma cylindrami oscylacyjnymi. Dokonując oględzin sprężarek trafiających do remontu okazało się, że uszczelnienia w części pneumatycznej oraz elementy grupy cylinder-tłok są całkowicie niesprawne, zęby kół zębatych w części przekładniowej są połamane i zużyte, łożyska zniszczone, a pasy rozerwane itp. Sprężarki kierowane do remontu są prawie zawsze ponownie montowane. W celu poznania przyczyn powstałych usterek, przeprowadzono analizę konstrukcyjno-kinematyczną instalacji sprężarkowej. W tym celu sporządzono schemat kinematyczny mechanizmu przekładniowego sprężarki, czyli mechanizmu korbowo-wodzikowego, metodą zamkniętych konturów z równań projekcyjnych względem osi X i Y, a opierając się na założeniu, że kąty obrotu osi tłoka i cylindra są równe, znaleziono wartości chwilowego kąta obrotu, prędkości kątowej i przyspieszenia kątowego obrotu cylindra oraz chwilowe wartości przemieszczenia, prędkości i przyspieszenia tłoka wewnątrz cylindra. Wykazano, że wyniki analizy kinematycznej i dynamicznej powinny stanowić podstawę przy wyborze przez konstruktora schematu konstrukcyjnego i określaniu wymiarów produktów, co zwykle stwarza największe problemy podczas projektowania mechanizmu.

Słowa kluczowe: sprężarka, mechanizm korbowo-wodzikowy, przemieszczenie tłoka, kąt obrotu, prędkość kątowa, przyspieszenie kątowe, kąt nacisku.

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Introduction

The collection of gas released into the atmosphere from oil-producing fields is considered to be one of the ways to provide the country's population with gas. It is known that when the amount of gas released from oil in wells is small, it is released into the atmosphere. This causes environmental pollution on the one hand and economic losses on the other. The supply of wells located at close distances to each other in a single system by means of compressors is an urgent issue today. In oil producing countries, this is done through various types of compressors (Shabana, 2010; Aliyev 2015; Aliyev and Aliyeva, 2017).

In the conditions of Azerbaijan, an electric drive compressor unit with two oscillating cylinders of NQK-7/1-5 type is widely used for collecting and transporting low-pressure oil gas inside the mine (Figure 1).

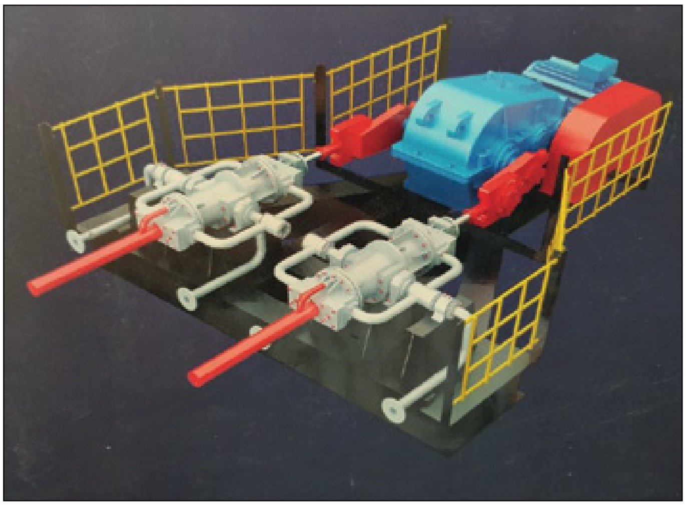

Figure 1. General view of double-acting piston compressor with two oscillating cylinders

Rysunek 1. Widok ogólny sprężarki tłokowej dwustronnego działania z dwoma cylindrami oscylacyjnymi

This unit has a capacity of 7 m^3/min and sucks in the total gas at a pressure of 0.08–0.12 MPa, raising its pressure to 0.5 MPa. The nominal diameter of the cylinder is 300 mm, the piston displacement is 630 mm. It is connected to the motor through a V-belt transmission and reducer. The compressor drive uses a two-stage Novikov transmission gear reducer with a transmission number equal to 16.94. In order to ensure the productivity within the specified limits, the number of belt transmissions was 0.89. The use of V-belt transmission reduces the dynamic forces that may arise during operation, absorbs shocks and protects the structure (Janahmadov et al., 2016; Aliyev, 2022; Rahimova and Mansurova, 2022). Four V-belts are used to transfer the 75 KW power generated by the engine to the working machine.

Visual inspection of compressors sent to repair showed complete damage to the clamps, piston-cylinder group elements in the pneumatic part of the compressor, broken teeth of the gears of the reducer in the drive part, chipped pads, broken belts, etc. Repaired compressors are almost always reassembled. To investigate the causes of all these malfunctions, it is required to carry out a structural, kinematic and dynamic analysis of the compressor unit.

Setting the issue

The converter mechanism of the compressor is made up of a slider-crank linkage mechanism. Instead of the generally accepted crosshead, an oscillating cylinder was used to give the piston a forward movement (Figure 2).

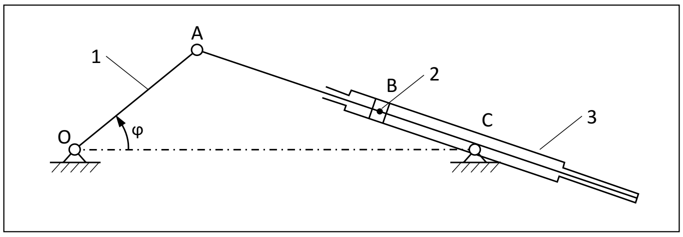

Figure 2. Structural scheme of the mechanism; 1 – crank, 2 – piston rod, 3 – cylinder

Rysunek 2. Schemat strukturalny mechanizmu; 1 – korba, 2 – trzon tłoka, 3 – cylinder

In order to have the same volume in the left and right chambers of the cylinder, the same driving rod of the piston was installed in the right chamber. This also performs a balancing function.

Two two-cylinder, two-acting piston-cylinder groups were applied to make more efficient use of the invested power. In order to adjust the dual movements of the piston in the cylinders, i.e., to perform the suction and tapping processes in each cycle of the crank, the movement of the crank is shifted 90°. When "suck-knock" tactics occur in cylinder I, "knock-suck" tactics are performed in cylinder II.

Solution of the issue

In order to determine the parameters of the dynamic model of the mechanism, it is necessary to calculate its kinematic parameters. (Frolov et al., 1987; Janahmadov et al., 2016). The first transfer function for mechanisms with a degree of mobility of the order of $w = 1$ is obtained by differentiating the state functions of the clauses according to the generalized coordinate. This can be determined by the method of closed

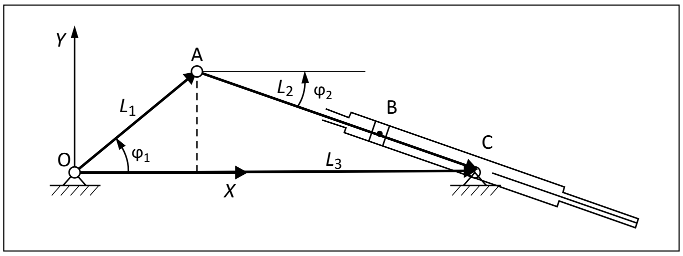

Figure 3. Calculation scheme for determining kinematic parameters (description in the text below)

Rysunek 3. Schemat obliczeń w celu ustalenia parametrów kinematycznych (opis w tekście poniżej)

contours, in which the clauses of the mechanism are expressed in terms of vectors (Figure 3).

When using the closed contour method, we get from the projection equation with respect to the *Y*-axis:

$$\mathrm{tg}\varphi_2 = \frac{L_1 \sin\varphi_1}{L_3 - L_1 \cos\varphi_1} \tag{1}$$

The angles of rotation of the axes of the piston and cylinder are equal.

$$\varphi_3 = \varphi_2 \tag{2}$$

From the projection of the clauses with respect to the *X*-axis, we get:

$$L_2 = \frac{L_3 - L_1 \cos\varphi_1}{\cos\varphi_2} \tag{3}$$

where L_3 is the distance between the centers.

Let's denote the right side of the expression (1) with $k(\varphi_1)$:

$$k(\varphi_1) = \frac{L_1 \sin\varphi_1}{L_3 - L_1 \cos\varphi_1}$$

From the joint solution of the expressions (1) and (3), we can determine the length L_2 variable.

$$L_2 = \frac{L_1 \sin\varphi_1}{\sqrt{\dfrac{k^2(\varphi_1)}{1 + k^2(\varphi_1)}}} \tag{4}$$

The way of differentiating the angular velocity rotation equation (2) of the cylinder can be derived from:

$$\omega_3 = \frac{d\varphi_3}{dt} = \frac{d\varphi_3}{d\varphi_1} \cdot \frac{d\varphi_1}{dt} = \frac{d\varphi_3}{d\varphi_1} \cdot \omega_1 \tag{5}$$

where:

$U_{31} = d\varphi_3/d\varphi_1$

U_{31} – is first derivative function, and ω_1 – leading is the angular velocity of the dam.

In most cases, when designing machines and mechanisms, the dependence of the generalized coordinate φ_1 on time is determined after a dynamic study of the movement, taking into account the characteristics of the forces, masses and moments of inertia acting on the points of the machine unit. In such cases, the actions of the output and intermediate points are studied in two stages: at the first stage, the dependence of the kinematic parameters of points and points on the generalized coordinate is determined, that is, relative functions (state function and transfer function of the mechanism) are studied, and at the second stage, the law of time-dependent change of the kinematic parameters.

From the expression (1), we define the φ_2:

$$\varphi_2 = \varphi_3 = \mathrm{arctg}\frac{L_1 \sin\varphi_1}{L_3 - L_1 \cos\varphi_1} \tag{6}$$

we use the expression (5) to define the expressions $(d\varphi_3)/(d\varphi_1)$ or $(d\varphi_2)/(d\varphi_1)$:

$$\omega_3 = \frac{d\varphi_3}{d\varphi_1} \cdot \omega_1 = \frac{L_1(L_3 \cos\varphi_1 - L_1)}{L_1^2 + L_3^2 - 2L_1L_3 \cos\varphi_1} \cdot \omega_1 \tag{7}$$

We also calculate the angular acceleration in an analogous manner.

$$\varepsilon_3 = \frac{d\omega_3}{d\varphi_1} \cdot \omega_1 = \frac{L_1L_3(L_1^2 - L_3^2)\sin\varphi_1}{(L_1^2 + L_3^2 - 2L_1L_3 \cos\varphi_1)^2} \cdot \omega_1^2 \tag{8}$$

Discussion of the solution

Based on the specified statements, a program was compiled and the following results were obtained according to the procedures performed. The dependence of the angle of the φ_2 on the angle of the φ_1 is described in Figure 4. The movement follows a sine wave in the group of cylinders I and a cosine wave in the group of cylinders II. In cylinders I and II, as mentioned above, the movement of the pistons is shifted in phase by 90°.

Graphs of the dependence of angular velocities on the angle of rotation of the crank during the rotation of the rollers around the support are shown in Figure 5. It can be seen from the graphs that in the slider on which cylinder 1 is located, the direction of ω_2 changes starting at 78° and returning to the previous one after 282°. These figures are 192° and 348° respectively. At these angles, the value of the pressure angle at the point of structural connection of the crank and the slider becomes zero, and the value of the transfer angle reaches a maximum, resulting in the most favorable connection conditions. In order for the mechanism to work in accordance with the conditions of transmission of forces in kinematic pairs, the maximum value of the pressure angle in rotational kinematic pairs should not exceed the permissible value. This value is about 30° per working run.

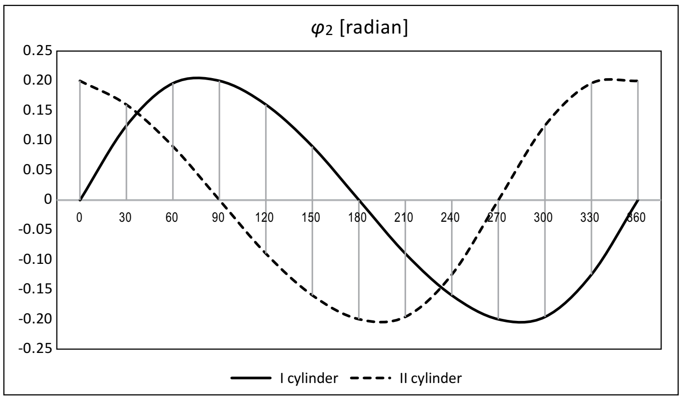

Figure 4. Dependence of the angle of the φ_2 on the angle of the φ_1

Rysunek 4. Zależność kąta φ_2 od kąta φ_1

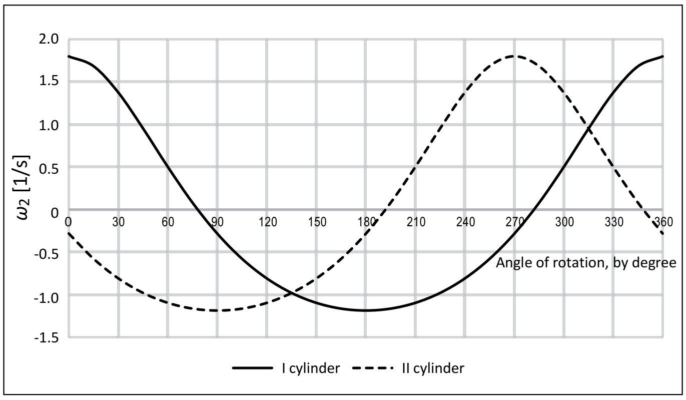

Figure 5. Graphs of the dependence of angular velocities on the angle of rotation of the crank during the rotation of the rollers around the support

Rysunek 5. Wykresy zależności prędkości kątowych od kąta obrotu tłoka podczas obrotu wałków wokół podpory

Since there is always friction in real mechanisms, riveting should occur at less than 90° of pressure angle. For initial calculations, only mechanisms with a capacity of $[\alpha] = 45° – 60°$ are accepted for rotation pairs, while for progression pairs – mechanisms with a capacity of $[\alpha] = 30° – 45°$. It should be noted that in the so-called "dead" states of the mechanism, the pressure angles are at the point of $\alpha = 90°$. In statics, the mechanism in this case can be riveted, while in dynamics the mechanism passes through these states using the kinetic energy accumulated by the moving dams (Frolov et al., 1987; Applied Mechanics, 2018).

The graphs of the angular acceleration changes of the sliders are respectively described in Figure 6.

As the piston moves inside the cylinder, the length of the slider changes. The variation of L_2 was calculated from the expression (4) and is shown in Figure 7. Depending on the time (or from the rotation angle of the crank arm), the current displacements of the piston have been determined. The difference between the maximum and minimum length is 630 mm, which determines the travel path of the piston.

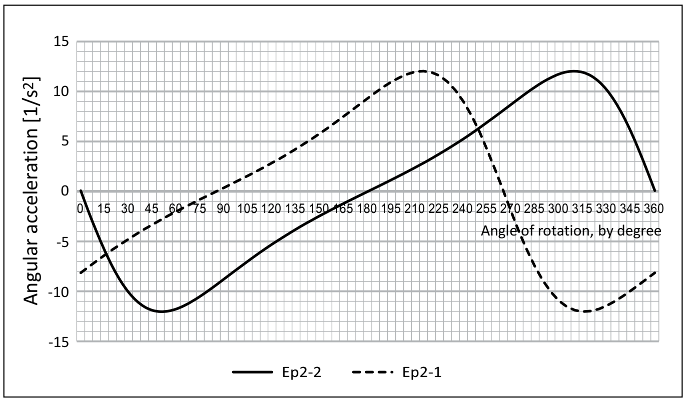

Figure 6. Graphs of the dependence of angular acceleration on the angle of rotation of the crank during the rotation of the cylinder around the support

Rysunek 6. Wykresy zależności przyspieszenia kątowego od kąta obrotu tłoka podczas obrotu cylindra wokół podpory

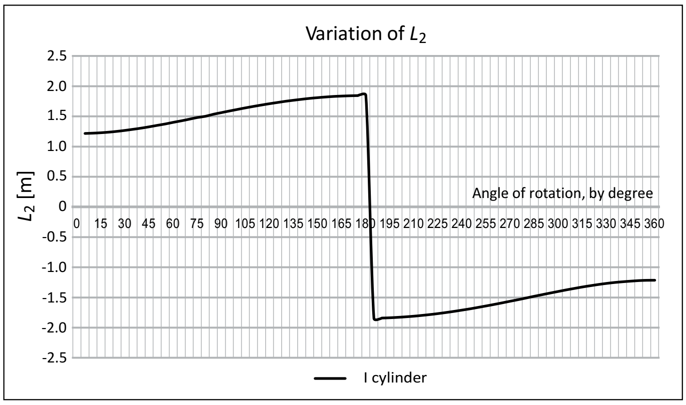

Figure 7. Current displacements of the length of the slider with the piston

Rysunek 7. Bieżące przemieszczenia na długości wodzika z tłokiem

The speed of the piston with respect to the cylinder can be found by differentiating the variable distance (L_2's) between points A and C.

$$V_{23} = \frac{dL_2}{dt} = \frac{dL_2}{d\varphi_1} \cdot \frac{d\varphi_1}{dt} = \frac{dL_2}{d\varphi_1} \cdot \omega_1$$

By analogy, the speed of the piston is also determined. Graphs of changes in velocity and acceleration are presented in Figures 8 and 9, respectively.

Determination of the values of kinematic parameters enables a dynamic analysis of the machine unit as a whole to be carried out.

Results

The instantaneous angle of rotation, angular velocity and angular acceleration of the rotation of the compressor cylinder

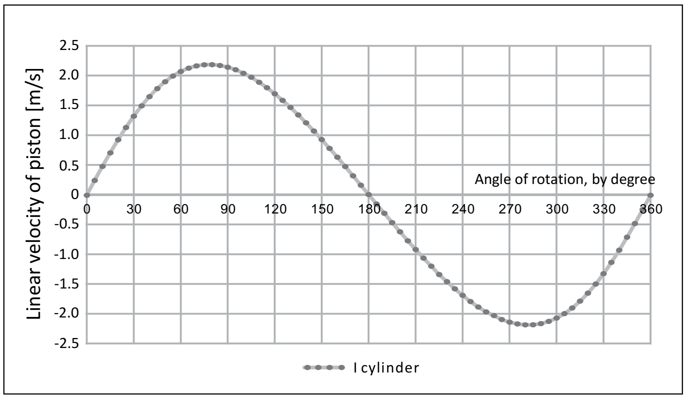

Figure 8. Graph of velocity change

Rysunek 8. Wykres zmian prędkości

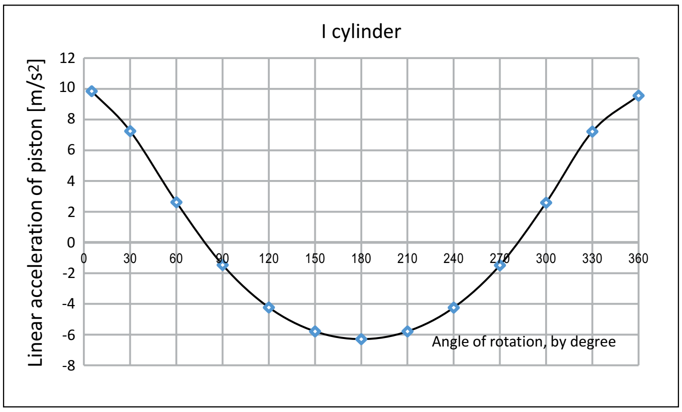

Figure 9. Graph of acceleration change

Rysunek 9. Wykres zmian przyspieszenia

around the support, as well as the instantaneous values of the displacement, velocity and acceleration of the piston inside the cylinder are found, which is very important for conducting dynamic analysis of the mechanism.

The results of the structural and kinematic analysis of the compressor unit allow the constructor to decide on the selection of the structural scheme, which is one of the main issues in the design of the mechanism, and the successful implementation of the procedure for determining the dimensions of the dams.

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