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Comparative analysis of different variants of installing rotary counterweights on the crank of the new design of beamless pumping unit

Analiza porównawcza różnych wariantów montażu przeciwwag obrotowych na korbie nowego rozwiązania konstrukcyjnego zespołów pompowych bez żurawia

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ABSTRACT: The article presents a comparative analysis of different variants of installing rotary counterweights on the crank in the mechanical drive of the new design of the pumping unit used in oil production. It also addresses the assessment of torque on the output shaft of the gearbox and the balancing coefficient of the mechanical drive. In examining the rotary balancing approach for the new design of the pumping unit, various options for installing counterweights on the crank during rotary balancing were analyzed. Analytical expressions were proposed to determine the torques on the output shaft of the gearbox. Calculations based on the technical parameters of classic pumping units of the CK series revealed that the installation of counterweights on the crank during rotary balancing in the new design of the beamless pumping unit machine significantly affects the torque on the output shaft of the gearbox and the balancing of the pumping unit. They also revealed that although the torque on the output shaft of the gearbox is small in the pumping unit equipped with two counterweights of the same weight and located at the same distance, in this configuration, the output shaft of the gearbox experiences a substantial cantilever load due to excessive weight of the counterweights, leading to a significant reduction in the durability of the gearbox. In the other two options, when installing a single counterweight on the crank, the torque on the output shaft of the pumping unit's gearbox is approximately from 5 to 10% greater than in the first variant, resulting in additional energy losses. In the pumping machine equipped with two counterweights of equal weight but located at different distances from the center of rotation, the torque on the output shaft of the gearbox is reduced, similar to the first variant. However, due to the weight of the counterweights, it also imposes a substantial cantilever load on the output shaft, leading to a significant reduction in the service life of the gearbox. Additionally, in this option, unlike the first, the balancing coefficient is approximately 3% less.

Key words: counterweight, crank, pumping unit, balancing, rotor, rods column.

STRESZCZENIE: W artykule przedstawiono analizę porównawczą różnych wariantów montażu przeciwwag obrotowych na korbach napędów mechanicznych w nowych rozwiązaniach konstrukcyjnych zespołów pompowych używanych w eksploatacji ropy naftowej. Poruszono również kwestię oceny momentu obrotowego na wale wyjściowym przekładni i współczynnika wyważenia napędu mechanicznego. W przypadku metody wyważania obrotowego nowego rozwiązania konstrukcyjnego zespołu pompowego przeanalizowano różne opcje montażu przeciwwag na korbie podczas wyważania obrotowego oraz zaproponowano wyrażenia analityczne do określenia momentów obrotowych na wale wyjściowym przekładni. W wyniku obliczeń przeprowadzonych z wykorzystaniem parametrów technicznych klasycznych zespołów pompowych serii CK stwierdzono, że montaż przeciwwag na korbie podczas wyważania obrotowego w nowym rozwiązaniu konstrukcyjnym maszyny bez żurawia znacząco wpływa na moment obrotowy na wale wyjściowym przekładni i wyważenie zespołu pompowego. Stwierdzono także, że chociaż moment obrotowy na wale wyjściowym przekładni jest niewielki w zespole pompowym wyposażonym w dwie przeciwwagi o tej samej masie i umieszczone w tej samej odległości, w tej konfiguracji wał wyjściowy przekładni doświadcza znacznego obciążenia wspornikowego z powodu nadmiernej masy przeciwciężarów, co prowadzi do znacznego obniżenia trwałości przekładni. W pozostałych dwóch wariantach, przy montażu pojedynczej przeciwwagi na korbie, moment obrotowy na wale wyjściowym przekładni zespołu pompowego jest w przybliżeniu od 5 do 10% większy niż w pierwszym wariancie, co powoduje dodatkowe straty energii. W zespole pompowym wyposażonym w dwie przeciwwagi o równej masie, umieszczonych w różnych odległościach od środka obrotu, moment obrotowy na wale wyjściowym przekładni ulega zmniejszeniu, podobnie

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jak w pierwszym wariancie, ale ze względu na masę przeciwwag, występuje także duże obciążenie wspornikowe na wale wyjściowym, co prowadzi do znacznego skrócenia żywotności przekładni. Ponadto w tym wariancie, w przeciwieństwie do pierwszego wariantu, współczynnik wyważenia jest o około 3% niższy.

Słowa kluczowe: przeciwwaga, korba, zespół pompowy, wyważanie, wirnik, kolumna żerdzi.

Introduction

The oil industry stands as the most critical and notable sector at every stage of the economic development of any oil-producing country. Consequently, formulating the general development plan of a state and predicting its future stages necessitate a thorough consideration of the situation and opportunities within this field. Currently, the most common equipment used in mechanized oil extraction methods is the sucker rod pumping unit. Given this perspective, there exists a significant need to study the pumping units employed in oil extraction and to develop more innovative designs for them (Najafov, 2013; Gabor, 2015; Ziuzev and Tecle, 2022).

This equipment constitute an integral part of the oil production complex. Pumping units are designed to ensure the forward movement of a deep-well pump installed at the bottom of a well. In order to reduce energy losses, this equipment must have a perfect kinematic scheme.

Currently, one of the main trends in developing new mechanical drive designs for deep-well pumps is to give preference to beamless pumping units. The elimination of the walking beam and horse head provides several advantages, including a reduction in metal capacity and alleviation of requirements for a concrete platform for unit installation, among others (Dennis, 2001; Najafov, 2013; Elias and Rutácio, 2020).

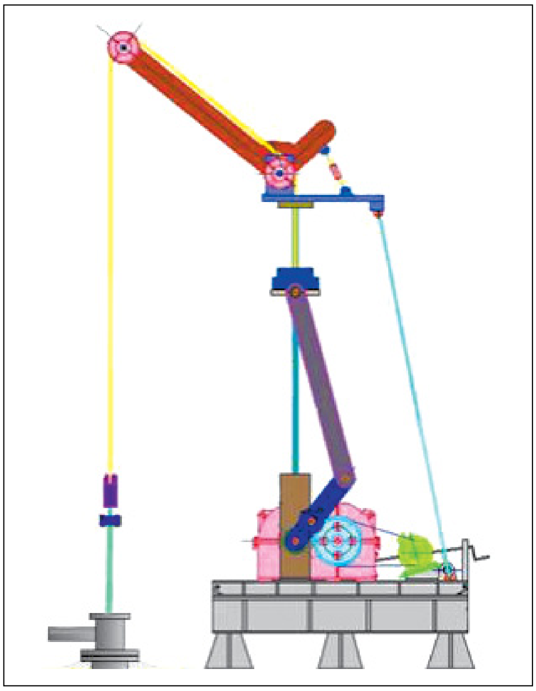

Figure 1. New design solution of the beamless sucker-rod pumping unit

Rysunek 1. Nowe rozwiązanie konstrukcyjne zespołów pomp żerdziowych bez żurawia

The department of “Machine Design and Industrial Technologies” at Azerbaijan Technical University has developed a new design solution for the beamless sucker-rod pumping unit. This unit has a smaller metal capacity, more compact overall dimensions, and a suspension point movement pattern that more closely adheres to the harmonic law (Sherif et al., 2021). Moreover, this original design has received approval from the Eurasian Patent Organization under No. 039650 in 2022 (Abdullaev et al., 2022) and the Intellectual Property Agency of the Republic of Azerbaijan under No. a2019-0162 in 2021 (Abdullaev et al., 2021). The new pumping unit comprises a rope-block and crank-slide converter mechanism (Figure 1).

Formulation of the problem

The experience with operating pumping units shows that they work under a cyclically changing load. Specifically, during the upstroke movement of the rods suspension point, the pumping unit lifts both the liquid column and the rods column. At this juncture, the engine of the pumping unit bears the maximum load. Conversely, during the downstroke movement, as the rods column descends under its own gravity, the pumping unit is loaded only by the weight of the rods column. At this phase, the engine does not operate; instead, it works in generator mode by absorbing energy.

This situation leads to uneven loading of both the engine and the gearbox, causing a significant decline in their performance. In such cases, the torque on the output shaft of the gearbox follows a sinusoidal pattern, which is undesirable (Figure 2a) (Liu and Liu, 2010; Ahmedov and Hajiyev, 2020; Ahmedov, et al., 2021, 2023; Yin et al., 2023). Therefore, in order to achieve a more uniform load on both the engine and the gearbox, pumping units are usually balanced (Figure 2b). Mechanical (using counterweights) and, less commonly, pneumatic (gas or air pressure) methods are primarily employed for balancing pumping units. In units balanced with a mechanical method, counterweights can be placed on the walking beam, the crank, or both structural elements. Accordingly, these balancing methods are referred to as beam, rotary or combined.

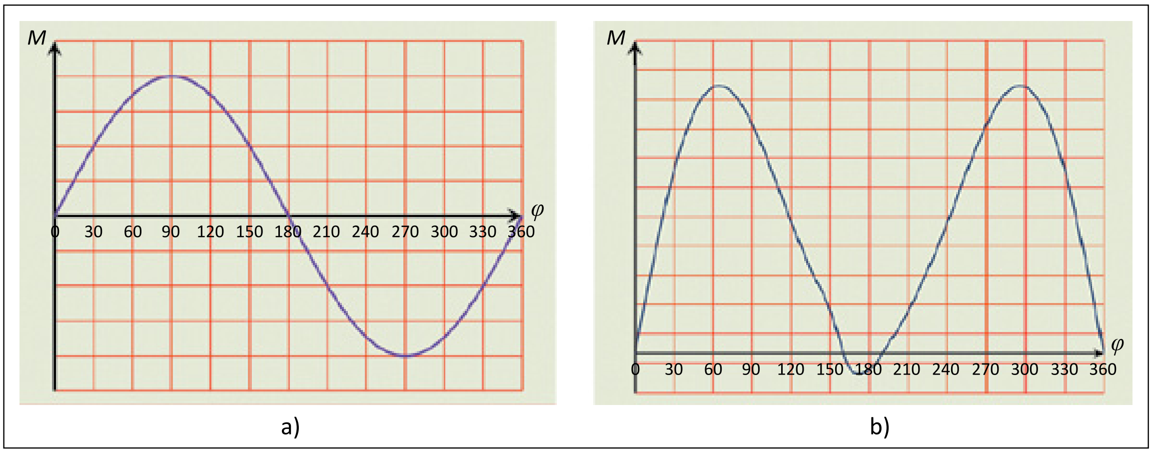

Figure 2. The diagrams of the torque change on the output shaft of the reducer; a) on an unbalanced pumping unit, b) on a balanced pumping unit

Rysunek 2. Wykresy zmiany momentu obrotowego na wale wyjściowym reduktora; a) na niewyważonym zespole pompowym, b) na wyważonym zespole pompowym

For beam pumping units with a substantial load capacity (80–200 kN), the rotor balancing method is usually the preferred choice (Ganzulenko and Petkova, 2023; Shishlyannikov et al., 2023).

Solution of the problem

In the rotary balancing method, balancing counterweights are mounted on the cranks. The weight of these counterweights is determined by ensuring equality of work done during the upward and downward movement of the rods column. In the rotary balancing method, the weight of balancing counterweights is standardized, and conventionally, these loads are divided into main and auxiliary loads. The placement of these counterweights on the cranks and their quantity significantly influences the maximum balancing moment on the output shaft of the reducer.

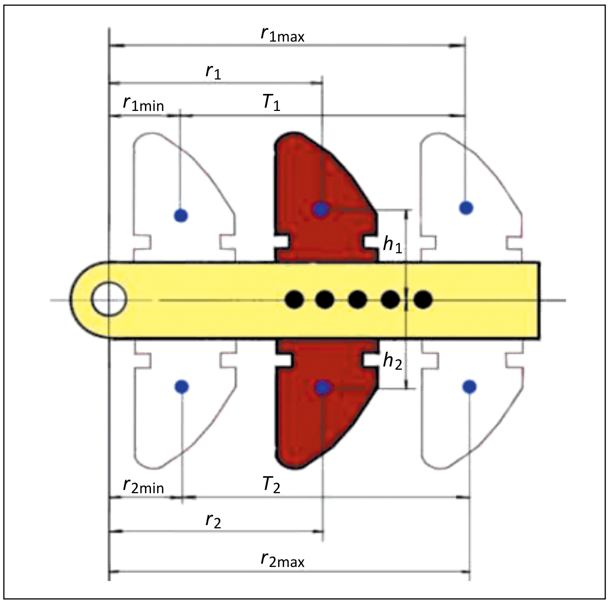

Figure 3. Locations of the counterweights on the crank

Rysunek 3. Lokalizacja przeciwwag na korbie

The arrangement of the main counterweights on the crank is shown in Figure 3. As can be seen from the scheme, the maximum travel of counterweights (T), their maximum distance from the rotation center of the crank (r_{1max}, r_{2max}) and the vertical components of the centers of gravity of counterweights (h_1, h_2) depend on the type of used counterweights. The choice of counterweights type depends on the construction of the crank of the pumping unit.

The maximum displacement of the center of gravity of the main counterweights mounted on the crank is usually known. Smaller counterweights on the same crank may have their center of gravity further away from the crank's axis of rotation, resulting in a much larger travel displacement. When the same counterweight is used on smaller cranks, the maximum distance and travel of the counterweight will be smaller. Figure 4 shows various configurations for the placement of the main counterweights on the cranks.

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If two counterweights are with the same weight and mounted at the same distance on the crank, then the torque on the output shaft of the reducer is determined by the following expression (Figure 4a):

$$M_b = F_1 \cdot r - G_d \cdot r_d \cdot \sin\varphi - 2G_r \cdot r_{1,2} \sin\varphi \qquad (1)$$

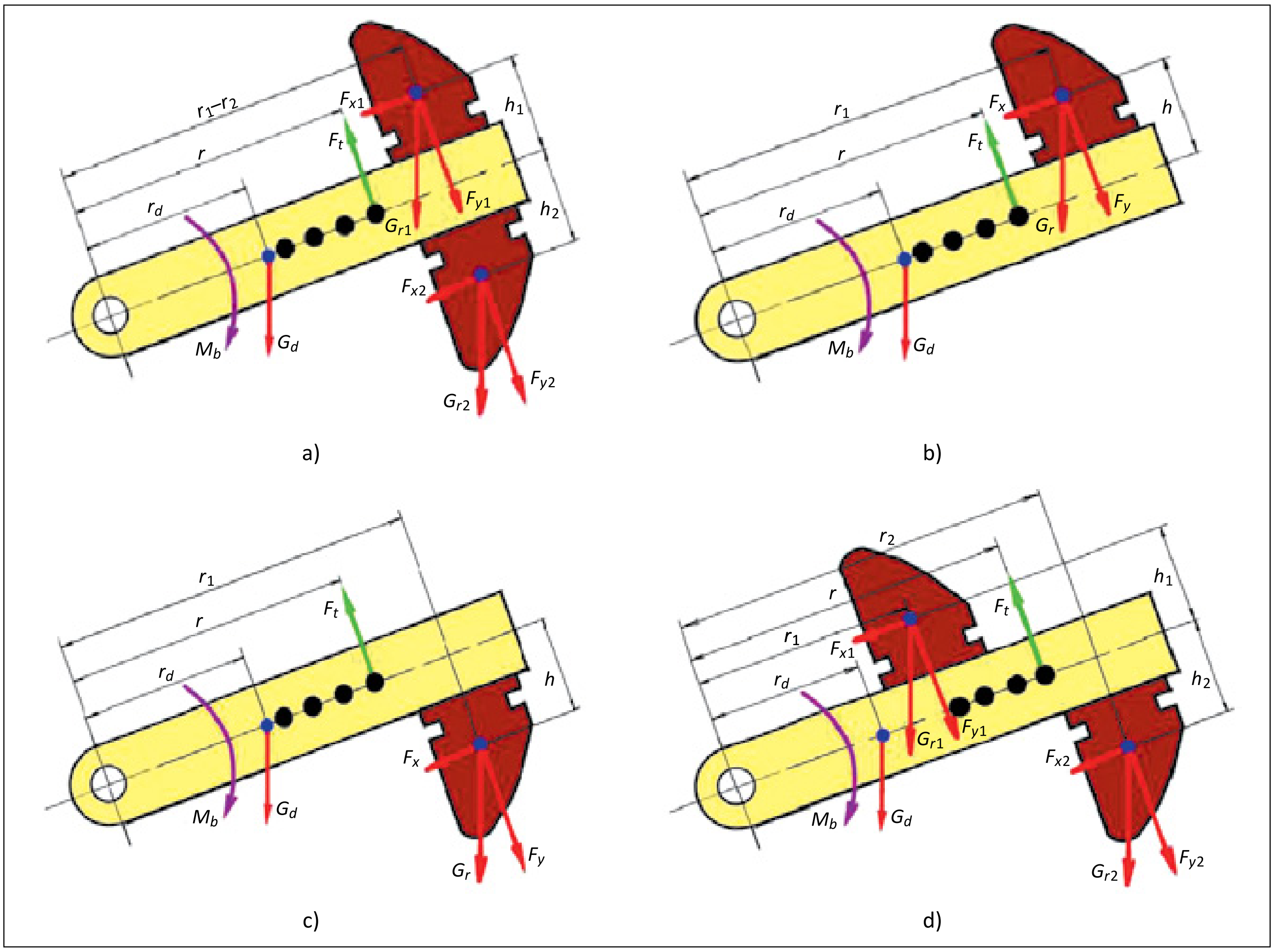


Figure 4. Various configurations for the placement of the main counterweights on the cranks

Rysunek 4. Różne konfiguracje umieszczenia głównych przeciwwag na korbach

If only one counterweight with the same weight is mounted at the same distance on the crank, then the torque on the output shaft of the reducer is determined by the following expression:

- if counterweight is mounted on the upper side of the crank (Figure 4b),

$$M_b = F_1 \cdot r - G_d \cdot r_d \cdot \sin\varphi - G_{r1}(r_1 \sin\varphi - h_1 \cos\varphi) \quad (2)$$

- if counterweight is mounted on the underside of the crank (Figure 4c).

$$M_b = F_1 \cdot r - G_d \cdot r_d \cdot \sin\varphi - G_{r2}(r_2 \sin\varphi + h_2 \cos\varphi) \quad (3)$$

If two counterweights with the same weight, but located at different distances, are mounted on the cranks, then the moment on the output shaft of the reducer of the pumping unit is determined by the following expression (Figure 4d):

$$M_b = F_1 \cdot r - G_d \cdot r_d \cdot \sin\varphi - G_{r1}(r_1 \sin\varphi - h_1 \cos\varphi) - \\ - G_{r2}(r_2 \sin\varphi + h_2 \cos\varphi) \quad (4)$$

where:

F_t – circumferential force on the crank,
G_d – weight of the crank,
G_{r1}, G_{r2} – weights of the counterweights,
r – the distance from the rotation center of the crank to the point where the mounted pitman arm,
r_d – the distance from the rotation center of the crank to its gravity center,
r_1, r_2 – distances from the rotation center of the crank to the gravity center of counterweights,
h_1, h_2 – vertical components of the gravity centers of the counterweights.

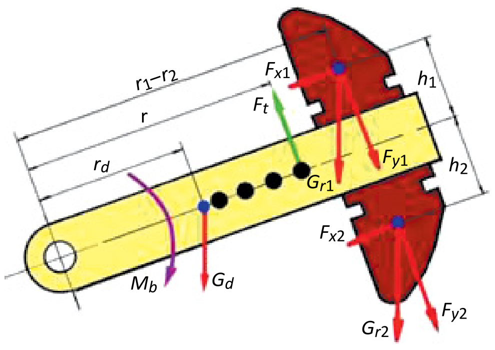
In order to quantify the impact of various main rotary counterweights placement options on the balancing of the pumping unit, calculations were made using the technical parameters of the CK 6-2,1-2500 beam pumping unit.

As can be seen from the results of the calculations, in the first variant, i.e., in the pumping unit equipped with two counterweights of the same weight and located at the same distance, the torque on the output shaft of the reducer at different values of the distance from the rotation center of the crank to the gravity center of the counterweights ($r_1 = r_2 = r_{max}$; $r_1 = r_2 = 0.8r_{max}$; $r_1 = r_2 = 0.6r_{max}$; $r_1 = r_2 = 0.4r_{max}$; $r_1 = r_2 = 0.2r_{max}$) varies from 24 065 to 30 833 N · m during the upward movement of the rods suspension point, and from 24 600 to 32 149 N · m during downward movement. Correspondingly,

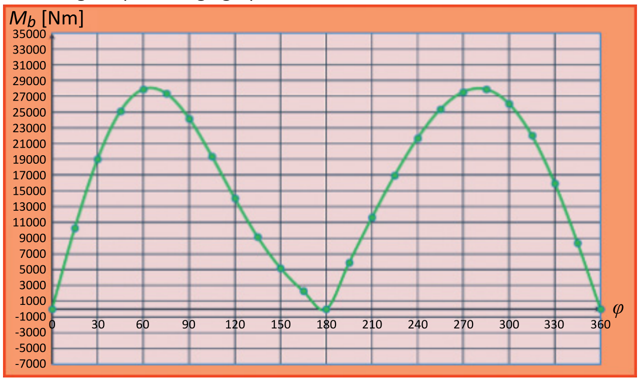
Table 1. Balancing of the pumping unit equipped with two counterweights located at the same distance of each crank
Tabela 1. Wyważanie zespołu pompującego wyposażonego w dwie przeciwwagi umieszczone w tej samej odległości od każdej korby

$r_1 = 0.548 r_{max}$ $r_2 = 0.548 r_{max}$	$M_{up}^{max} = 27892\,N \cdot m$ $M_{down}^{max} = 27880\,N \cdot m$	Balancing coefficent: $K_{bl} = 0.999$

1st variant:



Balancing torque change graph:



the balancing coefficent K_{bl} (perfect balancing when $M_{up} = M_{down}$) of the pumping unit varies accordingly from 0.748 to 0.999. In this rotary balancing method, the pumping unit is almost in complete equilibrium at the distance $r_1 = r_2 = 0.548 r_{max}$ to the gravity center of the counterweights, and in this case, the maximum torque on the output shaft of the reducer is 27 892 N · m (Table 1).

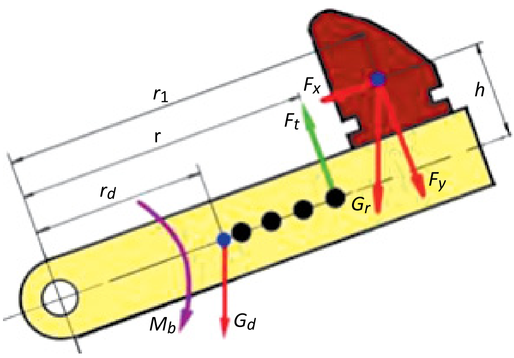
In the second studied variant, i.e., in the pumping unit equipped with one counterweigh on the upper side of each crank, the torque on the output shaft of the reducer at different values of the distance from the rotation center of the crank to the gravity center of the counterweight ($r_1 = r_{max}$; $r_1 = 0.8 r_{max}$; $r_1 = 0.6 r_{max}$; $r_1 = 0.4 r_{max}$; $r_1 = 0.2 r_{max}$) varies from 29 458 to 32 842 N · m during the upward movement of the rods suspension point, and from 24 258 to 28 033 N · m during downward movement. Correspondingly, the balancing coefficient of the pumping unit varies accordingly from 0.739 to 0.952. In this rotary balancing method, the pumping unit is almost in complete equilibrium at the distance $r_1 = r_{max}$ to the gravity center of the counterweights, and in this case the maximum torque on the output shaft of the reducer is 29 458 N · m (Table 2).

In the third studied variant, i.e., in the pumping unit equipped with one counterweight on the underside of each crank, the torque on the output shaft of the reducer at different values of the distance from the rotation center of the crank to the gravity center of the counterweight ($r_2 = r_{max}$; $r_2 = 0.8 r_{max}$; $r_2 = 0.6 r_{max}$; $r_2 = 0.4 r_{max}$; $r_2 = 0.2 r_{max}$) varies from 27 220 to 30 994 N · m during the upward movement of the rods suspension point, and from 23 187 to 27 095 N · m during downward movement. Correspondingly, the balancing coefficient of the pumping unit varies accordingly from 0.748 to 0.995. In this rotary balancing method, the pumping unit is almost in com-

Table 2. Balancing of the pumping unit equipped with one counterweight in the upper side of each crank
Tabela 2. Wyważanie zespołu pompującego wyposażonego w jedną przeciwwagę umieszczoną w górnej części każdej korby

$r_1 = r_{max}$	$M_{up}^{max} = 29458\,N \cdot m$ $M_{down}^{max} = 28033\,N \cdot m$	Balancing coefficent: $K_{bl} = 0.952$

2nd variant:



Balancing torque change graph:

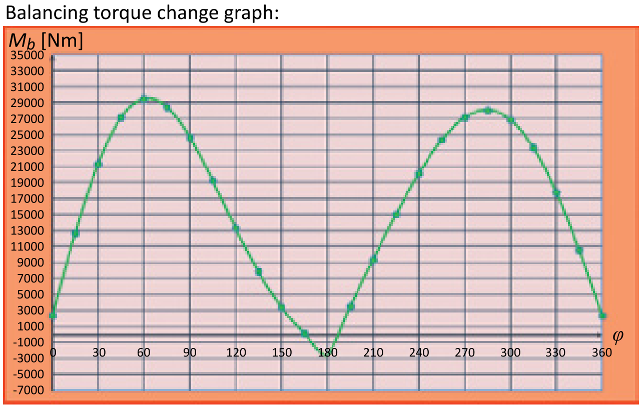
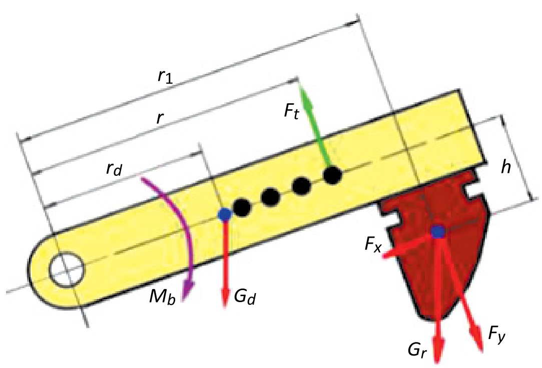


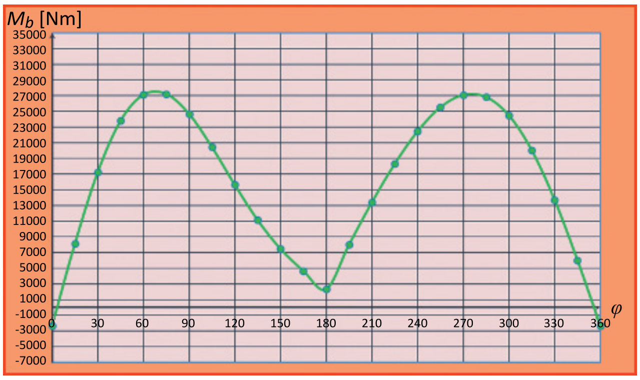
Table 3. Balancing of the pumping unit equipped with one counterweight on the underside of each crank
Tabela 3. Wyważanie zespołu pompującego wyposażonego w jedną przeciwwagę umieszczoną w dolnej części każdej korby

$r_2 = r_{max}$	$M_{up}^{max} = 30994\,N \cdot m$ $M_{down}^{max} = 27095\,N \cdot m$	Balancing coefficent: $K_{bl} = 0.995$

3rd variant:



Balancing torque change graph:



plete equilibrium at the distance $r_2 = r_{max}$ to the gravity center of the counterweights, and in this case the maximum torque on the output shaft of the reducer is 30 994 N · m (Table 3).

And finally, in last – the fourth variant, i.e., in the pumping unit equipped with two counterweights of the same weight but located at different distances, the torque on the output shaft of the reducer at different values of the distance from the rotation center of the crank to the gravity center of the counterweights ($r_1 = 0.8r_{max}$ & $r_2 = r_{max}$; $r_1 = 0.6r_{max}$ & $r_2 = r_{max}$; $r_1 = 0.4r_{max}$ & $r_2 = r_{max}$; $r_1 = 0.2r_{max}$ & $r_2 = r_{max}$) varies from 24 911 to 27 449 N · m during the upward movement of the rods suspension point, and from 28 374 to 31 205 N · m during downward movement. Balancing coefficent of the pumping unit varies accordingly from 0.798 to 0.967. In this rotary balancing method, the pumping unit is almost in complete equilibrium at the distance $r_1 = 0.2r_{max}$ and $r_2 = r_{max}$ to the gravity center of the counterweights, and in this case the maximum torque on the output shaft of the reducer is 28 374 N · m (Table 4).

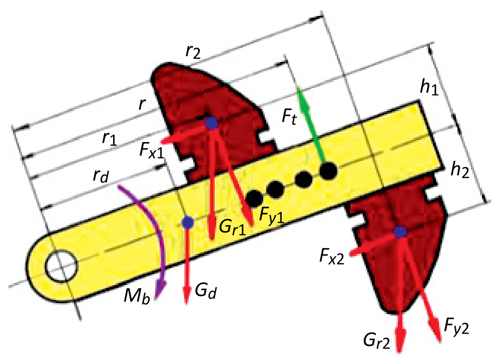
Summary

1. The torque on the output shaft of the reducer in the mechanical transmission of the new design solution of the sucker-rod pumping unit, as well as its balance ratio during operation, depends on the installation variants of the rotary counterweights on the crank.
2. Results from calculations reveal that in the pumping unit equipped with two identical counterweights located at the same distance, the maximum torque on the output shaft of the reducer is 27 892 N · m, and the balancing coefficient is up to 0.999. However, in this version, the high weight

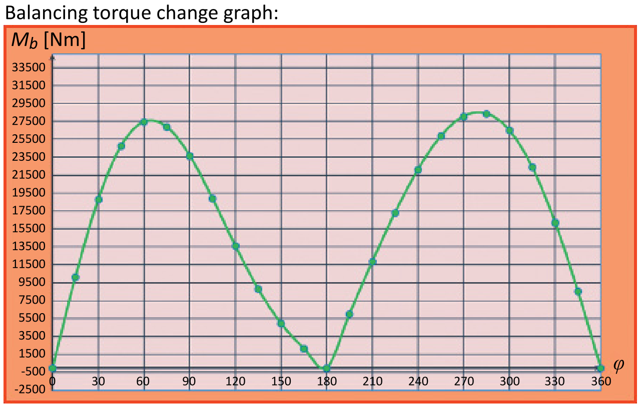
Table 4. Balancing of the pumping unit equipped with two counterweights located at different distances
Tabela 4. Wyważanie zespołu pompującego wyposażonego w dwie przeciwwagi umieszczone w różnych odległościach

$r_1 = 0.2r_{max}$ $r_2 = r_{max}$	$M_{up}^{max} = 27442\,N \cdot m$ $M_{down}^{max} = 28374\,N \cdot m$	Balancing coefficent: $K_{bl} = 0.967$

4th variant:



Balancing torque change graph:



of the counterweights leads to a significant cantilever load on the output shaft of the reducer, significantly reducing its longevity.

3. In both second and third variants, where pumping unit equipped with one counterweight located on the upper or underside of the crank, the maximum torques on the output shaft of the reducer are 29 458 N · m and 30 994 N · m, respectively, and the balancing coefficients are up to 0.952 and 0.995. However, in these variants, the torque on the output shaft of the gearbox is approximately 5 to 10% greater than in the first variant, resulting in additional energy losses.
4. In a pumping unit equipped with two same counterweights but located at different distances, the torque on the output shaft of the gearbox is 28 374 N · m, and the balancing coefficient is up to 0.967. In this variant, the torque on the output shaft of the gearbox is closer to the first variant, but due to the higher weight of the counterweights compared to the second and third variants, there is a substantial cantilever load on the output shaft of the gearbox. In addition, in this variant, unlike the first one, the balancing coefficient is approximately 3% less.

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