UDC [622.2/.8.002.5:539.3]:005.2 DOI: https://doi.org/10.15407/geotm2024.169.066

DEVELOPMENT OF METHODS FOR CALCULATING THE PARAMETERS OF STEEPLY INCLINED BELT CONVEYORS USED IN OPEN PIT MINING

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Abstract. As the depth of quarries grows and the overall angle of inclination of the sides increases, the transport support of stripping and mining operations becomes especially important due to the problems of using road transport. In this regard, steeply inclined conveyors (SIC) are particularly relevant for deep quarries. Steeply inclined belt conveyors are conveyors with an angle of inclination of more than 30 degrees, the main types of them are: conveyors with partitions on the belt, with pockets on the belt, double-circuit with a clamping belt, tubular conveyors. The methods for calculating parameters of various types of steeply inclined conveyors, which can be used in open-pit mining, namely the parameters of the drive and the design of the framework were developed. The calculation of the power of the SIC drive depending on the height of the lift and the angle of inclination of the conveyor is presented. It is shown, that the power of the SIC drive is directly proportional to the lifting height, while the value of the SIC inclination angle has a small effect on the drive power. Tubular belt conveyors (TBC) are one of the most promising types of conveyors. However, due to the complex processes, which occur during the operation of TBC, when a tubular belt with a load moves along roller supports, TBC failures and their consequences differ from those of other conveyors. Recommendations for selecting the distance between roller supports on the linear and curved sections of the TBC route were developed. The calculations showed that the greatest load on the rollers of a tubular conveyor falls on the lower side and central rollers. At the same time, the maximum loads on the rollers in the curved section of the tubular conveyor increase by 20%, and the average service life in the curved section of the framework decreases by 43%, when a distance between the roller supports is 0.5 m. Therefore, it is recommended to reduce the distance between the roller supports on the curved section of the tubular conveyor by four times compared to the distance between the roller supports on the linear section of the conveyor framework. TBC failures can occur as a result of the tubular belt scrolling, which leads to dusting and shedding of the load. These phenomena can be prevented by selecting the design parameters of the TBC. The formula was obtained for calculating the maximum distance between the roller support of the TBC, which will not lead to the divergence of the sides of the tubular belt if the central roller of the first roller support is rotated by certain angle.

Keywords. steeply inclined belt conveyor, drive power, roller service life, distance between roller supports.

1. Introduction

Currently, steeply inclined belt conveyors are increasingly being used in various industries abroad and in Ukraine: mining, chemical, light industry and agriculture. In particular, as the depth of quarries grows and the overall angle of the sides increases, the mining situation becomes more complicated. Transport support for stripping and mining operations is of particular importance due to the problems of using road transport. In this regard, the issue of using a steeply inclined conveyor for deep quarries is very relevant [1, 2].

To effectively solve the problem of delivering load in quarries at a high angle, it is important to choose the right type and parameters of a steeply inclined conveyor taking into account the nature of the route's ascent, mining and geological conditions, load properties, etc. [3].

Steeply inclined belt conveyors are conveyors with a belt inclination angle of more than 30 degrees. The main components of the conveyor are: an unloading device, a drive, a conveyor framework with roller bearings, a belt, a loading device, and a tensioning device.

Steeply inclined belt conveyors (SIC) are mainly used for transporting bulk load to a certain height. The following types of conveyors are used depending on the lifting height, the angle of inclination of the conveyor and the particle size distribution of the load transported by the SIC:

– with partitions on the belt;
– double-circuit with a clamping belt;
– with pockets on the belt or special containers;
– tubular conveyors.

At open pit mining, the criteria for selecting the type of steeply inclined conveyor are:

– angle of inclination of the conveyor;
– stability of the transported load;
– dynamic coefficient when the load moves along the conveyor framework;
– energy consumption;
– safety;
– environmental friendliness.

Currently, the most commonly used conveyors in quarries around the world are those with a clamping belt [4], the use of which for large lump rocks is associated with certain difficulties [5, 6]. Steeply inclined tube conveyors have a number of advantages. They can transport load along curved routes, have no harmful impact on the environment, and require fewer reloading points. The main disadvantages of TBC are the increased load on the rollers and belt torsion.

The aim of the study is to develop methods for calculating the main parameters of steeply inclined tubular conveyors used in open-pit mining.

2. Methods

Methods for calculating the parameters of various types of steeply inclined conveyors that can be used at open-pit mining were developed. Namely, the drive power and the distance between roller supports of the TBC framework.

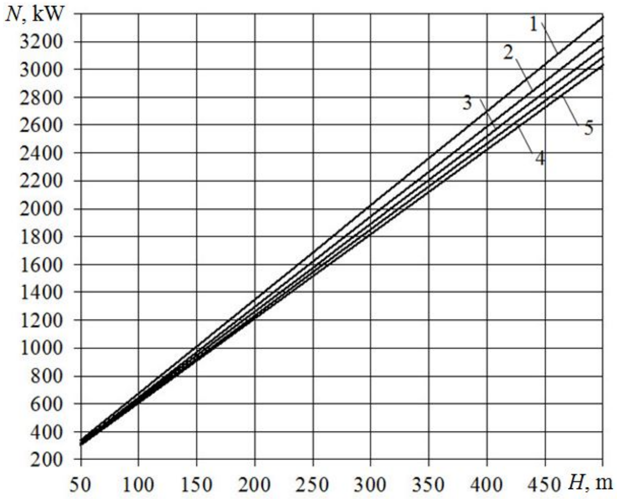
The drive power of a steeply inclined conveyor, depending on the height of the load and the angle of inclination of the conveyor, is calculated by the formula [7]:

$$N = \frac{W_0 v_b}{1000\eta}, \text{ kW},$$

$$W_0 = \left[k_1 (q_l + 2q_b + q'_r + q''_r)\omega' ctg\alpha + q_l\right]H,$$

where W_0 is the pulling force on the drum of a steeply inclined conveyor, N; k_1 is a coefficient that takes into account additional resistances when the belt and load are moving; q_l is the linear weight of the load, N/m; q_b is the linear weight of the belt, N/m; q'_r, q''_r are the linear weights of the rotating parts of the rollers of the loaded and empty branches, N/m; ω' is the coefficient of resistance when the load moves along the roller supports of a straight section of the conveyor framework; α is the angle of inclination of a steeply inclined conveyor ($\alpha \geq 30°$), degrees; H is the height of the load, m; v_b is the belt speed, m/s; η is the efficiency of the conveyor drive.

Figure 1 shows the graphs of the dependence of the drive power of a steeply inclined conveyor N on the lifting height H of the transported load at different angles of inclination of the conveyor. In this case, the parameters of the conveyor, roller bearings, belt and load had the following values: v_b = 2.5 m/s; q_l = 2000.0 N/m; q_b = 400.0 N/m; q'_r = 620.0 N/m; q''_r = 430.0 N/m; ω' = 0.04; k_1 = 1.1; η = 0.85. The lifting height H varied from 50 to 500 m. The calculation was performed for the conveyor inclination angles α = 30°, 40°, 50°, 60°, 70°.



1 – α = 30°; 2 – α = 40°; 3 – α = 50°; 4 – α = 60°; 5 – α = 70°

Figure 1 – Graphs of dependence of the drive power of a steeply inclined conveyor N on the lifting height H of the transported load at different conveyor inclination angles

It is established that with an increase in the height of lifting the load, the drive power of the belt conveyor increases according to a linear law. At the same time, the value of the angle of inclination of a steeply inclined belt conveyor has little influence on the power of the conveyor drive.

By rationally selecting distance between the roller supports of a tubular conveyor, it is possible to significantly reduce the load on the rollers, increase their service life, and thereby increase the reliability of the tubular conveyor.

Figure 2 shows a normal cross-section of the loaded tubular conveyor belt.

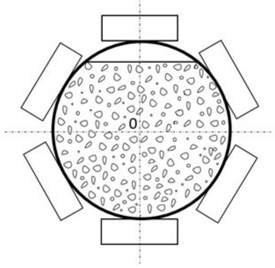


Figure 2 – Normal cross-section of the six-roller support of a tubular conveyor

Let's determine the average service life of a roller in the linear and curved sections of a tubular belt conveyor.

Average radial load on the roller bearing F_r (N) [8] is:

$$F_r = 0.5(P_{\partial k} + q_r l_r),$$

where $P_{\partial k}$ is the maximum load on the roller of the six-roller support [9], N; q_r is the linear weight of the rotating parts of the roller, N/m; l_r is the distance between the roller supports of the linear part of the conveyor framework, m.

Dynamic radial load on the roller bearing P_m (N) and 90 percent service life of the roller bearing (h) [8] are:

$$P_m = 1.2F_r;$$

$$L_{09} = \left(\frac{C_n}{P_m}\right)^p \cdot \frac{2\pi r 10^6}{3600 v_b} k_e,$$

where k_e is the coefficient that takes into account the operating conditions; C_n is the dynamic load capacity of roller bearings, N; v_b is the belt speed, m/s; r is the radius of the roller, m; p is the ball bearing degree index.

The average service life of a roller bearing in hours is equal to

$$t_a = 4.08 \cdot L_{09}, \text{ h.}$$

From a reliability point of view, the average life of a roller bearing is the average life of a roller, and, thus, it is possible to select the parameters of roller support with sufficient life and to predict repairs.

Several variants of calculating the parameters of the tubular conveyor and the roller life were made for different initial data.

The results of one of the calculations showed that the average service life of the lower central rollers is: linear section of the framework – t_a = 3829 hours; curved section of the framework – t_a = 2195 hours.

The calculations showed that the greatest load on the rollers of a tubular conveyor is on the lower side and central rollers (see Fig. 2). Regarding the curved section of the tubular conveyor, the maximum loads on the rollers increase by 20%, and the average service life of the rollers in the curved section decreases by 43%, when a distance between the roller supports is 0.5 m. Therefore, it is recommended to reduce the distance between the roller supports on the curved section of the tubular conveyor by four times compared to the distance between the roller supports on the linear section of the conveyor framework.

The operation of a tubular belt conveyor (TBC) can be accompanied by the phenomenon of belt scrolling. With a significant rotation of the belt, its sides may diverge and cause dusting and shedding of the load. The works [10, 11] were devoted to study of the stress-strain state of the TBC belt, but they do not contain specific recommendations for preventing these undesirable phenomena.

Preventing the phenomena of belt side divergence and load shedding during the operation of a tubular conveyor is possible by selecting the right TBC parameters. These are, first of all, the geometric dimensions and physical properties of the belt, its width, thickness, torsional stiffness, as well as the parameters of the conveyor's framework, the angle of inclination and radius of rotation of the route, and the distance between the roller bearings.

Torsion angle of the belt θ on the first roller support [12] is

$$\theta = \frac{Ml_r}{GI_p}, \tag{1}$$

where GI_p is the torsional stiffness of the belt cross-section, N·m^2; M is the moment applied to the belt cross-section on the first roller support, N·m.

To ensure the stable operation of a tubular conveyor, the following conditions must be met

$$\theta \le [\theta], \tag{2}$$

where $[\theta]$ is the permissible angle of the tubular belt torsion, i.e. the angle at which the belt sides do not divergence and the load does not shed.

Taking into account (2), expression (1) takes the form

$$\frac{l_r}{GI_p} \le \frac{[\theta]}{M}. \tag{3}$$

Suppose that the central roller of the first roller support is rotated in a plane perpendicular to the belt axis by an angle of β. Then the torque acting on the belt from the roller side is

$$M = PR\sin\beta, \tag{4}$$

where $P = \rho g S l_r$ is the weight of the belt with the load per one roller support, N; ρ is the bulk density of the load, kg/m^3; $S = k_l \pi R^2$ is the loading area of the belt cross-section, m^2; g is the acceleration of gravity, m/c^2; R is the radius of the belt cross-section, m; k_l is the loading coefficient of the belt cross-section.

Substituting the value of the moment M (4) into formula (3), we obtain

$$\frac{\rho g k_l \pi R^3 l_r^2 \sin\beta}{GI_p} \le [\theta],$$

whence

$$l_r \le \sqrt{\frac{[\theta] GI_p}{\rho g k_l \pi R^3 \sin\beta}}\,. \tag{5}$$

Formula (5) makes it possible to calculate the maximum distance between the roller supports that will not lead to side divergence and load shedding if the central roller of the roller support is rotated by an angle of β.

As an example, let's calculate the required distance between the roller support for a conveyor transporting coal, the parameters of which are given in [10]:

$$R = 0.108\ \text{m};\ \rho = 800\ \text{kg/m}^3;\ GI_p = 29.3\ \text{N·m}^2. \tag{6}$$

We take that $[\theta] = 20°$, $\beta = 2°$, $k_l = 0.75$. Substituting the values of parameters (6) into formula (5), we obtain

$$l_r \le 3.55\,\text{m}.$$

Thus, in order for a conveyor with parameters (6) to avoid divergence of the sides of the tubular belt when the central roller is rotated by an angle $\beta = 2$, the distance between its roller supports should not exceed 3.55 meters.

Formula (5) is recommended for use when selecting and justifying the main parameters of the tubular conveyor system operating in deep quarries.

Results:

– the factors affecting the torsion of a loaded tubular belt when it moves along a straight section of a conveyor are investigated. It is shown that the angle of rotation of the belt depends not only on the value of the applied torques, but also on the design of the TBC, the geometric characteristics of the cross section of the tubular belt, and the physical properties of its material;

– a formula was obtained for calculating the maximum distance between the roller supports of the TBC framework, which will not lead to the divergence of the sides of the tubular belt and load shedding if the central roller of the first roller support is rotated to the angle β.

3. Conclusions

1. It was established that the required drive power of a steeply inclined belt conveyor is directly proportional to the height of the load lift, while the angle of the load lift has little effect on the drive power.

2. Recommendations were developed for the selection of the main elements of the tubular conveyor – roller supports and rollers, which makes it possible to significantly reduce the load on the rollers and increase their service life. It is recommended to reduce the distance between the roller supports on the curved section of the tubular conveyor by four times compared to the distance between the roller supports on the linear section of the conveyor framework.

3. When selecting the main parameters of the tubular conveyor framework operating in deep quarries, it is recommended to use formula (5) to calculate the distance between the roller supports. This will help to avoid divergence of the belt sides if the central roller of the roller support is rotated by an angle β.

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РОЗРОБКА МЕТОДІВ РОЗРАХУНКУ ПАРАМЕТРІВ КРУТОПОХИЛИХ СТРІЧКОВИХ КОНВЕЄРІВ, ЩО ЗАСТОСОВУЮТЬСЯ ПРИ ВІДКРИТІЙ РОЗРОБЦІ КОРИСНИХ КОПАЛИН

Кірія Р., Смірнов А., Жигула Т., Міщенко Т., Куттибаєв А.

Анотація. Із зростанням глибини кар'єрів та збільшенням загального кута нахилу бортів транспортне забезпечення розкривних та видобувних робіт набуває особливої важливості через проблеми застосування автомобільного транспорту. У зв'язку з цим крутопохилі конвеєри (КПК) для глибоких кар'єрів є особливо актуальними. Крутопохилі стрічкові конвеєри – це конвеєри з кутом нахилу більше 30 градусів, основні типи таких конвеєрів: з перегородками на стрічці, з кишенями на стрічці, двоконтурні з притискною стрічкою, трубчасті конвеєри. Розроблено методи розрахунку параметрів різних типів крутопохилих конвеєрів, що можуть бути застосовані при відкритій розробці корисних копалин, а саме параметрів приводу і конструкцій става. Наведено розрахунок потужності привода КПК в залежності від висоти підйому і кута нахилу конвеєра. Показано, що величина потужності привода КПК прямо пропорційна висоті підйому, при цьому величина кута нахилу КПК має малий вплив на потужність приводу. Трубчасті стрічкові конвеєри (ТСК) є одним з перспективних типів КПК. Але через складні процеси, що відбуваються при експлуатації ТСК при русі трубчастої стрічки з вантажем по роликоопорах, відмови ТСК і їх наслідки відрізняються від тих, що характерні для інших КПК. Розроблено рекомендації щодо вибору відстані між роликоопорами на лінійній та криволінійній ділянках траси ТСК. Розрахунки показали, що найбільше навантаження на ролики трубчастого конвеєра припадає на нижні бічні та центральні ролики. При цьому на криволінійній ділянці трубчастого конвеєра максимальні навантаження на ролики збільшуються на 20 %, а середній ресурс роликів на криволінійній ділянці става зменшується на 43 % при відстані між роликоопорами, що дорівнює 0,5 м. Отже, відстань між роликоопорами на криволінійній ділянці трубчастого конвеєра рекомендовано зменшити у чотири рази порівняно з відстанню між роликоопорами на лінійній ділянці става конвеєра. Відмови ТСК можуть статися внаслідок прокручування трубчастої стрічки, що призводить до пиління та просипу вантажу. Запобігти цим явищам можна за допомогою вибору конструктивних параметрів ТСК. Отримано формулу для розрахунку максимальної відстані між роликоопорами ТСК, яка не приведе до розходження бортів трубчастої стрічки у випадку, якщо центральний ролик першої роликоопори повернеться на певний кут.

Ключові слова: крутопохилий стрічковий конвеєр, потужність приводу, термін служби роликів, відстань між роликоопорами.