DETERMINING BASIC GEOMETRIC PARAMETERS OF THE APRON CONVEYOR DRIVE

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Resume

The article provides an analysis of the classical design of the apron conveyor, which revealed the significant disadvantages. The study is devoted to the improvement of the plate conveyor design, which would provide an increase in the speed and performance of the conveyor, and allow to create conveyors of any required length and installation angle.

As a result of the research work, the following are determined: dependencies of the value of the specific traction force of the drive on the power of the engine and the value of its relative pole division; dependence of the active area of the linear asynchronous motors [LAM] on its power and conveyor speed; dependence of the distance between the intermediate drives on the power of the drive and the angle of installation of the conveyor.

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1 Introduction

Increasing the efficiency of the process of open-source mining is the use of conveyor transport. One of the types of conveyors used is a plate conveyor [1].

Figure 1 shows the general appearance of the plate conveyor.

The diagram of the classical plate conveyor is shown in Figure 2.

The carrier belt of the conveyor consists of separate plates connected to chains that are driven by drive sprockets.

In open-pit mining conveyor applications, the maximum possible conveyor angle and the possibility of transporting a lump rock mass are important.

Conveyor transport is the best positioned at the angle of natural derailment of the quarry, which is 30-40 °. The classical design of the plate conveyor allows its installation at a maximum angle of 25 °. At large installation angles, the load is sliding over the carrier surface under the action of its own weight. The use of partitions makes it possible to hold the load on the load-bearing sheet.

Load sliding dividers are welded to the conveyor carrier plate, allowing the conveyor to be installed at 35-40 ° inclinations. The plate conveyor makes it possible to transport large chunks of mountain mass, as the load moves together with the carrier cloth, so there is no impact of the load on stationary rollers, as in belt conveyors.

At the same time, the classical design of the plate conveyor has certain disadvantages - significant dynamic loads in the traction unit and the maximum length of the conveyor.

Significant dynamic loads that limit the speed of the traction and load-bearing organ of the plate conveyor are caused by the physical processes that occur when the traction chain of the drive sprocket is curved.

The drive sprocket circumference with the traction chain is shown in Figure 3.

Since the sprocket rotates at an angular speed ω, the start and end of the link of the traction chain move at a speed:

$$v_e = \omega R_e , \qquad (1)$$

where: ω - the speed at which the asterisk rotates;
R_e - start and end radius of the chain link.

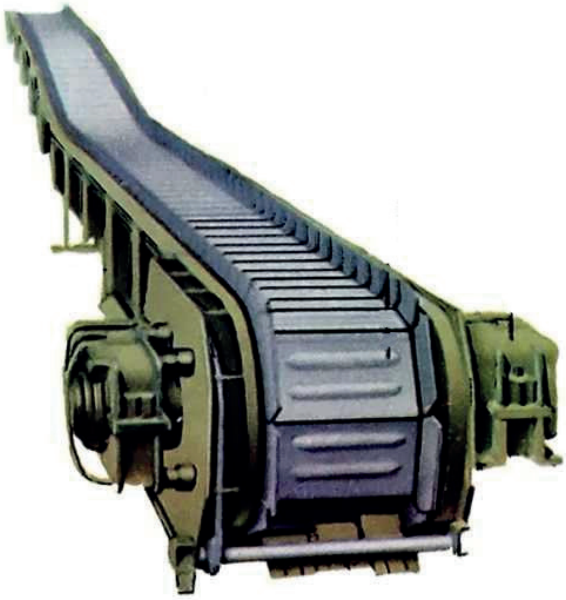
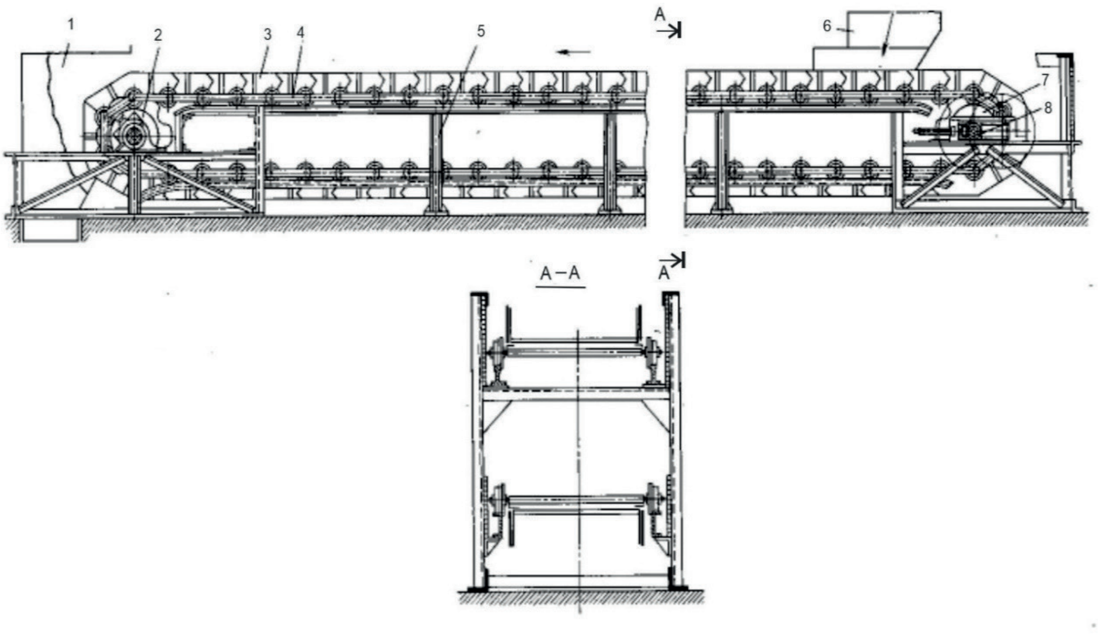


***Figure 1** Apron conveyor*



1- unloading funnel; 2 - drive sprocket; 3 - load-carrying belt; 4 - traction chains; 5 - bed; 6 - loading funnel; 7 - sprocket; 8 - tension device

***Figure 2** Structural design of a traditional apron conveyor*

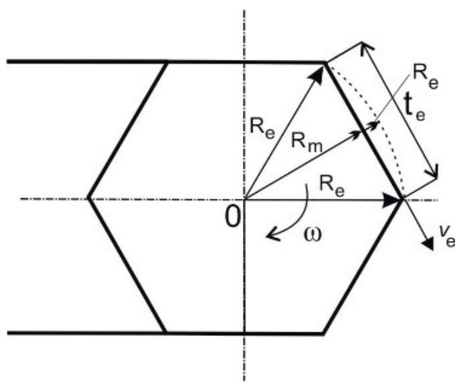


***Figure 3** The drive sprocket chain bending diagram*

If the drive sprocket is at rest, the middle of the link of the traction chain t_e is at a distance $R_m < R_e$ from the drive sprocket axis (Figure 3).

When the drive sprocket is rotated, the link of the traction chain is subjected to elastic deformation, leading the center of the link of the traction chain to a distance from the axis of the drive sprocket R_e.

This process causes additional loads on the traction unit, depending on the speed of rotation of the drive sprocket and the speed of the conveyor. Numerous experimental studies have shown that the maximum speed of these conveyors is 1.2 m/s. The tension at any point of the traction body of the plate conveyor can be determined by:

$$S_e = S_S + LA, \tag{2}$$

where: S_e, S_s - are the tension at the beginning and end of traction organ section, respectively; A - a coefficient depending on the linear load on the traction organ, the drag of the traction unit and the installation angle of the conveyor; L -length of the plot. At the same time, the force in the traction organ is limited by its strength ratio:

$$S_e < [S], \tag{3}$$

where: [S]- permissible traction tension.

From Equation (3) and Equation (4) follows:

$$L = \frac{[S] - S_e}{A}. \tag{4}$$

The single-drive conveyor is thus limited in length.

Research of the possibility of using the conveyor transport in difficult conditions of deep quarries shows a fairly high efficiency of these solutions [2]. The continuous technology of the conveyor process can be improved by using inclined conveyors [3].

Conveyor lifts can be placed both on the side of the quarry and in inclined shafts [4].

Scientific research has been carried out on many issues of improving the technology of mining, dedicated to improving the safety and intensity of mining operations, which is ensured by the use of the conveyor transport [5-8]. Cyclic-flow technology, along with the conveyor transport, involves the use of vehicles to deliver rock mass to the transfer point to the conveyor [9-10].

Various design solutions for conveyor lifts are being offered. For example, it is proposed to use a conveyor lift made of composite elastomer and diaphragm plates of the carrier body, which allows implementing a large angle of inclination and a significant lifting height of the transported material. An increase in the possible angle of installation of the conveyor, in some cases, is achieved by using the method of backing up the transported cargo, by using transverse dividing plates in the design of the carrier body of the conveyor [11]. In addition, there are known cases of using of linear induction motors in the conveyor transport drives, which makes it possible to provide a significant length of the conveyor line and lifting height [12].

The development of a special design of an apron conveyor that allows transporting cargo of a necessary height at the shortest length, while ensuring minimal costs, is a task of current interest.

Disadvantages of the plate conveyors can be eliminated by using intermediate drives with LAM.

In this way, the research is needed to determine the design parameters of intermediate conveyor drives with the LAM.

2 Problem statement

The aim of the study is to establish the basic geometric parameters of intermediate drives of plate conveyor with LAM. To achieve this objective, it is necessary to:

1. Set the value of the specific traction force of the drive the dependence on the power of the engine and the value of its relative pole pitch.
2. Set the area of the active part of a linear asynchronous motor the dependence on its power and conveyor speed.
3. Sett the distance between the intermediate drives the dependence on the drive power and the angle of setting of the conveyor.

3 An apron conveyor with linear drives

The structural design of an apron conveyor with a system of several drives with linear asynchronous motors is shown in Figure 4.

The apron conveyor operates as follows:

- plates *2* are interconnected through the swivel assembly of plates 3;
- interconnection of plates *2* allows performing the functions of a traction-bearing body of the apron conveyor;
- the traction force is transmitted to the traction-bearing plates *2* through the developed rotors of 8 linear motors, which are set in motion from interaction with the stators *9* of linear motors;
- rotors *8* of linear motors have a developed (linear) view and are fixed to axis *4* of travel rollers with flange *6*;
- stators *9* of linear motors also have a developed (linear) view and are fixed to frame *1* of the conveyor;
- the number and the distribution step of linear motors (rotors *8* and stators *9*) are determined by calculation depending on the operating conditions.

The developed apron conveyor with linear motors allows significant reducing dynamic loads and increasing the length of one conveyor, due to the absence of traction chain sprockets.

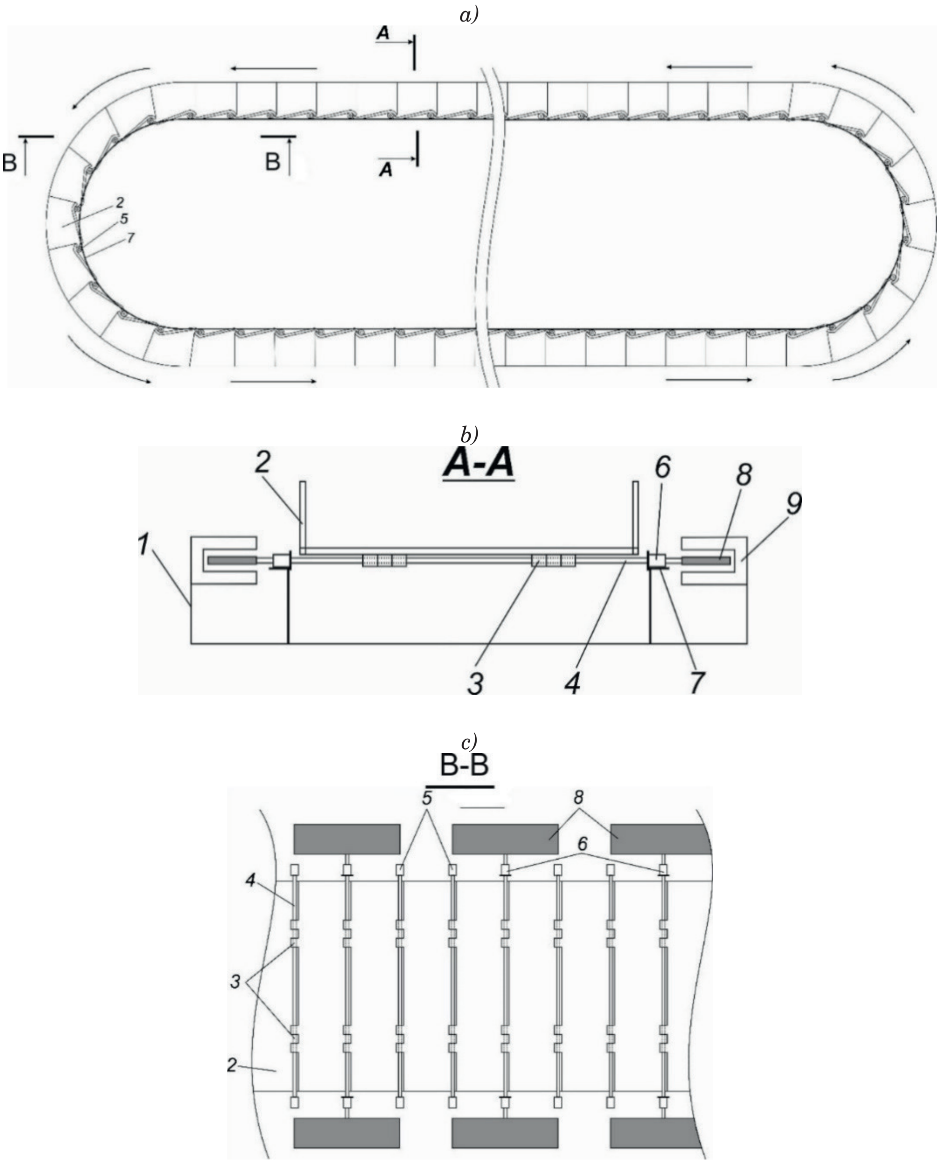


Figure 4 *Structural design of an apron conveyor with a system of several drives with linear asynchronous motors: a) The side view of the conveyor; b) A-A conveyor section; c) B-B conveyor section;1-frame, 2 - base, 3 - plate swivel assembly, 4 - axle, 5 - travel rollers, 6 - travel flanged rollers, 7 - travel roller guides, 8 - linear motor rotors, 9 - linear motor stators*

4 Determining the specific traction forces of the apron conveyor intermediate drive

A specific traction force can be determined according to the following dependence [13-14]:

$$f_{sp.force} = \frac{4\mu_0 A^2 \varepsilon \tau_\delta}{\pi[(1+\beta_x)^2 + \varepsilon^2]}, [Nm^{-2}], \tag{5}$$

where: μ_0 - is magnetic permeability in vacuum [H/m];
A - is the linear current load [A];
β_x - is the dimensionless coefficient accounting for the magnetic conductor steel saturation;
ε - is the electromagnetic slip;
τ_δ - is the relative pole pitch of the linear asynchronous motor (LAM).

The relative pole pitch is determined by the expression:

$$\tau_\delta = \frac{\tau}{\delta}, \tag{6}$$

where: δ -is the gap between inductor and secondary element LAM,
τ -is the pole pitch LAM calculated according to the dependence [m]:

$$\tau = \frac{v_1}{2(1-s)f}, m, \tag{7}$$

where: v_1 - is the secondary element speed (conveyor speed) [m/s];
s - is the slip amount;
f - is the alternate current frequency [1/s];
δ - is engine air gap [m].

Values of the specific traction force for various engine powers and the relative value of the pole pitch calculated by Equation (5), are shown in Figure 5.

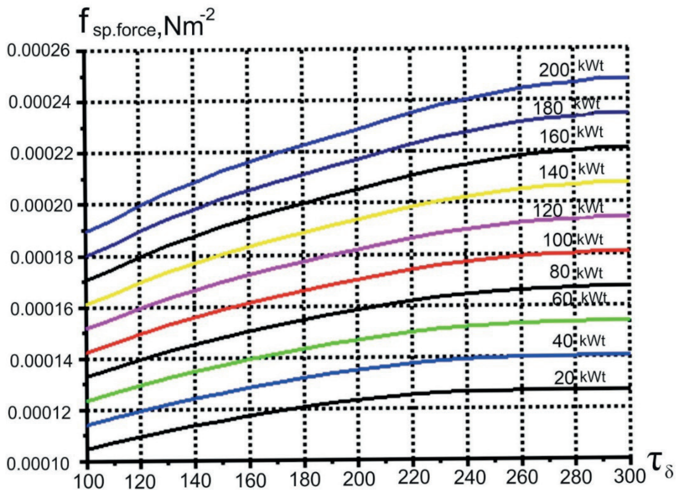


Figure 5 *Specific traction forces of a linear intermediate drive*

5 Determining the basic geometric parameters of the apron conveyor intermedicate linear drive

Using the dependences of the specific traction force, one next determines the geometric dimensions of a linear asynchronous motor for a steeply inclined apron conveyor.

Resistance to the cargo branch of the apron conveyor movement is determined by the formula:

$$W_w = [(q_w + q_{conv})w\cos\beta + (q_w + q_{conv})\sin\beta]gL, \quad [N], \tag{8}$$

where q_w, q_{conv} - are the load per unit length and the mass per unit length of the tension-skidding body, respectively, [kg/m];
w - is specific resistance to the tension-skidding body movement of the apron conveyor, w = 0.0025;
β - is the angle of the conveyor mounting [rad];
L - is the conveyor section length [m].

Taking the values of the relative pole pitch equal to τ_δ = 200, with the power values of the traction motors N = 50, 100, 200 kW, one obtains, according to the graphs (Figure 5), the values of specific traction forces that are respectively equal to 0.00014, 0.00017 and 0.00020 Nm^{-2}.

The drive pull force is determined by the ratio:

$$T_{i,j} = \frac{N_i}{v_j}, [N], \tag{9}$$

where: N_i - is the i- th value of the drive power LAM [kW];
v_j - is the j-th value of the conveyor speed [m/s].

The required area of the secondary element active part is determined by the expression:

$$S_{i,j} = \frac{T_{i,j}}{f_{sp.force}}, [m^2], \tag{10}$$

where $f_{sp.force}$ - is the specific traction force of LAM, [Nm^{-2}].

The areas of the secondary element active part dependences on the conveyor speed, with the drive power of 50, 100 and 200 kW, are presented in Figure 6.

The distance between the drives of the conveyor is determined from the ratio of the equality of the drive traction force to resistance of the traction-carrying body movement:

$$F = W_w, [N]. \tag{11}$$

Substituting Equation (8) and Equation (9) into Equation (11), one obtains the expression for calculating the distance between the drives:

$$L_{k,j} = \frac{N}{v_j\begin{bmatrix}(q_w + q_{conv})w\cos\beta_k + \\ +(q_w + q_{conv})\sin\beta_k\end{bmatrix}} g, [m]. \tag{12}$$

The dependences of the distance between the drives on the speed and the angle of setting the conveyor, calculated by Equation (12) for different performance and power, are shown in Figures 7-9.

A specific example of determining the geometric parameters of the LAM and the distance between the drives is considered next.

According to the dependences shown in Figures 3 and 4, at the speed v = 4 m/s, the load per unit length q_w = 200 kg/m, the conveyor performance will be:

$$Q = 3.6 * q_w * v = 2880 \text{ t/h} \tag{13}$$

With the angle of setting β = 35 degrees and the drive power N = 200 kW, the LAM active part area will be S = 2.5 m^2, the distance between the drives L= 20 m.

The obtained geometric parameters with a sufficient conveyor performance can well be implemented in the actual design of the conveyor.

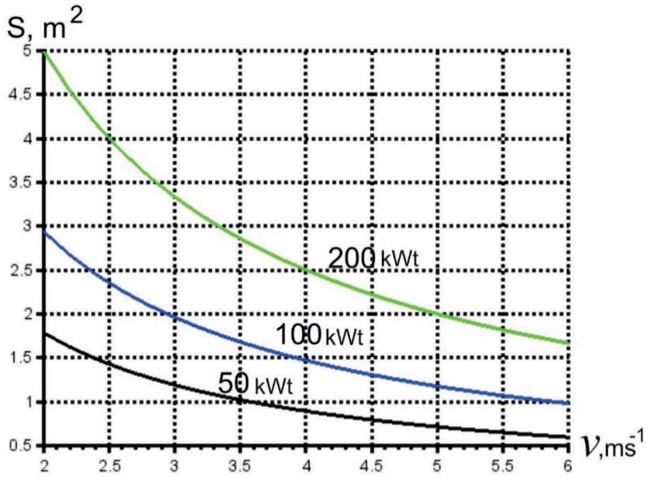


Figure 6 *Intermediate drive active part areas*

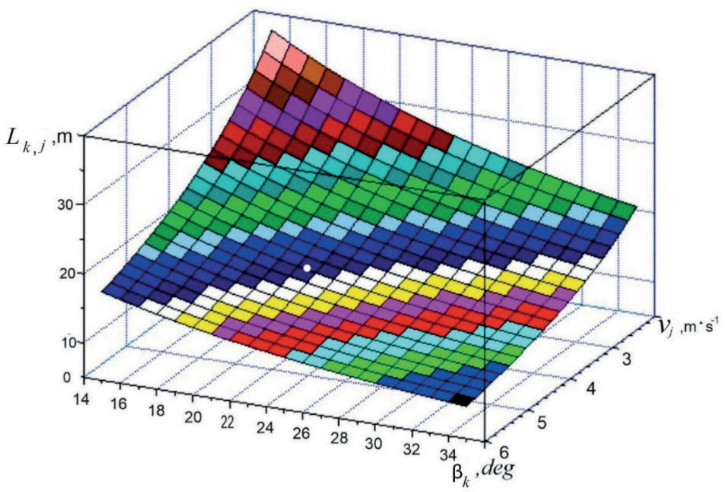


Figure 7 *The distance between the intermediate drives dependence on the conveyor speed and the angle of setting for Q = 1000 t/h, N = 50 kW*

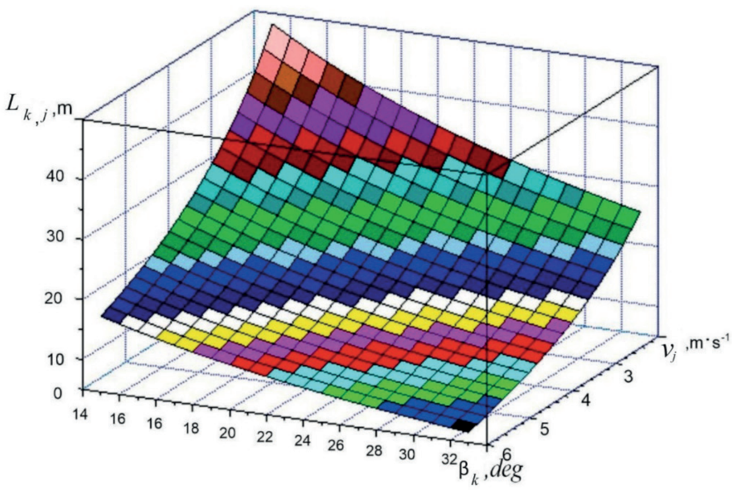


Figure 8 *The distance between the intermediate drives dependence on the conveyor speed and the angle of setting for Q = 1500 t/h, N = 100 kW*

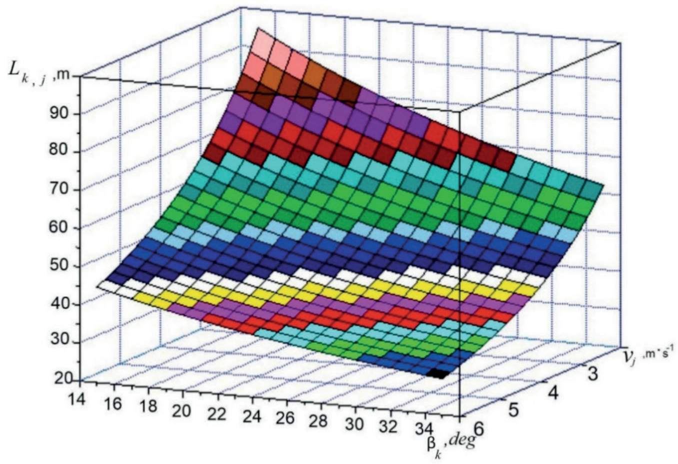


Figure 9 *The distance between the intermediate drives dependence on the conveyor speed and the angle of setting for Q = 2000 t/h, N = 200 kW*

Thus, the use of the LAM in intermediate drives of steeply inclined apron conveyors is quite achievable.

6 Conclusions

As a result of the study:

- The dependence of the value of the specific traction force of the intermediate drive of the plate conveyor with the LAM on the power of the engine and the value of the relative pole division thereof have been determined.
- At the value of the relative pole division τ = 200 and the change in engine power from 20 to 200 kW, the specific traction forces of the intermediate drive have values from 13*10-5 N m^{-2} to 25*10 N m^{-2}, which is acceptable for the geometrical parameters of the secondary LAM element.
- The area of the active part of the LAM is determined by its power and the speed of a conveyor. Within the range of changing the conveyor speed from 3 to 5 m/s and the drive power from 10 to 200 kW required by the operating conditions of conveyor transport in open mining, the area of the active part of the LAM has values from 2 to 3.2 m^2, which shows that it is possible to design the LAM with these parameters.
- The dependence of the distance between the intermediate drives of a plate conveyor with the LAM from the power of the drive and the angle of installation of the conveyor is determined.
- Within the range of change in conveyor performance from 1000 to 2000 t/h, average conveyor speed 4 m/s and drive power from 100 to 200 kW, the drive distance is between 15 and 50 m, which is quite feasible when designing a plate conveyor with intermediate drives based on the LAM, for specific operating conditions.
- Studies have shown that the use of intermediate-drive plate conveyors based on the LAM is very promising for increasing the efficiency of transport processes in open-pit mining.

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Conflicts of interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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