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ANALYSIS OF INFLUENCE OF DESIGN CHARACTERISTICS OF INCLINED BUCKET ELEVATOR ON THE POWER OF ITS DRIVE

Purpose. One of the main elements of the inclined belt bucket elevators is their drive. To determine the drive power, it is necessary to carry out calculations according to standard methods, which are described in the modern literature. The basic design parameters are the productivity, lifting height, type and properties of the transported material, the angle of inclination. It is necessary to build a parametric dependence of the driving power of the elevator on its design parameters, which takes into account the standard sizes and types of buckets and belts. **Methodology.** Using the methodology of traction calculation of inclined belt bucket elevator there were built parametric dependences of efforts in specific points of the route of the elevator, as well as the parametric dependences of the drive power of high-speed elevators with deep and shallow buckets on their design parameters and characteristics. **Findings.** On the basis of constructed parametric dependencies, it was found that the function of changing the value of the elevator's power from design capacity (at fixed lifting height, type of cargo, belt speed) is piecewise constant and monotonically increasing. It was built a graphical representation of elevator drive power on the angle of its inclination within acceptable limits of change. The resulting relationship is non-linear and monotonically decreasing. In general terms the intervals of project performance values, which provide a constant value of drive power of inclined elevator were defined. As an example of the obtained results it was observed the process of dependence construction of the drive power on design capacity and inclination angle of the elevator for transporting the fine coal. **Originality.** For the first time there were constructed the parametric dependences of drive power of inclined bucket elevator on its design parameters that take into account the standard sizes and types of buckets and belts. **Practical value.** Using the constructed dependencies enables relatively quick determination of the approximate value of the drive power of high-speed inclined elevators with deep and shallow buckets at the design stage and high-quality selection of its basic elements in the design of specific characteristics: type of cargo, productivity, lifting height, angle of inclination.

Keywords: inclined elevator; bucket; drive; power; productivity; cargo; angle of inclination

Introdiction

Increasing the pace of economic development is impossible without technical re-equipment of production. The successful solution of this problem is largely determined by implementation of new technologies with the use of stream-flow transportation machines. They have great performance and length of transportation and can replace batch machines in traditional application fields, such as hauling, handling and warehousing operations. These machines have become very popular in mass and high-volume production with wide use of automatic lines. A special type of stream-flow transportation machines is inclined belt bucket elevators. Generally, elevators are the lifts that are used for vertical and steeply inclined (at an angle 60–82°) displacement of bulk and piece cargo without intermediate loading and unloading. Their use when transporting materials increase the efficiency of the production process in many industries: chemical, metallurgical, engineering, etc.

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The main publications describing the structure, design features, performance and design parameters of elevators, including the inclined ones are the following works [5-9, 11-15]. To determine the drive power of inclined elevator it is necessary to conduct a detailed calculation of its elements and perform a selection of basic elements of the drive. The order of these calculations is described in detail in the works [8, 9]. It should be noted that the use of traditional calculation methodology of the elevator's drive requires a lot of time. To improve the design process of the inclined elevator's drive it is necessary to define a scheme that makes it possible to determine the required drive power value depending on the specific design parameters: the type of load, lifting height, track inclination angle and performance using simpler calculations. The works [2-4] of one of the authors include similar scheme for vertical elevators and conveyor belts. The natural generalization and continuation of these works will be the construction of schemes for inclined elevators. This is because the inclined elevators as opposed to the vertical ones include the component of tension force related to the force of belt friction on the support elements.

Purpose

The article is aimed to construct and analyze the parametric dependence of inclined elevator's drive power on its design parameters (type of load, lifting height, angle of inclination, performance) taking into account the standard sizes and parameters of buckets and belts.

Methodology

In general, for design of stream-flow transportation machines one should have the following basic data:

– diagram of machine track with indicated places of loading and unloading;

– appointment, conditions and operation mode of machine and the place of its installation;

– the required performance;

– characteristics of transported cargoes.

Thus, the initial data for design calculation of the elevator are such values as the transported material (its density and physical and mechanical properties) lift height of cargo, inclination angle of elevator to the horizon, required performance.

To construct general dependence of drive power on the performance there will be used the required coefficients at the values that make it possible to calculate the corresponding values of the required drive power for specific types of cargoes.

By analogy with [3] let us consider the value α that takes into account the properties of transported cargo for further studies:

$$\alpha = 3,6 v \rho \psi . \quad (1)$$

Linear content of the elevator's bucket:

$$\frac{i_0}{t} = \frac{\Pr}{3,6 v \rho \psi} = \frac{\Pr}{\alpha}, \quad (2)$$

where α – is a value that takes into account characteristics of the cargo and is calculated using dependence (1), t m/l h; ψ – is a coefficient of bucket fill (according to the physical and mechanical properties of cargo); t – is a spacing of the buckets, m; ρ – is a cargo density, t/m^3; v – is a speed of the belt movement, m/s.

According to the value of linear content of elevator's bucket calculated from the formula (2) the type and spacing of buckets in accordance with the table 1 recommended by the wok [9] are selected. Selection of buckets type depends on the properties of the material, which is being transported. Deep buckets are used for free-flowing, dusty and small pieced cargoes; the shallow ones – for non-free-flowing cargoes.

To take account physical and mechanical properties of the cargo, which is being transported in further calculations let us construct the correspondence tables of elevator parameters specified in the Table 1 to the performance value expressed by the formula (2) in the parts of coefficient α. The obtained data will be tabulated in the Tables 2, 3 for elevators with deep and shallow buckets respectively.

Based on the design value of elevator productivity and the type of material, which is being transported according to the Tables 2 and 3, the bucket parameters, their spacing on the belt and the required width of the belt are selected. Characteristics of deep and shallow buckets are shown in the Tab. 4.

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Table 1

Value of linear content of buckets

Bucket width B_b , mm	Belt width B , mm	Spacing of the buckets t , mm	Bucket			
			deep		shallow	
			i_0 , l	$\frac{i_0}{t}$, l/m	i_0 , l	$\frac{i_0}{t}$, l/m
1	2	3	4	5	6	7
100	125	200	0.2	1	0.1	0.5
125	150	320	0.4	1.3	0.2	0.66
160	200	320	0.6	2	0.35	1.17
200	250	400	1.3	3.24	0.75	1.87
250	300	400	2.0	5	1.4	3.5
320	400	500	4.0	8	2.7	5.4
400	500	500	6.3	12.6	4.2	8.4
500	650	630	12	19	-	-
650	800	630	18	28.6	-	-
800	1000	800	32	40	-	-
1000	1200	800	45	56.25	-	-

Table 2

Dependence of parameters of deep buckets on the elevator's productivity

Bucket width B_b , mm	Belt width B , mm	Spacing of the buckets t , mm	Bucket capacity i_0 , l	Elevator productivity, t/h
100	125	200	0.2	α
125	150	320	0.4	1.3α
160	200	320	0.6	2α
200	250	400	1.3	3.24α
250	300	400	2.0	5α
320	400	500	4.0	8α
400	500	500	6.3	12.6α
500	650	630	12	19α
650	800	630	18	28.6α
800	1000	800	32	40α
1000	1200	800	45	56.25α

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Table 3

Dependence of parameters of shallow buckets on the elevator's productivity

Bucket width B_b , mm	Belt width B , mm	Spacing of the buckets t , mm	Bucket capacity i_0 , l	Elevator productivity, t/h
100	125	200	0.1	0.5α
125	150	320	0.2	0.66α
160	200	320	0.35	1.17α
200	250	400	0.75	1.87α
250	300	400	1.4	3.5α
320	400	500	2.7	5.4α
400	500	500	4.2	8.4α

Table 4

Description of elevator buckets

Bucket type	Internal size of bucket, mm				Bucket capacity, l
	width B_b	outreach A_b	height	R	
Rounded deep one D	100	50	65	25	0.1
	100	75	80	25	0.2
	125	90	95	30	0.4
	160	105	110	35	0.6
	200	125	135	40	1.3
	250	140	150	45	2.0
	320	175	190	55	4.0
	400	195	210	60	6.3
	500	235	255	75	12
	650	250	275	80	18
	800	285	325	85	32
	1000	310	355	95	45
Bucket type	Internal size of bucket, mm				Bucket capacity, l
	width B_b	outreach A_b	height	R	
Rounded shallow one S	125	65	85	30	0.2
	160	75	100	35	0.35
	200	95	130	40	0.75
	250	120	160	55	1.4
	320	145	190	70	2.7
	400	170	220	85	4.2

For clearness of further research let us take the conveyor belt according to State Standard 20-85 of the type BKNL-150 as traction body of elevator. The actual number of spacer plates of the belt can be 3-6.

The belt thickness is determined by the formula

$$\delta_b = \delta_o + i\delta_f + \delta_n , \quad (3)$$

where $\delta_o = 3$ mm, $\delta_n = 1{,}5$ mm – is the thickness of rubber coatings from the working and non-working sides of the belt; $\delta_f = 1{,}6$ mm – is the thickness of fabric insert ply, i – is the number of fabric insert plies.

The weight of one running meter of belt is determined by the formula

$$q_b = 10^{-6} B\delta_b\rho_b g\,, \quad (4)$$

where $\rho_b = 1100$ kg/m^3 – belt density.

Involving the formulas (3)-(4) in the calculation let us present the table of correspondence of width and linear weight of the belt with a different number of insert plies to design values of elevator productivity for deep and shallow buckets.

Table 5

Linear weight of belts for deep buckets

Bucket width B, mm	Linear weight of the belt at $i = 3$, N/m	Linear weight of the belt at $i = 4$, N/m	Linear weight of the belt at $i = 5$, N/m	Linear weight of the belt at $i = 6$, N/m	Elevator productivity, t/h
125	12.5	14.7	16.8	19.0	α
150	15.0	17.6	20.2	22.8	1.3α
200	20.1	23.5	27.0	30.4	2α
250	25.1	29.4	33.7	38.0	3.24α
300	30.1	35.3	40.4	45.6	5α
400	40.1	47.0	53.9	60.8	8α
500	50.1	58.8	67.4	76.0	12.6α
650	65.2	76.4	87.6	98.8	19α
800	80.2	94.0	107.8	121.6	28.6α
1000	100.3	117.5	134.8	152.0	40α
1200	120.3	141.0	161.7	182.4	56.25α

Table 6

Linear weight of belts for shallow buckets

Bucket width B, mm	Linear weight of the belt at $i = 3$, N/m	Linear weight of the belt at $i = 4$, N/m	Linear weight of the belt at $i = 5$, N/m	Linear weight of the belt at $i = 6$, N/m	Elevator productivity, t/h
125	12.5	14.7	16.8	19.0	0.5α
150	15.0	17.6	20.2	22.8	0.66α
200	20.1	23.5	27.0	30.4	1.17α
250	25.1	29.4	33.7	38.0	1.87α
300	30.1	35.3	40.4	45.6	3.5α
400	40.1	47.0	53.9	60.8	5.4α
500	50.1	58.8	67.4	76.0	8.4α

Distributed weight of cargo per 1 m of belt is determined by the formula:

$$q_w = \frac{\Pr g}{3{,}6v} = \lambda \Pr\,, \quad (5)$$

where $\lambda = \frac{g}{3{,}6v}$ – coefficient depending on the belt speed, N·s/kg·m.

The dependence of value of distributed weight of cargo on the design productivity calculated by the formula (5) is given in the Table 7.

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Table 7

Distributed weight of cargo

Bucket width B_b, mm	Distributed cargo weight during operation of elevator with shallow buckets N/m	Elevator productivity with shallow buckets, N/m	Distributed cargo weight during operation of elevator with deep buckets N/m	Elevator productivity with deep buckets, N/m
100	0.5αλ	0.5α	αλ	α
125	0.66αλ	0.66α	1.3αλ	1.3α
160	1.17αλ	1.17α	2αλ	2α
200	1.87αλ	1.87α	3.24αλ	3.24α
250	3.5αλ	3.5α	5αλ	5α
320	5.4αλ	5.4α	8αλ	8α
400	8.4αλ	8.4α	12.6αλ	12.6α
500	-	-	19αλ	19α
650	-	-	28.6αλ	28.6α
800	-	-	40αλ	40α
1000	-	-	56.25αλ	56.25α

Linear weight of the belt with buckets is determined by the formula:

$$q_x = q_b + \frac{m_b g}{t}, \quad (6)$$

where m_b – bucket weight, kg (Tab. 8).

Linear burden on the loaded strand is determined using the formula:

$$q_o = q_x + q_w. \quad (7)$$

The estimated weight of deep and shallow buckets is given in the Table 8 [9].

Involving the formulas (6)-(7) in the calculation and taking into account data from the Table 8 let us determine the dependency of linear load on the loaded strand of elevator on the productivity values for deep and shallow buckets. The obtained results of calculations for belts with different number of insert plies is presented in the Tables 9, 10.

Table 8

Estimated mass of elevator's buckets

Bucket width, mm	Wall thickness, mm	Weight of one bucket, kg	
		Deep	Shallow
100	2	0.5	0.4
125	2	0.7	0.6
160	2	0.9	0.7
200	3	2	1.5
250	3	3	2
320	3	5	5
400	4	11	10
500	5	18	-
650	5	23	-
800	6	28	-
1000	6	33	-

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Table 9

The linear load on the loaded strand for deep buckets

Bucket width B_b, mm	Distributed weight of cargo q_w, N/m	Linear load on loaded strand at the belt with $i=3$ q_o, N/m	Linear load on loaded strand at the belt with $i=4$ q_o, N/m	Linear load on loaded strand at the belt with $i=5$ q_o, N/m	Linear load on loaded strand at the belt with $i=6$ q_o, N/m	Elevator productivity, t/h
100	αλ	37+αλ	39.2+αλ	41.3+αλ	43.5+αλ	α
125	1.3αλ	36.4+1.3αλ	39+1.3αλ	41.6+1.3αλ	44.2+1.3αλ	1.3α
160	2αλ	47.7+2αλ	51.1+2αλ	54.6+2αλ	58+2αλ	2α
200	3.24αλ	74.1+3.24αλ	78.4+3.24αλ	82.7+3.24αλ	87+3.24αλ	3.24α
250	5αλ	103.6+5αλ	108.8+5αλ	113.9+5αλ	119.1+5αλ	5α
320	8αλ	138.1+8αλ	145+8αλ	151.1+8αλ	158+8αλ	8α
400	12.6αλ	265.7+12.6αλ	274.4+12.6αλ	283+12.6αλ	291.6+12.6αλ	12.6α
500	19αλ	345.2+19αλ	356.4+19αλ	367.6+19αλ	378.8+19αλ	19α
650	28.6αλ	438+28.6αλ	451.8+28.6αλ	465.6+28.6αλ	479.4+28.6αλ	28.6α
800	40αλ	443.3+40αλ	460.5+40αλ	477.8+40αλ	495+40αλ	40α
1000	56.25αλ	524.6+56.3αλ	545.3+56.3αλ	566+56.3αλ	586.7+56.3αλ	56.25α

Table 10

The linear load on the loaded strand for shallow buckets

Bucket width B_b, mm	Distributed weight of cargo q_w, N/m	Linear load on loaded strand at the belt with $i=3$ q_o, N/m	Linear load on loaded strand at the belt with $i=4$ q_o, N/m	Linear load on loaded strand at the belt with $i=5$ q_o, N/m	Linear load on loaded strand at the belt with $i=6$ q_o, N/m	Elevator productivity, t/h
1	2	3	4	5	6	7
100	0.5αλ	32.1+0.5αλ	34.3+0.5αλ	36.4+0.5αλ	38.6+0.5αλ	0.5α
1	2	3	4	5	6	7
125	0.66αλ	33.4+0.66αλ	36+0.66αλ	37.8+0.66αλ	40.4+0.66αλ	0.66α
160	1.17αλ	41.5+1.17αλ	44.9+1.17αλ	48.4+1.17αλ	51.8+1.17αλ	1.17α
200	1.87αλ	61.9+1.87αλ	66.2+1.87αλ	70.5+1.87αλ	74.8+1.87αλ	1.87α
250	3.5αλ	79.1+3.5αλ	84.3+3.5αλ	89.4+3.5αλ	94.6+3.5αλ	3.5α
320	5.4αλ	138.1+5.4αλ	145+5.4αλ	151.1+5.4αλ	158+5.4αλ	5.4α
400	8.4αλ	246.1+8.4αλ	254.8+8.4αλ	263.4+8.4αλ	272+8.4αλ	8.4α

Traction calculation of inclined bucket elevator is performed by the method of encirclement, the basic principle of which is to identify specific points of the track where the belt tension is changed. At this tension in the next ($i+1$) point is equal to the sum of belt tension in this (i) point and the belt movement resistance in the area between these points:

$$S_{i+1} = S_i + W_{i,i+1}. \qquad (8)$$

In case of drive drum rotation (Fig. 1) in clockwise order the minimum tension will be at the point 2 – S_2. This tension in the belt during normal scooping satisfies the following condition:

$$S_2 = S_{\min} \geq 5q_w. \qquad (9)$$

The belt tension force at the point 3 consists of tension force S_2, drum resistance and resistance to scooping of cargo W_{2-3}:

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$$S_3 = kS_2 + W_{2-3}, \quad (10)$$

where $k = 1,08$ – coefficient of tension increase in the belt with buckets when bending around the drum.

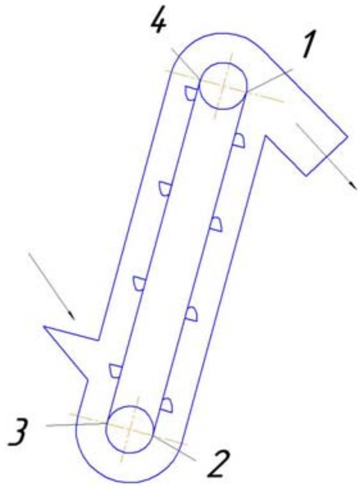

Fig. 1. Scheme of inclined bucket elevator

Resistance to material scooping is determined using the formula:

$$W_{2\text{-}3} = \frac{k_s q_w}{g}, \quad (11)$$

where k_s – is a coefficient of scooping (Nm/kg), which is determined by specific work expended for scooping of 1 kg of material. At the speed of buckets $v = 1,0...1,25$ m/s $k_s = 12,5...25$ Nm/kg for powdered and small pieced materials, and $k_s = 20...40$ Nm/kg – for medium pieced material.

Thus, substituting formulas (8) and (11) to (10), we have:

$$S_3 = q_w \left(5,4 + \frac{k_s}{g} \right). \quad (12)$$

Choosing the value $k_s = 25$ N m/kg (which meets all cargoes) we have:

$$S_3 = 7,95 q_w. \quad (13)$$

We assume that the belt with buckets at the track sections 3-4 and 1-2 (Fig. 1) is supported by direct roller supports.

The specific weight of moving parts of roller supports for loaded (section 3-4) and unloaded (section 1-2) strands is determined by the formulas:

$$q_{oo} = \frac{G_r'}{l_r'}. \quad (14)$$

$$q_{on} = \frac{G_r'}{l_r''}. \quad (15)$$

where G_r' – weight of rotating parts of the upper and lower rolers.

For further calculations the tables of estimated values of the distances between rollers of loaded strand (Tab. 11) and the characteristics and sizes of roller supports shown in the Table 12 will be used.

Ordinary roller supports of the strand 1-2 are set with the spacing l_r'', twice as high as l_r'. The dependence of the weight of ordinary roller supports on the belt width is presented in the Table 12.

To facilitate further studies, it is assumed that the cargo has a density in the range of 1 … 2 t/m³. Using the formulas (14)-(15) let us present the values of specific weight of moving parts of roller supports for loaded and unloaded strands depending on the belt width and width of the bucket. Calculated values of the specific weight will be presented in the Table 13.

Table 11

The estimated value of distances between supports of loaded strand l_r'

Material density ρ, t/m³	Distances between supports of loaded strand at the belt width, mm							
	400	500	650	800	1000	1200	1400…1600	1800…2000
1	1500	1500	1400	1400	1300	1300	1200	1100
1…2	1400	1400	1300	1300	1200	1200	1100	1100
more than 2	1300	1300	1200	1200	1100	1100	1100	900

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Table 12

Weight of ordinary direct roller supports

Belt width *B*, mm	Weight, kg
400	6.0
500	7.5
650	10.5
800	18.5
1 000	22.0
1 200	25.0

Table 13

The estimated values of the specific weight of moving parts of roller supports for loaded and unloaded strands

Specific weight of moving parts	Bucket width B_b, mm					
	320	400	500	650	800	1000
loaded strand q_{oo}, N/m	40	50	75	132	169	192
uloaded strand q_{on}, N/m	20	25	37.5	66	84.5	96

For clearness of further calculations at the buckets with width less than 320 mm, let us take the value of specific weight of moving parts of roller supports for loaded and unloaded strands branches $q_{oo} = 40$ N/m, $q_{on} = 20$ N/m, respectively. We also accept that working conditions of the elevator will be difficult; therefore, the resistance coefficient of the belt movement along the rollers in future will be equal to 0.03.

Traction forces at the points 1 and 4 are determined using the formulas:

$$S_4 = S_{nb} = S_3 + W_{3-4} = = 7,95q_w + \left(q_o + q_{oo}\right) H \cdot c \cdot \text{ctg}\beta + q_w H, \quad (16)$$

$$S_1 = S_{zb} = S_2 + W_{2-1} = = 5q_w + \left(q_x + q_{on}\right) H \cdot c \cdot \text{ctg}\beta + q_x H, \quad (17)$$

where H – lift height of cargo, m; β – inclination angle of elevator, degree; $c = 0,03$ – resistance coefficient of the belt movement along the rollers.

The dependence of traction forces values at the point 4 calculated by the formula (16) on the value of design productivity, bucket type and amount of insert plies are summarized in the Tables 14-15.

The dependence of the values of tension force at the point 1 calculated by the formula (17) on the value of design productivity, bucket type and amount of insert plies of the belt are summarized in the Tables 16-17.

Tractive effort accounting rotational resistance of the drive drum is determined using the formula:

$$F_o = S_4 - S_1 + (k' - 1)(S_4 + S_1), \quad (18)$$

where $k' = 1,08$ – is a resistance coefficient of drive drum rotation.

After algebraic transformations in the formula (18) we have:

$$F_o = 1,08S_4 - 0,92S_1. \quad (19)$$

The values of tractive effort taking into account the drum rotation resistance depending on the values of design performance, bucket type (deep and shallow) and the number of insert plies of the belt are summarized in the Tables 18-19.

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Table 14

Traction force at the point 4 at deep buckets

Bucket width B_b, mm	Traction force at the belt with $i = 3$ S_4, N	Traction force at the belt with $i = 4$ S_4, N	Traction force at the belt with $i = 5$ S_4, N	Traction force at the belt with $i = 6$ S_4, N	Elevator productiv-ity, t/h
100	37*H*+αλ (7.95+*H*)+ +(77+αλ) *cH*ctgβ	39.2*H*+αλ (7.95+*H*)+ +(79.2+αλ) *cH*ctgβ	41.3*H*+αλ(7.95+*H*)+ +(81.3+αλ) *cH*ctgβ	43.5*H*+αλ (7.95+*H*)+ +(83.5+αλ) *cH*ctgβ	α
125	36.4*H*+1.3αλ(7.95+*H*)+ +(76.4+1.3αλ)*cH*ctgβ	39*H*+1.3αλ (7.95+*H*)+ +(79+1.3αλ)*cH*ctgβ	41.6*H*+1.3αλ(7.95+*H*) +(81.6+1.3αλ) *cH*ctgβ	44.2*H*+1.3αλ (7.95+*H*) +(84.2+1.3αλ) *cH*ctgβ	1.3α
160	47.7*H*+2αλ (7.95+*H*)+ +(87.7+2αλ) *cH*ctgβ	51.1*H*+2αλ (7.95+*H*)+ +(91.1+2αλ) *cH*ctgβ	54.6*H*+2αλ(7.95+*H*)+ +(94.6+2αλ) *cH*ctgβ	58*H*+2αλ (7.95+*H*)+ +(98+2αλ) *cH*ctgβ	2α
200	74.1*H*+3.24αλ(7.95+*H*) +(114.1+3.24αλ)*cH*ctgβ	78.4*H*+3.24αλ(7.95+*H*)+ +(118.4+3.24αλ) *cH*ctgβ	82.7*H*+3.24αλ(7.95+*H*) +(122.7+3.24αλ)*cH*ctgβ	87*H*+3.24αλ (7.95+*H*)+ +(127+3.24αλ) *cH*ctgβ	3.24α
250	103.6*H*+5αλ (7.95+*H*)+ +(143.6+5αλ) *cH*ctgβ	108.8*H*+5αλ (7.95+*H*)+ +(148.8+5αλ) *cH*ctgβ	113.9*H*+5αλ(7.95+*H*)+ +(153.9+5αλ) *cH*ctgβ	119.1*H*+5αλ (7.95+*H*) +(159.1+5αλ) *cH*ctgβ	5α
320	138.1*H*+8αλ (7.95+*H*)+ +(178.1+8αλ) *cH*ctgβ	145*H*+8αλ (7.95+*H*)+ +(185+8αλ) *cH*ctgβ	151.1*H*+8αλ(7.95+*H*)+ +(191.1+8αλ) *cH*ctgβ	158*H*+8αλ(7.95+*H*)+ +(198+8αλ) *cH*ctgβ	8α
400	265*H*+12.6αλ(7.95+*H*) +(315.7+12.6αλ)*cH*ctgβ	274.4*H*+12.6αλ(7.95+*H*) +(324.4+12.6αλ) *cH*ctgβ	283*H*+12.6αλ(7.95+*H*) +(333+12.6αλ) *cH*ctgβ	291.6*H*+12.6αλ(7.95+*H*) +(341.6+12.6αλ) *cH*ctgβ	12.6α
500	345.2*H*+19αλ (7.95+*H*) +(420.2+19αλ) *cH*ctgβ	356.4*H*+19αλ(7.95+*H*) +(431.4+19αλ) *cH*ctgβ	367.6*H*+19αλ(7.95+*H*) +(442.6+19αλ) *cH*ctgβ	378.8*H*+19αλ(7.95+*H*) +(453.8+19αλ) *cH*ctgβ	19α
650	438*H*+28.6αλ (7.95+*H*) +(570+28.6αλ) *cH*ctgβ	451.8*H*+28.6αλ(7.95+*H*) +(583.8+28.6αλ) *cH*ctgβ	465*H*+28.6αλ(7.95+*H*) +(597.6+28.6αλ)*cH*ctgβ	479.4*H*+28.6αλ(7.95+*H*) +(611.4+28.6αλ) *cH*ctgβ	28.6α
800	443.3*H*+40αλ (7.95+*H*) +(612.3+40αλ) *cH*ctgβ	460.5*H*+40αλ(7.95+*H*) +(629.5+40αλ) *cH*ctgβ	477.8*H*+40αλ(7.95+*H*) +(646.8+40αλ) *cH*ctgβ	495*H*+40αλ(7.95+*H*)+ +(664+40αλ) *cH*ctgβ	40α
1 000	524*H*+56.3αλ(7.95+*H*) +(716.6+56.3αλ)*cH*ctgβ	545.3*H*+56.3αλ(7.95+*H*) +(737.3+56.3αλ) *cH*ctgβ	566*H*+56.3αλ(7.95+*H*)+ +(758+56.3αλ) *cH*ctgβ	586.7*H*+56.3αλ(7.95+*H*) +(778.7+56.3αλ) *cH*ctgβ	56.25α

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Table 15

Traction force at the point 4 at shallow buckets

Bucket width B_b, mm	Traction force at the belt with $i=3$ S_4, N	Traction force at the belt with $i=4$ S_4, N	Traction force at the belt with $i=5$ S_4, N	Traction force at the belt with $i=6$ S_4, N	Elevator productivity, t/h
100	32.1*H*+0.5αλ(7.95+*H*) +(72.1+0.5αλ) *cH*ctgβ	34.3*H*+0.5αλ(7.95+*H*) +(74.3+0.5αλ) *cH*ctgβ	36.4*H*+0.5αλ(7.95+*H*) +(76.4+0.5αλ) *cH*ctgβ	38.6*H*+0.5αλ(7.95+*H*) +(78.6+0.5αλ) *cH*ctgβ	0.5α
125	33.4*H*+0.66αλ(7.95+*H*)+(73.4+0.66αλ) *cH*ctgβ	36*H*+0.66αλ(7.95+*H*) +(76+0.66αλ) *cH*ctgβ	37.8*H*+0.66αλ(7.95+*H*) +(77.8+0.66αλ) *cH*ctgβ	40.4*H*+0.66αλ(7.95+*H*) +(80.4+0.66αλ) *cH*ctgβ	0.66α
160	41.5*H*+1.17αλ(7.95+*H*) +(81.5+1.17αλ) *cH*ctgβ	44.9*H*+1.17αλ(7.95+*H*) +(84.9+1.17αλ) *cH*ctgβ	48.4*H*+1.17αλ(7.95+*H*) +(88.4+1.17αλ) *cH*ctgβ	51.8*H*+1.17αλ(7.95+*H*) +(91.8+1.17αλ) *cH*ctgβ	1.17α
200	61.9*H*+1.87αλ(7.95+*H*) +(101.9+1.87αλ)*cH*ctgβ	66.2*H*+1.87αλ(7.95+*H*) +(106.2+1.87αλ)*cH*ctgβ	70.5*H*+1.87αλ(7.95+*H*) +(110.5+1.87αλ)*cH*ctgβ	74.8*H*+1.87αλ(7.95+*H*) +(114.8+1.87αλ)*cH*ctgβ	1.87α
250	79.1*H*+3.5αλ(7.95+*H*)+ +(119.1+3.5αλ)*cH*ctgβ	84.3*H*+3.5αλ(7.95+*H*)+ +(124.3+3.5αλ)*cH*ctgβ	89.4*H*+3.5αλ(7.95+*H*)+ +(139.4+3.5αλ)*cH*ctgβ	94.6*H*+3.5αλ(7.95+*H*) +(134.6+3.5αλ)*cH*ctgβ	3.5α
320	138.1*H*+5.4αλ(7.95+*H*) +(178.1+5.4αλ)*cH*ctgβ	145*H*+5.4αλ(7.95+*H*) +(185+5.4αλ)*cH*ctgβ	151.1*H*+5.4αλ(7.95+*H*) +(191.1+5.4αλ)*cH*ctgβ	158*H*+5.4αλ(7.95+*H*)+ +(198+5.4αλ)*cH*ctgβ	5.4α
400	246.1*H*+8.4αλ(7.95+*H*) +(296.1+8.4αλ)*cH*ctgβ	254.8*H*+8.4αλ(7.95+*H*) +(304.8+8.4αλ)*cH*ctgβ	263.4*H*+8.4αλ(7.95+*H*) +(313.4+8.4αλ)*cH*ctgβ	272*H*+8.4αλ(7.95+*H*)+ +(322+8.4αλ)*cH*ctgβ	8.4α

Table 16

Traction force at the point 1 at deep buckets

Bucket width B_b, mm	Traction force at the belt with $i=3$ S_1, N	Traction force at the belt with $i=4$ S_1, N	Traction force at the belt with $i=5$ S_1, N	Traction force at the belt with $i=6$ S_1, N	Elevator productivity, t/h
1	2	3	4	5	6
100	37*H*+5αλ+ +57*cH*ctgβ	39.2*H*+5αλ+ +59.2*cH*ctgβ	41.3*H*+5αλ+ +61.3*cH*ctgβ	43.5*H*+5αλ+ +63.5*cH*ctgβ	α
125	36.4*H*+6.5αλ+ +56.4*cH*ctgβ	39*H*+6.5αλ+ +59*cH*ctgβ	41.6*H*+6.5αλ+ +61.6*cH*ctgβ	44.2*H*+6.5αλ+ +64.2*cH*ctgβ	1.3α
160	47.7*H*+10αλ+ +67.7*cH*ctgβ	51.1*H*+10αλ+ +71.1*cH*ctgβ	54.6*H*+10αλ+ +74.6*cH*ctgβ	58*H*+10αλ+ +78*cH*ctgβ	2α

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End of table 16

Bucket width B_b, mm	Traction force at the belt with $i=3$ S_1, N	Traction force at the belt with $i=4$ S_1, N	Traction force at the belt with $i=5$ S_1, N	Traction force at the belt with $i=6$ S_1, N	Elevator productivity, t/h
1	2	3	4	5	6
200	74.1*H*+16.2αλ+ +94.1*cH*ctgβ	78.4*H*+16.2αλ+ +98.4*cH*ctgβ	82.7*H*+16.2αλ+ +102.7*cH*ctgβ	87*H*+16.2αλ+ +97*cH*ctgβ	3.24α
250	103.6*H*+25αλ+ +123.6*cH*ctgβ	108.8*H*+25αλ+ +128.8*cH*ctgβ	113.9*H*+25αλ+ +133.9*cH*ctgβ	119.1*H*+25αλ+ +139.1*cH*ctgβ	5α
320	138.1*H*+40αλ+ +158.1*cH*ctgβ	145*H*+40αλ+ +165*cH*ctgβ	151.1*H*+40αλ+ +171.1*cH*ctgβ	158*H*+40αλ+ +178*cH*ctgβ	8α

Table 17

Traction force at the point 1 at shallow buckets

Bucket width B_b, mm	Tractive effort at the belt with $i=3$ S_1, N	Tractive effort at the belt with $i=4$ S_1, N	Tractive effort at the belt with $i=5$ S_1, N	Tractive effort at the belt with $i=6$ S_1, N	Elevator productivity, t/h
100	32.1*H*+2.5αλ+ +52.1*cH*ctgβ	34.3*H*+2.5αλ+ +54.3*cH*ctgβ	36.4*H*+2.5αλ+ +56.4*cH*ctgβ	38.6*H*+2.5αλ+ +58.6*cH*ctgβ	0.5α
125	33.4*H*+3.3αλ+ +53.4*cH*ctgβ	36*H*+3.3αλ+ +56*cH*ctgβ	37.8*H*+3.3αλ+ +57.8*cH*ctgβ	40.4*H*+3.3αλ+ +60.4*cH*ctgβ	0.66α
160	41.5*H*+5.85αλ+ +61.5*cH*ctgβ	44.9*H*+5.85αλ+ +64.9*cH*ctgβ	48.4*H*+5.85αλ+ +68.4*cH*ctgβ	51.8*H*+5.85αλ+ +71.8*cH*ctgβ	1.17α
200	61.9*H*+9.35αλ+ +81.9*cH*ctgβ	66.2*H*+9.35αλ+ +86.2*cH*ctgβ	70.5*H*+9.35αλ+ +90.5*cH*ctgβ	74.8*H*+9.35αλ+ +94.8*cH*ctgβ	1.87α
250	79.1*H*+17.5αλ+ +99.1*cH*ctgβ	84.3*H*+17.5αλ+ +104.3*cH*ctgβ	89.4*H*+17.5αλ+ +109.4*cH*ctgβ	94.6*H*+17.5αλ+ +114.6*cH*ctgβ	3.5α
320	138.1*H*+27αλ+ +158.1*cH*ctgβ	145*H*+27αλ+ +165*cH*ctgβ	151.1*H*+27αλ+ +171.1*cH*ctgβ	158*H*+27αλ+ +178*cH*ctgβ	5.4α
400	246.1*H*+42αλ+ +271.1*cH*ctgβ	254.8*H*+42αλ+ +279.8*cH*ctgβ	263.4*H*+42αλ+ +288.4*cH*ctgβ	272*H*+42αλ+ +297*cH*ctgβ	8.4α

Table 18

Tractive effort on the drive drum at deep buckets

Bucket width B_b, mm	Tractive effort at the belt with $i=3$ F, N	Tractive effort at the belt with $i=4$ F, N	Tractive effort at the belt with $i=5$ F, N	Tractive effort at the belt with $i=6$ F, N	Elevator productivity, t/h
1	2	3	4	5	6
100	5.9H+αλ(4+1.08H)+ +(30.7+1.08αλ)cHctgβ	6.3H+αλ (4+1.08H)+ +(31.1+1.08αλ)cHctgβ	6.6H+αλ (4+1.08H)+ +(31.4+1.08αλ)cHctgβ	7H+αλ (4+1.08H)+ +(31.8+1.08αλ)cHctgβ	α

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End of table 18

Bucket width B_b, mm	Tractive effort at the belt with $i = 3$ F, N	Tractive effort at the belt with $i = 4$ F, N	Tractive effort at the belt with $i = 5$ F, N	Tractive effort at the belt with $i = 6$ F, N	Elevator productivity, t/h
1	2	3	4	5	6
125	5.82*H*+1.3αλ(4+1.08*H*) +(30.6+1.4αλ) *cH*ctgβ	6.2*H*+1.3αλ (4+1.08*H*) +(31+1.4αλ) *cH*ctgβ	6.7*H*+1.3αλ (4+1.08*H*) +(31.5+1.4αλ) *cH*ctgβ	7.1*H*+1.3αλ (4+1.08*H*)+ +(31.9+1.4αλ) *cH*ctgβ	1.3α
160	7.63*H*+2αλ (4+1.08*H*) +(32.4+2.16αλ) *cH*ctgβ	8.2*H*+2αλ (4+1.08*H*)+ +(33+2.16αλ) *cH*ctgβ	8.7*H*+2αλ (4+1.08*H*)+ +(33.5+2.16αλ) *cH*ctgβ	9.3*H*+2αλ (4+1.08*H*)+ +(34.1+2.16αλ) *cH*ctgβ	2α
200	11.9*H*+3.24αλ(4+1.08*H*) +(36.7+3.5αλ) *cH*ctgβ	12.5*H*+3.24αλ(4+1.08*H*) +(37.3+3.5αλ) *cH*ctgβ	13.2*H*+3.24αλ(4+1.08*H*) +(38 +3.5αλ) *cH*ctgβ	13.9*H*+3.24αλ (4+1.08*H*) +(38.7+3.5αλ) *cH*ctgβ	3.24α
250	16.6*H*+5αλ (4+1.08*H*)+ +(41.4+5.4αλ) *cH*ctgβ	17.4*H*+5αλ (4+1.08*H*)+ +(42.2+5.4αλ) *cH*ctgβ	18.2*H*+5αλ (4+1.08*H*)+ +(43+5.4αλ) *cH*ctgβ	19.1*H*+5αλ (4+1.08*H*)+ +(43.9+5.4αλ) *cH*ctgβ	5α
320	22.1*H*+8αλ (4+1.08*H*)+ +(46.9+8.64αλ) *cH*ctgβ	23.2*H*+8αλ (4+1.08*H*)+ +(48+8.64αλ) *cH*ctgβ	24.2*H*+8αλ (4+1.08*H*)+ +(49+8.64αλ) *cH*ctgβ	25.3*H*+8αλ (4+1.08*H*)+ +(50.1+8.64αλ) *cH*ctgβ	8α
400	42.5*H*+12.6αλ(4+1.08*H*) +(73.5+13.6αλ) *cH*ctgβ	43.9*H*+12.6αλ(4+1.08*H*) +(74.9+13.6αλ) *cH*ctgβ	45.3*H*+12.6αλ(4+1.08*H*) +(76.3+13.6αλ) *cH*ctgβ	46.7*H*+12.6αλ (4+1.08*H*) +(77.7+13.6αλ) *cH*ctgβ	12.6α
500	55.2*H*+19αλ (4+1.08*H*)+ +(101.7+20.5αλ) *cH*ctgβ	57*H*+19αλ (4+1.08*H*)+ +(103.5+20.5αλ) *cH*ctgβ	58.8*H*+19αλ (4+1.08*H*) +(105.3+20.5αλ) *cH*ctgβ	60.6*H*+19αλ (4+1.08*H*) +(107.1+20.5αλ) *cH*ctgβ	19α
650	70.1*H*+28.6αλ(4+1.08*H*) +(167.8+30.9αλ) *cH*ctgβ	72.3*H*+28.6αλ(4+1.08*H*) +(170+30.9αλ) *cH*ctgβ	74.5*H*+28.6αλ(4+1.08*H*) +(172.2+30.9αλ) *cH*ctgβ	76.7*H*+28.6αλ (4+1.08*H*) +(174.4+30.9αλ) *cH*ctgβ	28.6α
800	70.9*H*+40αλ (4+1.08*H*) +(196+43.2αλ) *cH*ctgβ	73.7*H*+40αλ (4+1.08*H*) +(198.8+43.2αλ) *cH*ctgβ	76.4*H*+40αλ (4+1.08*H*) +(201.5+43.2αλ) *cH*ctgβ	79.2*H*+40αλ (4+1.08*H*) +(204.3+43.2αλ) *cH*ctgβ	40α
1000	83.9*H*+56.3αλ(4+1.08*H*) +(202.9+60.8αλ) *cH*ctgβ	87.2*H*+56.3αλ(4+1.08*H*) +(206.2+60.8αλ) *cH*ctgβ	90.6*H*+56.3αλ(4+1.08*H*) +(209.6+60.8αλ) *cH*ctgβ	93.9*H*+56.3αλ (4+1.08*H*) +(212.9+60.8αλ) *cH*ctgβ	56.25α

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Table 19

Tractive effort on the drive drum at shallow buckets

Bucket width B_b, mm	Tractive effort at the belt with $i=3$ F, N $i=3$ F, N	Tractive effort at the belt with $i=3$ F, N $i=4$ F, N	Tractive effort at the belt with $i=3$ F, N $i=5$ F, N	Tractive effort at the belt with $i=3$ F, N $i=6$ F, N	Elevator productiv-ity, t/h
100	5.1*H*+αλ (4+1.08*H*)+ +(30+1.08αλ) *cH*ctgβ	5.5*H*+αλ (4+1.08*H*)+ +(30.3+1.08αλ)*cH*ctgβ	5.8*H*+αλ (4+1.08*H*)+ +(30.6+1.08αλ)*cH*ctgβ	6.2*H*+αλ (4+1.08*H*)+ +(31+1.08αλ) *cH*ctgβ	0.5α
125	5.3*H*+1.3αλ(4+1.08*H*) +(30.1+1.4αλ) *cH*ctgβ	5.8*H*+1.3αλ(4+1.08*H*) +(30.6+1.4αλ) *cH*ctgβ	6.0*H*+1.3αλ(4+1.08*H*) +(30.8+1.4αλ) *cH*ctgβ	6.5*H*+1.3αλ(4+1.08*H*) +(31.3+1.4αλ) *cH*ctgβ	0.66α
160	6.6*H*+2αλ (4+1.08*H*)+ +(31.4+2.16αλ)*cH*ctgβ	7.2*H*+2αλ (4+1.08*H*)+ +(32+2.16αλ) *cH*ctgβ	7.7*H*+2αλ (4+1.08*H*)+ +(32.5+2.16αλ)*cH*ctgβ	8.3*H*+2αλ (4+1.08*H*)+ +(33.1+2.16αλ)*cH*ctgβ	1.17α
200	9.9*H*+3.24αλ(4+1.08*H*) +(34.7+3.5αλ) *cH*ctgβ	10.6*H*+3.24αλ(4+1.08*H*) +(35.4+3.5αλ) *cH*ctgβ	11.3*H*+3.24αλ(4+1.08*H*) +(36.1+3.5αλ) *cH*ctgβ	12*H*+3.24αλ (4+1.08*H*)+ +(36.8+3.5αλ) *cH*ctgβ	1.87α
250	12.7*H*+5αλ (4+1.08*H*)+ +(37.5+5.4αλ) *cH*ctgβ	13.5*H*+5αλ (4+1.08*H*)+ +(38.3+5.4αλ) *cH*ctgβ	14.3*H*+5αλ (4+1.08*H*)+ +(39.1+5.4αλ) *cH*ctgβ	15.1*H*+5αλ (4+1.08*H*)+ +(39.9+5.4αλ) *cH*ctgβ	3.5α
320	22.1*H*+8αλ (4+1.08*H*)+ +(46.9+8.6αλ) *cH*ctgβ	23.2*H*+8αλ (4+1.08*H*)+ +(48+8.6αλ) *cH*ctgβ	24.2*H*+8αλ (4+1.08*H*)+ +(49+8.6αλ) *cH*ctgβ	25.3*H*+8αλ (4+1.08*H*)+ +(50.1+8.6αλ) *cH*ctgβ	5.4α
400	39.4*H*+12.6αλ(4+1.08*H*) +(70.4+13.6αλ) *cH*ctgβ	40.8*H*+12.6αλ(4+1.08*H*) +(71.8+13.6αλ) *cH*ctgβ	42.1*H*+12.6αλ(4+1.08*H*) +(73.1+13.6αλ) *cH*ctgβ	43.5*H*+12.6αλ(4+1.08*H*) +(74.5+13.6αλ) *cH*ctgβ	8.4α

Estimated kinematic scheme of the elevator's drive is shown in the Fig. 2.

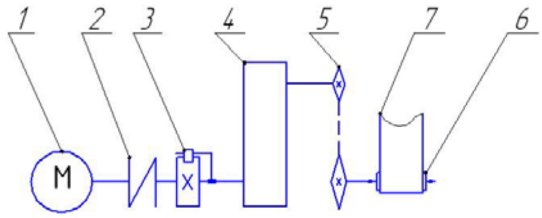

Fig. 2. Scheme of bucket elevator drive:
1 – engine; *2* – elastic clutch; *3* – locking device (ratchet); *4* – reducing gear; *5* – chain transmission; *6* – drive drum; *7* – belt

Efficiency coefficient of the drive is determined by the formula:

$$\eta = \eta_g \eta_{ch} \eta_c , \qquad (20)$$

where $\eta_g = 0,96$ – efficiency coefficient of reducing gear; $\eta_{ch} = 0,95$ – efficiency coefficient of chain transmission; $\eta_c = 0,98$ – efficiency coefficient of clutch.

Thus,

$$\eta = \eta_g \eta_{ch} \eta_c = 0,96 \cdot 0,95 \cdot 0,98 = 0,89 .$$

Engine power is determined by the formula:

$$P = \frac{F_o v}{1\,000\eta} . \qquad (21)$$

Calculated power of the engine is determined by the formula:

$$P_g = n_u P , \qquad (22)$$

where $n_u = 1,1...1,2$ – is the safety factor.

Since $\eta = 0,89$ and $n_u = 1,1$ then using the formulas (21) and (22) we obtain the following:

$$P_g = \frac{F_o v}{1\,000\eta} = 0,001 F_o v . \qquad (23)$$

Dependence of the calculated engine power on the values of design performance, bucket type, number of insert plies of the belt, speed of the belt movement and lifting height of cargo calculated using the formula (23) taking into account data from the Tables 18-19 are summarized in the Tables 20-21:

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Table 20

Calculated power of engine at deep buckets

Bucket width B_b, mm	Engine power at the belt with $i = 3$ P, W	Engine power at the belt with $i = 4$ P, W	Engine power at the belt with $i = 5$ P, W	Engine power at the belt with $i = 6$ P, W	Elevator productiv-ity, t/h
1	2	3	4	5	6
100	*v* [5.9*H*+αλ (4+1.08*H*)+ +(30.7+1.08αλ)*cH*ctgβ]	*v* [6.3*H*+αλ (4+1.08*H*)+ +(31.1+1.08αλ)*cH*ctgβ]	*v* [6.6*H*+αλ (4+1.08*H*)+ +(31.4+1.08αλ)*cH*ctgβ]	*v*[7*H*+αλ (4+1.08*H*)+ +(31.8+1.08αλ) *cH*ctgβ]	α
125	[5.82*H*+1.3αλ(4+1.08*H*) +(30.6+1.4αλ)*cH*ctgβ]*v*	[6.2*H*+1.3αλ (4+1.08*H*) +(31+1.4αλ) *cH*ctgβ]*v*	[6.7*H*+1.3αλ (4+1.08*H*) +(31.5+1.4αλ) *cH*ctgβ]*v*	[7.1*H*+1.3αλ(4+1.08*H*)+ +(31.9+1.4αλ)*cH*ctgβ]*v*	1.3α
160	*v*[7.63*H*+2αλ (4+1.08*H*) +(32.4+2.16αλ) *cH*ctgβ]	[8.2*H*+2αλ (4+1.08*H*)+ +(33+2.16αλ) *cH*ctgβ]*v*	[8.7*H*+2αλ (4+1.08*H*)+ +(33.5+2.16αλ)*cH*ctgβ]*v*	*v*[9.3*H*+2αλ (4+1.08*H*)+ +(34.1+2.16αλ) *cH*ctgβ]	2α
200	[11.9*H*+3.2αλ(4+1.08*H*) +(36.7+3.5αλ) *cH*ctgβ]*v*	[12.5*H*+3.2αλ(4+1.08*H*) +(37.3+3.5αλ) *cH*ctgβ]*v*	[13.2H+3.2αλ(4+1.08*H*) +(38 +3.5αλ) *cH*ctgβ]*v*	[13.9*H*+3.2αλ(4+1.08*H*) +(38.7+3.5αλ) *cH*ctgβ]*v*	3.24α
250	[16.6*H*+5αλ (4+1.08*H*)+ +(41.4+5.4αλ) *cH*ctgβ]*v*	[17.4*H*+5αλ (4+1.08*H*)+ +(42.2+5.4αλ) *cH*ctgβ]*v*	[18.2*H*+5αλ(4+1.08*H*)+ +(43+5.4αλ)*cH*ctgβ]*v*	[19.1*H*+5αλ (4+1.08*H*)+ +(43.9+5.4αλ) *cH*ctgβ]*v*	5α
320	[22.1*H*+8αλ (4+1.08*H*)+ +(46.9+8.64αλ)*cH*ctgβ]*v*	[23.2*H*+8αλ (4+1.08*H*)+ +(48+8.64αλ) *cH*ctgβ]*v*	[24.2*H*+8αλ (4+1.08*H*)+ +(49+8.64αλ)*cH*ctgβ]*v*	[25.3*H*+8αλ (4+1.08*H*)+ +(50.1+8.64αλ)*cH*ctgβ]*v*	8α
400	[42.5*H*+12.6αλ(4+1.08*H*) +(73.5+13.6αλ) *cH*ctgβ]*v*	[43.9*H*+12.6αλ(4+1.08*H*) +(74.9+13.6αλ) *cH*ctgβ]*v*	[45.3*H*+12.6αλ(4+1.08*H*) +(76.3+13.6αλ) *cH*ctgβ]*v*	[46.7*H*+12.6αλ(4+1.08*H*) +(77.7+13.6αλ) *cH*ctgβ]*v*	12.6α
500	[55,2*H*+19αλ (4+1,08*H*)+ +(101,7+20,5αλ)*cH*ctgβ]*v*	*v*[57*H*+19αλ (4+1,08*H*)+ +(103,5+20,5αλ) *cH*ctgβ]	*v*[58,8*H*+19αλ (4+1,08*H*) +(105,3+20,5αλ) *cH*ctgβ]	*v*[60,6*H*+19αλ (4+1,08*H*) +(107,1+20,5αλ) *cH*ctgβ]	19α
650	[70,1*H*+28,6αλ(4+1,08*H*) +(167,8+30,9αλ)*cH*ctgβ]*v*	[72,3*H*+28,6αλ(4+1,08*H*) +(170+30,9αλ) *cH*ctgβ]*v*	[74,5*H*+28,6αλ(4+1,08*H*) +(172,2+30,9αλ)*cH*ctgβ]*v*	[76,7*H*+28,6αλ(4+1,08*H*) +(174,4+30,9αλ)*cH*ctgβ]*v*	28,6α
800	[70,9*H*+40αλ (4+1,08*H*) +(196+43,2αλ) *cH*ctgβ]*v*	[73,7*H*+40αλ (4+1,08*H*) +(198,8+43,2αλ)*cH*ctgβ]*v*	[76,4*H*+40αλ (4+1,08*H*) +(201,5+43,2αλ)*cH*ctgβ]*v*	[79,2*H*+40αλ (4+1,08*H*) +(204,3+43,2αλ)*cH*ctgβ]*v*	40α

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End of table 20

Bucket width B_b, mm	Engine power at the belt with $i = 3$ P, W	Engine power at the belt with $i = 4$ P, W	Engine power at the belt with $i = 5$ P, W	Engine power at the belt with $i = 6$ P, W	Elevator productivity, t/h
1000	[83,9*H*+56,3αλ(4+1,08*H*) +(202,9+60,8αλ)*cH* ctgβ]*v*	[87,2*H*+56,3αλ(4+1,08*H*) +(206,2+60,8αλ)*cH*ctgβ]*v*	[90,6*H*+56,3αλ(4+1,08*H*) +(209,6+60,8αλ)*cH*ctgβ]*v*	[93,9*H*+56,3αλ(4+1,08*H*) +(212,9+60,8αλ)*cH*ctgβ]*v*	56,25α

Table 21

Calculated power of engine at shallow buckets

Bucket width B_b, mm	Engine power at the belt with $i = 3$ P, W	Engine power at the belt with $i = 4$ P, W	Engine power at the belt with $i = 5$ P, W	Engine power at the belt with $i = 6$ P, W	Elevator productivity, t/h
100	[5,1H+αλ (4+1,08H)+ +(30+1,08αλ) cHctgβ]v	v[5,5H+αλ (4+1,08H)+ +(30,3+1,08αλ) cHctgβ]	v[5,8H+αλ (4+1,08H)+ +(30,6+1,08αλ) cHctgβ]	v[6,2H+αλ (4+1,08H)+ +(31+1,08αλ) cHctgβ]	0,5α
125	[5,3H+1,3αλ(4+1,08H)+ +(30,1+1,4αλ) cHctgβ]v	[5,8H+1,3αλ(4+1,08H)+ +(30,6+1,4αλ) cHctgβ]v	[6,0H+1,3αλ(4+1,08H)+ +(30,8+1,4αλ) cHctgβ]v	[6,5H+1,3αλ(4+1,08H)+ +(31,3+1,4αλ) cHctgβ]v	0,66α
160	[6,6H+2αλ (4+1,08H)+ +(31,4+2,16αλ)cHctgβ]v	[7,2H+2αλ (4+1,08H)+ +(32+2,16αλ) cHctgβ]v	[7,7H+2αλ (4+1,08H)+ +(32,5+2,16αλ)cHctgβ]v	[8,3H+2αλ (4+1,08H)+ +(33,1+2,16αλ)cHctgβ]v	1,17α
200	[9,9H+3,24αλ (4+1,08H) +(34,7+3,5αλ) cHctgβ]v	[10,6H+3,2αλ(4+1,08H) +(35,4+3,5αλ) cHctgβ]v	[11,3H+3,2αλ(4+1,08H) +(36,1+3,5αλ) cHctgβ]v	[12H+3,24αλ(4+1,08H) +(36,8+3,5αλ) cHctgβ]v	1,87α
250	[12,7H+5αλ(4+1,08H)+ +(37,5+5,4αλ) cHctgβ]v	[13,5H+5αλ (4+1,08H)+ +(38,3+5,4αλ) cHctgβ]v	[14,3H+5αλ (4+1,08H)+ +(39,1+5,4αλ) cHctgβ]v	[15,1H+5αλ (4+1,08H)+ +(39,9+5,4αλ) cHctgβ]v	3,5α
320	[22,1H+8αλ (4+1,08H)+ +(46,9+8,6αλ) cHctgβ]v	[23,2H+8αλ (4+1,08H)+ +(48+8,6αλ) cHctgβ]v	[24,2H+8αλ (4+1,08H)+ +(49+8,6αλ) cHctgβ]v	[25,3H+8αλ (4+1,08H)+ +(50,1+8,6αλ) cHctgβ]v	5,4α
400	[39,4H+12,6αλ(4+1,08H) +(70,4+13,6αλ) cHctgβ]v	[40,8H+12,6αλ(4+1,08H) +(71,8+13,6αλ) cHctgβ]v	[42,1H+12,6αλ(4+1,08H) +(73,1+13,6αλ) cHctgβ]v	[43,5H+12,6αλ(4+1,08H) +(74,5+13,6αλ) cHctgβ]v	8,4α

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Findings

Let us analyze the influence of design parameters of inclined bucket elevator for transportation of fine coal on the power of required drive. Taking into account the physical and mechanical properties of fine coal according to the recommendations presented in the work [9] it was selected the belt elevator with spaced deep buckets and centrifugal discharge. The speed of belt movement is $v = 1,6$ m/s; fill factor of the bucket $v = 1,6$; t/m^3 –density of fine coal; lift height of the cargo $H = 10$ m; inclination angle of elevator to the horizontal $\beta = 75^{\circ}$.

Under these conditions the coefficient are:

$$\alpha = 3,6 v\rho\psi = 3,6 \cdot 1,6 \cdot 1,0 \cdot 0,8 = 4,61 \text{ (t m/l h);}$$

$$\alpha\lambda = 3,6 v\rho\psi \frac{g}{3,6v} = \rho\psi g =$$

$$= 1,0 \cdot 0,8 \cdot 9,8 = 7,84 . \quad \text{(N/m)}$$

At this the dependence of calculated power of electric engine of the elevator's bucket on the design performance is given in the Table 22.

Taking into account standard values of power of three-phase asynchronous squirrel cage motors of 4A series with synchronous frequency of rotation 1000 rev/min for the drive of inclined elevator for transportation of fine coal it was compiled the table of correspondence of design performance and the required engine power.

Analyzing results of calculations presented in the Table 23 it can be concluded that the dependence of elevator drive power on its design performance (at fixed lift height, type of cargo, the angle of inclination to horizontal) in general is a piecewise constant monotonically increasing function. At this the productivity values given in the last column of the Table 23 should be considered as such, in which the power value varies and is equal to the appropriate value given in the second column of the Table 23. But to the value of 4.61 t/h the power is 0.75 kW due to the minimum of such power in the engines of such class. According to calculations it was plotted the dependence of inclined elevator drive for fine coal transportation on the value of design productivity (Fig. 3).

To determine the graphic dependence of elevator drive power on its inclination angle we take the initial data: transported material – fine coal; productivity $\text{Pr} = 20$ t/h lift height $H = 10$ m; speed of the belt movement $v = 1,6$ m/sec.

Table 22

Calculated power of the engine at deep buckets

Bucket width B_b, mm	Engine power at the belt with $i = 3$ P, W	Engine power at the belt with $i = 4$ P, W	Engine power at the belt with $i = 5$ P, W	Engine power at the belt with $i = 6$ P, W	Elevator productivity, t/h
100	520	533	543	555	4.61
125	614	627	651	670	6
160	899	918	934	953	9.22
200	1438	1457	1480	1502	14.9
250	2158	2184	2210	2239	23.1
320	3306	3341	3373	3409	36.9
400	5452.5	5493	5538	5588	58.1
500	7935	7988	8045	8109	87.6
650	11533	11603	11673	11746	131.8
800	15251	15341	15430	15519	184.4
1000	20939	21039	21144	21261	259.3

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Taking into account the fact that $\alpha = 4{,}61\ t$ t m/l h $\cdot$ and $\Pr = 20$ t/h for calculation of drive power the dependency in the 5th line and first column will be used (Tab. 20).

Substituting the initial data for calculation into resulting dependence we obtain:

$$P = 76{,}3 \cdot \text{ctg}\beta + 1751{,}2\,. \qquad (24)$$

Graphic dependence of value of elevator drive power when transporting fine coal with design productivity $\Pr = 20$ t/h on the angle of its inclination within $\beta = \pi/3 \ldots \pi/2$ is presented in the Fig. 4.

Table 23

Engine power at shallow buckets

Bucket width B_b, mm	Engine power P, kW	Engine type	Elevator productivity, t/h
100	0.75	4A80A6У3	4.61
125	0.75	4A80A6У3	6
160	1.1	4A80B6У3	9.22
200	1.5	4A90L6У3	14.9
250	2.2	4A100L6У3	23.1
320	4.0	4A112MB6У3	36.9
400	5.5	4A132S6У3	58.1
500	11.0	4A160S6У3	87.6
650	15.0	4A160M6У3	131.8
800	18.5	4A180M6У3	184.4
1000	30	4A200M6У3	259.3

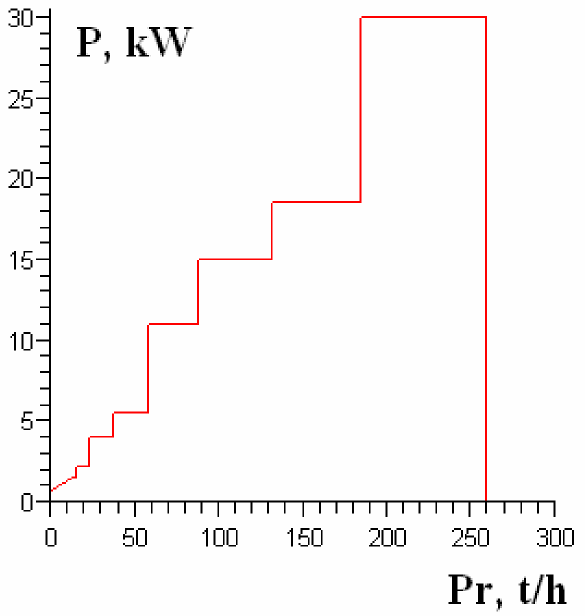

Fig. 3. Dependence of elevator drive power on the productivity

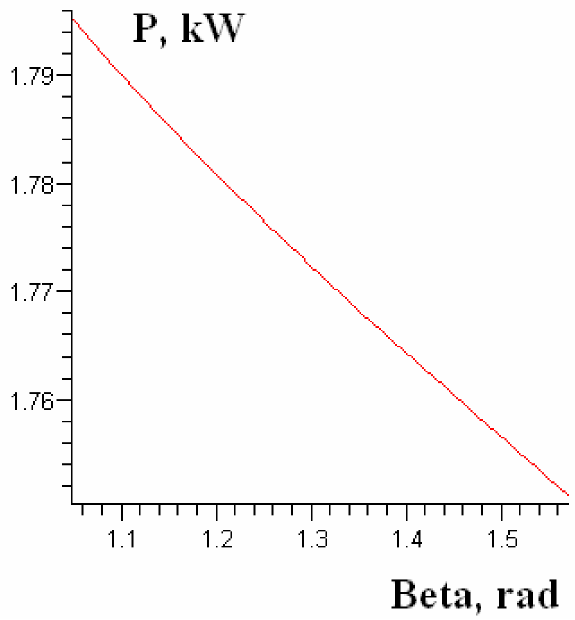

Fig. 4. Dependence of elevator drive power on the angle of inclination

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Originality and practical value

It was plotted the analytical dependence of elevator drive power on its design parameters (type and characteristics of the cargo, lifting height, inclination angle, productivity), which takes into account the standard sizes and types of buckets and belts.

Using this dependence makes it possible rapid determination of the approximate value of drive power of inclined elevators with deep and shallow buckets and performing high-quality selection of its key elements at the specific design characteristics.

Based on the proposed dependences it was plotted graphic dependence of power influence of required inclined elevator's drive on design productivity at the fixed lift height, inclination angle, and the type of cargo. It was also presented the graphic dependence of drive power on the inclination angle of elevator at the other fixed design parameters.

Conclusions

For inclined belt bucket elevators it was plotted analytical dependence of the drive power value on its design parameters. This makes it possible to obtain the required drive power value taking into account the type and physical and mechanical properties of the cargo, the value of lift height, inclination angle, design productivity and working conditions, involving only one calculation formula. As an example of involving the obtained results it was considered the process of plotting the dependence of drive power on the design productivity of elevator for fine coal transportation. For such elevator it was plotted the parametric and graphic dependence of drive power on design productivity and inclination angle of elevator taking into account the standard parameters of buckets and properties of electric engines. It was established that the function of varying the value of elevator power on the design productivity (at fixed lifting height, type of cargo¸ inclination angle) is piecewise and monotonically increasing, and the dependence of elevator power value on its inclination angle (at fixed design productivity, lift height, load type, the speed of belt movement) is non-linear and monotonically decreasing.

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АНАЛІЗ ВПЛИВУ ПРОЕКТНИХ ХАРАКТЕРИСТИК ПОХИЛОГО КІВШОВОГО ЕЛЕВАТОРА НА ПОТУЖНІСТЬ ЙОГО ПРИВОДУ

Мета. Одним із основних елементів похилих стрічкових ківшових елеваторів є їх привід. Для визначення потужності приводу необхідно виконати розрахунки за стандартними методиками, які наведені в сучасній літературі. Основними проектними параметрами таких елеваторів є продуктивність, висота підйому, тип та властивості транспортованого вантажу, кут нахилу. В роботі необхідно побудувати параметричну залежність потужності приводу елеватора від його проектних параметрів, яка враховувала б стандартні розміри і типи ковшів та стрічок. **Методика.** Використовуючи методику тягового розрахунку похилих стрічкових ківшевих елеваторів, побудовано параметричні залежності зусиль у характерних точках траси елеватора, а також залежності потужності приводу швидкохідних елеваторів із глибокими та мілкими ковшами від їх проектних параметрів та характеристик. **Результати.** На основі побудованих параметричних залежностей встановлено, що функція зміни величини потужності елеватора від проектної продуктивності (при фіксованих висоті підйому, типі вантажу, куті нахилу) є кусково-сталою та монотонно зростаючою. Побудовано графічну залежність потужності приводу елеватора від кута нахилу в допустимих межах його зміни. Отримана залежність є нелінійною та монотонно спадаючою. Визначені в загальному вигляді інтервали проектних значень продуктивності, що забезпечують постійну величину потужності приводу похилого елеватора. Як приклад залучення отриманих результатів розглянуто процес побудови залежностей потужності приводу від проектної продуктивності та кута нахилу елеватора для транспортування дрібного вугілля. **Наукова новизна.** Авторами вперше побудовані параметричні залежності потужності приводу похилого ківшевого елеватора від його проектних параметрів, які враховують стандартні розміри і типи ковшів та стрічок. **Практична значимість.** Використання побудованих залежностей дає можливість відносно швидкого визначення приблизного значення потужності приводу похилих швидкохідних елеваторів із глибокими та мілкими ковшами на стадії проектування, а також можливо виконати якісний підбір його основних елементів при конкретних проектних характеристиках: тип вантажу, продуктивність, висота підйому, кут нахилу.

Ключові слова: похилий елеватор;ківш; привід; потужність; продуктивність; вантаж; кут нахилу

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АНАЛИЗ ВЛИЯНИЯ ПРОЕКТНЫХ ХАРАКТЕРИСТИК НАКЛОННОГО КОВШОВОГО ЭЛЕВАТОРА НА МОЩНОСТЬ ЕГО ПРИВОДА

Цель. Одним из основных элементов наклонных ленточных ковшовых элеваторов является их привод. Для определения мощности привода необходимо провести расчеты по стандартным методикам, которые изложены в современной литературе. Основными проектными параметрами являются производительность, высота подъема, тип и свойства транспортированного материала, угол наклона. В работе необходимо построить параметрическую зависимость мощности привода элеватора от его проектных параметров, которая учитывала бы стандартные размеры и типы ковшей и лент. **Методика.** Используя методику тягового расчета наклонных ленточных ковшовых элеваторов, построены параметрические зависимости усилий в характерных точках трассы элеватора, а также зависимости мощности привода быстроходных элеваторов с глубокими и мелкими ковшами от их проектных параметров и характеристик. **Результаты.** На основе построенных параметрических зависимостей установлено, что функция изменения величины мощности элеватора от проектной производительности (при фиксированных высоте подъема, типе груза, скорости движения ленты) является кусочно-постоянной и монотонно возрастающей. Построена графическая зависимость мощности привода элеватора от угла наклона в допустимых пределах его изменения. Полученная зависимость является нелинейной и монотонно убывающей. Определены в общем виде интервалы проектных значений производительности, которые обеспечивают постоянную величину мощности привода наклонного элеватора. В качестве примера применения полученных результатов рассмотрен процесс построения зависимости мощности привода от проектной производительности и угла наклона элеватора для транспортировки мелкого угля. **Научная новизна.** Авторами впервые построены параметрические зависимости мощности привода наклонного ковшевого элеватора от его проектных параметров, которые учитывают стандартные размеры и типы ковшей и лент. **Практическая значимость.** Использование построенных зависимостей дает возможность относительно быстрого определения приблизительного значения мощности привода наклонных быстроходных элеваторов с глубокими и мелкими ковшами на стадии проектирования. А также можно выполнить качественный подбор его основных элементов при конкретных проектных характеристиках: типе груза, производительности, высоте подъема, угле наклона.

Ключевые слова: наклонный элеватор; ковш; привод; мощность; производительность; груз; угол наклона

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