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IMPROVEMENT OF CONSTRUCTIVE SCHEMES AND CALCULATIONS OF OSCILLATION TUBULAR CONVEYERS

Розглянуто питання створення вібраційних трубчастих конвеєрів, запропонована методика їх розрахунку. Вібраційні трубчасті конвеєри, дають змогу розширити технологічні можливості шляхом забезпечення транспортування сипких продуктів і утворення на їх базі транспортуючих ліній. Розглянуті основні вимоги на етапі проектування цих вібраційних конвеєрів, для забезпечення коливального руху і наведені різноманітні схеми кріплень пружних систем та експериментальні графічні залежності.

In this article the question of creation of oscillation tubular conveyers, has been considered and offered the method of their calculation. Oscillation tubular conveyers allow to extend technological possibilities by providing transportation of free-flowing products and to create on their base transport lines. The basic requirements are considered on the stage of designing these oscillation conveyers for providing fluctuating motion, and various schemes of mounting of the resilient systems and experimental graphic dependencies have been presented.

Introduction. Application of oscillation tubular conveyers with an electromagnetic drive in many industries for the purpose of transporting friable, lump, artificial materials and products in many cases is significantly more effective than the use of other types of conveyers. This is due to the following basic advantages:

1. Constructive simplicity of vibroconveyers.
2. Small consumption of power due to around resonance work mode.
3. Simplicity of adjusting and commissioning of conveyers working parameters.
4. Possibilities of reverse vibrotransportation for conveyers with elliptic law of motion of transported elements.
5. Reliable work in the conditions of gassed, dusted, and other similar environments.

Despite mentioned and also other advantages these devices have not received wide application. This can be explained due to the absence of serial standard highly productive conveyers on the market, lack of education of production workers and production managers about this effective technique and its possibilities, and extremely low amount of specialists in this field.

Also here can be added the unwillingness of private entities to implement the new technique, and to invest in this and many other directions of new and effective techniques.

Rising of problem. The first standards of small oscillation tubular conveyers with an electromagnetic drive appeared in 1960-ies of XX century. The biggest successes in creation of conveyers was attained by the scientific research laboratory NDL-40 of Lviv Polytechnic Institute (now National University «Lviv Politechnic»), where under the direction of the known specialists in the field of vibrotechnics V.O. Povidajlo and V.A. Shchigel' a range of effective samples of small oscillation tubular conveyers with electromagnetic drive was developed. Some constructive schemes of these machines are given in [1, 3, 5] and patented as copyright certificates.

In the 1980-ies of XX century there was done a considerable step in creation of more effective types of tubular conveyers, the productivity and speed of transporting of which and wide range of variable vibrations provided higher performance parameters in comparison with the before created. These works were conducted in NDL-40 of Lviv Polytechnic that in the 1980-ies became the lead in creation of vibrotechics in Soviet Union, making hundreds of models of various oscillation machines by the orders of state enterprises. As this direction of technique has a big future, there is a necessity of improving and optimization of flow, dynamic and constructive scheme of oscillation conveyers, method of their calculation, constructing, and adjusting.

Analysis of the latest researches. The latest researches are being done on double mass oscillation tubular conveyers with the electromagnetic drive of type "pipe in pipe" with the directed and independent (elliptic) vibrations of transporting element. These are small conveyers with the length of transporting pipe from a 0.15 m a to 3.5 m with the internal diameter of steel or duralumin pipes from 20 to 150 mm. These conveyers allow to transport various free-flowing, lump, pulverulent, artificial materials and products with the speed of 600-700 mm/sec at vibration frequencies of 12.5÷50Hz, consuming insignificant power for this purpose (40÷400 W).

Presentation of basic material. Productivity of transporting products by a tubular conveyer which is determined as (1) depends on the followings basic parameters:

1. Speed of transporting V_{tr}, m/h, [m/min.].
2. Area of layer of the transported material s, m^2.
3. Density of the transported material γ, kg/m^3.
4. Fillfactor k_f

$$Q = V_{tr} \cdot s \cdot \gamma \cdot k_f \quad (1)$$

Speed of transporting is determined by dependence (2).

$$V_{tr} = 2 \cdot \pi \cdot \nu \cdot A_H \cdot k_{sp} \quad (2)$$

where ν is the frequency of vibrations of transporting element, Hz; A_H is a horizontal constituent of amplitude of transported element vibrations, m; k_{sp} is a coefficient of speed, which shows efficiency of vibrations use, and depends on many factors, the main of which are the following:

1. Trajectory of transported element vibrations.
2. Parameter of the mode (overload) w [3].
3. A corner of vibration β for conveyers with the directed vibrations.
4. The angle of transported element to the horizon α.
5. Coefficient of friction between transporting surfaces and transported material.
6. Frequency of transported element vibrations.
7. Evenness of transporting speed through the length of transported element.
8. Methods of transported element loading.
9. Own frequency of transported elements vibrations.
10. Other features of the transported material, volume of the transported mass and feature of construction.

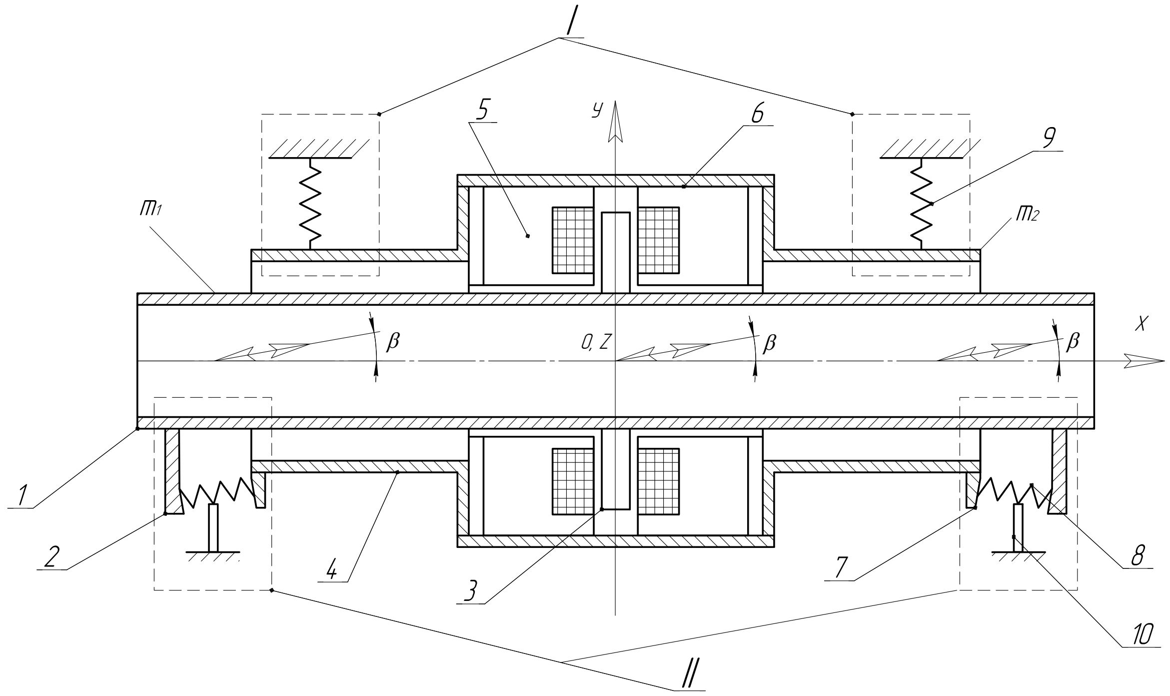
Analysis (1) and (2) show that for the increase of productivity it is necessary to increase the frequency of vibrations, amplitude of vibrations, coefficient of speed, and size of product layer area that is being transported.

We base our analysis on the most optimum flow diagram of tubular conveyer (drawing 1).

This scheme of "pipe in pipe" type is based on a double mass swaying scheme. Working transporting mass m_1 consists of transporting pipe 1 with holders 2 fastened hard to it for fastening of the resilient system, and circular anchor 3, forming together a hard construction. Reactive mass m_2 consists of two pipes 4, to which two circular electromagnets 5 are connected and holders 7 for fastening the springs. Through a ring 6 two symmetric halves of mass of m_2 are connected hard between themselves, forming a reactive swaying (non-transporting) mass. The masses m_1 and m_2 are connected between themselves with flat resilient elements 8, which set vibration direction at an angle β to the horizon. The construction can have two methods of hanging:

I – Hanging by the reactive mass with the resilient elements 9;

II – Hanging in neutral ("zero") zones of the resilient oscillating system through supports 10.



Drawing 1. Scheme of tubular vibroconveyer

The proposed scheme is symmetric in three co-ordinates of x, y, z; barycenters m_1 p. 0_1 and m_2 p. 0_2 match. Due to it parasite angular vibrations around general centre of mass are practically fully removed, which provides the even field of vibrations with the set vibration angle β through the length of working transporting element. In this scheme electromagnets 5 work in a double tact mode, and together with the resilient systems 8 create antiphase vibrations of masses of m_1 and m_2 with the amplitudes back proportional to the masses. Products are being transported in the direction of vibrations at an angle β (from left to right) in the detachable or non-detachable modes that are determined by overload parameter of *w* (w<1 – non-detachable; w>1 – detachable).

$$W=\frac{A_V \cdot \omega^2}{q \cdot \cos\alpha} \qquad (3)$$

where A_V is a vertical constituent of vibrations amplitude; $\omega = 2\pi\nu$ is the angular vibrations frequency; α – is the angle of conveyer slope to the horizon.

In order to increase conveyers technical parameters we offer a number of constructive schemes of resilient system, on which the conveyer work largely depends (drawing 2).

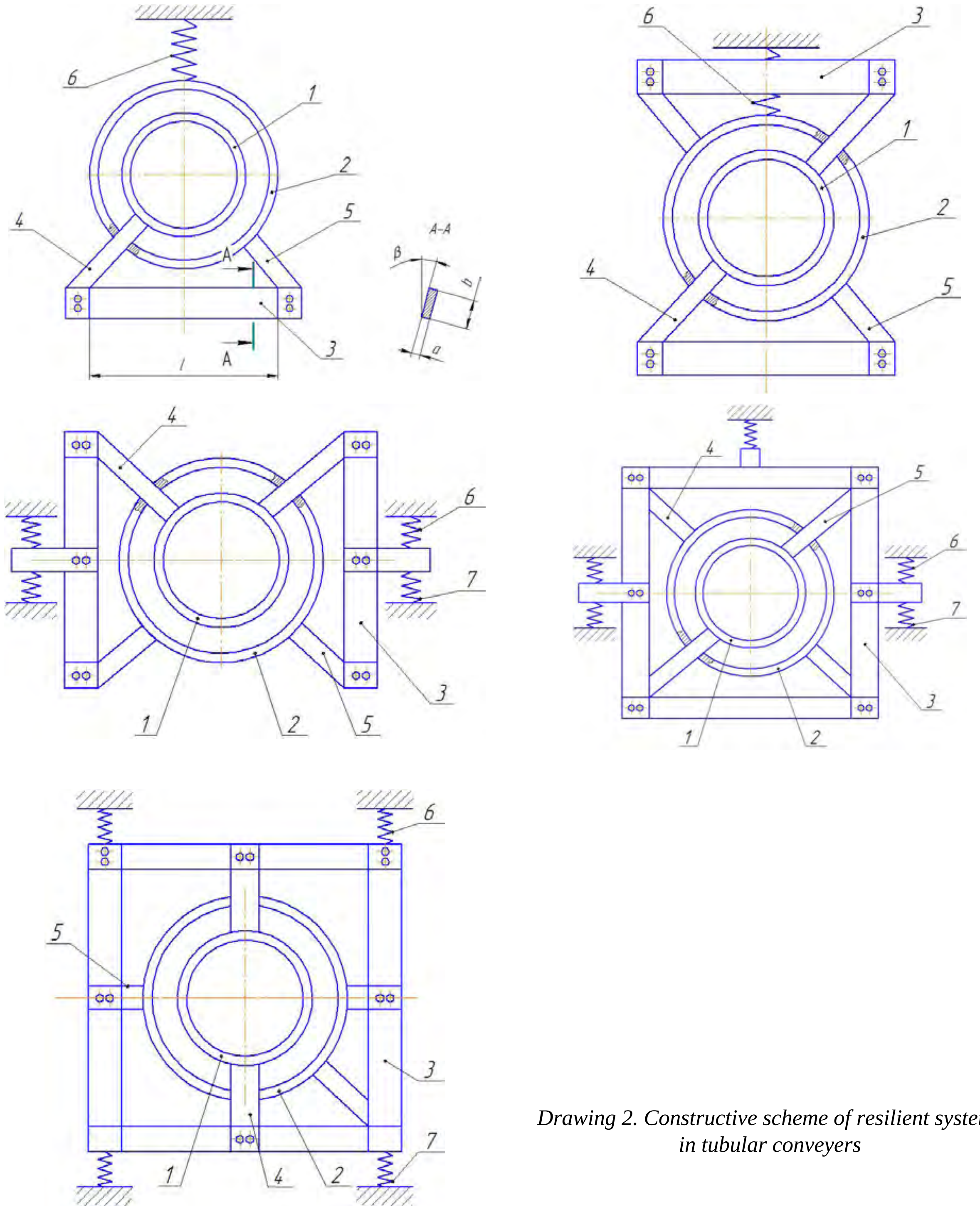
On the scheme (drawing 2.1) masses 1 and 2 with values m_1 and m_2 are connected with flat springs 3, placed underneath at an angle β to the vertical plane. Springs are fastened with their ends to the holders of swaying masses 4 and 5.

The conveyer is being hung up by reactive mass 2 through a resilient suspension 6 to the hard base.

On the scheme (drawing 2.2), besides springs 3 underneath, there are symmetrically mounted springs above; the suspension is analogical to 2.1. Such scheme provides matching the center-of-mass with the center of a hard spring system; identical hardness above and underneath can provide more even field of vibrations through the transporting length.

On the scheme (drawing 2.3) resilient elements 3 placed from two sides of conveyer and fastened to swaying masses 1 and 2 through hard brackets 4 and 5. The pendant of conveyer in the neutral areas of resilient elements can be executed in two variants: through elastic pendants 6, or supporting type – through resilient supports 7.

On the scheme (drawing 2.4) the resilient systems are assembled in the form of squares, where flat springs 3 fastened to the masses 1 and 2 through hard brackets 4 and 5. The ends of springs are united between itself. Due to it in the areas of connections of springs of oscillation is near to the zero, and they serve to support conveyer through resilient elements 7, or pendants through resilient elements 6.



Drawing 2. Constructive scheme of resilient systems in tubular conveyers

On the scheme (drawing 2.5) spring 3 fastened by central bulges to swaying masses, and pendants of conveyer – in the areas of springs fastening.

To provide even transportation on the length of conveyer, an amount of springs pair units, or packages of springs, must be binate *n= 4,6,8.*

The calculation of the resilient systems, as known, is calculation on inflexibility and durability [2, 3].

General inflexibility of the resilient system on the schemes (drawing 2.1÷2.5) is equal:

$$C = 4\pi^2 \cdot \nu_0^2 \cdot M_{\text{con}}, \tag{4}$$

Where v_0 is an own frequency of vibrations of the system which differs from forced v and set the parameter of z.

$$z = \frac{\nu}{\gamma_0} = 0.92 \div 0{,}95$$

$M_{con} = \dfrac{m_1 \cdot m_2}{m_1 + m_2}$ – consolidated mass of the system, kg;

For flat springs, fastened on ends (drawings 2.1÷2.4), inflexibility of one spring is equal:

$$C_1 = \frac{12 \cdot \mathrm{E} \cdot I}{l^3}, \tag{5}$$

where E is the module of resiliency of material of springs. $I = \dfrac{a^3 \cdot в}{12}$ is a moment of inertia the cut of spring, where a is a thickness of spring, m; $в$ – is a width of spring, m; l – is working (unjammed) length of spring 3, m.

According to (4) and (5), taking into account the amount of springs of conveyer (i), the thickness of spring is determined according to (6).

$$a = l\sqrt[3]{\frac{4 \cdot \pi^2 \cdot \nu_o^2 \cdot M_{con}}{E \cdot в \cdot i \cdot k_j}} \tag{6}$$

In (6) we introduced the coefficient of jamming k_j, which shows a relation between actual and ideally hard jamming (experiments show that k_j is within the limits of $k_j = 0{,}7 \div 1{,}0$).

Tension which arises in springs [3]

$$\sigma = \frac{3 \cdot E \cdot a \cdot y}{l^2} \le [\sigma_{-1}], \tag{7}$$

where $y = (A_1 + A_2)$ – is bending of spring, m; A_1 i A_2 accordingly amplitudes of swaying the masses of m_1 i m_2; $[\sigma_{-1}]$ – is maximum possible sign alternating tension, which arises in materials of springs, Pa.

From (7) to provide durability of springs, length of spring must be not less than $l_{\min}$ (8).

$$l_{min} = \sqrt{\frac{3 \cdot E \cdot a \cdot y}{[\sigma_{-1}]}} \tag{8}$$

Putting (8) in (6) we will get the relation to determine thickness of spring (9).

$$a = \frac{y}{[\sigma_{-1}]} \cdot \sqrt[3]{\frac{432 \cdot E \cdot \pi^4 \cdot \nu_o^4 \cdot M_{con}^2}{в^2 \cdot i^2 \cdot k_j^2}}. \tag{9}$$

Let us enter in (9) some correlations, namely:

1. Parameter of tuning $z = \dfrac{\nu}{\nu_0}$

2. $\dfrac{m_1}{m_2} = \lambda$ – correlation of masses.

3. Amplitudes of vibrations back proportional to the masses $\dfrac{A_1}{A_2} = \dfrac{m_2}{m_1}$, from where $A_2 = A_1 \cdot \lambda$

4. Consolidated mass $M_{con} = \dfrac{m_1 \cdot m_2}{m_1 + m_2} = \dfrac{m_1}{1 + \lambda}$.

5. Correlation of width $в$ and thickness of spring $\dfrac{в}{a} = \gamma$, from where $в = a \cdot \gamma$.

Taking into account the entered correlations we will get dependence

$$a = \sqrt[5]{\frac{432 \cdot E \cdot \pi^4 \cdot \nu^4 \cdot m_1^2 \cdot A_1^3 (1+\lambda)}{[\sigma_{-1}]^3 \cdot z^4 \cdot \gamma^2 \cdot i^2 \cdot k_j^2}} \qquad (10)$$

Putting in (10) dependence (8) and making transformation, we will get a formula to determine minimum length of spring (11) which provides the condition of durability

$$l_{\min} = 5,0 \cdot \sqrt[10]{\frac{(1+\lambda)^{6} \cdot E^6 \cdot A_1^8 \cdot \nu^4 \cdot m_1^2}{[\sigma_{-1}]^8 \cdot z^4 \cdot \gamma^2 \cdot i^2 \cdot k_j^2}} \qquad (11)$$

For the most widespread flat steel springs for which E = 2,1. 10^{11} Pa (10) and (11) we will get the following final values:

$$a = 1,55 \cdot 10^3 \sqrt[5]{\frac{(1+\lambda) \cdot m_1^2 \cdot A_1^3 \cdot \nu^4}{[\sigma_{-1}]^3 \cdot z^4 \cdot \gamma^2 \cdot i^2 \cdot k_j^2}} \qquad (12)$$

$$l_{min} = 3,1 \cdot 10^7 \sqrt[10]{\frac{(1+\lambda)^6 \cdot A_1^8 \cdot \nu^4 \cdot m_1^2}{[\sigma_{-1}]^8 \cdot z^4 \cdot \gamma^2 \cdot i^2 \cdot k_j^2}} \qquad (13)$$

Taking into account (2) and the angle of vibration β, we will get dependence (14) to determine minimum length of spring from speed of transportation and dependence (15) to determine V_{tr} for different l.

$$l_{min} = 7,13 \cdot 10^6 \cdot \sqrt[10]{\frac{(1+\lambda)^6 \cdot m_1^2 \cdot \upsilon_{tr}^8}{[\sigma_{-1}]^8 \cdot \nu^4 \cdot z^4 \cdot \gamma^2 \cdot i^2 \cdot k_j^2 \cdot k_{sp}^8 \cdot \cos^8\beta}} \qquad (14)$$

$$V_{\text{tr}} = 2,714 \cdot 10^{-9} \sqrt[8]{\frac{l^{10} \cdot [\sigma_{-1}]^8 \cdot \nu^4 \cdot z^4 \cdot \gamma^2 \cdot i^2 \cdot k_j^2 \cdot k_{sp}^8 \cdot cos^8\beta}{(1+\lambda)^6 \cdot m_1^2}} \qquad (15)$$

Let us introduce *k*:

$$K = \frac{[\sigma_{-1}]^8 \cdot z^4 \cdot \gamma^2 \cdot k_j^2}{(1+\lambda)^6} \qquad (16)$$

Then (15) will transform into (17).

$$V_{\text{tr}} = 2,714 \cdot 10^{-9} \sqrt[8]{K \cdot \frac{l^{10} \cdot \nu^4 \cdot i^2 \cdot k_{sp}^8 \cdot \cos^8\beta}{m_1^2}} \qquad (17)$$

Analysis [1, 2, 4] shows that the most optimum frequencies of conveyers' vibrations with an electromagnetic occasion is within range 12,5÷25 Hz; angles of vibrations $\beta = 15^\circ \div 45^\circ$.

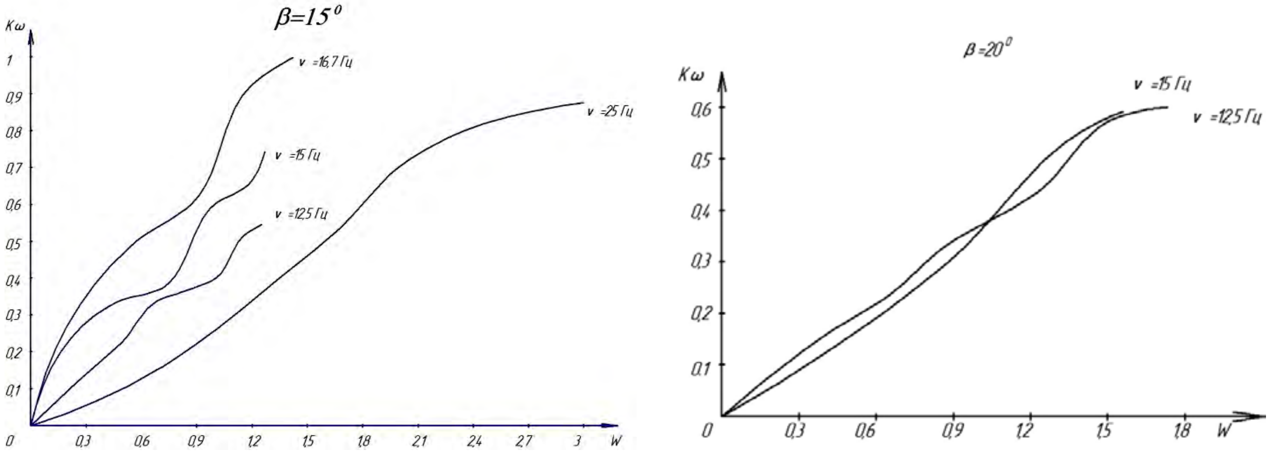
Mass of m_1 depends on length of transporting pipe of conveyer, thickness of pipe's wall material. Important is a condition [3], according to which an own frequency of vibrations of pipe must be not less than three times more from working frequency of conveyer. The amount of springs are defined structurally, optimum numbers are i = 4,6,8,12,16, and their amount depends on the features of construction.

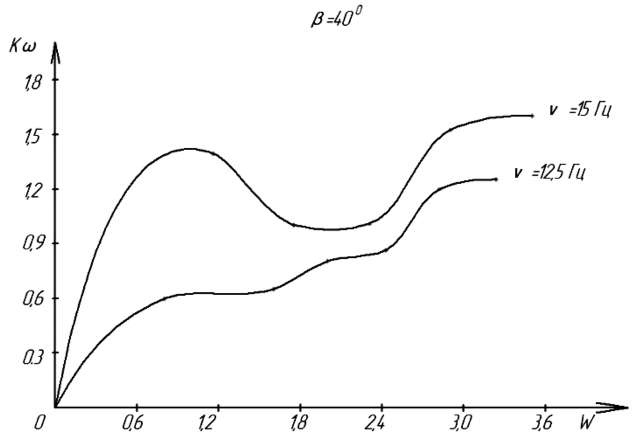
For the brand of a spring steel 60C2, which is recommended [1, 2, 4] as the best $[\sigma_{-1}] = 3 \cdot 10^8$ Pa.

Experimental researches give optimum settings z_{op}= 0,935.

Correlations of the masses chosen $\lambda = \frac{m_1}{m_2} = \frac{1}{3} \div 1$, k_j = 0,75÷0,85.

Average value k_{sp} is taken on the basis of experimental researches [1, 2, 4], which are executed in transportation of free-flowing and lump products, it is discovered that coefficient of speed k_{sp}, which for artificial products makes k_{sp}=0,3÷0,5, for transportation of free-flowing and lump products differs significantly. Its value at high amplitudes of vibrations and angles of vibration β is approaching to 1,0 and is even some bigger than one [4]. Therefore to interpret this coefficient, as correlation of actual speed of transporting to theoretical, it is illogical for transportation of free-flowing and lump products.





Drawing 3. Graphics of dependences of coefficient of speed from the parameter of overload at the different angles of vibration and enforcing frequencies in sand transportation

This parameter can be interpreted, as a coefficient of transmission of vibrations, or as a coefficient of volume speed. It reaches values considerably higher than known coefficient, due to the high parameters of overload (w=10÷15) and due to volume transportation of free-flowing and lump products.

On the drawing 3 there are some dependences of k_{sp} from w for the list of angles of vibrations β and enforcing frequencies v in sand transportation.

Conclusions. 1. Received dependences (12), (13) enable to calculate independently thickness and minimum lengths of springs at the stage of designing, having basic parameters (mass m_1, amplitude, or speed of transportation) only, easily combining other parameters.

2. Dependences (15) or (17) allow to calculate maximum speeds of transportation, which will be provided by a conveyer, at the stage of sketch designing, taking into account the angles of vibration β and volume coefficients of speed from experimental graphs (Drawing 3).

3. Dependences (14), (15) (17) allow to get graphic dependences, using them at the stage of designing, or to make calculations which will speed up analysis of possibilities of the created tubular vibroconveyers constructions.

4. Offered variety of structural modifications of conveyers and formulas of calculation give developers more materials at the stage of designing and calculation.

5. Most of the shown materials were used in the process of wok on creation several decades of models of vibroconveyers of different sizes with lengths of transporting elements of l=0,15÷3,5 m and with the diameters of transporting pipes of d=20÷150 mm.

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