UKRAINIAN JOURNAL OF MECHANICAL ENGINEERING AND MATERIALS SCIENCE
Vol. 2, No. 1, 2016

Vladyslav Shenbor[1], Petro Koruniak[2], Vitaliy Korendiy[1], Volodymyr Brusentsov[1], Marta Brusentsova[3]
[1] Lviv Polytechnic National University, Lviv, Ukraine
[2] Lviv National Agrarian University, Dubliany, Ukraine
[3] Lviv State College of Food and Processing Industry, Lviv, Ukraine

ANALYSIS AND IMPROVEMENT OF TWO-MASS VIBRATING TUBULAR CONVEYERS WITH TWO-CYCLE ELECTROMAGNETIC DRIVE

Received: July 16, 2016 / Revised: August 20, 2016 / Accepted: August 24, 2016

Abstract. The analysis of structure diagrams of the two-mass tubular conveyer with two-cycle electromagnetic drive is carried out in the article. Two methods of designing of the elastic system are considered; calculation formulas for elastic system computation are deduced according to different requirements for the operating parameters of conveyers; the comparative analysis of two types of elastic systems is conducted. The recommendations for designing the lengthy tubular vibrating conveyers are presented. According to the considered structural diagrams of the elastic system the models of the lengthy vibrating conveyers with the transporting distances $l = 1,5...4\ m$ are developed. On the basis of these models the lengthy vibrotransporting systems with the transporting distance $l = 5...30\ m$ were developed, manufactured and applied in industry. The considered systems ensures the transportation of loose materials (porcelain composition, sand etc.) with maximal speeds up to 800 *mm/sec* and productivities (for tubes with internal diameter $D = 100\ mm$) up to 30 tons per hour for. It is reasonable to use both considered schemes of elastic systems for development of vibrating tubular conveyers with electromagnetic drive. The conveyers may be designed in two modifications: overhead and supporting. The obtained dependencies allow optimization of the structure and considering the important technical parameters of conveyers (maximal permissible oscillations amplitude, springs stresses, productivity etc.) in the design phase.

Introduction

Vibrating tubular conveyers with electromagnetic drive may be effectively used in various branches of industry due to their specific features and advantages. The main advantages of these machines are structure simplicity, total absence of the parts which may rub and quickly work out of true, small wearing of the internal surface of the transporting element, simple regulation of the vibrotransportation modes, possibility of conveying of various types of loads (piece, lumpy, loose, granular including powder-like, gaseous etc.), simplicity of setting-up and repairing, possibility of combining conveying and technological operations (i.e. drying and cooling), simplicity of feeding and unloading with the possibility of dosing in wide range of diameters of transporting element. The important advantage of these conveyers is small power consumption due to specific setting-up of the devices with the aim to achieve near-resonance operation modes.

Problem statement

Despite of a number of advantages tubular conveyers are rather rarely used in manufacturing industry nowadays. First of all, this situation may be explained by the fact that they are not commercially manufactured. Single models designed and manufactured by different companies for their own needs are little known. That is why the development of this type of conveyers is being limited. The important problem consists in development of new and improvement of known model of tubular conveyers with the aim to increase their efficiency, to simplify the methods of calculation, designing and setting-up.

Analysis of modern information sources on the subject of the article

The information about vibrating tubular conveyers with electromagnetic drive is not very widespread. Literature sources and publications in special journals are scattered over several decades and do not allow the possibility to obtain the sufficient representation about this type of conveying equipment. The most accessible sources are [3–5; 7; 8; 10] which present the number of structures of vibrating tubular conveyers with electromagnetic drive.

The analysis and research of these sources show that a great deal of investigations and developments dedicated to tubular conveyers were conducted in the scientific and research laboratory NDL-40 (Department of mechanics and automation engineering of Lviv Polytechnic National University). The developments and investigations carried out in the laboratory NDL-40 for many years gave the considerable results in the field of vibrotechnics, which were realized in a number of effective models and designs including the models of vibrating tubular conveyers and transporting systems formed of separate modules. In recent years, these developments have undergone some modernization and the methods of their designing and calculation have been improved [4; 12].

Statement of purpose of research

The development of new machines for solving some technological problems usually starts with choosing the structure diagram of the machine. If there is a necessity to develop the machine which should perform tasks that were solved by earlier developed machines it is necessary to use the accumulated experience. If there is a necessity to improve technical parameters (increase productivity, reduce power or materials consumption) the analysis of analogues becomes less expedient. In that case the analysis of the known structure diagrams is carried out and on the basis of practical experience the machines structures are being improved or sometimes fundamentally changed.

The purpose of this paper consists in analysis of the two-mass tubular conveyer with two-cycle electromagnetic drive of “tube in tube” type and in investigation of conveyers with various combinations of elastic systems.

Main material presentation

The most optimal structures of vibrating conveyers with electromagnetic drive are the structures based on two-mass oscillatory schemes [8]. These schemes allow creating almost totally symmetrical structures with superposition of centers of oscillating masses. This factor makes supplementary parasitic oscillations the whole structure or relative oscillations of both oscillating masses impossible. Thus the uniformity of transportation along the transporting element increases and the efficiency and productivity of vibrotransportation also raise.

The structure shown in Fig. 1 may be considered as near-optimal one from the point of view of the requirements to forming of two-mass mechanical oscillating systems. This model was developed and applied in several industrial enterprises for conveying loose, lumpy and piece products. The conveyer is designed according to the two-mass oscillating scheme with lengthy working transporting element 1 coated with polyurethane or rubber inside. The reactive (nontransporting) mass consists of two tubes 2 to face flanges of which two circular electromagnets 3 are fastened. Two parts of the reactive mass are attached by pins and ring 7 and form in such a way the rigid structure. The working and reactive masses are fastened by four flat elastic elements 8 the ends of which are attached to the brackets of the working and reactive masses. The elastic elements are placed in the bottom part of the conveyer at the inclination angle with the vertical plane (vibration angle) β. The brackets of the working mass are fastened to the tube with a help of terminal clamps and the brackets of reactive mass are welded to two reactive tubes. Between the electromagnets of the reactive mass the double-sided circular armature is placed and rigidly fastened to the transporting tube. Two rings 5, which are attached by the terminal clamps to the tube 1, and four brackets 4 with elastic (rubber) buffers limit the vibrations amplitude if it is necessary and do not allow the transition of the oscillating system to the vibroimpact mode by stabilization of the operation on high amplitudes.

The conveyer operates as follows. If the alternating current is applied according to the two-cycle feed circuit the inclined harmonic oscillations with opposite phases of the working and reactive masses excite on the circular electromagnets 3 at the angle of vibration β, which is defined by the inclination angle of the attachment planes of the springs brackets. Due to this structure the parts (products) are being transported along the internal surface of the tube 1 (Fig. 1, from the right to the left) [8].

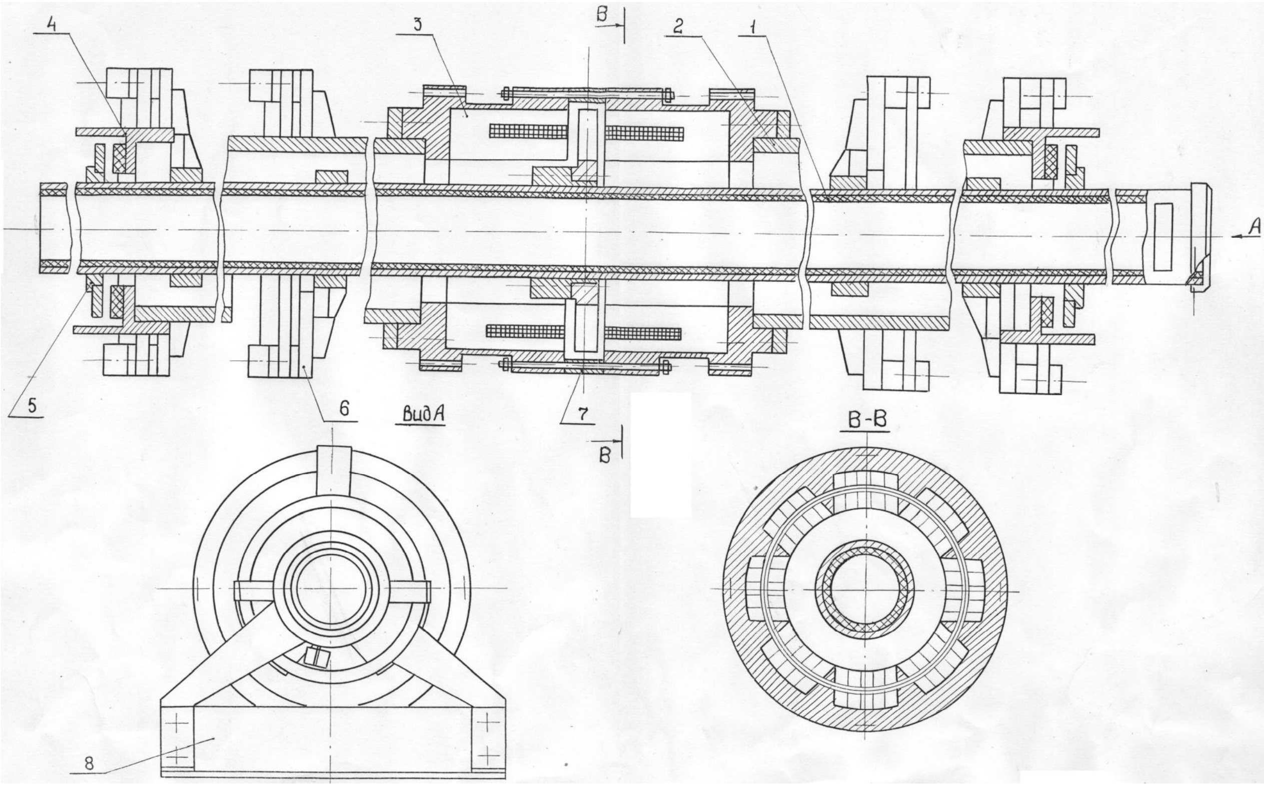

Fig. 1. *Structural diagram of vibrating tubular conveyer*

The efficiency and productivity of conveying depends on the following factors:
– the structure symmetry along three coordinates;
– the matching of the centers of oscillating masses;
– the natural oscillation frequency of the transporting tube;
– the rigidity of the fastening brackets;
– the placement of the elastic elements;
– the rigidity of all fastening connections;
– the method of the working element loading with products.

Based on the above material the multifunctional problem raises before the developer during the machines designing. Taking into account the experience of manufacturing, setting-up and applying of analogous vibrating equipment, this problem may be finally solved by means of setting-up of such machines.

Let us analyze some requirements to the development of conveyers of such type and some methods of their assurance.

1. The model of the conveyer (Fig. 1) is almost symmetrical due to using the scheme of “tube-in-tube” type. The masses of brackets for springs fastening are very small in comparison with the total value

of oscillating masses of the conveyer. That is why the brackets do not influence the symmetry of the structure and the matching (nonmatching) of the mass centers.

2. The choosing of parameters of the transporting tube (length, diameter, material) depends on the natural frequency ω_0 of oscillations of the tube considered as the beam without any supports, which should be 3–4 times larger than the frequency of forced oscillations of the conveyer. This fact ensures the oscillations of the tube as one rigid body when every point of the transporting element oscillates with the same frequency and along the same trajectories. By ensuring the uniform field of oscillations of the transporting element (Fig. 2) we obtain the uniformity of the longitudinal transporting and the necessary productivity. For the beam (tube) without supports the natural frequency of oscillations may be calculated as [8]:

$$\omega_0 = 22,4 \cdot \sqrt{\frac{E \cdot I}{m \cdot l^3}}, \tag{1}$$

where E is the modulus of elasticity of the tube material, *Pa*; I is the moment of inertia of the tube cross-section, m^4; m is the mass of the tube, *kg*; l is the length of the tube, *m*.

The dependence (1) is simple for calculating and ensures the absence of additional transversal natural oscillations of the working conveyer. However it is necessary to mention that for every specific structure the natural frequency of the transporting element is a little different because of the different attachments of elastic elements and different placement of these attachments. The determination of this frequency is a very complicated task and the solving of this problem allows optimization of the structure and decreasing of its materials consumption. The problem of choosing the parameters of the working tubular element is also very important task relating to the influence of the loading on the stability of oscillations parameters of the transporting element. With a help of special sensors places along the tube it is possible to define the components of oscillations and introduce corrections into the conveyer structure.

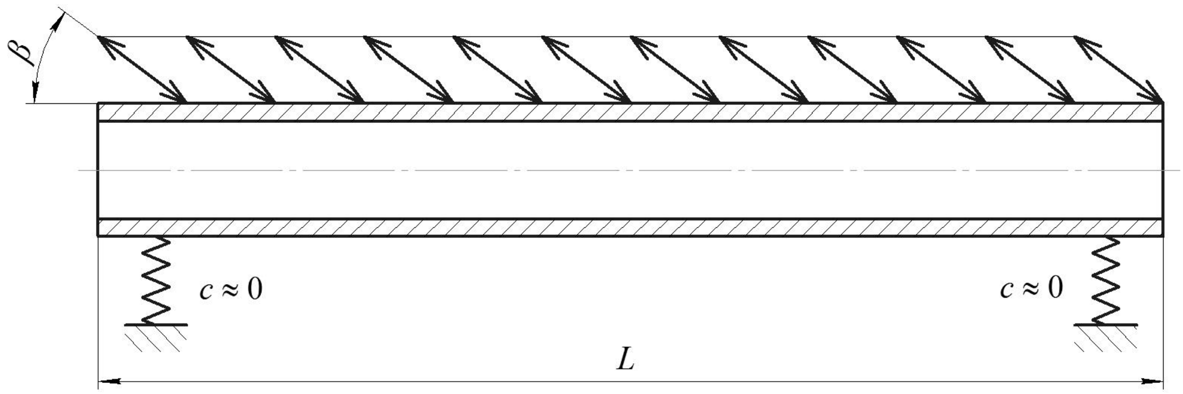

Fig. 2. *The scheme of the beam and the distribution of oscillations*

3. The rigidities of the fastening elements (brackets, terminal clamps etc.) should ensure the parameters of oscillations of the working and reactive masses as single rigid bodies. The methods of fastening of elastic elements essentially influence the coefficient of restraint k_3, which lies in the range of $k_3 = 0,7 - 1,0$ for typical conveyers and $k_3 = 0,55 - 0,6$ for large-sized lengthy conveyers with the lengths of transporting elements 3–5 *m* and with diameters of transporting tubes 100–200 *mm*.

4. The loading of the working transporting tube from the only hopper essentially influences the parameters of oscillations in the loading zone. That is why it is reasonable to design several loading zones especially for 3–5 m long transporting tubes. As the experimental investigations showed, in the lines of tubular conveyers, where the working transporting tubes are fastened with a help of special couplings and intermediate tubular inserts, the loading through the inserts cause very small influence on the parameters of oscillations of the transporting elements.

5. The placement of elastic elements for models, which are analogous to the presented model (Fig. 1), should solve two problems (requirements):

– do not allow the appearing of angular oscillation about the axis $Z-Z$ (Fig. 3);

– to create some zones on the elastic elements for fastening the inserts with dampers when the supporting method of conveyer installation is used.

The first problem is solved due to the fastening of flat springs 1 to oscillating masses m_1 and m_2 in staggered order (in the Fig. 3 the elastic system and its fastenings are presented on the upper view of the conveyer). Experimental investigations on several models of conveyers substantiated the expediency of such design (structure).

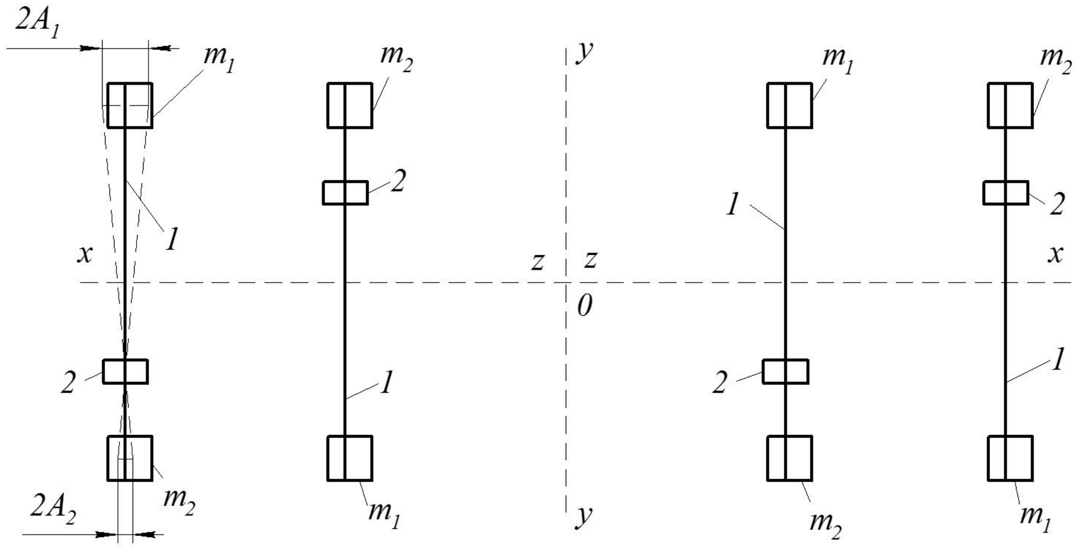

Fig. 3. *Simplified scheme of the conveyer elastic system: m_1 is transporting (active) mass; m_2 is reactive mass; 1 – flat elastic(spring) elements; 2 – supporting elements*

The second requirement taking into account the fact that the oscillations of both masses are with opposite phases and are carried out with amplitudes, which vary in inverse proportion to masses

$$\frac{A_1}{A_2}=\frac{m_2}{m_1} \tag{2}$$

allows the use of so-called “zones of neutral cross-section” which are defined by the lengths of springs and in which there are no oscillation (or they are close to zero). This phenomenon is shown on the left end support 2 (Fig. 3).

The most important calculations, which are carried. out during the designing of conveyers, are the computations of rigidity and strength of elastic elements. This process ensures the setting-up of the device according to the stationary close to resonance operation mode and does not allow the failures of the spring during their exploitation with ensuring the specified productivity Q.

In this paper we consider two methods of development (designing) of elastic system for lengthy vibrating two-mass tubular conveyers:

I. The springs are fastened on the ends to oscillating masses m_1 and m_2 (Fig. 4);

II. The springs form blocks which consist of two springs. The centers of the springs are fastened to the oscillating masses m_1 and m_2 and the ends are rigidly attached together (Fig. 5).

Let us carry out the analysis of methods of designing of elastic systems. According to the method I the calculated thickness of the spring which ensures the necessary resonance setting-up may be defined as [3; 8]:

$$a_I = l\cdot\sqrt[3]{\frac{4\cdot\pi^2\cdot\nu_0^2\cdot\mathrm{M}_{np}}{E\cdot b\cdot i\cdot k_з}}\,, \tag{3}$$

where l is the working (not restrained) length of the flat elastic element, *m*; ν_0 is natural frequency of conveyer vibrations, *Hz*. This frequency is defined according to the recommendations of close to resonance setting-up of the conveyer [8]:

$$\nu_0 = \frac{\nu}{Z}, \tag{4}$$

where ν is the frequency of forced oscillations, *Hz*; $Z = 0{,}92 - 0{,}94$ is the correction coefficient; $M_{np} = \frac{m_1 \cdot m_2}{m_1 + m_2}$ is the reduced mass of the oscillating system, *kg*. In [8] it is recommended to take the ratio $m_1/m_2 = 2 - 4$. Taking into account modern investigations this ratio cannot be absolutely unchangeable because of the fact that it is not reasonable to increase the reactive mass m_2 and the ratio of $m_1/m_2 = 1 - 2$ also may be adopted; E is the modulus of elasticity of the springs material, *Pa*. For steel spring $E = 2{,}1 \cdot 10^{11}$ Pa. Nowadays the springs made of glass-fabric-base laminate are also widely used but their modulus of elasticity varies in wide range and this material is not investigated when the oscillations amplitudes are very high. Also it is not recommended to use this material in hot and other operation conditions; b is the working width of the spring, *m*; i is the number of springs; $k_з$ is the coefficient of spring restraint.

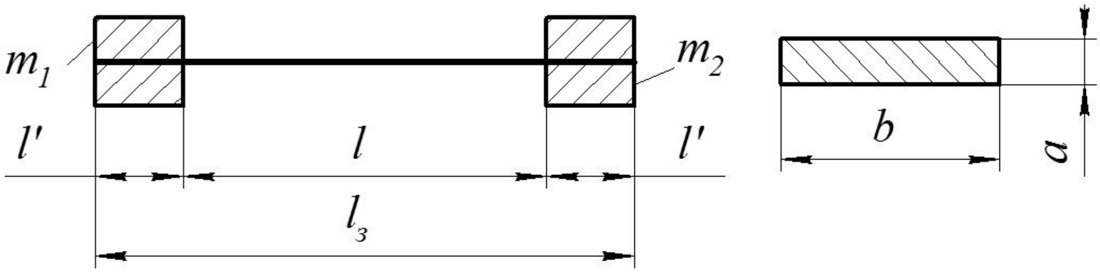

Fig. 4. *The scheme of the spring fastening (method I) and the cross-section of the spring: l is working (not restrained) spring length; l' is the width of the fastening; $l_з$ is total (overall) length of the spring; a is working spring thickness; b is the spring width*

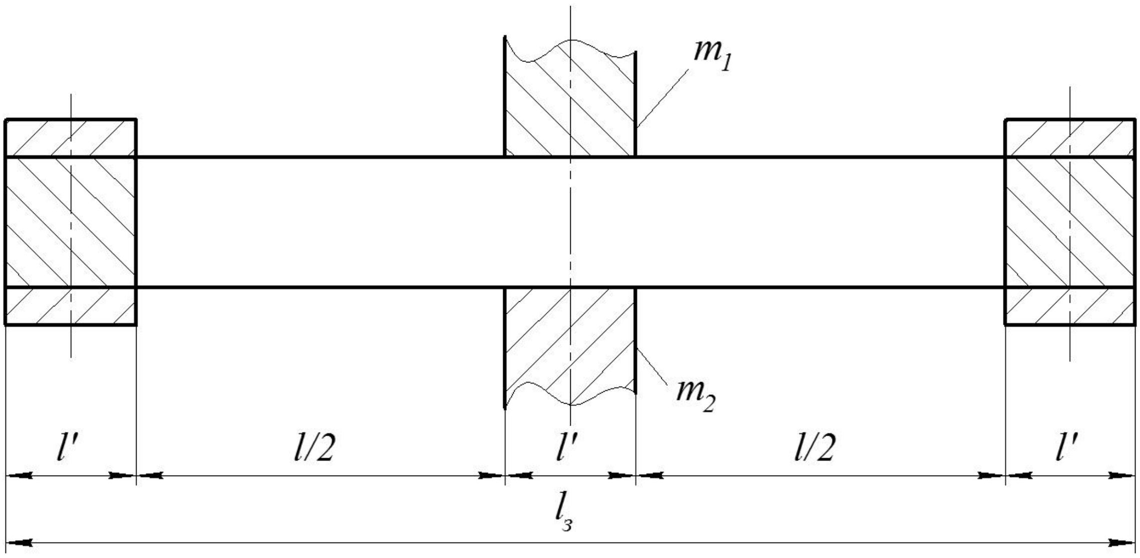

Fig. 5. *The scheme of fastening of the spring of block type (method II)*

In the dependence (3) all the parameters except l may be taken structurally or on the basis of logical consideration. The working length l and the total length of the spring $l_з$ may be defined according to one of the following requirements:

– do not allow the failures of the spring;

– to ensure the specified (limited) overall sizes of equipment which depend on manufacture conditions.

It is known that the spring stress, which is generated during the machine operation, may be defined as [8]:

$$\sigma = \frac{3 \cdot E \cdot a \cdot y}{l^2}, \tag{5}$$

where $y = A_1 + A_2$ is the spring deflection, *m*.

The stress should not exceed the permissible bending stress $[\sigma_{-1}]$:

$$\sigma \le [\sigma_{-1}]. \tag{6}$$

For quenched grinded spring made of high-quality spring steel of the 60C2 grade $[\sigma_{-1}] = 3 \cdot 10^8\ Pa$. The permissible stress may be increased on 15–20 % by using vibrational processing technologies: vibrogrinding, vibropolishing and vibrohardening.

Let us substitute the parameter a from the formula (3) into the formula (5) for calculation of the spring stress:

$$\sigma = \frac{3 \cdot E \cdot y}{l} \cdot \sqrt[3]{\frac{4 \cdot \pi^2 \cdot \nu_0^2 \cdot E^2 \cdot M_{np}}{b \cdot i \cdot k_з}}. \tag{7}$$

By calculation of the stress with a help of the formula (7) we may predict the durability of the conveyer elastic system operation.

So far as according to the formula (2) the deflection $y = A_1 \cdot \frac{m_1 + m_2}{m_2}$ and considering that $\lambda = \frac{m_1}{m_2}$ and $\nu_0 = \frac{\nu}{Z}$ we obtain the following formula:

$$\sigma = \frac{3 \cdot A_1}{l} \cdot \sqrt[3]{\frac{4 \cdot \pi^2 \cdot \nu^2 \cdot m_1 \cdot (1+\lambda)^2}{b \cdot i \cdot Z \cdot k_з}}. \tag{8}$$

In that case according to the formula (8) we may obtain the plot of the dependence $\sigma = \sigma(A_1)$ and during the designing we may protect the structure from the failures of elastic elements by combining the parameters of the specific model.

The amplitude of oscillations of the transporting mass of the conveyer is related with the transporting speed V (*m/s*) by the following dependence:

$$A = \frac{V}{2 \cdot \pi \cdot \nu \cdot \cos\beta \cdot k_ш}, \tag{9}$$

where β is the angle of vibration; $k_ш$ is the coefficient of transporting speed (for the conveyer which are being investigated when the amplitude is $A_1 = 4 - 5\ mm$ during the transportation of loose and lumpy products of various types $k_ш = 0{,}6 - 0{,}8$).

The productivity Q and speed of transportation are related by the dependence:

$$Q = V \cdot S \cdot \gamma, \tag{10}$$

where S is the area of the cross-section of the tube along which the products are being transported, m^2. For loose and lumpy products $S = 0{,}5 - 0{,}6$ from the total area of the transporting tube cross-section; γ is the bulk density of the material being transported, kg/m^3.

Let us simplify the dependence (8) by substitution of the known parameters E and π and by adopting for lengthy conveyers with electromagnetic drive the most optimal for electromagnetic vibration exciter oscillations frequency $\nu = 25\ Hz$, the correction coefficient $Z = 0,93$, the coefficient of restraint $k_з = 0,7$. In that case if the number of springs $i = 4$ for the model (Fig. 1) the following stresses arise:

$$\sigma = 2,3 \cdot 10^9 \cdot \frac{A_1}{l} \cdot \sqrt[3]{\frac{m_1 \cdot (1+\lambda)^2}{b}}. \tag{11}$$

By equalizing $\sigma = [\sigma_{-1}] = 3 \cdot 10^8\ Pa$ we obtain the dependence which allows the calculation of the length of the spring taking into account the strength conditions:

$$l = 7,7 \cdot A_1 \cdot \sqrt[3]{\frac{m_1 \cdot (1+\lambda)^2}{b}}. \tag{12}$$

From the equation (12) we may deduce the dependence for calculation of the maximal permissible oscillations amplitude of the transporting mass:

$$A_1 = 0,13 \cdot l \cdot \sqrt[3]{\frac{b}{m_1 \cdot (1+\lambda)^2}}. \tag{13}$$

By using the formulas (9) and (10) and by adopting the coefficient $k_ш = 0,7$ we obtain the approximate dependences for speed and productivity calculation:

$$V = 14 \cdot l \cdot \cos\beta \cdot \sqrt[3]{\frac{b}{m_1 \cdot (1+\lambda)^2}}; \tag{14}$$

$$Q = 14 \cdot l \cdot S \cdot \gamma \cdot \cos\beta \cdot \sqrt[3]{\frac{b}{m_1 \cdot (1+\lambda)^2}}. \tag{15}$$

When the number of springs rises from $i = 4$ to $i =$ 6, 8, 10, 12... the numerical coefficient 14 in the formulas (14) and (15) increases by 1,14; 1,26; 1,36; 1,44… times. For example, if $i = 8$ (paired springs) the numerical coefficient equals 17,6.

For designing the elastic system (four blocks with two springs in each block) (Fig. 5) the thickness of the spring, when the oscillating masses are equal ($m_1 = m_2$), may be defined as [13]:

$$a_{II} = a_1 = a_2 = l \cdot \sqrt[3]{\frac{\pi^2 \cdot \nu_0^2 \cdot m_1}{16 \cdot E \cdot b \cdot k_з}}. \tag{16}$$

If we put $m_1 = m_2$ in the formula (3) we obtain $M_{np} = \frac{m_1}{2}$ and the dependence for the method I looks like:

$$a_1 = l \cdot \sqrt[3]{\frac{2 \cdot \pi^2 \cdot \nu_0^2 \cdot m_1}{E \cdot b \cdot i \cdot k_з}}. \tag{17}$$

For exact comparison of the schemes I and II it is necessary to adopt $i = 8$ in the formula (17):

$$a_1 = l \cdot \sqrt[3]{\frac{\pi^2 \cdot \nu_0^2 \cdot m_1}{4 \cdot E \cdot b \cdot k_з}}. \tag{18}$$

The stress, which is generated in the spring when using the scheme I, equals:

$$\sigma_I = \frac{3 \cdot E \cdot y}{l} \cdot \sqrt[3]{\frac{\pi^2 \cdot \nu_0^2 \cdot m_1}{4 \cdot E \cdot b \cdot k_з}} = \frac{6 \cdot A_1}{l} \cdot \sqrt[3]{\frac{\pi^2 \cdot \nu_0^2 \cdot m_1 \cdot E^2}{4 \cdot b \cdot k_з}}. \tag{19}$$

If the elastic system is composed according to the scheme II the stress, which is generated in the spring, equals:

$$\sigma_{II} = \frac{12 \cdot E \cdot A_1 \cdot a_{II}}{l^2} = \frac{12 \cdot A_1}{l} \cdot \sqrt[3]{\frac{\pi^2 \cdot \nu_0^2 \cdot m_1 \cdot E^2}{16 \cdot b \cdot k_3}}. \tag{20}$$

In that case, the stress ratio of the two schemes equals:

$$\frac{\sigma_I}{\sigma_{II}} = \frac{6}{12} \cdot \sqrt[3]{\frac{16}{4}} = 0,8. \tag{21}$$

The calculations, made according the formula (21), show that springs of the scheme I are less loaded than springs of the scheme II.

Conclusions

1. According to the considered structural diagrams of the elastic system the models of the lengthy vibrating conveyers with the transporting distances $l = 1,5 - 4\ m$ are developed. On the basis of these models the lengthy vibrotransporting systems with the transporting distance $l = 5 - 30\ m$ were developed, manufactured and applied in industry.

2. The considered systems ensures the transportation of loose materials (porcelain composition, sand etc.) with maximal speeds up to 800 *mm/sec* and productivities (for tubes with internal diameter $D = 100\ mm$) up to 30 tons per hour.

3. It is reasonable to use both considered schemes of elastic systems for development of vibrating tubular conveyers with electromagnetic drive. The conveyers may be designed in two modifications: overhead and supporting.

4. The scheme II is simpler one considering structure design. The brackets for springs fastening are more rigid due to smaller lengths of structure elements.

5. The obtained dependencies allow optimization of the structure and considering the important technical parameters of conveyers (maximal permissible oscillations amplitude, springs stresses, productivity etc.) in the design phase.

6. For the scheme II it is possible to obtain the whole range of parameters by using the similar calculations as for the scheme I.

References

[1] Блехман И. И. Исследования вынужденных колебаний некоторых вибрационных машин со многими вибраторами / И. И. Блехман, Г. Ю. Джанелидзе // Известия АН СССР. Механика и машиностроение. – 1988. – № 3. – С. 51–64.

[2] Беспалов А. Л. Деякі результати створення і дослідження вібраційних конвеєрів-сепараторів / А. Л. Беспалов, В. С. Шенбор // Автоматизація виробничих процесів у машинобудуванні та приладобудуванні : Український міжвідомчий науково-технічний збірник. – 2005. – № 39. – С. 39–44.

[3] Боровець В. М. Аналіз і дослідження структурних і конструктивних схем вібраційних трубчастих конвеєрів з електромагнітним приводом / В. М. Боровець, О. Р. Серкіз, В. С. Шенбор // Вібрації в техніці та технологіях : Всеукр. наук.-техн. журн. – 2009. – № 2 (54). – С. 9–14.

[4] Боровець В. М. Прикладні проблеми розробки двомасних вібраційних трубчастих конвеєрів / В. М. Боровець, В. Г. Брусенцов, О. Р. Серкіз, В. С. Шенбор // Автоматизація виробничих процесів у машинобудуванні та приладобудуванні : Український міжвідом. наук.-техн. зб. – 2011. – № 45. – С. 168–174.

[5] Вибрации в технике : справочник : в 6 т. / под ред. Э. Э. Лавендела. Т. 4: Вибрационные процессы и машины. – М. : Машиностроение, 1981. – 509 с.

[6] Зенков Р. Л. Машины непрерывного транспорта / Р. Л. Зенков и др. – М. : Машиностроение, 1987. – 432 с.

[7] Повідайло В. О. Протяжні вібраційні транспортні системи / В. О. Повідайло, В. С. Шенбор // Автоматизація виробничих процесів у машинобудуванні та приладобудуванні : Український міжвідом. наук.-техн. зб. – 1999. – № 34. – С. 23–27.

[8] Повідайло В. О. Вібраційні процеси та обладнання : навч. посіб. / В. О. Повідайло. – Львів : Вид-во Національного університету "Львівська політехніка", 2004. – 248 с.

[9] Спиваковский А. О. Транспортирующие машины / А. О. Спиваковский, В. К. Дьячков. – М. : Машиностроение, 1983. – 487 с.

[10] Тропман А. Г. Вибрационные конвейеры для транспортирования горячих материалов / А. Г. Тропман, Н. М. Бельков, Ю. Н. Макеева. – М. : Машиностроение, 1972. – 120 с.

[11] Шенбор В. С. Розробка структурних схем вібраційних конвеєрів-сепараторів / В. С. Шенбор, А. Л. Беспалов // Автоматизація виробничих процесів у машинобудуванні та приладобудуванні: Український міжвідом. наук.-техн. зб. – 1998. – Вип. 33. – С. 20–23.

[12] Shenbor V. S. Improvement of constructive schemes and calculations of oscillation tudular conveyers / V. S. Shenbor, V. M. Borovets, V. G. Brusentsov. O. R. Serkiz, P. I. Fita / Вісник Національного університету "Львівська політехніка". – 2013. – № 760: Оптимізація виробничих процесів і технічний контроль у машинобудуванні і приладобудуванні. – С. 77–84.

[13] Шоловій Ю. П. Вивчення конструкції, онов розрахунку та дослідження вібраційного трубчастого конвеєра з електромагнітним приводом : метод. вказівки / Ю. П. Шоловій, В. М. Боровець, В. С. Шенбор. – Львів : Вид-во Нац. ун-ту "Львівська політехніка", 2005. – 20 с.