JUSTIFICATION OF RATIONAL PARAMETERS OF A PNEUMOCONVEYOR SCREW FEEDER
/
ОБГРУНТУВАННЯ РАЦІОНАЛЬНИХ ПАРАМЕТРІВ ЖИВИЛЬНИКА ПНЕВМО-ШНЕКОВОГО ТРАНСПОРТЕРА

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ABSTRACT
The article presents the justification of rational parameters of the operating parts of a pneumoconveyor screw. The dependences of pressure screw efficiency on the change in design and kinematic parameters and process variables of a screw and on loose material characteristics have been considered. In addition, general analysis of the choice substantiation of a feeder pressure screw diameter and the cross-sectional area of a hopper loading opening, which provides the required efficiency of a pneumoconveyor screw, has been conducted.

РЕЗЮМЕ
В статті наведено обґрунтування раціональних параметрів робочих органів пнемо-шнекового транспортера. Представлено залежності продуктивності напірного шнека від зміни конструктивно-кінематичних і технологічних параметрів шнека та властивостей сипкого матеріалу. Також проведено обґрунтування вибору діаметра напірного шнека живильника та площі поперечного перерізу завантажувального отвору бункера, при якому буде забезпечуватися необхідна продуктивність пнемо-шнекового транспортера.

INTRODUCTION
The conducted analysis on the state of current technologies and the review of recent scientific and patent literature, which cover the design of machinery and mechanisms for conveying loose materials along curvilinear routes *(Lyashuk O.L., et.al., 2015; Rogatynska O. et.al., 2015; Rohatynskyi R.M. et.al., 2016)* shows that they satisfy most of the requirements to a certain extent, but most of the designed operating parts of conveyers perform not only translational axial movement of material, but they also perform rotary motion, which causes material damage and reduces the efficiency of such mechanisms. This paper is the follow-on study to the work, covered in papers *(Hevko R.B. et.al., 2009; Hevko R.B. et.al., 2014; Hevko R.B. and Klendiy O.M., 2014; Hevko R.B. et.al., 2016)*, and is aimed at improving the efficiency of pneumoconveyor screw in order to provide loose material flow along technological lines of various space configurations applying power-operated material feed by means of a screw feeder and additional pneumatic reinforcement. The developed designs of pneumoconveyor screws have been patented *(Hevko R.B. et.al., 2011; Halka R.I., 2001; Hevko R.B. et.al., 2012)*. When doing theoretical research, we were based on the approaches set out in the work *(Voityuk V.M., 2005)*.

MATERIALS AND METHODS
In order to formalize loose material transportation process and further substantiate the rational parameters of a pneumoconveyor screw operating elements, let us consider its flow-sheet, presented in Fig.1.

The main system points of a pneumoconveyor screw are: a screw feeder *1* and a pneumoconveyor screw *2*. A screw feeder consists of a loading hopper *3*, which has a loading opener *4* in its underflow and a screw *5*, which is made in the form of an axle *6* with spiral flights *7*. A pneumoconveyor consists of a pneumatic nozzle section *8* and a pneumatic pipeline *9*.

The technological process is the following. Loose material from a loading hopper *3* of a screw feeder *1* is emptied on a screw *5* through a loading opening *4*. Screw flights *7* convey loose material to the outlet end of a screw and further to the initial pneumoconveyor *2* coverage. By means of high-pressure air, a

pneumoconveyor transports loose material through a pneumatic nozzle section *8* into a pneumatic pipeline *9*. In order to provide effective transportation of loose materials in a flexible casing (it is not shown in Fig.1), the guides, which are equally-spaced on a circle of a nozzle, are directed along with material movement. Thus, the process of material transportation is provided due to its forced feed by a feeder and further pneumatic pressure.
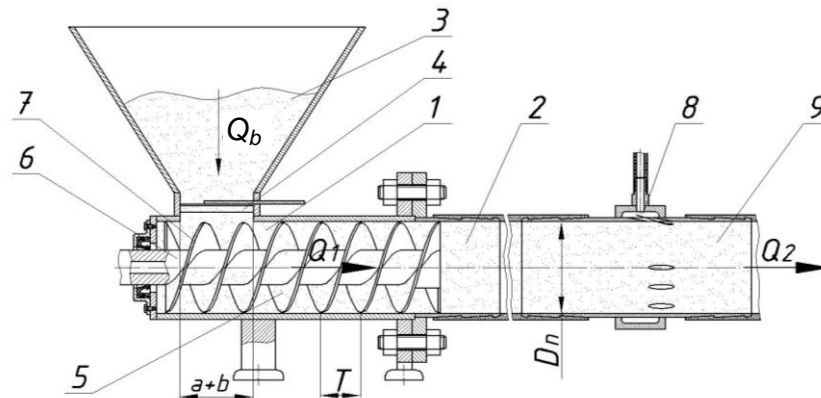
Fig.1 - A technological process flow-sheet of pneumoconveyor screw operation:
1 –screw feeder; 2 –pneumoconveyor; 3 –loading hopper; 4 –loading opener; 5 –screw; 6 – screw axle;
7 –screw flight; 8 –pneumatic nozzle section; 9 –pneumatic pipeline

To formalize the pneumoconveyor screw operation, let as assume the following:

- cross sectional area of a loading opening is rectangular in shape and its sides equal to *a, b*;

- patterns of loose material movement in a loading hopper are given by overall patterns of loose materials free outflow from a tank; here, material consumption through a loading opening is done in the process of its flowing down into a funnel at an angle of a natural slope before the height breakdown time of a dynamic unloading bridge;

- loose material consumption through a loading opening to screw flights in total is not less than the screw efficiency.

Justification of the operating elements parameters of a pneumoconveyor screw is based on the analysis of the technological material flow along the surface of the main element of a screw feeder structure. Design and kinematic parameters of a screw and dimensional specifications of a pneumoconveyor screw, which is formed by a pneumatic nozzle section and a pneumatic pipeline, have been interrelated based on the analytical analysis of the required capacity or design efficiency of a pneumoconveyor screw.

In total, the efficiency of a pneumoconveyor screw is regulated by the efficiency of its last basic block, namely, by the efficiency of a pneumatic conveyor, which, in its turn, is functionally dependent on the efficiency of a screw and, respectively, on loose material consumption through the opening of a loading hopper.

Let us set cross-sectional area of a loading opening 4 by S_o (sm^2); loose material consumption through opening 4 or per-second feeding of loose material to the flights 7 of a screw 5 is denoted by Q_b (kg/s); Q_1 (kg/s) stands for screw efficiency; pneumoconveyor efficiency is denoted by Q_2 (kg/s).

Then, it is possible to provide efficient operation of a pneumoconveyor screw under the following condition:

$$Q_b \leq Q_1 \leq Q_2 \tag{1}$$

According to *(Voityuk V.M., 2005),* loose material consumption Q_b (kg/s) through a hopper opening and the made assumption are determined from the following formula:

$$Q_b = \frac{1.47 k_n \rho S_o \sqrt[4]{r_z}}{\sqrt{f}} \tag{2}$$

where k_n – coefficient of resistance; ρ – bulk weight of loose material, kg/sm^3;. r_z – composite radius of

the opening, m; f – coefficient of internal friction.

In a general case of load transportation, in order to determine the efficiency of screw conveyers Q the following dependences are used:

$$Q = \rho F V_c \qquad (3)$$

where F – cross-sectional area of material flow, m^2; V_c average velocity of flow, m/s; ρ – load bulk weight, kg/m^3.

Here, the cross-sectional area of material flow F is determined from the flow area of a screw F_n and space filling coefficient of a trough φ_k, or

$$Q = 0.25\pi\varphi_k V_c \gamma \left(D^2 - d^2\right) \qquad (4)$$

where φ_k – space filling coefficient of a screw; D–screw diameter, m; d–diameter of a screw beater, m.

In this case, average velocity V_c of axial movement of loose material by a pressure screw can be rationally determined using a correction coefficient, which takes into account V_c reduction relative to theoretical velocity of axial movement of screw flights V_m, or slip coefficient k_k, which regulates the reduction in the design capacity of a pressure screw as consequence of the friction of loose material particles on the surface of its constructional elements

$$V_c = V_m k_k = \frac{T'\omega}{2\pi} k_k = \frac{T' k_k}{2\pi} \frac{d\varphi}{dt} = 0{,}5 k_k \left[D tg\left(45^0 - 0{,}5\alpha_k\right) - \delta_n \right] \frac{d\varphi}{dt} \qquad (5)$$

where T' is determined as the difference between the last pressure flight pitch T_n and its thickness; $\beta' = 45^0 - 0.5\alpha_k$ – a helix angle of the last pressure flight screw, deg.; α_k –angle of friction from loose material slipping on helical surface of a flight screw, deg.; δ_n –thickness of the last pressure flight screw, m.

Then, according to (4) and (5), the efficiency of a pressure screw is determined by the following formula

$$Q_1 = 0.125\pi\varphi_k k_k \rho \left(D^2 - d^2\right) \left[D tg\left(45^0 - 0.5\alpha_k\right) - \delta_n \right] \frac{d\varphi}{dt} \qquad (6)$$

The efficiency of screw conveyers depends to a great extent on the value of a space filling coefficient of a pressure screw φ_k, which is one of the main criterion (together with screw diameter and its angular velocity). That is why, when calculating the desired maximum capacity of a pressure screw it is important to take into account the volume of flights in the overall space of a screw.

For this purpose, let us introduce the coefficient, which is denoted as k_z; here k_z is formulated as the relation of the effective volume with the overall volume of a pressure screw inter-flight space, namely:

$$k_z = V_k / V_n ; \; V_k = V_n - V_z, \text{ or } k_z = 1 - V_z / V_n \qquad (7)$$

where V_k – volume of inter-flight space, m^3; $V_n = F_n l_n$ –overall volume of inter-flight space, m^3; $F_n = 0.25\left(D^2 - d^2\right)$ – cross-sectional area of a screw, m^2; l_n – length of a pressure pump, m; $V_z = V_z' z_n$ – overall space occupied by flights, m^3; $V_z' = F_z l_z$ – volume, which is occupied by the flights of one screw entry, m^3; $F_z = \delta_n h_n$ – cross-sectional area of a flight screw blade, m^2; l_z –length of a helical line over mean screw diameter, m; $h_n = 0.5\left(D - d\right)$ –height of a flight screw blade, m; z_n –number of screw entries, pc.

The length of a helical line over mean screw diameter l_z is determined by the following dependence:

$$l_z = \frac{\pi l_n (D+d)}{2T} \cos arctg \frac{2T}{D+d} = \frac{0.5 l_n (D+d)}{D tg\beta} \cos arctg \frac{2\pi D tg\beta}{D+d} \qquad (8)$$

Then, taking into consideration (7) and (8), after transformation and simplification, the coefficient k_z is determined by the following formula:

$$k_z = 1 - \frac{\delta_n z_n}{Dtg(45^0 - 0.5\alpha_k)} \cos arctg \frac{2\pi Dtg(45^0 - 0.5\alpha_k)}{D + d} \qquad (9)$$

Taking into account k_z, actual value of a space filling coefficient of a pressure screw φ_{kz} is determined as $\varphi_{kz} = \varphi_k k_z$, or

$$\varphi_{kz} = \left[1 - \frac{\delta_n z_n}{Dtg(45^0 - 0.5\alpha_k)} \cos arctg \frac{2\pi Dtg(45^0 - 0.5\alpha_k)}{D + d} \right] \varphi_k \qquad (10)$$

In fact, in the process of practical implementation of loose material transportation by a screw, the value of a space filling coefficient of a pressure screw φ_{kz} is not constant and mainly depends on the uniformity of granular loose material flow from a loading hopper and on other factors as well, for example on dimensional features of material, its moisture content, ability of firming material particles during their conveying by a screw and so on.

A slip coefficient k_k depends on many factors, such as a helix angle of flights winding, a screw diameter, a coefficient of loose material compacting by a screw, etc. and is determined by the following formula:

$$k_k = k_\beta k_y \qquad (11)$$

where k_β, k_y – respectively, the coefficients, which show the influence of a helix angle β' over a mean radius of the last pressure flight screw, compacting coefficient of a screw and a screw diameter D on a slip coefficient k_k.

Having substituted the values from formulas (10) and (11), in (6) we obtain a dependence needed for determination of the required design efficiency Q_1 (kg/s) of a pressure screw:

$$Q_1 = 0.125\pi\varphi_k k_\beta k_y \rho (D^2 - d^2) \left[Dtg(45^0 - 0.5\alpha_k) - \delta_n \right] \times$$
$$\times \left[1 - \frac{\delta_n z_n}{Dtg(45^0 - 0.5\alpha_k)} \cos arctg \frac{2\pi Dtg(45^0 - 0.5\alpha_k)}{D + d} \right] \frac{d\varphi}{dt} \qquad (12)$$

For practical implementation, when calculating design efficiency of a pressure screw Q_1' (kg/s), taking into consideration, that $\frac{d\varphi}{dt} = \omega = \frac{\pi n}{30}$, where n – rotation frequency of a screw (rpm), the dependence (12) may be written as:

$$Q_1' = 0.25\pi^2 n\varphi_k k_\beta k_y \rho (D^2 - d^2) \left[Dtg(45^0 - 0.5\alpha_k) - \delta_n \right] \times$$
$$\times \left[1 - \frac{\delta_n z_n}{Dtg(45^0 - 0.5\alpha_k)} \cos arctg \frac{2\pi Dtg(45^0 - 0.5\alpha_k)}{D + d} \right] \qquad (13)$$

or, when determining Q_1'' (t/h), it may be written as:

$$Q_1'' = 0.015\pi^2 n\varphi_k k_\beta k_y \rho (D^2 - d^2) \left[Dtg(45^0 - 0.5\alpha_k) - \delta_n \right] \times$$
$$\times \left[1 - \frac{\delta_n z_n}{Dtg(45^0 - 0.5\alpha_k)} \cos arctg \frac{2\pi Dtg(45^0 - 0.5\alpha_k)}{D + d} \right]. \qquad (14)$$

RESULTS

Change dependences of pressure screw efficiency Q_1'' on its diameter D and rotation frequency of a screw n as functional $Q_1'' = f(D,n)$ at $\varphi_k = 1, k_\beta = 0.8$; $k_y = 1.3$; $d = 0.5D$, $\alpha_k = 0.522$ rad $z = 1$, $\delta = 0.02$ m, $\rho = 1300$ kg/m^3 and on bulk material weight ρ and rotation frequency of a screw n as functional $Q_1'' = f(\rho,n)$ for D=0.15 m have been graphed (according to 14) in the form of surfaces and their two-dimensional sections, which are illustrated in Fig. 2 and Fig. 3 respectively.
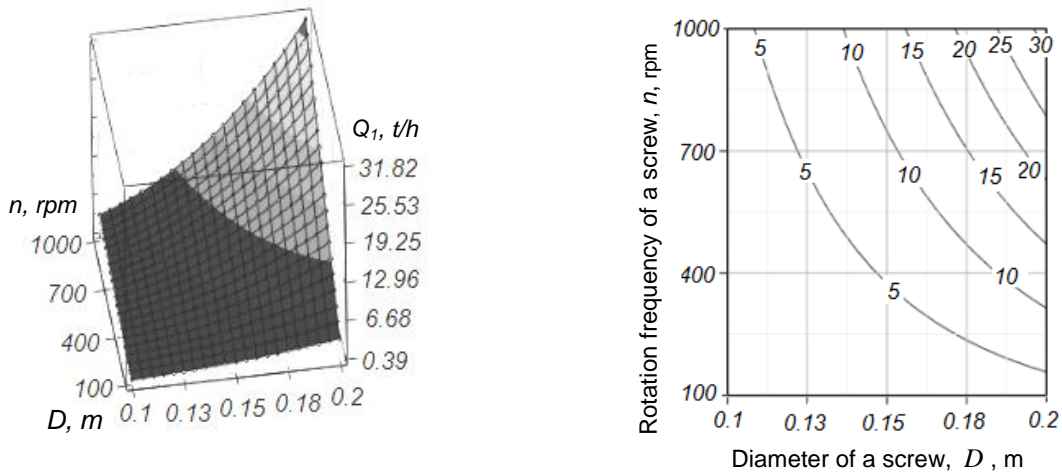
Fig. 2 - Change dependence of pressure screw efficiency Q_1'' on its diameter D and rotation frequency of a screw n as functional $Q_1'' = f(D, n)$
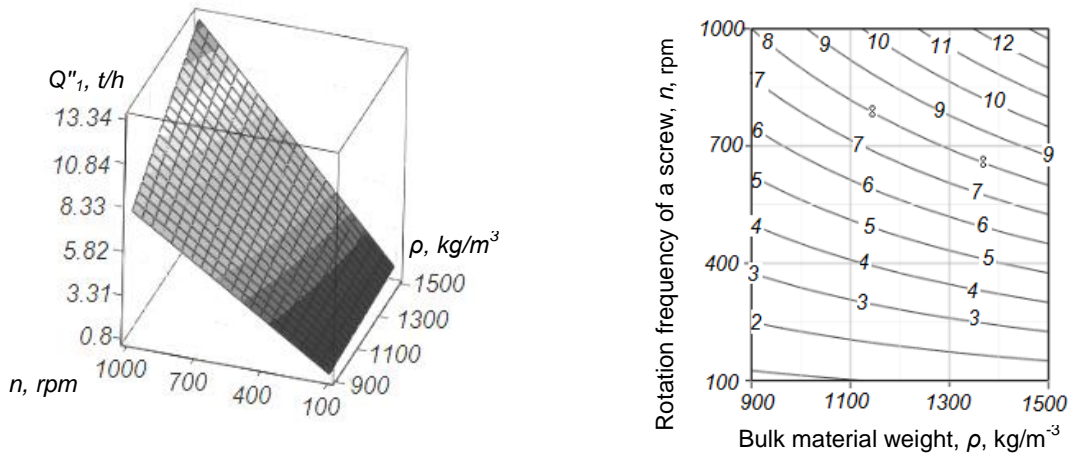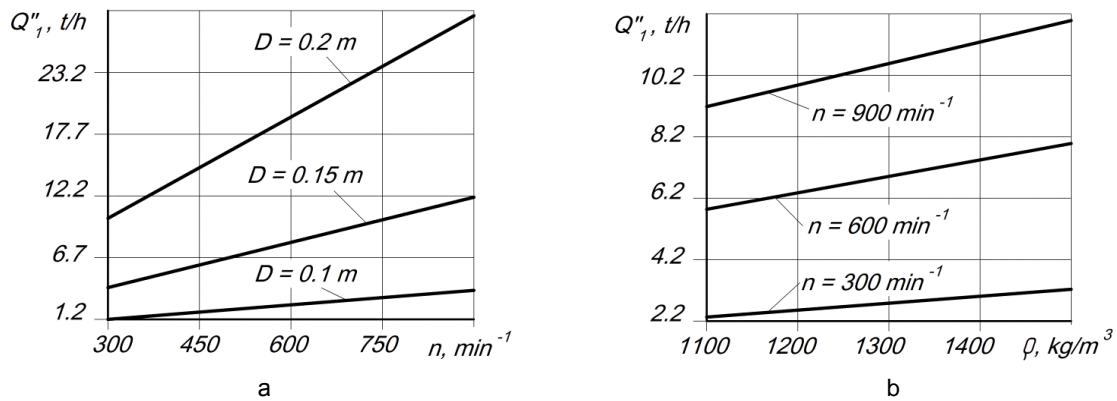

Fig. 3 - Change dependence of pressure screw efficiency Q_1'' on bulk material weight ρ and rotation frequency of a screw n as functional $Q_1'' = f(\rho, n)$

The analysis of the surfaces reveals, that pressure screw efficiency Q_1'' changes within the range of 0.4…32 (t/h) depending on the change in design and kinematic parameters and process variables of a screw and on loose material characteristics within the following limits: screw diameter D=0.1…0.2 (m); rotation frequency of a screw n=100…1000 (rpm); bulk material weight ρ = 900…1500 (kg/m³).

General tendency of the change in pressure screw efficiency Q_1'' depending on rotation frequency of a screw n at specified limits of screw diameter D variation and at ρ=1300 kg/m³ is represented by characteristic curves in Fig. 4, a, and depending on bulk material weight ρ within the specified variation limits of rotation frequency of a screw n and at D=0.15 m is represented by characteristic curves, which are shown in Fig.4, b. The change in pressure screw efficiency Q_1'' depending on rotation frequency of a screw n and bulk weight of loose material ρ is described by a linear function and is of directly proportional character. The main values of Q_1'' are within the range of $Q_1'' = 1.2…28.6$ and $Q_1'' = 2.3…11.9$ (t/h).

In order to determine the required design diameter D (cm) of a pressure screw, the values of which provide transportation of loose material, which comes from a loading opening *4* of a hopper *5* of a screw feeder *1*, let us write condition (1) according to (12) as:

$$Q_b \le 0.125\pi\varphi_k k_\beta k_y \rho \left(D^2 - d^2\right)\left[Dtg\left(45^0 - 0.5\alpha_k\right) - \delta_n\right]\times$$

$$\times\left[1 - \frac{\delta_n z_n}{Dtg\left(45^0 - 0.5\alpha_k\right)}\cos arctg\frac{2\pi Dtg\left(45^0 - 0.5\alpha_k\right)}{D+d}\right]\frac{d\varphi}{dt} \tag{15}$$

Fig. 4 - Change dependence of pressure screw efficiency Q_1'' :

a – on rotation frequency n of a screw at ρ=1300 kg/m³ as functional $Q_1'' = f(n)$;

b – on bulk material weight ρ at D=0.15 m as functional $Q_1'' = f(\rho)$

After transformation (15) and according to (2), (9) we obtain:

$$D^3 tg\beta - D^2 \delta_n - Dd^2 tg\beta + d^2 \delta_n - \frac{1.47 k_n S_o \sqrt[4]{r_z}}{A\sqrt{f}\ \frac{d\varphi}{dt}} \geq 0 \tag{16}$$

where $A = 0.125\pi\varphi_k k_\beta k_y k_z$.

According to formula (9), the value of coefficient k_z has been determined in relation to the number of screw entries z_n , at design values $\delta_n = 0,002$ m, $d = 0,45D$ and β = 0.26 rad (Table 1).

Table 1

Value of coefficient k_z depending on pressure screw design parameters

Number of entries	1			2		
Screw diameter, m	12	16	20	12	16	20
k_z	0.999	0.996	0.992	0.988	0.984	0.979

The obtained dependence (16) is the equation of the third degree relative to the diameter *D* of a pressure screw. Firstly, let us reduce it to a canonical form by dividing each term of the dependence by $tg\beta$.

$$D^3 - D^2 \frac{\delta_n}{tg\beta} - Dd^2 + d^2 \frac{\delta_n}{tg\beta} - \frac{1.47 k_n S_o \sqrt[4]{r_z}}{Atg\beta\sqrt{f}\ \frac{d\varphi}{dt}} \geq 0 \tag{17}$$

Let us set the following in equation (17) $r = -\frac{\delta_n}{tg\beta}$; $s = -d^2$; $t = \frac{1}{tg\beta}\left(d^2\delta_n - \frac{1.47 k_n S_0 \sqrt[4]{r_z}}{A\sqrt{f}\ \frac{d\varphi}{dt}}\right)$ and let us

replace unknown $y = D + (r/3)$, here canonical form of the equation is brought into a reduced equation by

the replacement of $D = y - (r/3)$, where $p = \frac{3s - r^2}{3}$, $q = \frac{2r^3}{27} - \frac{rs}{3} + t$, according to *(Hevko R.B. et.al., 2016)*.

Here, we obtain

$$\left.\begin{array}{l} D^3 + rD^2 + sD + t \geq 0; \qquad y^3 + py + q \geq 0; \\[2mm] p = \frac{\delta_n^2}{3tg^2\beta} - d^2; \qquad q = \frac{2}{3tg\beta}\left(d^2\delta_n - \frac{2.2 k_n S_o \sqrt[4]{r_z}}{A\sqrt{f}\ \frac{d\varphi}{dt}} - \frac{\delta_n^3}{9tg^2\beta}\right) \end{array}\right\} \tag{18}$$

Let us determine the discriminant D' of the reduced cubic equation (18), here

$$D' = (p/3)^3 + (q/2)^2 = \frac{1}{9tg^2\beta}\left[\frac{1}{3}\left(\frac{\delta^2}{3} - d^2tg^2\beta\right)^3 + \frac{1}{2}\left(d^2\delta_n - \frac{2.2k_nS_o\sqrt[4]{r_z}}{A\sqrt{f}\frac{d\varphi}{dt}} - \frac{\delta_n^3}{9tg^2\beta}\right)^2\right] > 0 \qquad (19)$$

That is to say, in this case equation (18) possesses one real solution of unknown in a reduced cubic equation $y^3 + py + q \geq 0$.

The reduced cubic equation $y^3 + py + q \geq 0$ in unknown y is solved using Cardano formula and taking into account that the discriminant D' of the canonical form of cubic equation (18) $D^3 + rD^2 + sD + t \geq 0$ is greater than zero, that is to say $D' > 0$, we obtain one real solution of unknown y by the expression $y = u + \upsilon$, where $u = \sqrt[3]{-(q/2)+\sqrt{D'}}$, $\upsilon = \sqrt[3]{-(q/2)-\sqrt{D'}}$, that is to say

$$y \geq \frac{1}{\sqrt[3]{3tg\beta}}\left(\sqrt[3]{\sqrt{\frac{1}{3}\left(\frac{\delta_n^2}{3} - d^2tg^2\beta\right)^3 + 0.5\left(d^2\delta_n - \frac{2.2k_nS_o\sqrt[4]{r_z}}{A\sqrt{f}\frac{d\varphi}{dt}} - \frac{\delta_n^3}{9tg^2\beta}\right)^2} - d^2\delta_n + \frac{2.2k_nS_o\sqrt[4]{r_z}}{A\sqrt{f}\frac{d\varphi}{dt}} + \frac{\delta_n^3}{9tg^2\beta}} + \sqrt[3]{\frac{\frac{2.2k_nS_o\sqrt[4]{r_z}}{A\sqrt{f}\frac{d\varphi}{dt}} + \frac{\delta_n^3}{9tg^2\beta} - d^2\delta_n - }{-\sqrt{\frac{1}{3}\left(\frac{\delta_n^2}{3} - d^2tg^2\beta\right)^3 + 0.5\left(d^2\delta_n - \frac{2.2k_nS_o\sqrt[4]{r_z}}{A\sqrt{f}\frac{d\varphi}{dt}} - \frac{\delta_n^3}{9tg^2\beta}\right)^2}}}\right). \qquad (20)$$

Using inverse substitution of value $D = y - (r/3)$ into inequality (18), taking into consideration dependence (20), and substituting value A, after corresponding calculations, let us determine a design value of the required diameter of a pressure screw D (cm) of a feeder:

$$^*D \geq \frac{1}{\sqrt[3]{3tg\beta}}\left(\sqrt[3]{\sqrt{\frac{1}{3}\left(\frac{\delta_n^2}{3} - d^2tg^2\beta\right)^3 + 0.5\left(d^2\delta_n - \frac{528k_nS_o\sqrt[4]{r_z}}{\pi^2n\varphi_kk_\beta k_yk_z\sqrt{f}} - \frac{\delta_n^3}{9tg^2\beta}\right)^2} - d^2\delta_n + \frac{528k_nS_o\sqrt[4]{r_z}}{\pi^2n\varphi_kk_\beta k_yk_z\sqrt{f}} + \frac{\delta_n^3}{9tg^2\beta}} + \sqrt[3]{\frac{\frac{528k_nS_o\sqrt[4]{r_z}}{\pi^2n\varphi_kk_\beta k_yk_z\sqrt{f}} + \frac{\delta_n^3}{9tg^2\beta} - d^2\delta_n - }{-\sqrt{\frac{1}{3}\left(\frac{\delta_n^2}{3} - d^2tg^2\beta\right)^3 + 0.5\left(d^2\delta_n - \frac{528k_nS_o\sqrt[4]{r_z}}{\pi^2n\varphi_kk_\beta k_yk_z\sqrt{f}} - \frac{\delta_n^3}{9tg^2\beta}\right)^2}}}\right) + \frac{\delta_n}{3tg\beta}. \qquad (21)$$

The obtained equation (21) describes a change in the required design diameter D of a feeder pressure screw depending on design parameters of a hopper loading opening, process parameters of a screw and loose material characteristics and the process of its transportation.

In order to determine the required design diameter D of a feeder pressure screw, the surface of change dependence of D and its two-dimensional section on the cross-sectional area S_o of a hopper loading opening of a screw feeder and rotation frequency of a screw n have been constructed in the form of a functional $D = f(S_o, n)$ (Fig. 5) and the surface of change dependence of D and its two-dimensional

section on rotation frequency of a screw n and a helix angle β have been constructed in the form of a functional $D = f(n, \beta)$ (Fig. 6) at average values $f = 0.5$, $d = 0.01$ m, $k_z = 0.984$ (Table 1.), $\beta = \pi / 12$ deg.
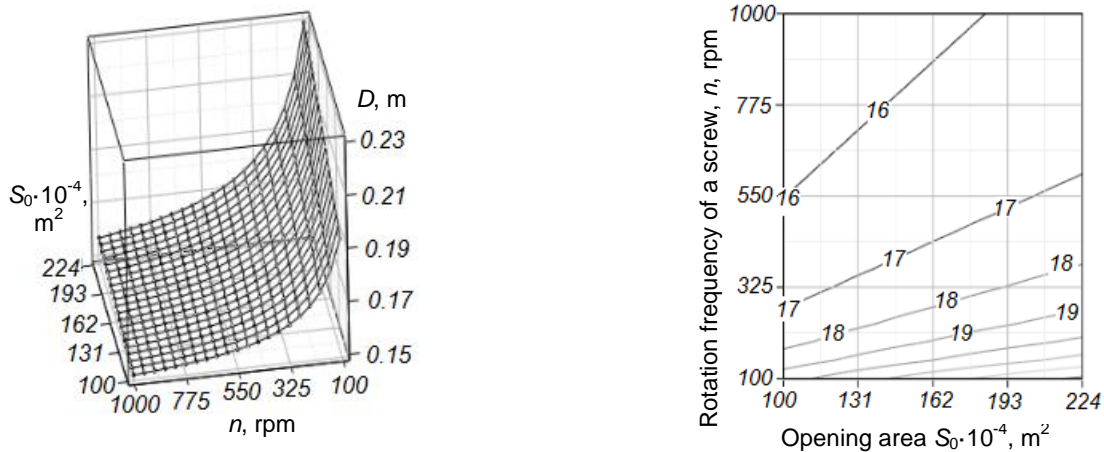

Fig. 5 - Change dependence of a screw diameter D on opening area S_o
and rotation frequency of a screw n as a functional $D = f(S_o, n)$

The analysis of the presented surfaces and their two-dimensional sections of the dependences $D = f(S_o, n)$ and $D = f(n, \beta)$ shows that with the increase in rotation frequency of a screw n from 100 to 1000 min^{-1} (Fig. 6) and a helix angle β from $\pi/18$ rad (or 10 deg) to $\pi/9$ rad (or 20 deg) the diameter D of conveyor pressure screw decreases and is within the range of $D = 0.15…0.36$ (m), which is also confirmed by characteristic curves $D = f(n)$ and $D = f(\beta)$, which are represented in Fig. 7.

Besides, diameter D change of a pressure screw depending on the change of the cross-sectional area S_0 of a hopper opening is of minor nature, here, when S_0 increases from 100 to 224 cm^2 (Fig. 5, 7, a), increment $\Delta D \cong 0.01…0.02$ m
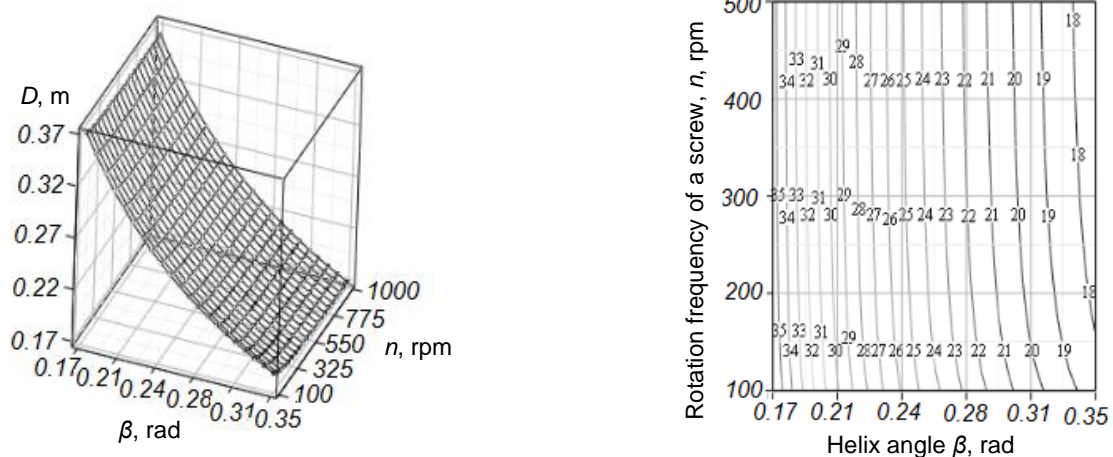

Fig. 6 - Change dependence of a screw diameter D on rotation frequency of a screw n
and a helix angle β as a functional $D = f(n, \beta)$

The conducted general analysis of the justification of a pressure screw feeder diameter D has shown that design values of D are within the range of $D = 0.15…0.36$ (m) at variations of rotation frequency of a screw being $100 \le n \le 1000$ (rpm) and a helix angle being $0.17 \le \beta \le 0.34$ (rad).

The required cross-sectional area S_o of a loading opening of a hopper, at which the desired efficiency of a screw feeder can be obtained, is determined from equation (16)

$$S_o \le \frac{A\sqrt{f}\left(D^3 tg\beta - D^2 \delta_n - D d^2 tg\beta + d^2 \delta_n\right)\dfrac{d\varphi}{dt}}{1.47 k_n \sqrt[4]{r_z}} \tag{22}$$

or

$$S_o \leq \frac{0.003\pi^2 n\varphi_k k_\beta k_y k_z \sqrt{f}\left(D^3 tg\beta - D^2\delta_n - Dd^2 tg\beta + d^2\delta_n\right)}{k_n \sqrt[4]{r_z}} \qquad (23)$$

or according to (1), (2)

$$S_o \leq \frac{Q_1\sqrt{f}}{1.47\rho k_n \sqrt[4]{r_z}} \qquad (24)$$

a   b
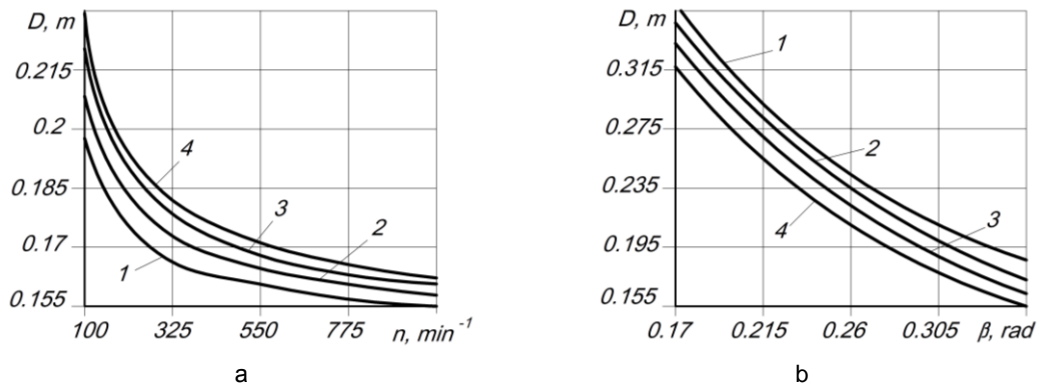
Fig. 7 - Dependence of screw diameter change D

a – on rotation frequency of a screw n at $\beta = \pi/12$ rad as a functional $D = f(n)$, 1, 2, 3, 4 – respectively,

$S_0 = 100, 140, 180, 220$ $(10^{-4} m^2)$; b – on a helix angle β at $S_0 = 160 \cdot 10^{-4} m^2$ as a functional $D = f(\beta)$,

1, 2, 3, 4 – respectively, $n = 400, 700, 1000$ (rpm[1])

According to equations (23) and (24), change dependence of the cross-sectional area S_0 of a feeder hopper loading opening on screw efficiency Q_1 and a screw diameter D, the limits of which are determined from the previous analysis, has been constructed (Fig. 8).

a   b
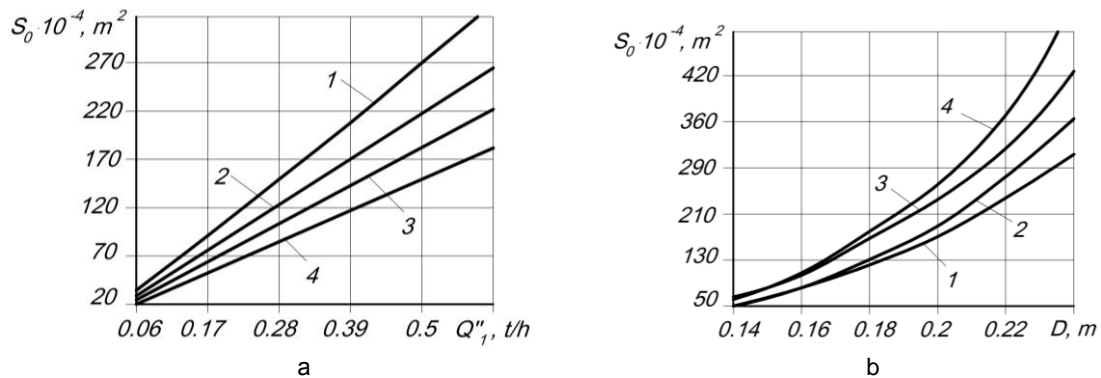
Fig. 8 - Change dependence of loading opening area S_0

a – on screw efficiency Q_1, 1, 2, 3, 4 – at ρ=900; 1100; 1300; 1500 кg/m³ respectively;

b – on screw diameter D, at n=100, 300, 700, 1000 rpm respectively

The cross-sectional area S_0 of an opening, which provides the required loose material consumption through the feeder hopper loading opening of pneumoconveyor screw, is within the range of 70…250 $(10^{-4} m^2)$ and provides designed screw capacity within the limits of 2,0…9,0 t/h (Fig. 8, a). At screw diameter variation within the range of $D = 0.15…0.22$ (m), cross-sectional area S_0 of an opening should be approximately from 70 to 350 $(10^{-4} m^2)$ (Fig. 8, b).

CONCLUSIONS

Based on the conducted constructive and technological analysis of a pressure pump operation process and under the condition of provided technological effectiveness of pneumoconveyor screw, a mathematical model, which describes a change in operation efficiency and loose material consumption through a screw feeder hopper depending on design and kinematic parameters and process variables of a screw and on loose material characteristics, has been obtained.

It has been determined that screw feeder efficiency Q_1 changes within the range of 0.4…32 (t/h) depending on process variations within the following limits: a screw diameter D= 0.1…0.2 (m); rotation frequency of a screw n = 100…1000 (min^{-1}); bulk weight of loose material ρ= 900…1500 (kg/m^3). A change in feeder efficiency Q_1'' depending on its rotation frequency and bulk weight of loose material ρ is described by a linear function and is of directly proportional character, while the values of Q_1'' are within the range of $Q_1'' = 1.2…28.6$ and $Q_1'' = 2.3…11.9$ (t/h).

Under the condition of proper operation of a screw feeder, the dependence for determining the required screw diameter, which can provide loose material consumption from a hopper to a screw has been developed. Here, it has been determined that the values of D are within the limits of D= 0.15…0.36 (m) at the variations of screw rotation frequency being 100≤ n ≤1000 (rpm) and screw helix angle being 0.17≤ β ≤0.34 (rad).

The cross-sectional area of a feeder hopper opening S_0 should be within the limits of 70…250 (10^{-4} m^2) and it provides the required screw efficiency within the range of 0.1…0.5 t/h. At diameter change within the limits of D= 0.15…0.22 (m), cross-sectional area S_0 of an opening should range from 70 to 350 (10^{-4} m^2).

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