УДК 658.562: 658.516:006.063 https://doi.org/10.34185/tpm.2.2019.01

Должанський А.М., Бондаренко О.А., Клюєв Д.Ю., Ломов І.М., Петльований Є.О., Мосьпан Н.М.

Інтегральна оцінка енергоефективності застосування шнекового нагнітача сухого технологічного мастила при волочінні

Dolzhanskiy A., Bondarenko O., Klyuev D., Lomov I., Petlovanyi Ye., Mos'pan N.

Integral assessment of energy efficiency of lubricant screw supercharger application when drawing

Зниження тертя під час волочіння забезпечується застосуванням технологічних мастил, що наближує режим тертя у волоці до рідинного. Критерієм переходу до рідинного тертя вважають рівень відношенням товщини мастильного шару до середньоарифметичної висоти мікрорельєфу не менше трьох. Мастило втягується в передосередкову зону волоки заготовкою. При цьому воно розігрівається, ущільнюється і переходить з порошкоподібного в в'язко-пластичний стан. Однак захоплюючої здатності дроту і відповідного природного гідродинамічного підпору мастила може виявитися недостатнім для формування режиму рідинного тертя. Для поліпшення умов тертя в контакті волоки з металом, що деформується, можна використовувати шнекові нагнітачі, які створюють необхідний підпір мастила в передосередковій зоні волоки. Проблемним питанням стає визначення енергетичних витрат на привід шнекового нагнітача, визначення енергетичної доцільності його використання при волочінні з урахуванням його геометричних характеристик та параметрів мастила. ***Метою*** *даної роботи стало розв'язання задачі визначення раціональних параметрів шнекового нагнітача, включаючи його геометрію і характеристики приводу: потужність, момент і швидкість з урахуванням закономірностей плину неньютонівського в'язко-пластичного технологічного мастила для забезпечення рідинного режиму тертя у волоці, а також - оцінка енергоефективності застосування такого пристрою під час волочіння. Аналіз виявив, що в рамках поставленої задачі однією з ключових величин для створення у волоці рідинного режиму тертя є тиск у мастилі, який необхідно створити шнековим нагнітачем перед волокою.* ***Методика****. Розроблена авторами ітераційна модель суміщення реологічних характеристик (пластична в'язкість та початкова межа напруження плину мастила за законом Бінгама-Шведова), закономірностей процесу волочіння та геометричних співвідношень параметрів шнекових нагнітачів побудована на основі аналізу розмірностей при урахуванні основних відповідних чинників.* ***Результати.*** *Використання моделі дозволило визначити раціональні характеристики конструкції та приводу шнекового нагнітача, а результати тестових розрахунків вперше теоретично показали, що економія електроенергії при використанні такого пристосування в кілька разів більше, ніж її витрати на привід пристрою. Одночасне зниження сил тертя під час волочіння сприяє зменшенню на 5 ... 15% енерговитрат на деформацію металу, підвищенню стійкості волок до 3 разів і зменшенню обривності дроту в 1,5 ... 3 рази. Це робить застосування привідного шнекового нагнітача в практиці сухого волочіння сталевого дроту доцільним.*

Ключові слова: волочіння, сухе технологічне мастило , шнековий нагнітач, енергоефективність

Summary. The reduction of friction during drawing is ensured by the use of technological lubricants that approximate the friction regime to liquid in dies. The criterion for the transition to liquid friction is considered the level by the ratio of the lubricating film thickness to the microrelief arithmetic average height at least three. The lubricant is drawn into the prefocal zone of the die by the wire-rod. At the same time, it is heated, compacted, and passes from a powdery to a viscoplastic state. However, the capture ability of the wire and the corresponding natural hydrodynamic lubricant affluent may not be sufficient to form a liquid friction regime. To improve the friction conditions in the contact of the die with the metal, which is deformed, supercharger can be used, which create the necessary lubricant support in the prefocal zone of the die. The task of energy costs determining for supercharger driving, determining the energy feasibility of its use when drawing, taking into account its geometric characteristics and lubrication parameters, becomes a problematic issue. The aim of this work was to solve the problem of determining the rational parameters of a supercharger, including its geometry and drive characteristics: power, torque and speed, taking into account the regularities of non-Newtonian viscous-plastic technological lubricant flow to ensure a fluid friction regime in the die, and also to evaluate the energy efficiency of using such a device in drawing. The analysis revealed that within the framework of the task, one of the main values for creating a fluid friction regime in the die is the pressure in the lubricant, which must be created with a supercharger in front of the die. The iterative model of combining rheological characteristics (plastic viscosity and the lubricant initial yield strength according to the Bingham-Shvedov regularity), the laws of the drawing process, and the geometric relationships of the parameters of supercharger was developed based on an analysis of dimensions, taking into account the main relevant factors. Using the model allowed to determine the rational characteristics of the design and drive of the supercharger.The results of test calculations theoretically for the first time showed that the energy savings when using such a device are several times greater than its costs for driving the device. A simultaneous decrease in the friction forces during drawing contributes to a 5 ... 15% reduction in energy consumption for metal deformation, an increase in die resistance up to 3 times and a decrease in wire breakage by 1.5 ... 3 times. This makes appropriative the use of a drive supercharger in the dry drawing steel wire practice.

Introduction. The efficiency of the drawing process substantially depends on the friction level in a die. Friction reduction is provided by the technological lubricants use that approach the friction regime to liquid. At the same time, they try to separate the die and wire surfaces by a lubricant film. As a technolog-

Должанський Анатолій Михайлович – д.т.н., проф. НМетАУ
Бондаренко Оксана А натоліївна – к.т.н., доц. НМетАУ,
Клюєв Дмитро Юрійович – к.т.н., доц., м.Кривий Ріг
Ломов Ілля Миколайович – к.т.н., доц. НМетАУ,
Петльований Євген Олександрович – к.т.н., доц.. НМетАУ,
Мосьпан Наталя Миколаївна – к.т.н., доц.. НМетАУ

Dolzhanskij Anatolij – d.t.s., prof. NMetAU
Bondarenko Oksana – c.t.s, doc, NMetAU
Klyuyev Dmitro– c.t.s, doc, Kr Rog
Lomov Illya– c.t.s, doc, NMetAU
Petlovanij Yevgen– c.t.s, doc, NMetAU
Mospan Natalya– c.t.s, doc, NMetAU

ical lubricant while the so-called "dry" wire drawing of steel, copper and a number of other materials are usually used powders of alkali or alkaline-earth metal salts of fatty acids (soaps) of natural or artificial origin in pure form or with functional additives (talc, lime, mica, etc.). Such a lubricant is drawn into the die preface zone by a moving wire-rod, warms up, compacts and transferees from a powdery to a viscous-plastic state. The movement of such a substance is described by the nonisothermal laws of hydrodynamics [1].

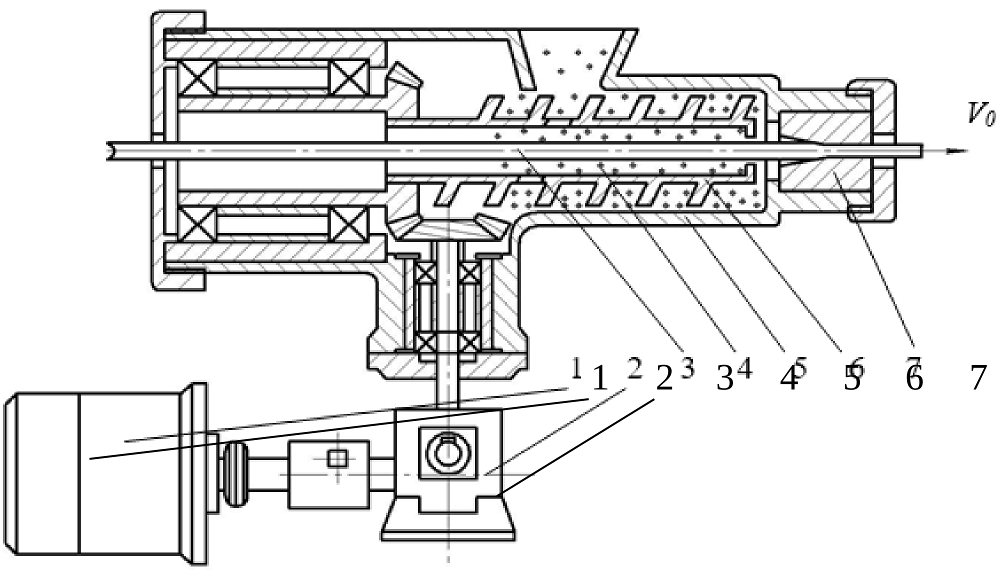
Literature data analysis. The friction regime, which is formed in the die deformation zone, is usually characterized by the dimensionless indicator [1]:

$$k = \frac{\xi}{R_a}, \qquad (1)$$

where ξ is thickness of separation lubricant film; R_a is the microrelief height in the deformation center.

In case of $k \geq 3$, lubricant almost completely separates the wearing surfaces of drawing die and wire, and the mode of friction becomes liquid. Experimental and theoretical data demonstrate that drawing in such mode is followed by minimization of friction that leads to reduction of energy consumption for metal deformation by 5…15%, increase of die life and also breakage reduction by 1.5 … 3 times [2].

However fascinating ability of wire and natural lubricant hydrodynamic head can be not enough for formation of liquid friction mode. This is accompanied by an increase in energy consumption and a decrease in the technical and economic indicators of the adopted drawing method. To improve the friction conditions in the contact of the die with the metal, various blowers are often used, which create the necessary lubricant support in the preface zone of the die. Such devices are placed in front of the die and hermetically attached to it [3]. In some cases, drive superchargers are especially effective. In particular, one of typical designs having proved itself to be positive is screw supercharger (Figure 1) [4].



1 – electric motor; 2 – reducer with belt drive; 3 – wire-rod; 4 – lubricant (soap powder); 5 – superchargers body; 6 – screw; 7 – drawing die; V_o - drawing speed

Figure 1. Drawing process scheme using a lubricator's screw

In such a device, it is necessary to bring the screw into rotation by a separate motor. This should provide pressure and lubricant flow rates at the outlet of the supercharger to the necessary conditions in the die preface zone. In this case, the question of the energy costs ratio to drive the supercharger working body and the possible energy savings in its application due to the reduction of friction in the die remains is open.

A number of approaches are known that describe the movement of a substance, in particular, consisting of discrete particles (sand, grain, flour, etc.) or liquid (water, oil, tar, suspensions, etc.) using screw blowers [5].

A feature of soap drawing lubricant movement by the auger screw is its gradual transformation from a powder state to a state of a viscous non-Newtonian fluid [1]. This is characterized by a complex and previously unknown relationship between the kinematic and geometric parameters of a screw blower, flow rate, pressure, and temperature that are created in the lubricant by the device (and, accordingly, at the entrance to the die).

The aim of the work was to solve the problem of determining the rational parameters of a supercharger, including its geometry and drive characteristics: power, torque and speed, taking into account the parameters of visco-plastic technological lubricant to ensure a fluid friction regime in the die, and in the future, based on this, an assessment of the energy efficiency of using such devices when drawing.

The research results. The necessary power N of the screw supercharger was determined by the formula [6]:

$$N = (1{,}2...1{,}4) \cdot \frac{M_{sc} \cdot \omega_{sc}}{\eta_1 \cdot \eta_2}, \qquad (2)$$

where M_{sc} is the moment in the superchargers shaft, **N**·m; ω_{sc} is an angular speed of its rotation, c^{-1}; η_1 is an efficiency coefficient of supercharger that can be accept equal to 0.70 … 0.80, taking into account use of bevel gear of rotation from the gear to a shaft of screw, countercurrent of lubricant substance and its leak flow in compressions; η_2 is the efficiency coefficient of gear, which is formed by efficiency of components of electromechanical gear of screw shaft).

Further comparison of shaft speed of the screw (rpm)

$$n_{sc} = 30 \cdot \omega_{sc} / \pi \qquad (3)$$

on rotation frequency of industrial standard electric motors ($n_{sc} =$ 600…3000 rpm and efficiency coefficient $\eta_m = 0{,}8...0{,}9$) shows that they differ by several orders of magnitude. As a result of this, in addition to the belt drive (at gear ratio $i_{bel} \le 5...7$ and efficiency coefficient $\eta_{bel} = 0{,}95...0{,}96$) a worm gear (at gearing ratio i_{gear} of several thousands and efficiency coefficient $\eta_{gear} = 0{,}7...0{,}75$ [7]) and conical reduction gear (at gear ratio $i_s = 0{,}5...3$ [8]) are used to drive the screw.

As a result we received: $\eta_2 = \prod_{u=1}^{v} \eta_u$, where $1 \le u \le v$; v, η_u are components of u-element electromechanical transmission of shaft rotation of the screw supercharger.

A torque M_{sc} in screw shaft according to data [9] approximately is:

$$M_{sc} = 0{,}131 \cdot m \cdot p_0 \cdot (D_{sc}^3 - d_{sc}^3) \cdot tg\varphi_{av}, \qquad (4)$$

and discharge Q_{sc} of lubricant substance (kg/s) in outlet of cylindrical screw supercharger with constant step H_{sc} (m) according to data [5] is determined as:

$$Q_{sc} = 0{,}127 \cdot (D_{sc} - d_{sc}) \cdot (\mathrm{H}_{sc} - \delta_{sc}) \cdot (1 - k_{ab}) \cdot \rho_m \cdot \psi \cdot \omega_{sc}, \qquad (5)$$

where $m = 2...6$ is quantity of working steps of the screw; D_{sc}, d_{sc} are diameters of the screw in outer generating line and diameter of its shaft respectively, m (Figure 2); φ_{av} is an average angel with an axis of surface screw of its helical blade in outer generating line and shaft respectively (in order to avoid a material separation from internal surface of device body, it is suggested: $\varphi_{av} \ge 10^0 = 0{,}174$ radian):

$$\varphi_{av} = 0{,}5 \cdot (\varphi_{\mathrm{D}} + \varphi_d); \; \varphi_{\mathrm{D}} = arctg\frac{H_{sc}}{\pi \cdot D_{sc}}; \; \varphi_{\mathrm{d}} = arctg\frac{H_{sc}}{\pi \cdot d_{sc}}; \qquad (6)$$

p_0 is a greasing pressure which at the outlet is developed by the supercharger, and simultaneously, lubricant pressure before the inlet to the drawing die, N/m^2; k_{ab} is an absentee ratio, which characterizes distinction of speeds of lubricant substance stream in a screw shaft and in the field of its external diameter:

$$k_{ab} = 1 - \left[\cos(2\varphi_{av}) - 0{,}5 \cdot f_{1ub} \cdot \sin(2\varphi_{av})\right]; \qquad (7)$$

indicator of internal friction in lubricant layer:

$$f_{1ub} = \frac{\tau_{liq}}{p_0}; \qquad (8)$$

τ_{1ub} is a shearing stress in a layer of viscoplastic liquid, N/m^2; $\rho_m \approx 1000$ kg/m^3 is a density of slightly compressed soap lubricant [2]; $\psi \to 1{,}0$ is a filling coefficient of interturn space of the screw; δ_{sv} is a screw flight thickness in the axial direction by outer diameter, m.

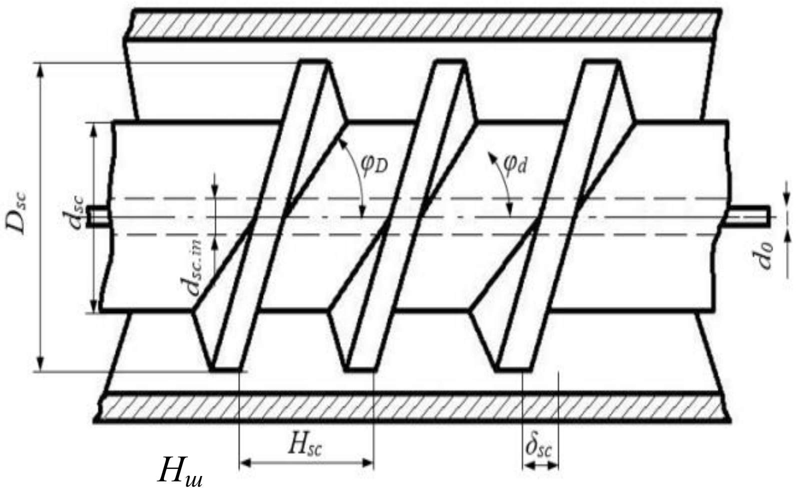


Figure 2. Scheme and screw sizes characteristics

When determining dimensional characteristics of screw supercharger the following factors have been kept in mind.

It is usually accepted:

$$H_{sc} = (0{,}7...0{,}8) \cdot D_{sc}\,. \qquad (9)$$

In this case, working length of screw is determined as follows:

$$L_{sc} = m \cdot H_{sc} < L_{des}\,, \qquad (10)$$

where L_{des} is the design limitation length, for example, connected with the sizes of equipment of drawing machine.

In case of determination of d_{sc}, it is necessary to consider presence in screw shaft of an axial hole with diameter of $d_{sc.in}$ for wire-rod passing of the maximum size $d_{0.\max}$, which is characteristic for this machine:

$$d_{\mathrm{sc.in}} = (1{,}2...1{,}4) \cdot d_{0.\max}\,. \qquad (11)$$

Considering this, the value $d_{sc} > d_{sc.in}$ is selected constructively fulfilling condition of absence of selfbraking of the moving substance: $d_{sc} \geq f \cdot H_{sc}$ and screw shaft durability condition:

$$\sigma_{eq} \leq [\sigma]_{com}\,, \qquad (12)$$

where equivalent stress of compression and shearing stress $\sigma_{\mathrm{eq}} = \sqrt{\sigma_{com}^2 + 4 \cdot \tau_r^2}$ of shaft rotation are: $\sigma_{com} = \dfrac{4 \cdot P_{sc}}{\pi \cdot (d_{sc}^2 - d_{sc.in}^2)}$ and $\tau_r = \dfrac{M_{sc}}{w}$ respectively; axial force P_{sc} (in newtons) affecting the shaft is approximately is determined by a formula [4]:

$$P_{sc} = 0{,}392 \cdot m \cdot p_0 \cdot (D_{sc}^2 - d_{sc}^2)\,; \qquad (13)$$

the moment of resistance of round shaft with through axial hole is $w = \dfrac{\pi \cdot d_{sc}^3}{16} \cdot (1 - \dfrac{d_{sc.in}^4}{d_{sc}^4})$ [5] and considering yield σ_{y} point of material of screw shaft and coefficient of margin of safety under compression $k_{mar} = 2...3$; permissible stress is $[\sigma]_{com} = \dfrac{\sigma_y}{k_{mar}}$.

The value δ_{sc} can be determined by bending strength condition of the helical blade of screw in its attachment point to the shaft. In this regard, similar to data [5] with an assumption that one flight of the screw can be assimilated to circular plate fixed on the internal contour of shaft, the maximum bending moment M_{ben} is determined by a formula:

$$M_{ben} = \frac{P_{sc} \cdot (D_{sc} - d_{sc}) \cdot \left[1{,}9 - 0{,}7 \cdot a^{-4} - 1{,}2 \cdot a^{-2} - 5{,}2 \cdot \ln a\right]}{32 \cdot (1{,}3 + 0{,}7 \cdot a^{-2})}\,, \qquad (14)$$

where $a = D_{sc}/d_{sc} = 1{,}5...3{,}0$, and according to [6]:

$$\delta_{sc} \geq \sqrt{\frac{6 \cdot M_{ben}}{[\sigma]_{ben}}}\,, \qquad (15)$$

and the allowed stress in case of bending considering coefficient of margin of safety k_{marben} can be accepted as $[\sigma]_{ben} = \dfrac{\sigma_y}{k_{marben}}$ for corresponding steel.

At that, according to recommendations [4], it is necessary to provide fulfillment of the following condition in order to avoid slipping in an internal surface of the device body of the material moved with screw supercharger:

$$S_b > S_{fl}, \qquad (16)$$

where the area of internal surface of the supercharger body along the length of one flight is $S_b = \pi \cdot D_{sc} \cdot (H_{sc} - \delta_{sc})$; area of one side of surface of one flight of the screw is $S_{fl} = \frac{1}{4\pi} \cdot (\pi \cdot \mathrm{D_{sc}} \cdot l_\mathrm{D} - \pi \cdot d_{sc} \cdot l_d + H_{sc}^2 \cdot \ln \frac{\mathrm{D}_{sc} + 2 \cdot l_\mathrm{D}}{\mathrm{d_{sc}} + 2 \cdot l_\mathrm{d}}$,

l_D, l_d are the length of development of screw lines on a surface and screw shaft respectively: $l_D = D_{sc} / \cos\varphi_\mathrm{D}$; $l_d = d_{sc} / \cos\varphi_\mathrm{d}$, and angels φ_D and φ_d are determined by formulas (6).

For achievement of constant consumption condition of lubricant substance in system "screw supercharger – drawing die", in addition to formula (5) and taking into account formula (1) for the liquid mode of friction, discharge of Q_PL (kg/s) of process lubricant in die has been determined as:

$$Q_{PL} = V_0 \cdot \rho_{PL} \cdot \frac{\pi}{4} \cdot \left[(d_{0.\max} + 2 \cdot \xi)^2 - d_0^2) \right] \approx \pi \cdot \rho_{PL} \cdot V_0 \cdot d_{0.\max} \cdot \xi = 3 \cdot \pi \cdot \rho_{PL} \cdot V_0 \cdot d_{0.\max} \cdot R_a, \qquad (17)$$

where V_0 is a drawing speed; $\rho_{PL} = 1300...1400$ (kg/m^3) is a lubricant density after its passing into the center of deformation [2].

In drawing practice, the discharge q of process lubricant is usually determined by kg/ton of the wired metal with density ρ_{met}. For such evaluation of calculation results by formula (17), it is possible to determine productivity Q_{met} (kg/s) of drawing machine as:

$$Q_{met} = \frac{\pi \cdot d_0^2}{4} \cdot V_0 \cdot \rho_{met}. \qquad (18)$$

Comparison of formulas (17) and (18) gave the following result:

$$q_{PL} = \frac{12 \cdot 10^3 \cdot R_a \cdot \rho_{PL}}{d_0 \cdot \rho_{met}} \ (kg / ton). \qquad (19)$$

Further, comparing equalities (5) and (17), after transformations, we obtained an expression for calculating the angular velocity of the screw shaft:

$$\omega_{sc} = \frac{74{,}2 \cdot V_0 \cdot d_0 \cdot R_a \cdot \rho_{PL}}{(D_{sc} - d_{sc}) \cdot (H_{sc} - \delta_{sc}) \cdot (1 - k_{ab}) \cdot \rho_m}. \qquad (20)$$

A joint consideration of formulas (2), (4), (13) and (14) showed that, within the framework of the problem to be solved, the parameters of the screw blower are related to the technological conditions of drawing. At the same time, one of the main values for creating a fluid friction regime in a die is the pressure that must be created by a screw blower in front of it.

To determine the values included in the above formulas, we took into account that the one-dimensional motion of a dry soap drawing lubricant obeys the Shvedov -Bingham law for a viscous plastic fluid [10]:

$$\tau_l = A + B \frac{\partial V}{\partial y}, \qquad (21)$$

where τ_l, $\partial V / \partial y$ are shearing stress and velocity gradient in the lubricant film respectively; A and B are boundary (minimal) stress of initial flow and plastic viscosity of process lubricant respectively, which depend on temperature and pressure. At the wire-drawing speed V_0 for the process of lubricant flow with the speed V_s at the output of screw supercharger, approximately accepted:

$$\frac{\partial V}{\partial y} \approx \frac{2 \cdot (V_0 - V_s)}{D_s - d_s},$$

and with the geometrical ratios (q.v. Fig. 2), it follows that:

$$V_s = \pi \cdot \omega_s \cdot \frac{D_s - d_s}{2} \cdot tg\varphi_{av}.$$

In this case, after simple conversions:

$$\tau_l \approx A + B \frac{2 \cdot V_0 - \pi \cdot (D_s - d_s) \cdot \omega_s \cdot tg\varphi_{av}}{D_s - d_s}. \qquad (22)$$

According to the investigated data [2] for wire conditions with dry soap lubricant at temperature T (K) and pressure p_0 (N/m^2), the value A (N/m^2) is determined by the empirical formula:

$$A = 0{,}067 \cdot 10^6 + 4{,}44 \cdot 10^{-4} \cdot p_0 + \frac{65{,}54 \cdot 10^6}{T - 273}. \qquad (23)$$

However, plastic viscosity of B ($Pa \cdot s$) can be demonstrated by the computational scheme:

$$B = \frac{1 + 1{,}273 \cdot 10^{-9} \cdot p_0}{T - 273} \cdot G, \qquad (24)$$

where:

$$T = \frac{1240 + 2{,}25 \cdot 10^{-4} \cdot \frac{V_0}{R_a}}{1 + 86{,}7 \cdot (\frac{d_0}{R_a})^{-0{,}503}}; \qquad (25)$$

$$G = 9{,}78 \cdot 10^{-5} \cdot G(\frac{V_0}{R_a}) \cdot G(\frac{d_0}{R_a}); \quad (26) \qquad G(\frac{d_0}{R_a}) = -1{,}5 \cdot \frac{d_0}{R_a} + 12000; \quad (27)$$

$$\left.\begin{array}{l} G(\frac{V_0}{R_a}) = 35{,}6 \cdot 10^3 - 9{,}88 \cdot (\frac{V_0}{R_a})^{0{,}641} \text{ at } 0{,}02 \cdot 10^6 \le \frac{V_0}{R_a} \le 0{,}2 \cdot 10^6; \\ G(\frac{V_0}{R_a}) = 11{,}66 \cdot 10^3 - 3{,}32 \cdot 10^{-3} (\frac{V_0}{R_a}) \text{ at } 0{,}2 \cdot 10^6 \le \frac{V_0}{R_a} \le 3{,}0 \cdot 10^6; \\ G(\frac{V_0}{R_a}) = 1700 \text{ at } \frac{V_0}{R_a} > 3{,}0 \cdot 10^6. \end{array}\right\} \quad (28)$$

During the "wet" wire-drawing with the use of the liquid process lubricant, the viscosity of the last one can be determined according to the data of technical literature [2].

In order to simplify the practical evaluation of p_0, the proximal change of the "accurate model" of the lubricant film forming under the "dry" wire-drawing process [2] is allowable by the empirical formula, the general form of which can be obtained as a result of dimensional analysis [11…13].

For this, we took as a response function the value: the p_0 rate with its dimensions (Pa=N/m^2 =kg·m^{-1}·s^{-2}) was accepted as response function. And as arguments the following values were accepted with their dimensions:

- index of plastic ("effective") viscosity of lubricant B (Pa·s= kg·m^{-1}·s^{-1});
- speed V_0 (m/s= m·s^{-1});
- index of friction mode $k = \frac{\xi}{R_a}$ (non-dimensional) by formula (1);
- index of annular space at the output of super-charger h (m) (q.v. fig. 1);
- diameter of the wire-rod passing through super-charger d_0 (m);
- temperature of the lubricant T (K);
- coefficient of heat capacity (for "compensation" of the absolute temperature grade) c_g (J/kg/K= m^2 ·s^2 ·K^{-1}).

The corresponding parameters equation is of the form:

$$p_0 = W \cdot \left[B^a \cdot V_0^b \cdot k^e \cdot h^x \cdot d_0^g \cdot T^z \cdot c_p^r\right], \quad (29)$$

and the formula with the dimensions:

$$(kg \cdot m^{-1} \cdot s^{-2}) = (kg \cdot m^{-1} \cdot s^{-1})^a \cdot (m \cdot s^{-1})^{\mathrm{b}} \cdot (1)^e \cdot (m)^{\mathrm{x}} \cdot (m)^{\mathrm{g}} \cdot (K)^{\mathrm{z}} \cdot (m^2 \cdot s^{-2} \cdot K^{-1})^r. \quad (30)$$

Formulas of their "balance":

- at the dimension «kg»: $1 = a$;
- at the dimension «m»: −1 = −a + b + x + g + 2r;
- at the dimension «s»: − 2 = −a − b − 2r;
- at the dimension «K» 0 = z − r.

After simple transformations, the solving of this equations has determined that a =1; b =1− 2·z; x = −1− g ; therefore, that the equation (29) obtained the following form:

$$p_0 = N \cdot \left[B^1 \cdot V_0^{1-2q} \cdot k^e \cdot h^{-1-g} \cdot d_0^g \cdot T^z \cdot c_g^r\right].$$

From this equation, after connection of the parameters with equivalent power coefficients into non-dimensional combinations (criterias), it was found that:

$$(\frac{p_0 \cdot h}{B \cdot V_0}) = W \cdot (\frac{T \cdot c_g}{V_0^2})^z \cdot (k)^e \cdot (\frac{h}{d_0})^g. \quad (31)$$

The formula (31) was linearized by logarithm:

$$\ln(\frac{p_0 \cdot h}{B \cdot V_0}) = \ln W + z \cdot \ln(\frac{T \cdot c_g}{V_0^2}) + e \cdot \ln(k) + g \cdot \ln(\frac{h}{d_0}). \quad (32)$$

Thus, with application of the method of linear regression analysis to the field of design data obtained for the dry soap lubricant with the index $c_g = 2300 \frac{\mathrm{J}}{\mathrm{kg \cdot K}}$ by computer model [2] within the limits of values changes: $5 \cdot 10^6 \le p_0 \le 200 \cdot 10^6$ (Pa); $6 \le B \le 300$ (Pa·s); $0{,}2 \le V_0 \le 5$ (m/s); $1 \cdot 10^{-6} \le \xi \le 20 \cdot 10^{-6}$ (m); $1 \cdot 10^{-6} \le R_a \le 20 \cdot 10^{-6}$ (m); $1 \le k \le 10$; $0{,}02 \cdot 10^{-3} \le h \le 10 \cdot 10^{-3}$ (m); $3{,}0 \cdot 10^{-3} \le d_0 \le 6{,}5 \cdot 10^{-3}$ (m); $320 \le T \le 580$ (K), or $47 \le t \le 307$ (°C), - it has been determined that: $\ln W = 4{,}59$; $z = 0{,}074$; $e = 2{,}80$; $g = 1{,}09$ (with the multiple correlation coefficients 0.93 and under probability 0.95).

As a result, the formula (29) has obtained the form:

$$(\frac{p_0 \cdot h}{B \cdot V_0}) = 98{,}5 \cdot (\frac{T \cdot c_g}{V_0^2})^{0,074} \cdot (k)^{2,80} \cdot (\frac{h}{d_0})^{1,09}. \quad (33)$$

As a result:

$$p_0 = 98{,}5 \cdot \frac{B \cdot V_0}{h} \cdot (\frac{T \cdot c_g}{V_0^2})^{0,074} \cdot (k)^{2,80} \cdot (\frac{h}{d_0})^{1,09}. \quad (34)$$

For the level $k = \frac{\xi}{R_a} \to 3$ that leads to the hydronamic friction mode, the formula (34) could be slightly simplified:

$$p_0 = 711 \cdot \frac{B \cdot V_0}{h} \cdot (\frac{T \cdot c_g}{V_0^2})^{0,074} \cdot (\frac{h}{d_0})^{1,09}. \quad (35)$$

Joint solution of formulas 24…28) and (35) allowed to define levels of mutually dependent parameters p_0, B and, as a result, taking into account the formulas (21) and (23) - viscous shear τ_l, and according to the formula (8), the index of friction in lubricant film f_{lb}; according to the formula (4) - the moment on the screw shaft M_s and, as the result, taking into account all necessary parameters, power N according to the formula (2).

Thus, on the basis of necessary power value N (kW) of the driving mechanism from the ordinary line of electromotors [5], it is possible to capture the certain type of the motor and determine its nominal speed n_n (r/min).

Taking this data into consideration, the total ratio i_Σ from the motor to the screw shaft is:

$$i_\Sigma = \frac{n_n}{n_s} = \prod_{u=1}^{v} i_u , \quad (36)$$

where i_u is the gear ratio of *u* -element of speed transmission from the motor to the screw shaft; n_s is the screw shaft speed which conforms to the angular rate of the screw supercharger ω_s according to formula (3).

Accordingly, it is necessary to check whether the rated-load torque M_{load} (N·m) of the motor is sufficient comparing it to torque rating M_{rat}, which is determined by formula (4):

$$M_{rat} = \frac{M_s}{i_\Sigma} \le M_{load} , \quad (37)$$

where $M_{load} = \frac{9554 \cdot N_n}{n_n}$.

For implementation of calculated gear system ratio i_Σ taking into account conic gearing "embedded" into the supercharger (q.v. Fig.1), it is reasonably to use gear system and driving mechanism. The letter will allow "soft" keeping the necessary rotation frequency n_s of the screw during the pressure fluctuation. In accordance with this, taking into account formula (36) and $\prod_{u=1}^{v} i_u = i_{bel} \cdot i_s \cdot i_{gear}$ at $i_s = 0{,}5...3$ of the conic gear, the selection and calculation of the gear system with gear ratio i_{gear} and the driving mechanism with gear ratio $i_{bel} < 5...7$ were conducted:

$$i_{gear} = \frac{i_\Sigma \cdot i_s}{i_{bel}}. \quad (38)$$

The calculations showed that in such driving mechanism scheme, it is preferable to use the dual-stage worm gear system which can provide a great value of gear i_{gear} (up to several thousand).

With the purpose of test calculation of screw supercharger of dry soap process lubricant in actual practice of its use under the wire-drawing process, the dimensions of value were the following: $V_0 = 5$ m/s; $d_0 = 7{,}0 \cdot 10^{-3}$ m; $R_a = 4 \cdot 10^{-6}$ m; the length of screw was $L = 120 \cdot 10^{-3}$ m; the efficiency coefficient of conic gear was $\eta_s = 1{,}0$; the efficiency coefficient of the supercharger was $\eta_1 = 0{,}75$; the efficiency coefficient of the motor was $\eta_{motor} = 0{,}55$; the efficiency coefficient of the gear system was $\eta_{gear} = 0{,}30$; the efficiency coefficient of the driving mechanism was $\eta_{bel} = 0{,}95$; the coefficient of heat capacity of the lubricant $c_g = 2300 \frac{J}{kg \cdot K}$; the density of the initial soap lubricant was $\rho_s = 1000 \frac{kg}{m^3}$; the density of lubricant after temperature and power impact was $\rho_{TP} = 1300 \frac{kg}{m^3}$; the density of wrought metal was $\rho_{met} = 7850 \frac{kg}{m^3}$; the coefficient of safety of screw shaft material was $k_s = 2$; the screw spade material (Steel 10) with flow limit

$\sigma_{T.spad} = 210 \cdot 10^6 \frac{N}{m^2}$; the load factor was $k_{spad} = 3$.

The following results were obtained according to the calculations, which are presented by the arrows, with the dimensions in square brackets and the use of represented formulas in the round brackets: $d_{s.in} = 10 \cdot 10^{-3}[m]$ (11); $\rightarrow d_s = 26 \cdot 10^{-3}[m]$; $\rightarrow D_s = 40 \cdot 10^{-3}[m]$ when $a = 2{,}3$; $\rightarrow H_s = 30 \cdot 10^{-3}[m]$ (9); $\rightarrow m = 4$ from the formula (10); $\rightarrow V_0/R_a = 1{,}25 \cdot 10^6 [s^{-1}]$; $\rightarrow \frac{d_0}{R_a} = 1{,}75 \cdot 10^3$; $\rightarrow T = 503[K]$ (25); $\rightarrow G(V_0/R_a) = 7510$ (28); $\rightarrow$; $G(d_0/R_a) = 9375$ (27); $\rightarrow G = 6886$ (26); $\rightarrow B = 29{,}94 + 38{,}1 \cdot 10^{-9} \cdot p_0[Pa \cdot s]$ (24); $\rightarrow h = 7 \cdot 10^{-3}[m]$; $\rightarrow p_0 = 1{,}122 \cdot 10^6 \cdot B$ [N/m^2] (35); $\rightarrow p_0 = 33{,}72 \cdot 10^6$ [N/m^2] and $B = 31{,}2$ [Pa·s] as a result of simultaneous consideration of (24…28) and (35); $\rightarrow A = 0{,}367[N/m^2]$ (23); $\rightarrow P_s = 49000$ [N] (13); $\rightarrow \varphi_D = 13°{,}4$ (6); $\rightarrow \varphi_d = 20°{,}2$ (6); $\rightarrow \varphi_{aw} = 16°{,}8 \succ 10^0$ (6); $\rightarrow M_s = 248$ [N·m] (4); $\rightarrow w = 3{,}45 \cdot 10^{-6}$ [m^3] – for (22); $\rightarrow \tau_{kr} = 71{,}9 \cdot 10^6$ [N/m^2] – for (22); $\rightarrow \sigma_{sg} = 100 \cdot 10^6$ [N/m^2] – for (22); $\rightarrow \sigma_{eg} = 175 \cdot 10^6$ [N/m^2] – for (22); $\rightarrow \sigma_T = 350 \cdot 10^6$ [N/m^2], on the basis of which the screw shaft material is Steel 40X with $\sigma_T \geq 400 \cdot 10^6$ [N/m^2] $\rightarrow M_{ben} = -30{,}4[N \cdot m]$ (14); $\rightarrow \delta_s \geq 1{,}61 \cdot 10^{-3}[\mathrm{m}]$ (15), on its base it was accepted that $\delta_s = 2 \cdot 10^{-3}[\mathrm{m}]$; $\rightarrow S_{corp} = 3{,}52 \cdot 10^{-3}[\mathrm{m}^2]$ - for (16); $\rightarrow l_D = 41{,}1 \cdot 10^{-3}[m]$ - for (16); $\rightarrow l_d = 27{,}7 \cdot 10^{-3}[m]$ - for (16); $\rightarrow S_{vit} = 0{,}5 \cdot 10^{-3}[\mathrm{m}^2]$ - for (16); $\rightarrow \mathrm{S_b} > \mathrm{S_{fl}}$ - the condition (16) was performed; $\rightarrow$ it was accepted in advance $f_{sm} = 0{,}2$ - for (7) $\rightarrow k_{ots.1} = 0{,}222$ - the first base to the stepwise approximation according to (7) $\rightarrow \omega_s = 44 \cdot 10^{-3}[s^{-1}]$ - the first base to the stepwise approximation according to (20); $\rightarrow \tau_g = 0{,}39 \cdot 10^6$ [N/m2] – the first base to the stepwise approximation according to (22); $\rightarrow f_{ub} = 0{,}012$ (8) $\rightarrow$ the condition of absence of self-braking of lubricant substance is satisfied: $f_s \cdot H_s << d_s$; $\rightarrow k_{ost.2} = 0{,}17$ - the second base to the stepwise approximation according to (7); $\rightarrow \omega_s = 41{,}5 \cdot 10^{-3}[s^{-1}]$ - the second base to the step-wise approximation according to (20); $\rightarrow n_{sc} = 0{,}013$[sp/s]=0,4 [sp/min] (3); $\rightarrow q_{PL} = 2$ [kg/tn] (19), that is conformed with the practice of wire-drawing process; $\rightarrow N = 0{,}105$ [kW] (2), on the basis of which the asynchronous three-phrase motor of AIR 56A4 type with the nominal rating power $N_n = 0{,}22$ [kW], is selected, nominal speed is $n_n = 1360$[sp/min], efficiency of the $\eta_{mot} = 66\%$, with weight 3,8 [kg]; $\rightarrow i_\Sigma = 3375$; $\rightarrow M_{rat} = 0{,}073$[N·m]; $\rightarrow M_n = 0{,}573$[N·m]; $\rightarrow \mathrm{M_n} >> \mathrm{M_{rat}}$; $\rightarrow$ for the driving mechanism of the screw supercharger; screw type double-stage regulator 5CH80/40 with $i_{gear} = 1250$, $\mathrm{M} = 355$[N·m]>> M_s and mass of 38 [kg]; $\rightarrow i_{gear} = 2{,}7$ (38).

The necessary strength calculations for all transmission elements were carried out according to known methods [14].

Next, we compared the calculated value $N = 0{,}105$ [kW] of the required drive power of the screw blower with the difference $\Delta N = N_0 - N_1$ (between the values of the drawing N_0 power without using a blower and N_1 with its use):

$$E = \frac{N}{\Delta N}(\cdot 100\%). \tag{39}$$

This corresponds to the "performance indicator" of the supercharger use:

$$I = \frac{1}{E}. \tag{40}$$

The power of the drawing process was determined by the expression:

$$N_{draw} = \frac{\pi \cdot d_0^2}{4} \cdot V_0 \cdot \sigma_{draw}, \tag{41}$$

where according to the formula Dolzhanskiy A. [15]:

$$\sigma_{draw} \approx \sigma_T \cdot \left[\ln \mu + 0,77 \cdot \alpha + \frac{f_{draw} \cdot \ln \mu}{\alpha + \alpha^2 + f_{draw} \cdot \ln \mu} + \frac{2,6 \cdot f_{draw} \cdot \alpha}{\alpha + f_{draw} \cdot \ln \mu + 2 \cdot f_{draw} \cdot \alpha} \right], \quad (41)$$

were $\sigma_{draw}, \sigma_{\theta}$ are wire-drawing stress and flow limit of the wrought material; μ is the metal elongation ratio during wire-drawing process; α is a half of the conic zone angle of a drawing die; f_{draw} is the friction ratio during wire drawing.

Usually, $\alpha = 0,1...0,12$ rad; $\mu = 1,25...1,40$ at rough average steel wire-drawing. According to the data of [15] for ordinary drawing conditions (without a supercharger, index «0») with a friction mode index $k = \frac{\xi}{R_a} << 3$ we have accepted $f_{draw.0} = 0,2$. When testing the effectiveness of the use of a screw blower (fluid friction drawing for $k = \frac{\xi}{R_a} = 3$, index «1») the friction coefficient level became $f_{draw.1} = 0,16$.

Corresponding value ΔN for $\alpha = 0,12$; $\mu = 1,35$; $\sigma_T = 250 \cdot 10^6$ *N/m²* amounted to $\Delta N \approx 0,7$ *kW*, which are in the formulas (39) and (40), determined $E \approx 0,14$ (14%) and $I \approx 7$ respectively.

The discussion of the results. The presented data suggest that the energy saving during the advanced wire-drawing process is several times hire than the calculated power necessary for the screw supercharger driving mechanism. It indicates the applicability of such tool in practice of wire-drawing. In addition, due to a decrease in the level of friction forces during drawing, a significant increase in the wear resistance of the fibers and a decrease in wire breakage can be expected, which makes the use of a supercharger in the practice of dry wire drawing appropriate.

Conclusions. A method has been developed for calculating the parameters of a screw blower of a dry soap technological lubricant to ensure a fluid friction during wire-drawing. The method is based on the use of the laws of non-Newtonian fluid flow, drawing process, application of dimensional analysis and is presented in the form of an appropriate calculation algorithm. Test calculations showed that energy savings when using a screw lubricator in the drawing process are several times greater than its consumption for this device driving. A simultaneous decrease of the friction forces during drawing contributes to a significant increase of the die wear resistance and to a wire breakage decrease, which makes the use of a supercharger in the practice of dry wire-drawing efficient.

Бібліографічний опис

1. Колмогоров Г.Л. Гидродинамическая смазка при обработке металлов давленим/ Г. Колмогоров. – М.: Металлургия, 1986. – 166 с.
2. Должанский А.М. Теоретический учет неизотермических явлений при захвате сухой волочильной смазки/ А. Должанский// Известия вузов. Черная металлургия. – 1998. – № 9. – С. 38-41.
3. Kelly, A.L. The effect of screw geometry on melt temperature profile in single screw extrusion/ A.L. Kelly, E.C. Brown, P.D. Coates // Polymer engineering and science. – 2006. – Vol. 12. (Vol. 46). – P. 1706-1714; **doi**: 10.1002/pen.20657.
4. Декларацiйний патент 58916А Україна. МПК 7В21С3/00. Пристрій для волочіння круглої заготовки/ Должанський А.М., Голобурда Ю.В., Ломов І.М. – Заявка 2002119301. – Бюл. №18. – 15.08.2003 р.
5. Ostroukh, A.S. Calculation of performance for mechanical deboning screw presses considering counterpressure / A.S. Ostroukh, V.A. Abaldova // Theory and practice of meat processing. – 2016. – Vol. 3. – P. 66-80. **doi**: 10.21323/2414-438X
6. Nikola Stosic Review of mathematical models in performance calculation of screw compressors/ Nikola Stosic, Ian K. Smith, Ahmed Kovacevic and Evedin Mujic// International journal of fluid machinery and systems. – April-June, 2011. – Vol. 4. – P. 200-217; **doi**: 10.5293/ijfms.2011.4.2.200.
7. Roman Rogatynskyi The cargo movement model by the screw conveyor surfaces with the rotating casing/ Roman Rogatynskyi, Ivan Hevko, Andrii Diachun, Olena Rogatynska, Andrii Melnychuk// Ternopil National Technical University. – 04.2018. – Vol. 4 (92). – P. 34-41; **doi:** 10.33108/visnyk_tntu2018.04.034.
8. Kotlov, A. Influence of suction port parameters on integral characteristics of screw-type compressor/ International Conference on Screw Machines 2018IOP; Conf. Series: Materials Science and Engineering 425 (2018) 012014; **doi**: 10.1088/1757-899X/425/1/012014 P. 1- 10.
9. Справочник по оборудованию зерноперерабатывающих предприятий. Изд. 2-е, перераб. и доп. / А.Б. Демских, М.А. Борискин, Е.В. Тамаров [и др.]. – М.: Колос, 1980. – 383 с.
10. Исследование установившегося течения вязкопластического битумного вяжущего, описываемого моделью Шведова-Бингама, в цилиндрической трубе/ О.В. Матвиенко, В.П. Базуев, Н.Р. Сабылина [и др.]// Вестник Томского государственного архитектурно-строительного университета. – 2019. – Т. 21. – № 3. – С. 158-177; **doi**: 10.31675/1607-1859-2019-21-3-158-177.
11. Frishter, L. Application of the methods of the theory similarity and dimensional analysis for research the local stress-strain state in the neighborhood of an irregular point of the boundary/ L. Frishter// Procedia Engineering. – Vol. 153. – 2016. – P. 151 – 156; **doi**: 10.1016/j.proeng.2016.08.095.

12. Bhaskar, R. Qualitative physics using dimensional analysis/ R. Bhaskar, Anil Nigam// Artificial Intelligence. – 1990. – Vol. 45. – P. 73-111; **doi**: 10.1016/0004-3702(90)90038-2.
13. Protsenko, Vyacheslav. Application of dimensional analysis and similarity theory in electrochemistry: a review/ Protsenko Vyacheslav// Issues of Chemistry and Chemical Technology (Voprosy khimii i khimicheskoi technologii). – January, 2014. – Vol. 3. – P. 5-9.
14. Анурьев, В.И. Справочник конструктора машиностроителя: в 3 т. / В.И. Анурьев. – 5-е изд., перераб. и доп. – М.: Машиностроение, 1981. – Т. 3. – 557 с.
15. Должанский, А.М. Определение тягового напряжения и оптимального угла волоки с учетом критерия формы очага деформации/ А. Должанский// Металлургическая и горнорудная промышленность. – 2003. – Сообщение 1. – № 2. – С. 70-73; Сообщение 2. – № 3. – С. 71-73; Сообщение 3. – № 4. – С. 61-63.

References

1. Kolmohorov H.L. Hidrodynamichna mastylo pry obrobtsi metaliv tyskom / H. Kolmohorov. - M.: Metalurhiya, 1986. - 166 p.
2. Dolzhanskiy A.M. Teoreticheskiy uchet neizotermicheskikh yavleniy pri zakhvate sukhoy volochil'noy smazki/ A. Dolzhanskiy// Izvestiya vuzov. Chernaya metallurgiya. – 1998. – № 9. – P. 38-41.(in Ukraine).
3. Kelly, A.L. The effect of screw geometry on melt temperature profile in single screw extrusion/ A.L. Kelly, E.C. Brown, P.D. Coates // Polymer engineering and science. – 2006. – Vol. 12. (Vol. 46). – P. 1706-1714; **doi**: 10.1002/pen.20657.
4. Deklaratsiyniy patent 58916A Ukrayina. MPK 7V21S3/00. Prystriy dlya volochinnya kruhloyi zahotovky / Dolzhans'kiy A.M., Holoburda YU.V., Lomov I.M. - Zayavka 2002119301. - Byul. №18. - 15.08.2003 y.
5. Ostroukh, A.S. Calculation of performance for mechanical deboning screw presses considering counterpressure / A.S. Ostroukh, V.A. Abaldova // Theory and practice of meat processing. – 2016. – Vol. 3. – P. 66-80. **doi**: 10.21323/2414-438X.
6. Nikola Stosic Review of mathematical models in performance calculation of screw compressors/ Nikola Stosic, Ian K. Smith, Ahmed Kovacevic and Evedin Mujic// International journal of fluid machinery and systems. – April-June, 2011. – Vol. 4. – P. 200-217; **doi**: 10.5293/ijfms.2011.4.2.200.(in Russian).
7. Roman Rogatynskyi The cargo movement model by the screw conveyor surfaces with the rotating casing/ Roman Rogatynskyi, Ivan Hevko, Andrii Diachun, Olena Rogatynska, Andrii Melnychuk// Ternopil National Technical University. – 04.2018. – Vol. 4 (92). – P. 34-41; **doi:** 10.33108/visnyk_tntu2018.04.034.
8. Kotlov, A. Influence of suction port parameters on integral characteristics of screw-type compressor/ International Conference on Screw Machines 2018IOP; Conf. Series: Materials Science and Engineering 425 (2018) 012014; **doi**: 10.1088/1757-899X/425/1/012014 P. 1- 10.
9. Dovidnyk shchodo obladnannya zernopererobnykh pidpryyemstv. Vyd. 2-e, pererab. i dop. / A.B. Dems'kyy, M.A. Boryskyn, E.V. Tamarov [y dr.]. - M.:Kolos, 1980. - 383 p.
10. Issledovaniye ustanovivshegosya techeniya vyazkoplasticheskogo bitumnogo vyazhushchego, opisyvayemogo model'yu Shvedova-Bingama, v tsilindricheskoy trube/ O.V. Matviyenko, V.P. Bazuyev, N.R. Sabylina [i dr.]// Vestnik Tomskogo gosudarstvennogo arkhitekturno-stroitel'nogo universiteta. – 2019. – T. 21. – № 3. – S. 158-177; **doi**: 10.31675/1607-1859-2019-21-3-158-177. (in Russian).
11. Frishter, L. Application of the methods of the theory similarity and dimensional analysis for research the local stress-strain state in the neighborhood of an irregular point of the boundary/ L. Frishter// Procedia Engineering. – Vol. 153. – 2016. – P. 151 – 156; **doi**: 10.1016/j.proeng.2016.08.095.
12. Bhaskar, R. Qualitative physics using dimensional analysis/ R. Bhaskar, Anil Nigam// Artificial Intelligence. – 1990. – Vol. 45. – P. 73-111; **doi**: 10.1016/0004-3702(90)90038-2.
13. Protsenko, Vyacheslav. Application of dimensional analysis and similarity theory in electrochemistry: a review/ Protsenko Vyacheslav// Issues of Chemistry and Chemical Technology (Voprosy khimii i khimicheskoi technologii). – January, 2014. – Vol. 3. – P. 5-9.
14. Anur'yev, V.I. Spravochnik konstruktora mashinostroitelya: v 3 t. / V.I. Anur'yev. – 5-ye izd., pererab. i dop. – M.: Mashinostroyeniye, 1981. – T. 3. – 557 p.
15. Dolzhanskiy, A.M. Opredeleniye tyagovogo napryazheniya i optimal'nogo ugla voloki s uchetom kriteriya formy ochaga deformatsii/ A. Dolzhanskiy// Metallurgicheskaya i gornorudnaya promyshlennost'. – 2003. – Soobshcheniye 1. – № 2. – P. 70-73; Soobshcheniye 2. – № 3. – P. 71-73; Soobshcheniye 3. – № 4. – P. 61-63.

Стаття поступила:15.01.2019