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THEORY OF THE WIRE-DRAWING DIE DESIGN FOR THE FILM LUBRICATION REGIME

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Examples of die design for the wire-drawing in the film lubrication regime are presented. Calculation methods for die parameters taking into account pumping effect of both the force feed nozzle and the die itself have been developed for the case of liquid and viscous-plastic lubricants. Basic formulae necessary for the die parameters calculation providing the separation of the workpiece and die surface by the lubricant film are quoted. The optimal opening value of the force feed nozzle gap assuring the maximal lubricant film thickness in the deformation zone has been determined.

Notation

A, B	rheological constants,
b_1, b_2, k_1, k_2	dimensionless constants,
c	lubricant specific heat,
d_0	workpiece diameter,
F	tensile counter-force due to the lubricant action,
H	gap opening between the workpiece and nozzle,
H_1, H_2	rigid lubricant layer parameters,
H_d	gap opening at the entry to working die channel,
H_n	lubrication gap opening at the force feed nozzle entry,
H_{opt}	optimal working die gap opening,
H_0	lubricant layer thickness at the deformation zone entry,
K, b, m	temperature coefficients,
K_1, K_2	lubricant slip coefficients,
L	nozzle length,
L_d	working die working cone length,
L_n	force feed die working cone length,
L_0	optimal nozzle length,
h_1, h_2, h_n, l_d, l_n	dimensionless quantities,
p	lubricant layer pressure,
p_1	lubricant pressure at the nozzle exit,

p_2 lubricant pressure at the deformation zone entry,
p_3 lubricant pressure due to working die action,
Q volumetric lubricant flow rate per unit length of the workpiece contour,
q relative lubricant flow rate,
Rz height of the workpiece surface micro-roughness,
t lubricant layer temperature,
v lubricant flow velocity,
v_0 drawn workpiece velocity,
v_{sl} lubricant slip velocity at the workpiece or nozzle surface,
x, y Cartesian coordinates,
t_0 wire temperature at the nozzle entry,
t_c ambient temperature,
α coefficient of piezo-viscosity,
α_d generating angle of the working cone,
α_n generating angle of the nozzle cone,
γ heat exchange coefficient,
β_0, β_1, β_2, σ dimensionless parameters,
μ lubricant dynamic viscosity,
μ_0 lubricant dynamic viscosity for $p = 0$,
μ_{50} lubricant dynamic viscosity at $t = 50°$C,
ρ lubricant density,
σ_0 tensile stress produced by the counter-force,
σ_S limit stress of the workpiece material plastic deformation,
τ_R tangential stress in the lubricant layer at the surface,
τ_S critical shear stress.

1. Introduction

The intensification of the wire-drawing process needs the working tool design to improve the technical and economical process parameters by reducing the friction forces in the deformation zone. The external friction arising at the contact between the tool and workpiece is the main factor obstructing the plastic deformation during the wire-drawing process. The action of friction forces decreases the attained diameter reduction per one draft, reduces the drawing velocity, causes the worked metal stick to the die surface and deteriorates the quality of the product surface. Contact friction gives rise to non-uniform temperature distribution over the drawn wire cross-section what results in the thermal stresses causing deterioration of the final product characteristics. E.g., intensive heating of the surface layer observed during drawing of the high strength metals and alloys may result, in the case of age-hardening alloys, in deterioration of the plastic characteristics due to the ageing process.

The most effective way to reduce the drawing process friction forces leads through attaining of the film lubrication regime, i.e. separation of the wire and die surfaces by the continuous lubricant film [1]. Recently, an advanced technique of wire manufacturing based on the so-called compound dies is frequently employed [2]. The compound die consists of the force feed lubricator supplying the lubricant into the deformation zone and the working die, where the plastic deformation occurs. For the lubricant force feeder one uses special force feed nozzles or simply the worn-out dies taken from the previous stage of the draft operation. The lubricant pressure growth is attained due to the hydrodynamic effect of the lubricant force fed by the moving wire into the working zone of the device. The lubricants used for the wet drawing are usually very viscous, thus the lubricant is dragged by the surface of the moving wire into the gap between the nozzle wall and the workpiece, with the resulting force fed into the device. When the appropriate pressure is attained, the separation of the interacting surfaces of the working die and the workpiece takes place. The efficiency of the compound dies depends to a great extent on the appropriate choice of the device design parameters, mainly on the gap opening between the force feed nozzle and the drawn workpiece. The compound dies make it possible to increase the process output by 30 %, three to four times the die durability, to reduce the electric energy consumption by 10-20 %, and to increase by 40-45 % the diameter reduction ratio per one draft. At the same time, the preliminary preparation of the workpiece surface becomes much simpler or even, in some cases, it can be eliminated. The main feature of the compound die design is the presence of one or more force feed nozzles (or dies) assembled in the hermetically sealed housing together with the working die. External dimensions of the compound die housing enable us, in the most cases, to use the existing drawbench die holders without any changes.

In the course of operation of the compound dies insufficient reliability of the working space sealing has been disclosed. The lubricant is squeezed out through the sealing what reduces the device efficiency due to the transition from the film lubrication to the boundary lubrication regime, worsens maintenance conditions and increases the costs due to excess losses of the lubricant.

The compound dies deficiency mentioned above has been removed in the film lubrication wet drawing device shown in Fig.1 cf. [3]. This device differs from the compound die by the presence of the soft insert of plastic material,

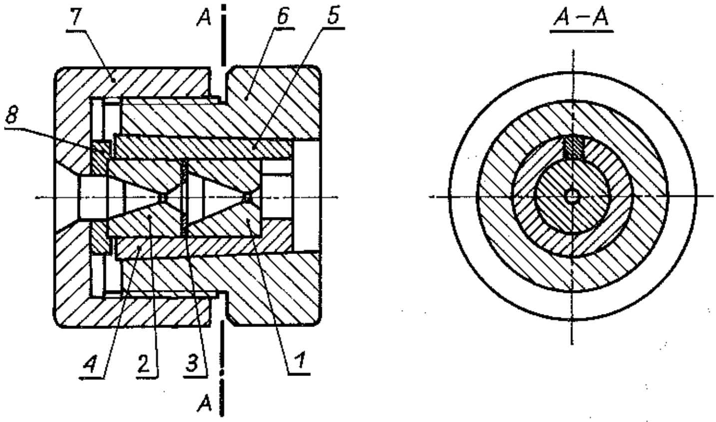

FIG. 1. Improved compound die design: *1* and *2* – working and force feed hard-alloy inserts, *3* – thickening washer, *4* – collet, *5* – sealing insert , *6* – housing, *7* – flanged nut, *8* – thrust washer.

e.g. copper, placed into the collet slot. The drawing device assembling has to be performed as described in the paper [4]. The force feed (*2*) and the working (*1*) dies separated by the thickening washer (*3*) are placed into the collet (*4*), the insert is placed into the collet slot and the whole set is placed in the housing (*6*). The thrust washer (*8*) preventing heading die (*2*) edges from spalling is placed on the front side of the die (*2*) and the flanged nut (*7*) is screwed on at the housing (*6*) until it reaches the thrust washer (*8*). Then the axial pressure is exerted by the punch on the dies (*1*) and (*2*), the collet is press-fitted into the housing (*6*) and the flanged nut is tightened, while the pressure on the dies is still exerted. The plastic deformation of the soft insert fills the collet (*4*) slot, thus assuring reliable sealing of the device. Application of the modified compound die makes it possible to assure reliable sealing of the working space and to prevent both the lubricant squeezing off and reduction of the pressure attained. The lubricant expense is also reduced and thus the device efficiency is raised.

Polish researchers [5] proposed another method of sealing of the wet drawing device providing proper working space sealing tightness up to 2 GPa lubricant pressure. It is based on application of some kind of labyrinth seal. Two metallic concentric rings are placed between the die surfaces, the inner one being of rectangular cross-section and the outer one – circular. Before assembling the device, the space between the rings is filled with powdered zinc oxide.

Compound wet drawing devices are employed for manufacturing of the aluminum wire designed for bare cables with aluminum conducting core, particularly a drawing device with hydrodynamic lubricant feed shown in Fig.2. Axial compression of the working die (*2*) and the force feed nozzle (*3*) placed in the housing (*1*) and separated by the sealing ring (*4*) is ensured with a hollow pressure plug (*7*) connected with a flanged nut (*6*). During assembling the device, annular flange of the flanged nut (*6*) deforms the elastic clamping bush (*5*) giving rise to an external force pressing together the die and the nozzle. Changing the tightening of the flanged nut one can control the external force according to the strength of the drawn workpiece.

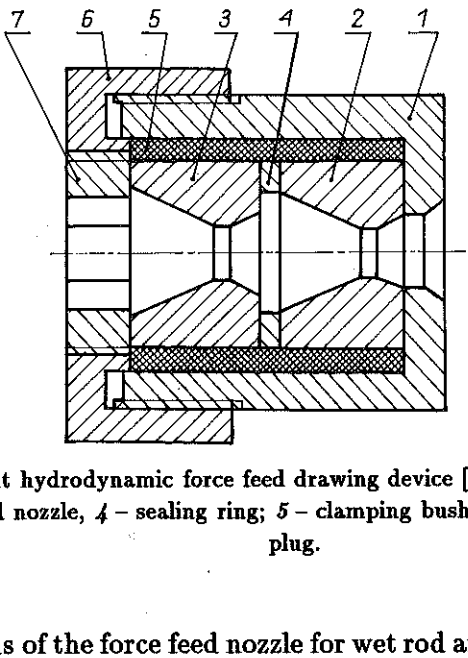

FIG. 2. Lubricant hydrodynamic force feed drawing device [6]: *1* – housing, *2* – working die, *3* – force feed nozzle, *4* – sealing ring; *5* – clamping bush, *6* – flanged nut, *7* – hollow plug.

Applications of the force feed nozzle for wet rod and wire drawing through the rotating die as well as for coating of the wire by the uniform thickness lubricant layer, and also for providing the proper tensile counter-force in the case of roll drawing are also known.

2. CALCULATION OF THE COMPOUND DEVICE PARAMETERS FOR THE FLUID LUBRICANT DRAWING

As it has already been mentioned, for the wet drawing in the fluid film lubrication regime the force feed nozzles or heading dies can be used for assuring proper hydrodynamic conditions. The lubricant force feed into the gap between the wire and the nozzle is provided by the workpiece motion. When a certain lubricant pressure is attained, the separation of the wire and tool surfaces in the deformation zone occurs. From the practice of hydrodynamic lubricant feed it follows, that the ratio of the lubricant film thickness to the force feed nozzle diameter is small, and the gap opening between the nozzle surface and the workpiece is smaller than the other two dimensions of the lubricant channel. Thus the liquid lubricant flow in the nozzle can be described by the following differential equation

$$\frac{dp}{dx} = \mu \frac{d^2 v_x}{dy^2}, \tag{2.1}$$

where dp/dx is the component of the lubricant pressure gradient in the direction of drawing, v_x is the lubricant flow velocity component and μ denotes the dynamic viscosity coefficient of the lubricant.

Using the boundary conditions: $v_x(0) = 0$, $v_x(H) = 0$ and the flow rate formula $Q = \int_0^H v_x \, dy$, one can obtain from Eq. (2.1) the following expressions for the distribution of the flow velocity and the pressure gradient:

$$\begin{aligned} v_x &= v_0 \left(1 - \frac{y}{H}\right) + \frac{1}{2\mu} \frac{dp}{dx} y(y - H), \\ \frac{dp}{dx} &= \frac{6\mu v_0 (1 - 2q)}{H^2}, \end{aligned} \tag{2.2}$$

where v_0 is the workpiece motion velocity; H is the nozzle gap opening; $q = Q/v_0 H$ is the relative flow rate and Q denotes the volumetric flow rate per unit length of the workpiece contour.

The pressure gradient is determined by the workpiece velocity, gap opening and the relative lubricant flow rate. When $q = 0.5$, then the pressure gradient is equal to zero and shearing flow takes place.

Dynamic viscosity coefficient μ significantly depends on the temperature and pressure in the lubricant film. For hydrocarbon oils, joint pressure and temperature influence on the dynamic viscosity is expressed by the following relation:

$$\mu = \mu_{50} \left(\frac{50}{t}\right)^m \exp(\alpha p), \tag{2.3}$$

where μ – dynamic viscosity coefficient at 50 C°, m – a coefficient depending on the kinematic viscosity of oil, t, p – temperature and pressure in the lubricant film, α – piezo-viscosity coefficient.

For the lubricants obeying the law (2.3) which describes the variation of viscosity due to the temperature and pressure changes, differential equality (2.2) yields the following expression for the force feed nozzle pumping effect:

$$p_1 = -\frac{1}{\alpha} \ln \left[1 - \frac{6\alpha\mu_{50}\left(\frac{50}{t}\right)^m v_0 L(1-2q)}{H^2} \right], \tag{2.4}$$

where L denotes the nozzle length.

Using relation (2.4) one takes into account the lubricant heating; to this end, appropriate heat balance equation has to be solved, all terms of such equation are quoted in the paper [2].

Force feed nozzle pumping effect is determined by rheological properties of the lubricant and the drawing process conditions, but mainly by the nozzle geometry. Lubricant pressure in the nozzle rises with the lubricant viscosity, with the drawing velocity and nozzle length as well as with decreasing gap opening.

Stress distribution in the drawn workpiece, total drawing force and the lubricant pressure, necessary for the plastic deformation, depend on the tensile counter-force arising in the nozzle. The stress caused by this force is given by the following formula:

$$\sigma_0 = \frac{4Hp_1(2-3q)}{3d_0(1-2q)},$$

where d_0 is the diameter of the wire. If emulsions are used, their viscosity does not depend on the pressure and the nozzle pumping effect is given by the following formula:

$$p_1 = \frac{6\mu_0 v_0 L(1-2q)}{H^2}.$$

The pressure attained in the nozzles in the case of water-oil emulsions is not very high. E.g., for 25 % emulsion if $v_0 = 50\text{m/s}$, $q = 0$, $L = 100$ mm and $H = 0.02$ mm, then the calculated pressure value equals 34 MPa. This is not enough to attain the film lubrication regime, however even such a pressure is able to improve the friction condition and reduce the working tool wear during drawing of low strength metals and alloys.

Application of force feed devices for liquid lubricants is impeded by the difficulties in manufacturing the force feed nozzles, especially in the case of high strength alloys, for which the necessary length of the nozzle may be considerable. Nozzle length can be reduced by the optimal choice of its parameters particularly of the gap opening H [8]. Optimal nozzle gap opening is equal to $H_{\text{opt}} = 1.5H_0$ (H_0 is the thickness of the lubricant film at the deformation zone entry). The optimal gap choice corresponds to the relative flow rate $q = 1/3$, and the maximal pressure gradient equals

$$\left(\frac{dp}{dx}\right)_{\max} = \frac{2\mu v_0}{H_{\text{opt}}^2}.$$

Pumping effect of the optimally designed force feed nozzle, in the case of variable viscosity lubricant, is determined by the following equality:

$$p_1 = -\frac{1}{\alpha}\ln\left(1 - \frac{2\alpha\mu_0 v_0 L_0}{H_{\text{opt}}^2}\right). \tag{2.5}$$

From the Eq. (2.5) one determines the optimal nozzle length assuring the pressure p_1:

$$L_0 = \frac{H_{\text{opt}}^2\left[1 - \exp(\alpha p_1)\right]}{2\alpha\mu_0 v_0}.$$

The stress due to the tensile counter-force under optimal nozzle design parameters, is expressed by the formula

$$\sigma_0 = -\frac{4H_{\text{opt}}}{\alpha d_0}\ln\left(1 - \frac{2\alpha\mu_0 v_0 L_0}{H_{\text{opt}}^2}\right).$$

For the emulsions, optimal length of the nozzle can be expressed as follows:

$$L_0 = \frac{H_{\text{opt}}^2 p_1}{2\mu_0 v_0},$$

and for the tensile counter-force stress we have the expression

$$\sigma_0 = \frac{8\mu_0 v_0 L_0}{H_{\text{opt}} d_0}.$$

In the case of high viscosity lubricants their slip at tool and workpiece surfaces is possible. The lubricant slip along the surface begins when tangential stress in the film layer, adjacent to the nozzle and/or workpiece, reaches some critical shear stress level. The slip velocity is then described as follows

$$v_{\text{sl}} = \begin{cases} \dfrac{K_{\text{sl}}\tau_R}{H} & \text{when} \quad \tau_R \geq \tau_S, \\ 0 & \text{when} \quad \tau_R < \tau_S, \end{cases}$$

where τ_R is the tangential stress in the lubricant layer adjacent to the surface, τ_S is the critical shear stress, $K_{\rm sl}$ denotes a slip coefficient.

It has been shown in the papers [9, 10] that the slip along the walls can considerably decrease the lubricant pressure attained due to the action of the force feed nozzle. It has been assumed, that the law controlling the lubricant slip velocity as a function of the shear stress is the same for the tool and workpiece surfaces. In fact, due to the differences in the states of contact surfaces and different adhesive interaction with the workpiece and the tool, the lubricant slip coefficients may differ.

Using the following boundary conditions modeling the lubricant slip at the walls:

$$v_x(0) = v_0 + \frac{K_1\tau(0)}{H}, \qquad v_x(H) = -\frac{K_2\tau(H)}{H}$$

one can derive from Eq. (2.1) the following relation for the velocity distribution across the lubricant layer and for the lubricant pressure gradient in the channel:

$$\begin{aligned} v_x &= \frac{1}{2\mu}\frac{dp}{dx}\left[y^2 - \frac{(Hy+\mu K_1)(H^2+2\mu K_2)}{H^2+\mu(K_1+K_2)}\right] \\ &\qquad + v_0\left[1 - \frac{Hy+\mu K_1}{H^2+\mu(K_1+K_2)}\right], \end{aligned} \tag{2.6}$$

$$\frac{dp}{dx} = \frac{6\mu v_0}{H^2}\cdot\frac{1+2k_2-2q(1+k_1+k_2)}{1+4k_1+4k_2+12k_1k_2}.$$

where

$$k_1 = \frac{\mu K_1}{H^2}, \qquad k_2 = \frac{\mu K_2}{H^2}.$$

Equality $(2.6)_2$ for $k_1 = k_2 = k$ reduces to the relation:

$$\frac{dp}{dx} = \frac{6\mu v_0(1-2q)}{H^2(1+6k)},$$

(c.f. [10]); if, moreover, $k_1 = k_2 = 0$, then we get Eq. (2.2).

The calculated values of the ratio $\operatorname{grad} p/\operatorname{grad} p_0$ are plotted in Fig.3 against coefficients k_1 and k_2 for different rates q; $\operatorname{grad} p$ and $\operatorname{grad} p_0$ denoting, respectively, the pressure gradient in the nozzle for the cases of boundary slip and for perfect sticking at the wall. One can see that the slip at the wall, particularly at the wire surface, results in an essential drop of the pumping effect.

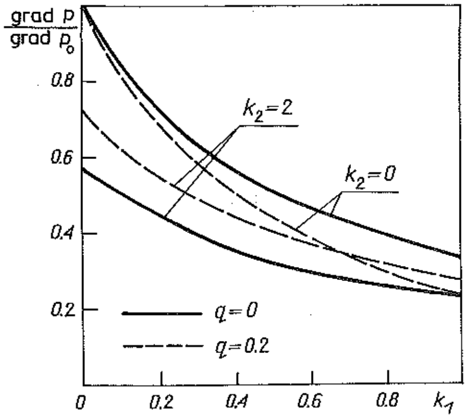

FIG. 3. k_1 and k_2 quantities impact on the force feed nozzle pumping effect.

It follows from the Eq. (2.6) that increasing flow rate q reduces the nozzle pumping effect. For $dp/dx = 0$ we get q value corresponding to the transition to the shear flow

$$q = \frac{1+2k_2}{2(1+k_1+k_2)} \,. \tag{2.7}$$

From Eq. (2.7) in the case of $k_1 = k_2$ we obtain $q = 0.5$, which corresponds to the lubricant flow rate for perfect sticking condition ($k_1 = k_2 = 0$).

The stress originated by the tensile counter-force arising due to the lubricant, in the case of boundary slip, should be calculated according to the following formula:

$$\sigma_0 = \frac{4Hp_1}{3d_0} \cdot \frac{2(1+3k_2) - 3q(1+2k_2)}{1+2k_2-2q(1+k_1+k_2)} \,.$$

Discussion of Eq. (2.6) has shown that, for a given lubricant volumetric flow rate, there exist the optimal gap opening assuring the maximal pumping effect of the drawing device. Demanding the derivative of the pressure gradient with respect to gap opening to vanish, we obtain the following condition for the optimal gap opening $H_{\rm opt}$:

$$\begin{aligned}(2h_{\rm opt} - 3)(h_{\rm opt}^4 + 4b_2h_{\rm opt}^2 + 8b_2^2 - 4b_1b_2)h_{\rm opt}^2 \\ -3\left[h_{\rm opt}^4(3b_1 - b_2) + 4h_{\rm opt}^2(b_1^2 + 2b_1b_2 - b_2^2) + 4b_1b_2(b_1+b_2)\right] = 0,\end{aligned} \tag{2.8}$$

where

$$h_{\rm opt} = H_{\rm opt}/H_0, \qquad b_1 = \mu K_1/H_0^2, \qquad b_2 = \mu K_2/H_0^2.$$

h_{opt} variation versus b_1 and b_2 is shown in Fig.4. One can see, that h_{opt} rises with b_1 increase and with b_2 decrease, the effect of b_1 being better pronounced.

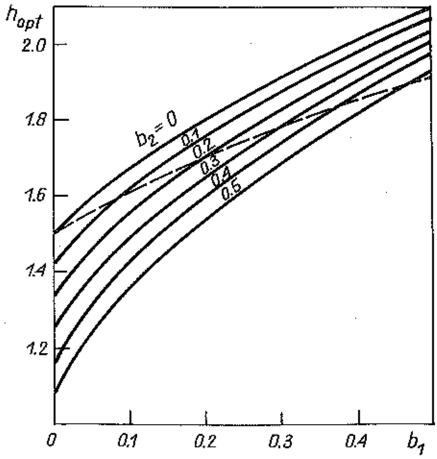

FIG. 4. Nomograph for determination of h_{opt} value.

From equation (2.8) for the case $b_1 = b_2 = b$ one can obtain the following equation [11]:

$$2h_{\rm opt}^3 - 3h_{\rm opt}^2 - 6b = 0.$$

Solution of this equation is depicted in the Fig.4 (dashed line).

In the case of optimal gap opening, expressions for the lubricant pressure gradient and the stress induced by the counter-force (due to the lubricant) take the following form:

$$\left(\frac{dp}{dx}\right)_{\max} = \frac{6\mu v_0}{H_0^2} \cdot \frac{h_{\rm opt}^2 + 2b_2 - 2q(h_{\rm opt}^2 + b_1 + b_2)}{h_{\rm opt}^4 + 4b_1 h_{\rm opt}^2 + 4b_2 h_{\rm opt}^2 + 12 b_1 b_2},$$

$$\sigma_0 = \frac{4H_{\rm opt} p_1}{3d_0} \cdot \frac{2(h_{\rm opt}^2 + 3b_2) - 3q(h_{\rm opt}^2 + 2b_2)}{h_{\rm opt}^2 + 2b_2 - 2q(h_{\rm opt}^2 + b_1 + b_2)}.$$

The relations quoted above enable the calculations of the optimal parameters of the force feed nozzle.

At present, force feed nozzles find only limited application. Application of force feed sockets has been discussed in the paper [12].

In exploitation of drawing devices important role plays the pumping effect of the working die (Fig.5). Liquid lubricant flow in the working die channel

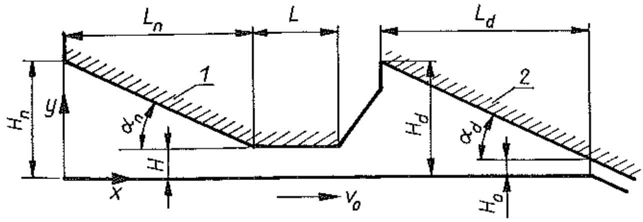

FIG. 5. Calculation scheme for the compund drawing device design parameters: 1 – force feed die, 2 – working die.

is similar to the flow in the case of the variable nozzle gap opening $H = H_d - x\tan\alpha_d$, and the relative flow rate of the lubricant equal to $q = Q/(H_d - x\tan\alpha_d)$. At the entry to the deformation zone, the maximal pressure is attained and the volumetric flow rate is equal to $Q = v_0 H_0/2$. Gap opening changes from H_d to H_0, besides $q_{\min} = H_0/2H_d$ and $q_{\max} = 0.5$. For variable gap opening the pressure gradient is described by the formula

$$\frac{dp}{dx} = \frac{6v_0\mu_0 \exp(\alpha p)\left[(H_d - x\tan\alpha_d) - H_0\right]}{(H_d - x\tan\alpha_d)^3}\,. \tag{2.9}$$

Integrating first Eq. (2.9) and then taking into account the condition $p(0) = p_1$, one can determine the lubricant pressure distribution in working cone of the die. At the deformation zone entry, for $H_d \gg H_0$ the pressure is equal to

$$p = -\frac{1}{\alpha}\ln\left[\exp(-\alpha p_1) - \frac{3\alpha\mu_0 v_0}{H_0\tan\alpha_d}\right].$$

The film lubrication friction regime is ensured when the pressure $p = p_2$ is attained. It occurs under the condition of sufficiently thick separating layer between the surfaces of the working die and the workpiece. The critical pressure p_2 is determined from the yield condition

$$p_2 = \sigma_S - \sigma_0\,,$$

where σ_S denotes the plastic deformation strength of the material.

Taking into account the condition $p = p_2$, thickness of the lubricant layer at the inlet to the deformation zone is determined,

$$H_0 = \frac{3\alpha\mu_0 v_0 \exp(\alpha p_1)}{\{1 - \exp\left[-\alpha(\sigma_S - \sigma_0)\right]\}\tan\alpha_d}\,.$$

Knowing the value of H_0, one can determine the friction regime of the drawing process. In the case of absence of the force feed nozzle or die, i.e. for single die drawing, we have $p_1 = 0$.

Taking into account the yield condition we obtain for $p_1 = 0$ the thickness of the lubricant layer at the entry to the deformation zone characterizing the pumping effect of the working die alone

$$H_0 = \frac{3\alpha\mu_0 v_0}{\{1 - \exp[-\alpha(\sigma_S - \sigma_0)]\}\tan\alpha_d}. \tag{2.10}$$

It follows from Eq. (2.10) that the hydrodynamic lubricant wedge effect of the working die can be promoted by higher drawing velocity, application of highly viscous lubricants and the reduction of the tool cone opening angle. Results of calculations of H_0 indicate that, for high strength materials, it may be assumed that $\exp[-\alpha(\sigma_S - \sigma_0)] = 0$. Pumping effect of the working die drastically decline for wide opening angles, when $\alpha > 6°$. Value of H_0 increases with the lubricant viscosity; for the drawing conditions characteristic for the contemporary production techniques the use of the lubricants with viscosity higher than 1 Pa s ensures a friction regime fairly close to the film lubrication regime. The presence of the thick lubricant film is typical, for example, for drawing of the aluminum wire with the use of heavy cylinder oils at the early stages of the multiple drawing process when the lubricant viscosity is still high enough. It is also known that employing of the dies with reduced working cone opening angle results in the greater wear resistance due to action of the hydrodynamic effect of the lubricant.

Choosing the value of H_0 one should depart from the prerequisite, that the lubricant film thickness should exceed the maximal height of the workpiece surface micro-roughness (Rz according to ST SEW 638-77 standard), i.e. $H_0 \geq Rz$. The roughness parameter Rz characterizes the workpiece surface quality and depends on the preliminary machining process and on the properties of the material. One has to bear in mind that for cold drawn steel rod of circular cross-section $Rz = 1.6 \div 10\ \mu m$, for a similar brass rod – $Rz = 0.8 \div 6.3\ \mu m$. In the most cases value $H_0 = 10\ \mu m$ can be considered to be sufficient for attaining the film lubrication regime. In the engineering calculations of the device parameters for definite real materials, the preliminary estimation of surface quality is performed and appropriate value of separation H_0 of the tool and workpiece surfaces is more precisely determined.

In the course of preparing the film lubrication drawing techniques, two kinds of calculations can be performed. The first consists in determination of the thickness of the lubricant film at the entry to deformation zone H_0; this value serves for the estimation of the tool and workpiece surfaces separation possibility. For these calculations both the pumping effects: of the force feed nozzle and of the working die should be taken into account. The second choice is to start from the surfaces separation condition and to calculate, according to the known value of H_0, the working die pumping effect and then to determine the parameters of the force feed nozzle assuring necessary increase of the lubricant pressure.

Compound dies are usually employed in the case of liquid lubricants, for example for production of aluminum alloys conducting wire in the electric cable manufacturing industry. Calculations of compound die parameters are also based on the Newtonian liquid flow laws. Device channel geometry (Fig.4) is determined by dies configuration and the diameter reduction ratio per one draft. The necessary lubricant pressure is attained due to pumping action of the working cone and the calibrating part of the force feed die as well as due to the action of the free part of the cone. For each of these zones the lubricant flow is analyzed and the pumping effect is determined. Thus for the working die, the pressure gradient is equal to

$$\frac{dp}{dx} = \frac{6\mu\left[v_0(H_d - x\tan\alpha_d) - 2Q\right]}{(H_d - x\tan\alpha_d)^3}.$$

From the condition of the necessary pressure at the deformation zone entry $p = p_2$, the pressure which has to be assured by the force feed die is determined,

$$p_1 = -\frac{1}{\alpha}\ln\left[\exp(-\alpha p_2) + 3\beta_1\right], \tag{2.11}$$

where

$$\beta_1 = \frac{\alpha\mu_0 v_0 L_d}{H_0 H_d}$$

is the dimensionless parameter comprising lubricant rheology parameters (α, μ_0), working die geometry (L_d, H_d), inlet velocity (v_0) and necessary lubricant film thickness at the the entry to the deformation zone (H_d).

The lubricant pressure necessary for the plastic deformation, is determined in accordance with the known plastic properties of the drawn metal σ_S and the counter-force stress σ_0.

Starting from the value of p_1, calculated according to Eq. (2.11), the force feed die parameters are determined. Pressure p_1 is secured due to the pumping effect of the working cone and of the calibrating part of the force feed die. The formula for the working cone pressure gradient has the form

$$\frac{dp}{dx} = \frac{6\mu\left[v_0(H_n - x\tan\alpha_n) - 2Q\right]}{(H_n - x\tan\alpha_n)^3}, \tag{2.12}$$

and for the calibrating part – the

$$\frac{dp}{dx} = \frac{6\mu_0 v_0 \exp(\alpha p)(1-2q)}{H^2}\,. \tag{2.13}$$

Solving piece-wise the differential equations for pressure gradients (2.12) and (2.13) and equating the pressures at the interval boundaries, we determine the lubricant pressure at the outlet of the calibrating part of the force feed die,

$$p_1 = -\frac{1}{\alpha}\ln\left\{1 - 6\beta_2 h_n\left[l_n(1-q) + h_n(1-2q)\right]\right\}, \tag{2.14}$$

where

$$h_n = H_n/H, \qquad q = Q/v_0 H, \qquad l_n = L_n/H_0, \qquad \beta_2 = \alpha\mu_0 v_0 L/H_n^2.$$

Relations (2.11) and (2.14) make it possible to calculate the gap opening in the cylindrical part of the force feed die, starting from the condition of the necessary pressure p_1.

3. Calculation of compound die pameters for powdered lubricant drawing

We shall consider the drawing device employing the powdered soap lubricant, governed by the following rheological flow law of the Bingham viscous-plastic medium:

$$\mathbb{T} = A + B\mathbb{H}\ , \tag{3.1}$$

where $\mathbb{T}$ an $\mathbb{H}$ are intensities of the tangential stress and of the shear rate, respectively; A is the limiting shear stress and B denotes the plastic viscosity. The necessary calculations include the determination of the dimensions of the nozzle entry cone and the cylindrical part parameters, as well as the lubricant flow calculations at the free part of the working die cone.

Dimensions of the force feed nozzle entry cone (see Fig.5) are determined from the condition of preliminary compression of the powdered mixture preliminary compression up to 15 MPa; this allows to consider the lubricant in the cylindrical part as incompressible, and its rheological properties as pressure-independent.

Necessary dimensions of the entry cone are determined according to the following formula:

$$\frac{H_n}{H} = \exp\left(91.3\,\tan(\alpha/K)\right),$$

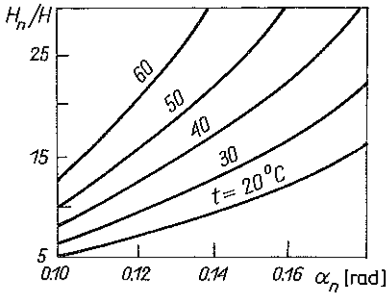

FIG. 6. Determination of the dimensions of the nozzle entry cone.

where K is a coefficient describing the temperature effect on the rheological properties of the soap lubricant.

The calculated values of the H_n/H ratio assuring the necessary compression of the powdered lubricant are depicted for different temperatures in Fig.6. Lubricant temperature is determined by the temperature of the wire at the compound die entry. It should be mentioned that the dimensions of the die working cone are sufficient for the preliminary compression of the lubricant. Therefore, in calculating the pumping effect of the force feed die it is sufficient to confine the calculations to the determination of the pressure rise due to the action of the calibrating part of the force feed die.

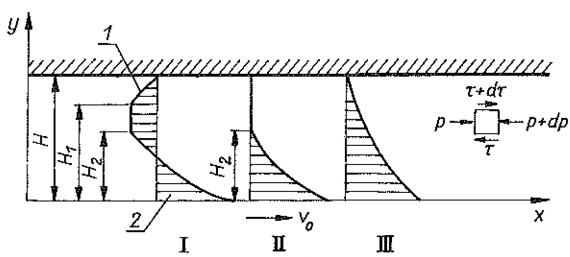

FIG. 7. Viscous-plastic lubricant flow calculation schemes: *1* and *2* – shearing flow zones, *3* – rigid undeformable layer.

The compressed lubricant proceeds into the uniform opening gap H. Three different viscous-plastic lubricant flow schemes can be realized in the nozzle gap (Fig.7). Pressure gradients along the nozzle are constant for all of them, therefore the pressure at the end of the nozzle of length L is equal to

$$p_1 = L \frac{dp}{dx} , \tag{3.2}$$

where $$\frac{dp}{dx} = \begin{cases} \dfrac{2A}{(h_1 - h_2)H} & \text{for Scheme I,} \\ \dfrac{2Bv_0}{H_2^2} & \text{for Scheme II,} \\ \dfrac{6Bv_0(1-2q)}{H^2} & \text{for Scheme III,} \end{cases}$$

and $h_1 = H_1/H$, $h_2 = H_2/H$, H_1 and H_2 are the parameters determining the position of the rigid lubricant layer.

For the viscous-plastic lubricant flow the presence of the rigid undeformable layer is typical. For the Scheme I the rigid layer is bounded by H_1 and H_2 $(0 < H_1 < H < H_2)$. For the pressure gradients calculations, in the case of Scheme I, quantities h_1 and h_2 are determined according to Fig.1 using the known value of $\sigma = Bv_0/AH$ parameter value and the preliminary assumed value of relative flow rate $q = H_0/2H$. For the Scheme II the rigid layer adjoins to the nozzle surface and is bounded by the value H_2 which is determined by the flow laws and equals $H_2 = 3Hq$. For the last Scheme III, viscous flow over the whole layer width takes place, similarly to the case of liquid lubricant (plastic viscosity B being involved in this case, however).

Selection of the flow scheme and appropriate choice of expression for the pressure gradient are to be made depending both on the relative lubricant flow rate q and on the dimensionless parameter σ. Knowing the assumed value of the flow rate q and the value of σ, one can determine the appropriate flow scheme (Fig.8).

For the viscous-plastic lubricant, there exist also optimal nozzle parameters. The optimal parameters correspond to Scheme III, for which absence of the rigid layer is typical. Optimal gap opening is equal to $H_{\mathrm{opt}} = 1.5\ H_0$, the nozzle length is then equal to $L_0 = H_{\mathrm{opt}}^2 p_1/2Bv_0$.

Counter-force induced tensile stress, due to the action of viscous-plastic lubricant in the nozzle, is determined by the sum of the tangential stresses over the surface of the drawn wire.

$$\sigma_0 = \begin{cases} \dfrac{4AL(h_1 + h_2)}{d_0(h_1\, h_2)} & \text{for Scheme I,} \\ \dfrac{4AL}{d_0}\left(1 + \dfrac{2\sigma}{h_2}\right) & \text{for Scheme II,} \\ \dfrac{4AL_0}{d_0}\left(1 + \dfrac{H_{\mathrm{opt}}^2 p_1}{AL_0}\right) & \text{for Scheme III.} \end{cases}$$

It should be mentioned here, that Scheme I occurs most frequently in the practice of wire drawing. For this scheme such flow regimes take place, for

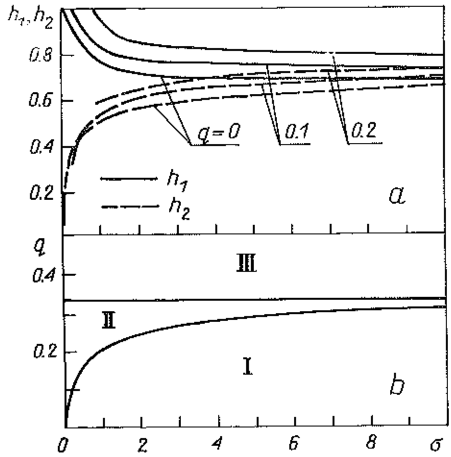

FIG. 8. Calculated values of h_1 and h_2: a) – for the Scheme I, b) – in the viscous-plastic lubricant flow schemes validity range.

which the lubricant flow rate attains several percent only of the flow rate corresponding to the shear flow. Besides, between the inlet and outlet of the lubricant channel, strong lubricant circulation associated intense heat release due to viscous dissipation takes place.

Temperature effect on the constants entering Eq. (3.1) is described by the following expressions: $A = A_0 \exp(-\vartheta)$; $B = B_0 \exp(-\vartheta)$, where $\vartheta = b(t - t_0)$, t_0 is the wire temperature at the nozzle entry. The mean temperature t of the lubricant should be determined from the following equality [13]:

$$3\sigma\left[(D + Mq)\vartheta - DN\right](h_1 - h_2)^2 \exp(\vartheta) \\ -(4 - 3h_2 - h_1)(1 - h_1)^2 + h_2^2(3h_1 + h_2) = 0,$$

where

$$D = \gamma / b v_0 A_0, \qquad M = \rho c H / b L A_0, \qquad N = b(t_c - t_0),$$

γ – heat exchange coefficient, b – viscosity temperature dependence coefficient, ρ and c – lubricant density and specific heat, t_c – ambient temperature.

The calculated values of the dimensionless quantity ϑ versus parameter σ, plotted for various values of q, D and M are shown in Fig.9. One can see that increasing q, D and M reduce the value of ϑ, while increasing parameter σ results in growth of ϑ.

In Fig.10 the calculated values of dimensionless lubricant pressure ($\Pi = p_1 H / A_0 L$) at the nozzle end are shown for the cases of isothermal and non-isothermal flow, in the course of film lubrication regime drawing. It can be

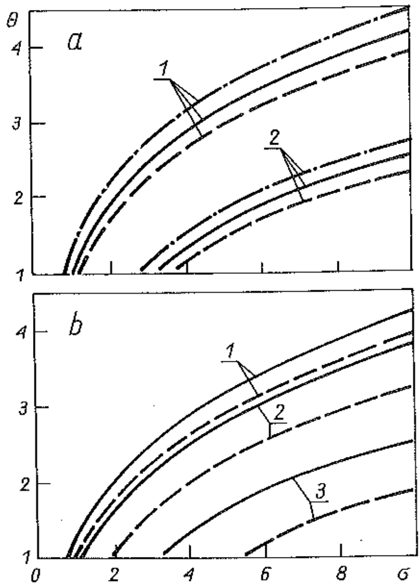

FIG. 9. Variation of the dimensionless quantity ϑ versus parameter σ for different D and M and $N = 0$:
a – $M = 1$, 1 – $D = 1$, 2 – $D = 10$,
b – $D = 1$, 1 – $M = 1$, 2 – $M = 10$, 3 – $M = 100$,
– · – · – $q = 0$, ——— $q = 0.1$, — — — $q = 0.2$.

seen that, in the case of non-isothermal lubricant flow, the pressure produced by to the force feed nozzle is lower than that in the isothermal case, this pressure difference rising with increasing σ and decreasing q. It should be mentioned here that the presence of pressure peak is typical for the non-isothermal flow.

In calculating the device parameters one has to take into account the pumping effect of the free part of the working nozzle.

In the course of the viscous-plastic lubricant flow along the working cone of the die, the velocity distribution profile across the channel changes, these changes yield the transition from Scheme I to Schemes II and III. Pressure growth in the working die is determined by the sum of the pressures arising in the working cone intervals

$$p_3 = \frac{A}{3\tan\alpha_d}\left[6\left(1 + \frac{\ln a}{h_1 - h_2}\right) + \beta_0\right],$$

where

$$a = \beta_0\left((1 + l_d \tan\alpha_d)/2.25 + 1.5\right), \qquad \beta = Bv_0/AH_0; \qquad l_d = L_d/H_0.$$

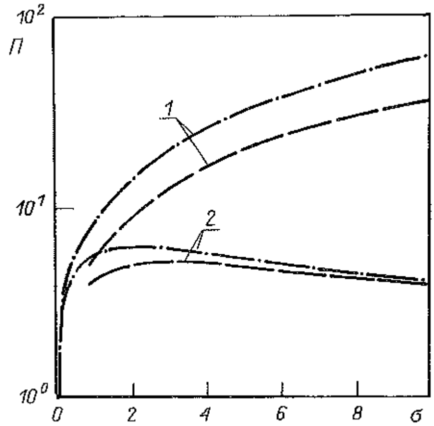

FIG. 10. Calculated dimensionless pressure Π versus σ and q, for $K = 10$, $M = 1$ and $N = 0$:

· — · — · — $q = 0$, — — — $q = 0.1$
1 – isothermal flow, *2* – non-isothermal flow.

The difference $h_1 - h_2$ is averaged over the length of the working cone and is determined using the mean values of σ and q for Scheme I:

$$\sigma = \frac{2\beta_0^2}{\beta_0(2.5 + l_d \tan \alpha_d) + 2.25},$$

$$q = \frac{\sqrt{\sigma^2 + 4\sigma} - \sigma}{12}.$$

The total pressure generated by the compound drawing device is equal to the sum of the pressures arising due to the working die and the action of the force feed nozzle.

Knowing the value of the lubricant pressure p_2 at the deformation zone entry and the pressure p_3 generated by the working die, one can find the pressure $p_1 = p_2 - p_3$. Then the force feed nozzle parameters, appropriate for maintaining the film lubrication friction regime, can be found with the use of relation (3.2).

The presented set of calculation rules enables the determination of the compound drawing device parameters for the film lubrication regime drawing for both liquid and powdered soap lubricants, assuring high effectiveness of the process.

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