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The theoretical analysis and optimization of the cutting knife-grille pair parameters in the screws

V.V. Pelenko, E.I. Verboloz* and A.V. Baranenko

St. Petersburg National Research University of Information Technologies, Mechanics and Optics, Institute of Refrigeration and Biotechnology, Department of Food Engineering and Automation, 9 Lomonosov St., 191002 St. Petersburg, ITMO University Russia; *Correspondence: elenaverboloz@mail.ru

Abstract. We show how energy-force knife-grille pair parameters depend on their tightening torque which also indicate the node which is the most dynamically and thermally tensed. The research demonstrates that the temperature, at the junction of the knife-grille, varies in the 10°C, and therefore this is a significant factor in the rate increasing of the grids and knives deterioration. From the condition of the screw grille and the knife blade compatibility deformations, we are shown the analytical dependence between the structural and technological characteristics, which allows us to minimize the depreciation value of the grille and the knife, as well as to reduce the energy intensity of the grinding process.

Key words: lattice, the cutter knife, friction, bending, wear, temperature, pressure, distortion, efficiency, performance, etc.

INTRODUCTION

The mathematical model of the screw (Aret et al. 2012; Pelenko et al., 2014c; 2014d) and the results of experiments of the dependence of the energy-force knife-grille pair parameters on their tightening torque indicate that the specified node is the most dynamically and thermally tensed.

The experience in the industrial exploitation of the screws confirms this fact, as the frequent resharpening of the cutting knives and changing the sets of the screws are the common weak point of grinding-cutting equipment (Gorbatov et al., 1977; Chizhikova, 1978; Andrianov, 1982; Pelenko et al., 2006; Pelenko et al., 2008a; 2008b; 2008c; Voronenko et al., 2009).

The research demonstrates that the temperature, at the junction of the knife-grille, varies in the 10 °C, and therefore this is a significant factor in the rate increasing of the grids and knives deterioration. Moreover, uncontrolled increase in torque and a knife blade lattice nut shell lead to the increasing of the blade node parts mutual friction moment and therefore it increases the power consumption by 20–25%. (Pelenko et al., 2014a; 2014b).

Advanced analysis of the essence of physical processes which occur in the operation of the screw type meat grinders shows that the exerted pressure of the raw meat on the bars and a knife reaches $(3 \div 7) \cdot 10^5$ Pa and over. Therefore, based on the terms of the way it is constructed and due to the under the influence of the pressure, the grille is bent outwardly with the screw body and the knife being flexed bulge inward. Thus, the knife, on the peripheral annular surface, while rotating, creates a significant stress concentration at the junction of the knife-grille, which leads to accelerated deterioration of both the knife and the grille as well as a rapid and premature blunting of the knife.

The foregoing leads to the necessity of the correct mathematical description of the interaction process between the knife and the knife grille of the screw, in order to optimize structural and technological parameters of the cutting node.

MATERIALS AND METHODS

For the implementation of the formulated problem, the authors have reviewed the literature materials describing this problem, but no specific analytical solutions on how to fix these circumstances were found. Thus, our problem was solved using the theory of elasticity and strength of materials by making differential equations for the deflection of the knife and the lattice of their analytical solutions by taking into account the temperature of the bending moment, using the compatibility conditions of deformation and lattice cutter knife.

RESULTS

From the perspective view of the theory of elasticity, the knife screw grille is a thin circular perforated annular plate. In the most general case of the consideration, more than two dozen options of the fixing with clamping nut of the knife grille in the screw body should be subjected by the analysis (Vlasov, 1958; Smolentsev, 1963; Timoshenko & Voynovskiy-Krieger, 1963; Tymoshenko, 1965; Weinberg, D.V. & Weinberg, E.D., 1970; Savruk, 1980; Dozhzhel, 1982; Birge & Mavlyutov, 1986; Savruk & Timoshuk, 1987).

Fig. 1 shows 20 fixing options (hinged annular support, the ring rigid closure) and the loading round plate (uniform loading over the entire area, the annular uniform load without bending moment and with him). In addition, the plate can be approximately considered as a continuous disc, a disc with a central hole or as a perforated disc with a central hole.

Of the considered variants of the schemes of the grille fixing we put attention on the most advantageous, from a practical point of view, the case (Fig. 1., Scheme 14), when fixing the annular plate is carried on the inner edge of the ring by the tight clamp, and external influence is uniformly distributed as the load 'q' over the area of the perforated plate and thermal bending moment 'M_T'.

This choice is based on the analysis of the process of the mutual deformation of the grille and the knife during the screw operation. Indeed, in this case, the deformation of the plate is carried out with the bulge going inside the body of the screw, as well as the knife, so it is possible to ensure (with the given temperature of the grille deformation) the equal displacement values of the annular peripheral sections of the grille and the knife blade. It provides a uniform force of the knife and the grille interaction in the place of the junction, the exclusion of the stress concentration and the reducing of the deterioration rate of the grille and the blade, as well as the blunting knife. Besides, there is the decreasing of the energy losses to friction and the thermal load at the junction.

For the realization of the reasonable effective concept of the conditions of the knife and grille interaction we need to solve two problems.

Task 1: We need to define the deformation parameters of the selected option for the grille (Fig. 1, Scheme 14).

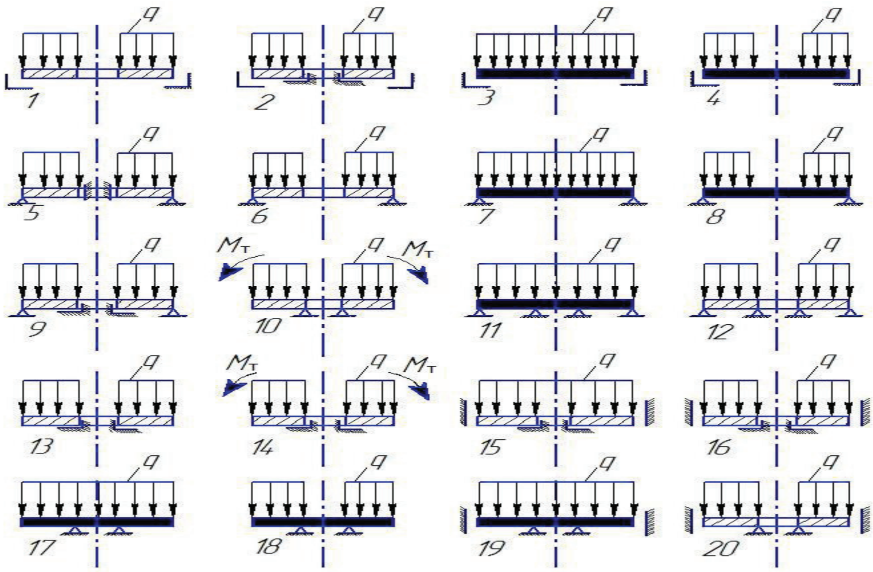

Figure 1. Loading scheme and fix lattice.

Solutions to the problem defined above are known, but the theory of the plates and shells was highly developed and is currently applicable (Vlasov, 1958; Smolentsev, 1963; Timoshenko & Voynovskiy-Krieger, 1963; Tymoshenko, 1965; Weinberg, D.V. & Weinberg, E.D. 1970; Savruk, 1980; Dozhzhel, 1982; Birger & Mavlyutov, 1986; Savruk & Timoshuk, 1987) and it has been widely used in practice, especially in high-tech industries, such as hydro, turbine construction, shipbuilding, aviation, astronautics. However, specific solutions to these problems are similar to those stated above, with mentioned fixing conditions, had not been found.

For the development of the desired mathematical model of the screw grille bending, we consider the calculation scheme that is shown in Fig. 2.

The proposed scheme shows a circular ring perforated plate (Fig. 2, pos. A-d) with a rigid closure on the inner edge of the ring (Fig. 2, pos. a, c, d) in the conditions of the transverse load q which is uniformly distributed over the grid area (Fig. 2, pos. a-D). And the bending moment M_t (Fig. 2, pos. a-d) which is caused by thermal deformation, linearly distributed over the plate thickness h (Fig. 2, pos. d) due to the frictional heating in the knife-grille junction.

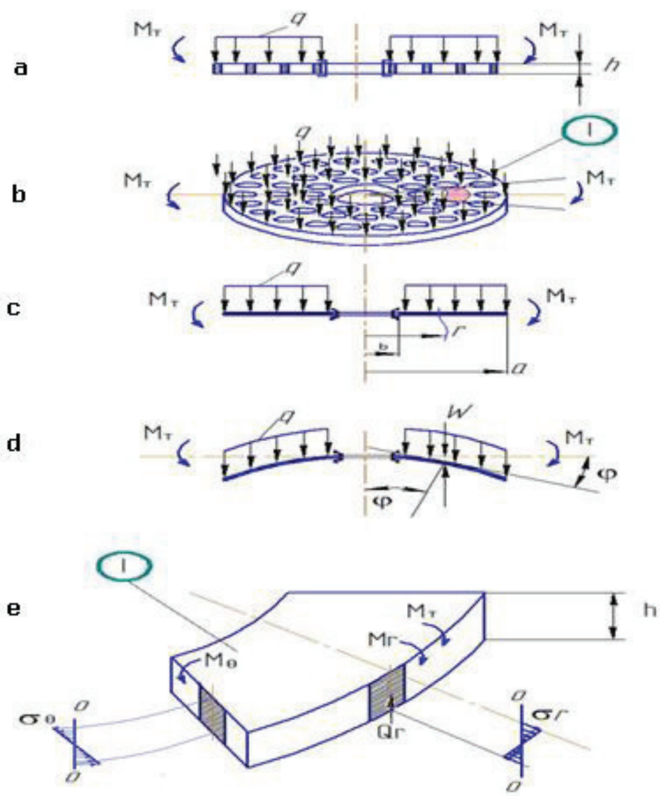

Figure 2. Scheme of loading and deformation of the lattice.

As it is known (Timoshenko & Voynovskiy-Krieger, 1963), the equilibrium equation for the plate element (Fig. 2, Pos. D) can be written as following:

$$\frac{d^2\varphi}{dr^2}+\frac{1}{r}\frac{d\varphi}{dr}-\frac{\varphi}{r^2}=-\frac{Q}{D} \tag{1}$$

or, considering the dependence of *φ=φ(w):*

$$\varphi=-\frac{dW}{dr}, \tag{2}$$

in another form:

$$\frac{d^3W}{dr^3}+\frac{1}{r}\frac{d^2W}{dr^2}-\frac{1}{r^2}\frac{dW}{dr}=\frac{Q}{D}, \tag{3}$$

where: r – the distance from the point under consideration to the center of the plate, m; φ – the rotation angle of the normal to the section under consideration with the coordinate ‘r’; Q – cutting force per length unit of the cylindrical section of the radius ‘r’, N m; D – cylindrical rigidity of the perforated plate, N m.

Equations (1) and (3) may be prescribed in a more compact and easy to integrate form:

$$\frac{d}{dr}\left[\frac{1}{r}\frac{d}{d^2}(ry)\right] = -\frac{Q}{D}, \tag{4}$$

or

$$\frac{d}{dr}\left[\frac{1}{r}\frac{d}{dr}\left(r\frac{dW}{dr}\right)\right] = \frac{Q}{D}, \tag{5}$$

From the equilibrium equation above, we can obtain the dependence of $Q = Q(q)$ for a plate with an external load 'q" which is distributed over the area. It should be mentioned that in the plate has a radius center hole 'b':

$$Q \cdot 2\pi r = \int_b^r q 2\pi r dr\,. \tag{6}$$

Under outward transverse load which is evenly distributed over the area of the plate $q = const$, from the equilibrium equation (6) we obtain:

$$Q = \frac{q}{2}(r - b)\,. \tag{7}$$

Now the equation of the deflection of the plate (5) takes the form:

$$\frac{d}{dr}\left[\frac{1}{r}\frac{d}{dr}\left(r\frac{dW}{dr}\right)\right] = \frac{q}{2D}(r - b)\,. \tag{8}$$

Integrating this equation for three times, we get:

$$W = \frac{qr^4}{64D} - \frac{qbr^3}{18D} + \frac{c_1 r^2}{4} + c_2 \ln r + c_3\,. \tag{9}$$

The integration constants c_1, c_2, c_3 found from the fixing plate condition.

1. At the inner annular edge of the round plate $(r = b)$ there is no movement $W = 0$; as well as the angle of the rotation of the section $\varphi = -\frac{dW}{dr} = 0$, so we can write for the displacement:

$$W|_{r=b} = -\frac{23qb^4}{576D} + \frac{c_1 b^2}{2} + c_2 \ln b + c_3 = 0\,. \tag{10}$$

Given that

$$\frac{dW}{dr} = \frac{qr^3}{16D} - \frac{qbr^2}{6D} + \frac{c_1 r}{2} + \frac{c_2}{r}, \tag{11}$$

for the rotation angle in the cross section $r = b$, we obtain:

$$\varphi|_{r=b} = -\frac{dW}{dr}|_{r=b} = \frac{-5qb^3}{48D} + \frac{c_1 b}{2} + \frac{c_2}{b} = 0. \tag{12}$$

2. On the outside, the plate annular round edge ($r=a$), that is free from the connections, only temperature bending moment M_T is applying, so we can write:

$$M_r|_{r=a} = M_T \tag{13}$$

where M_r – a radial bending moment intensity per length unit of the circular cylindrical section of the the annular plate, $H\ m\ m^{-1}$.

As it is known [13 ÷ 9] the radial bending moment is written in the form:

$$M_r = -D\left(\frac{d^2W}{dr^2} + \frac{v}{r}\frac{dW}{dr}\right), \tag{14}$$

where v – the Poisson's ratio for the material of the plate.

For the curvature of the plate according to (11) the following relation:

$$\frac{d^2W}{dr^2} = \frac{3qr^2}{16D} - \frac{qbr}{3D} + \frac{c_1}{2} - \frac{c_2}{r^2}. \tag{15}$$

Substituting (11) and (15) into the equation (14) yields:

$$M_r = -\frac{qr^2}{16}(3+\nu) + \frac{qbr}{6}(2+\nu) - c_1(1+\nu)\frac{D}{2} + c_2(1-\nu)\frac{D}{r^2}. \tag{16}$$

In view of (13) we can write:

$$M_T = -\frac{qa^2}{16}(3+\nu) + \frac{qba}{6}(2+\nu) - c_1\frac{D}{2}(1+\nu) + c_2\frac{D}{a^2}(1-\nu)\,, \tag{17}$$

Solving the system of three equations (10) and (12), (17) with three unknowns, we find the integration constants c_1, c_2, c_3.

$$c_1 = \frac{5qb^4(1-\nu) + 8qba^3(2+\nu) - 3qa^4(3+\nu) - 48M_T a^2}{24D[a^2(1+\nu) + b^2(1-\nu)]} = c_1(b, M_T) \tag{18}$$

$$c_2 = \frac{5qb^4(1+\nu) - 8qb^3a^3(2+\nu) + 3qb^2a^4(3+\nu) + 48M_T a^2 b^2}{48D[a^2(1+\nu) + b^2(1-\nu)]} = c_2(b, M_T) \tag{19}$$

$$c_3 = \frac{23qb^4}{576D} - \frac{5qb^4[b^2(1-\nu) + a^2(1+\nu)\ln b]}{48D[a^2(1+\nu) + b^2(1-\nu)]} - \frac{8qb^3a^3(2+\nu)(1-\ln b)}{48D[a^2(1+\nu) + b^2(1-\nu)]} +$$
$$+ \frac{3qb^2a^4(3+\nu)(1-\ln b)}{48D[a^2(1+\nu) + b^2(1-\nu)]} - \frac{M_T a^2 b^2 (1-\ln b)}{D[a^2(1+\nu)b^2(1-\nu)]} = c_3(b, M_T) \quad (20)$$

When structuring latter relation, after the numbering of each successive summand in c_3, we write:

$$c_3 = c_3(b, M_T) = c_{31}(b) - c_{32}(b) - c_{33}(b) + c_{34}(b) - c_{35}(b, M_T)\,,$$

where:

$$c_{31}(b) = \frac{23qb^4}{576D}\;;$$

$$c_{32}(b) = \frac{5qb^4[b^2(1-\nu) + a^2(1+\nu)\ln b]}{48D[a^2(1+\nu) + b^2(1-\nu)]}\;;$$

$$c_{33}(b) = \frac{qb^3a^3(2+\nu)(1-\ln b)}{6D[a^2(1+\nu) + b^2(1-\nu)]}\;;$$

$$c_{34}(b) = \frac{qb^2a^4(3+\nu)(1-\ln b)}{16D[a^2(1+\nu) + b^2(1-\nu)]}\;;$$

$$c_{35}(b, M_T) = \frac{M_T a^2 b^2 (1-\ln b)}{D[a^2(1+\nu)b^2(1-\nu)]}\;;$$

The necessity for such structuring is caused by the need to assess the accentuation effect on the individual values diameter deflection of the central hole of the grille as well as the temperature of the bending moment.

For the further functional dependence between the thickness of the knife blade $\delta_н$, and blade grille *h,* we introduce, for the constants of the integration, the modified expressions in accordance with the dependencies:

$$c_i = \frac{B_i}{D} \quad (21)$$

Inserting the obtained values of c_1, c_2, c_3 in relation (9), we obtain the general solution for the bending of the circular ring grille in the form of:

$$W = \frac{qr^4}{64D} - \frac{qbr^3}{18D} + c_1(b, M_T)\frac{r^2}{4} + c_2(b, M_T)\ln r + c_3(b, M_T)$$

or

$$W = \frac{1}{D}\left[\frac{qr^4}{64} - \frac{qbr^3}{18} + B_1(b, M_T)\frac{r^2}{4} + B_2(b, M_T)\ln r + B_3(b, M_T)\right] \quad (22)$$

Taking into account that we are interested in the maximum values of the deflections which are reached in the coordinates of the ring $r = a$, finally we obtain the desired quantity in the form of:

$$W_{max_p} = W|_{r=a} = \frac{qa^4}{64D} - \frac{qba^3}{18D} + c_1(b, M_T)\frac{a^2}{4} + c_2(b, M_T)\ln a + c_3(b, M_T) \quad (23)$$

The first feature of this solution is that when we are using the obtained ratio to estimate the deflection of the perforated plate, it is necessary to consider the influence of the significant amount of the holes in the output of the screw grille on the cylindrical rigidity value, which substantially reduce the value of the moment of the resistance to the bending.

Considering the macro deformation processes and taking into account the symmetry of the arrangement of the perforation holes, the additivity of the moment of inertia in the bending, as well as neglecting the edge effects of the local stresses concentration at the boundaries of the holes, the corrected value of the cylindrical rigidity can be written as follows:

$$D = \frac{Eh^3}{12(1-\nu^2)}\left(\frac{a - n_r d - b}{a}\right), \quad (24)$$

where: n_r – number of holes in the perforated section of the annular circular plate (grille); d – diameter of the holes of the screw output grille, m; E – Young's modulus of the material of the plate, Pa; h – thickness of the grille (the plate), m.

Taking into account the ratios (23) and (24), the equation (22) can be written in the form which is convenient for analysis:

$$W_{max_p} = \frac{12(1-\upsilon^2)a}{E(a - n_r d - b)h^3}\left[\frac{qa^4}{64} - \frac{qba^3}{18} + B_1\frac{a^2}{4} + B_2 \ln a + B_3\right]. \quad (25)$$

The second feature of the obtained mathematical model of the perforated circular ring grille deformation is a consideration of the effect of the temperature bending moment M_T on the plate deflection.

In general, the temperature dependence of the bending moment on the temperature difference at the outside and the inside of the screw output grille can be written as:

$$M_T = \frac{E}{1-\nu}\int_{-h/2}^{h/2} \alpha \Delta T_{(z)} Z dZ\,, \quad (26)$$

where: α – the coefficient of the thermal expansion of the plate material, K^{-1}.

Suggesting that the temperature distribution T in the plate thickness is linear, we can write:

$$\Delta T = \frac{T_B - T_H}{h} Z\,,$$

where: T_B, T_H – the temperature of the inner and the outer surfaces of the plate, К.

Substituting ΔT in (26) and integrating, we obtain the desired value of the temperature bending moment in the form of:

$$M_T = \frac{E\alpha(T_B - T_H)}{12(1-\nu)} h^2 \qquad (27)$$

DISCUSSION

In order to calculate the obtained mathematical model (23) of the annular circular perforated plate deflection, under a uniformly effect of the distributed over the area lateral load and temperature bending moment, which is applied on the outer ring contour, we use the following parameters:

$q = 3{\cdot}10^5$ Pa; $b = 5{\cdot}10^{-3}$ m; $a = 30{\cdot}10^{-3}$ m; $\nu = 0{,}3$; $E = 2{,}1{\cdot}10^{-11}$ Pa; $h = 4{\cdot}10^{-3}$ m; $d = 3{\cdot}10^{-3}$ m; $n_r = 3$; $T_v = 301$ К; $T_n = 296$ К; $\alpha = 10^{-5}$ К$^{-1}$.

According to the equation (27), we determine the value of the temperature bending moment as following:

$$M_T = \frac{2.1\cdot 10^{11}\cdot 5\cdot 10^{-3}}{12(1-0{,}3)} 9\cdot 10^{-6} = 11.25\ \mathrm{H}$$

The cylindrical rigidity of the plate is, according to the ratio (24), as follows:

$$D = \frac{2.1\cdot 10^{11}\cdot (4\cdot 10^{-3})^3\cdot (30\cdot 10^{-3} - 3\cdot 3\cdot 10^{-3} - 5\cdot 10^{-3})}{12(1-0{,}3^2)\cdot 30\cdot 10^{-3}} = 656.41\ \mathrm{H\cdot m}$$

For the convenience calculation of the deflection we find the characteristic value of the denominators in the c_1, c_2, c_3:

$$[a^2(1+\nu) + b^2(1-\nu)] = [(30\cdot 10^{-3})^2(1+0.3) + (5\cdot 10^{-3})^2(1-0.3)] = \\ = 11.7\cdot 10^{-4} + 0.175\cdot 10^{-4} = 11.875\cdot 10^{-4}$$

Reference values are the values of:
$\ln b = \ln(5{\cdot}10^{-3}) = -5.298$
$\ln a = \ln(30{\cdot}10^{-3}) = -3.507$

The ratios (18–21) give:

$C_1 = -0.1147$; $B_1 = -75.290$;
$C_2 = 1.4635 \cdot 10^{-6}$; $B_2 = 960.656 \cdot 10^{-6}$;
$C_{31} = 0.0114 \cdot 10^{-6}$; $C_{32} = -0.1549 \cdot 10^{-6}$; $C_{33} = 3.136 \cdot 10^{-6}$;
$C_{34} = 10.124 \cdot 10^{-6}$; $C_{35} = 2.045 \cdot 10^{-6}$;
$C_3 = 5.109 \cdot 10^{-6}$; $B_3 = 3353.6 \cdot 10^{-6}$.

The maximum deflection value of the plate we calculate as the following mention in Formula (23)

$$W_{max_p} = \frac{3 \cdot 10^{+5} \cdot 81 \cdot 10^{-8}}{64 \cdot 656.41} - \frac{3 \cdot 10^5 \cdot 5 \cdot 10^{-3} \cdot 27 \cdot 10^{-6}}{18 \cdot 656.41} +$$
$$+(-0.1147)\frac{9 \cdot 10^{-4}}{4} = +1.4635 \cdot 10^{-6} \cdot (-3.507) + 5.109 \cdot 10^{-6} =$$
$$= 5.784 \cdot 10^{-6} - 3.42 \cdot 10^{-6} - 25.8075 \cdot 10^{-6} -$$
$$-5.132 \cdot 10^{-6} + 5.109 \cdot 10^{-6} = 23.466 \cdot 10^{-6}(\text{m}) \cong 23.5 \text{ mkm}$$

Task 2. Determine the deflection of the knife blade.

The second problem is not so difficult and its solution can be broken down into (Bezukhov, 1968; Pisarenko et al., 1975; Sokolov, 1983) two components.

The first component of the movement f_1 is caused by the knife blade bending, which is under the action of the uniformly distributed over the surface load (pressure of raw meat):

$$f_1 = \frac{q \cdot S(a-b)^4}{8EJ}, \tag{28}$$

where: S – the blade width, *m*; J – The moment of inertia of the cross section of the blade in the bending, m^4;

The value of φ is defined by the well-known (Bezukhov, 1968) ratio:

$$J = \frac{S\delta_H^3}{12}, \tag{29}$$

Here $\delta_н$ – the thickness of the knife blade, m.

Then the expression (28), for the maximum deflection, can be written as following:

$$f_1 = \frac{3q(a-b)^4}{2E\delta_H^3}, \tag{30}$$

The second component of the movement is caused by the bending of the blade which is under the influence of the temperature bending moment.

The feature of the deformation of the knife blade is the omni directional movements owing to the load $f_1 > 0$ and the temperature bending moment: $f_2 < 0$.

Thus, we write for M_T (Birger & Mavlyutov, 1986):

$$f_2 = \int_b^a \int_b^a \frac{M_T}{EJ} drdr = \frac{-M_T}{2EJ}(a-b)^2 . \tag{31}$$

The total deflection of the blade takes the value:

$$f = f_1 + f_2 = \frac{qS(a-b)^4}{8EJ} - \frac{M_T S(a-b)^2}{2EJ}$$

or

$$f = \frac{(a-b)^2 S}{2EJ}\left[\frac{q}{4}(a-b)^2 - M_T\right]. \tag{32}$$

After the solution to the second problem (32), we can write the conditions of the compatibility of the grille and the knife blade deformation, providing the uniform internal stresses in their junction, and, thus, the minimum deterioration rate will be as following:

$$f = W_{max_p}. \tag{33}$$

$$\frac{(a-b)^2 S}{2EJ}\left[\frac{q}{4}(a-b)^2 - M_T\right] = W_{max_p}$$

Taking into account Formula (29), we obtain the ratio for the calculation of the required knife blade thickness:

$$\frac{3(a-b)^2}{2E\delta_H^3}[q(a-b)^2 - 4M_T] = W_{max_p} \tag{34}$$

$$\delta_H = \sqrt[3]{\frac{3(a-b)^2[q(a-b)^2 - 4M_T]}{2EW_{max_p}}} \tag{35}$$

Calculations for the same kind of materials of the grille and the knife give the value of:

$$\delta_H = \sqrt[3]{\frac{3 \cdot 625 \cdot 10^{-6}[3 \cdot 10^5 \cdot 625 \cdot 10^{-6} - 4 \cdot 11.25]}{2 \cdot 2.1 \cdot 10^{11} \cdot 23.5 \cdot 10^{-6}}} = \sqrt[3]{2707 \cdot 10^{-11}}$$
$$= \sqrt[3]{27.07} \cdot 10^{-3} = 3 \cdot 10^{-3} m = 3 \text{ mm}$$

Thus, to ensure the same deformations of the cutter grille and the knife blade, that are the equidistant elastic lines, which can ensure the minimal deterioration and the minimal knife blunt, should be used the mutual ratio (25), expressed in the terms of 'h' and (34) in the general case, to write functional iteration (relative M_T and B_i) dependence $\delta_н = \delta_н(h)$:

$$\delta_H = \frac{h}{2}\sqrt[3]{\frac{(a-b)^2[q(a-b)^2-4M_T](a-n_r d-b)}{a(1-\nu^2)\left[\frac{qa^4}{64}-\frac{qba^3}{18}+B_1\frac{a^2}{4}+B_2\ln a+B_3\right]}}\,. \tag{36}$$

In actual load conditions of the cutter grille and the knife of the screw, the values $\delta_н$ and h, as numerical calculations, are close.

Considering the special case when we can neglect the presence of the thermal bending moment ($M_T = 0$), the center hole ($b = 0$) and the grille perforation ($n_r = 0$), from the ratio (36) we obtain:

$$\delta_н = 2.06\ h.$$

CONCLUSION

The mathematical model of the process of the raw meat grinding in the screw (Pelenko & Kuzmin, 2009; Pelenko et al., 2012; Pelenko et al., 2013) allows us to estimate the level of the influence of each of more than 20 designs – technology and kinematic factors on the efficiency (productivity) of the meat grinder.

We were able to demonstrate that It was revealed that one of the factors affecting the quality of the grinding process is the tightening torque of the screw output grille, the value of which at this point is only calculated qualitatively. It results in the rapid blade blunting and depreciation of the knife – grille pair.

The methods of system analysis were able to provide the optimal design scheme of the screw output grille fixing (Scheme 14).

The mathematical model of the deflection of the perforated grille has been developed, that grill is loaded by the uniformly distributed mechanical stress across its surface and the thermal bending moment on the outer annular border. The equation of the cutting blade elastic line of the screw knife has been generated.

Based the condition of the screw grille and the knife blade compatibility deformations, we obtained the analytical dependence between the structural and technological characteristics, which allowed us to minimize the depreciation value of the grille and the knife, as well as to reduce the energy intensity of the grinding process.

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