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Effect of deformation on the stress-strain state of a honey extraction centrifuge flexible thread

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Abstract. Composite materials are becoming increasingly useful in engineering practice. They allow considering work aspects in the elastoplastic field. This is especially important for the systems that are modeled on geometric non-linearity. These systems include constructions which bearing capacity is provided by flexible threads, such as wires in bee honeycombs. An example of such a construction is the described centrifuge for honey or beebread mass extraction from a honeycomb (honey extractor), using frames for honeycombs with reinforced flexible wire. Comparative studies were conducted using previously tested computer programs based on the methods of linear and non-linear calculation of flexible wires. The deformation values of flexible wires were determined with consideration of the rotation frequency of the centrifuge rotor. The diagrams of changes in maximum expansion stresses, elongation of the wire, sagging of the flexible wire, the moment of inertia of the elastic body, equivalent loads for the methods of linear and non-linear calculation of flexible wires at different rotor rotation frequency were developed. When modeling objects with a rotating flexible wire, it is possible to use methods of both physically non-linear calculation and more simple and accessible linear calculation. Using the latter method, the calculation results should be divided by nonlinearity coefficient.

1. Introduction
Modern agricultural production is aimed at obtaining a variety of products. Products of beekeeping are very popular. Among beekeeping products are: honey, beebread, wax, propolis and others. Honey and beebread are used as ready-made products and are also used in food industry. [1] All the mentioned above products are raw materials for pharmacological [2] and chemical [3] industries. There are certain difficulties in honey and beebread extraction. Special devices are used: - honey extractor or combined units, which pump honey, perform scarification of beebread honeycombs and release waxy-beebread mass [4,5] from them.
The design of these devices uses centrifugal forces to extract honey or beebread from wax cells, and further extraction to the collection sites [4, 5]. At the same time, cells built up by bees on waxy mass, located on a wooden frame with internal span dimensions, are different in different constructions. Along the long side of the frame, reinforced wires with a certain pitch are stretched. In the process of cell rotation, frames together with the reinforced wires, bend in the direction of centrifugal forces. The wire is placed at some distance from the vertical axis of the rotor rotation. In addition, it is effected by a changeable side load.

Figure 1 shows a centrifuge for pumping honey and extracting waxy mass [5]. It has a working camera 1, inside which is placed rotor 2 with cassettes 3 for honeycombs and blades 4, to remove the extracted product through tray 5. Rotor 2 rotates from electric drive 6. When rotating rotor 2, the product, honey or waxy mass is released from the honeycombs installed in cassettes 3, which is removed by the blades 4 through unloading tray 5 beyond the working chamber 1.

On the one hand, increase in the rotation frequency of the frame accelerates the release of honey or removal of waxy mass from the inner space of the honeycomb due to the growth of the centrifugal force, and on the other hand, a convex surface of the honeycomb is formed. Displacement of the central part of the honeycomb from the rotation axis results in pulling the wires with the increasing effect of centrifugal forces on them.

Different design of centrifuges and cassette parameters used for honeycombs and reinforced wire features [4,5] require different rotation frequencies of the rotors to ensure removal of the product. At the same time, strength related features of the wires in the used frames [5] should ensure performance of the equipment without breaking the reinforced wire [6].

![Figure 1](image)

**Figure 1.** Constructive-technological scheme of centrifuges for pumping honey and extracting of waxy mass from cells: 1 – working camera; 2 – rotor; 3 – cassette with a cell; 4 – blades; 5 – unloading tray; 6 – electric drive.

The calculation of the reinforced wire parameters can be implemented with the help of various theoretical approaches. The calculation methods based on the finite element method have become most popular recently [7, 8]. However, these methods are applied through special programs that require expensive computers [9]. Classic methods for calculating flexible elements [10, 11] require much more simple IT capabilities.
Known methods of calculating flexible threads or wires \[10,12,13\] determine the location parameters of the stationary flexible elements. In the studied case, the dynamic load is in action and it changes the transverse load - centrifugal forces, as the wire position shifts from the original location. This is a significant aspect of the calculation method used.

One of the most important factors that significantly affects the reliability of constructions that use flexible threads (wires) as the main bearing elements of cells is emergence of plastic deformations in them. Modeling the work and studying the behavior of such systems under load is complicated by the fact that in these constructions, in addition to the physical non-linearity of the thread features, it is necessary to take into account geometric non-linearity of the shift of the thread coordinates from its initial position.

The purpose of this study is to assess the effect of the development of plastic deformations on the stress-deformed state of rotating flexible wire under changing the rotation frequency of the rotor. A comparative analysis of the calculation results obtained by linear and physically non-linear calculation methods at different rotation frequencies of the centrifuge rotor was performed.

2. Research methods
Taking into account the possibility of easier calculation by the linear model, as well as emergence of significant errors in such calculations in the non-linear zone of flexible thread features and nonlinearity of the applied load change, it is necessary to determine the values of proportionality coefficients between the methods of calculating flexible elements by linear dependence and with consideration of non-linearity features and loads.

When implementing the analytic calculations, the authors assumed that in their theoretical studies of the modeling process the cell mass remained unchanged and equal to the initial value. Although in fact, the density of honeycomb (and the degree of filling the cells) during the acceleration of the centrifuge rotor actively decreased. At the time of achieving the studied rotation frequency of the centrifuge, the cell mass pressing the wire was significantly less (and, accordingly, the created moment of inertia of the flexible wire was reduced), which allowed the reinforced wire to withstand the real current load to a higher rotation frequency compared to the model.

In the theoretical description, the term "reinforced wire" is replaced by a common term - flexible thread in calculations of such elements. Modeling of plastic deformation development was performed in the computer program MathCAD.

The research methods are based on the theory of mathematical modeling with the help of differential and integral function calculus of one and several variables, regulations of theoretical mechanics and are presented in detail in the works \[12, 13\]. The error of the values when calculating by these methods compared with the finite element method does not exceed 5% \[10,12\].

As the object of the study, a rotating flexible thread with a firmly attached elastic body is considered, the calculation model of which is presented in Figure 2.

The elastic body with density $\rho=500 \text{ kg/m}^3$, length $d=0.415 \text{ m}$, width $b=0.055 \text{ m}$ and thickness $h=0.025 \text{ m}$ is firmly fixed along its entire length to the flexible thread which stretches inside the body through its mass center. In its turn, the flexible thread with the supports installed in the vertical surface with the span $l=0.415 \text{ m}$, cross-section $A=12.56\times10^{-8} \text{ m}^2$, is located parallel to the axis with the distance $R=0.18 \text{ m}$. The initial arrow of the sag and the load causing the initial shape are absent. The characteristics of the flexible thread material are determined: temporary resistance $- \sigma_2=800 \text{ MPa}$; conditional fluidity limit $- \sigma_1=600 \text{ MPa}$, relative elongation after breakage $- \varepsilon_2 = 0.25$, elastic module $E=2.1\times10^5 \text{ MPa}$. Hardness of resiliently pliable supports $\nu=36.63 \text{ kN/m}$.

Dependencies between stresses and deformations in the flexible thread material are described as a piece-linear function shown in figure 3 \[12,13\]. At stresses to the conditional limit of fluidity, the stress-deformed state of the thread is elastic, and when this value is exceeded - elastoplastic. When the temporary resistance is reached, the thread breakage is caused by the material distraction.
Figure 2. Calculation model of the rotating flexible thread.

Where: – the initial state of the equilibrium line; – the final state of the equilibrium line after the external impact; \( H_1 \) – axial stress, N; \( T \) – thread tension, N; \( R_A \) – horizontal projection of the support reaction, N; \( R \) – radius of the thread dislocation from the rotation axis, m; \( l \) – length of the honeycomb frame span, m; \( d \) – length of the applied load, m; \( q_1 \) – distributed load on the thread, N\-m; \( h \) and \( b \) – thickness and width of the elastic body (wax cell construction), m; \( w(H_1, q_1, x) \) – sagging (dislocation) of the thread from the original position, m; \( I(H_1, q_1, x) \) – the moment of inertia of the flexible thread and elastic body, N\-m; \( R_B \) – the coordinates of the support points of the frame, m.

Figure 3. Diagram of the flexible thread deformation at the boundaries of elementary sections; \( \sigma_1, \sigma_2 \) – the values of the stresses in 1,2 sections, Pa; \( \varepsilon_1, \varepsilon_2 \) – values of the relative elongation of the thread, operating in 1,2 sections, m; \( E_1, E_2 \) – values of the elastic module operating in 1,2 sections.
3. Research results

Under the impact of centrifugal forces, the flexible thread is deformed, creating the ordinate shift from the initial position \( w(H_1,q_1,x) \) – sagging (dislocation) of the thread from the original position (maximum value, usually in the middle of the span). Such dislocation of the thread provides additional thread inertia. This causes additional displacement of the ordinate. As a result, all this leads to an additional increase in the centrifugal force of inertia and deformation (elongation) of the flexible thread. The original theoretical justification is presented in paper [14].

In addition, due to the high stresses (above the fluidity limit), the thread suffers residual deformations. To limit the thread load, it is important to achieve the strength condition of the thread to the breakage (p.C on Figure 4) or the firmness conditions (\( \Delta l \leq [l_B] \)) of the thread. At the same time, \( \sigma \) – calculated stresses for specific load (for example, for p.P it is the value of \([\sigma_P]\)=600 MPa), MPa; \( \sigma_B \) – allowed stresses that cause breakage under the temporary resistance of the thread (\( [\sigma_B]=800 \) MPa), MPa; \( \Delta l \) – thread elongation under the specific load, m; \( [l_B] \) – allowable elongation, causing thread breakage during its temporary resistance, m. On the charts (Figure 4 –Figure 8) the frequency of the rotor rotation is shown, which is equivalent to the centrifugal force of inertia.

If in the interval of compliance with the Hook’s law for stresses in flexible threads, the linear dependence of deformations and stresses remains, then with an additional load increase above the proportionality limit point (p. P), the specified linearity is destructed. And at the stress values above the fluidity point (p. T), residual deformations also occur after the load is removed. We are considering only the option of loading the thread.

When using the linear calculation method, the critical elongation \([l]\) is achieved at the calculated load \([\sigma_B]=2141 \) MPa (p.C), which inadequately describes the strength condition. Critical is the load of \([\sigma_B]=800 \) MPa (p.B). For the linear calculation model, the specified thread load (p.T) corresponds to the rotation frequency of the rotor \( n=104 \) min\(^{-1}\).

It should be noted that in case of a physically non-linear task setting, breakage of the flexible thread at the rotation frequency of 104 min\(^{-1}\) does not occur, although in the linear setting of the task expansion stresses in the material of the flexible thread already reach temporary resistance to the breakage (p.T. Figure 4). This state is proposed to be called a "conditional breakage".

In this case, when using in practice the principals of the linear calculation in sections of non-linearity of stresses, it is required to make an appropriate adjustment coefficient clarifying calculation results in accordance with the received result (Figure 4) for the principals of the physically non-linear calculation. As a result, in the sections of non-linearity of stresses, real existing stresses will be less than the estimated values received with the help of the linear model. Their ratio in the linearity section will be less than “1” (Figure 5) and it is shown with the bar line. At the same time, the ratio of real stresses (i.e. not linearly altered ones beyond the section of Hook’s law) and the linear model stresses...
with the same thread deformation will also differ from “1”. The increase in thread deformation will impact the accelerated increase in thread stresses. The adjustment coefficient will be over “1” – the full line graph (Figure 5) in the zone of non-linearity of features.

To clarify the calculation results, it is proposed to introduce a nonlinearity coefficient. The coefficient of non-linearity behavior of the flexible thread in the physically non-linearity calculation can have only a positive value equal to “1”. The values equal to “1” correspond the situation when the stresses or deformations defined in both linear and physically non-linear settings are equal. This indicates the absence of plastic deformations in the material of flexible thread. Any other value shows how many times stress or deformation of the thread length, determined outside the elastic work zone of the material differ from the same parameters defined by linear calculation which does not consider inelastic deformations.

Thus, with the selected parameters of the physical model for the rotation frequency of the rotor 250 min⁻¹ the reduction of expansion stresses with the linear model will be calculated with the coefficient 2141/800=2.68 (dividing by 2.68). And the calculated by the linear calculation model value of the thread elongation should be increased with the coefficient 0.41 (dividing by 0.41).

![Figure 5. Graphs of changes in non-linearity stresses k_{non}](image)

Among the external conditions of the load impact on the non-linearity behavior of the flexible thread (fog. 5) the centrifugal forces are the most significant, they arise depending on the squared angular rate and its derivative - the rotation frequency of the rotor. This is due to the formation and then further development of plastic deformations with increase in the rotation frequency. Moreover, the higher the rotation frequency is, the more there are deviations in the calculated values. Thus, with the specified geometric and physical parameters of the studied flexible thread, already at the rotation frequency more than 79min⁻¹, the modelling of its stress-deformed state should be implemented taking into account the physical nonlinearity of the thread features.

Analysis of the maximum sagging of the thread span in the middle of the span on the graphs (Figure 6), depending on the expansion stresses of the maximum loaded support cross-sections of the flexible thread in the linear and physically non-linear setting of the task with the consideration of the rotor rotation frequency, shows the non-linear nature of the sag value change. Calculation by the linear features compared to the physical nonlinearity undervalues the size of sagging from 135 to 55 mm, i.e. increase of 2.455 times is required. Thus, the non-linearity coefficient is $k_{sag}=1/2.455\approx0.41$.

In this case, plastic deformations begin to emerge at the rotation frequency over 79min⁻¹. When the specified value of rotation frequency is reached in a physically non-linear setting, there is a significant increase in the values of the sag deformations, which include elastoplastic deformations and kinematic movements of the thread. Similar to the plastic deformation of traditional steel constructions working under sagging, such state of flexible threads is proposed to be called “conditional plastic hinge”.
The power and strength calculation of centrifuge elements requires determining the value of the moment of inertia of the entire rotor, including the moment of inertia attached to the flexible thread of the elastic body (wax cell structure) during rotation. From the graphs presented in figure 7, it is clear that the moment of inertia grows correspondingly faster with the higher rotational frequency. At the same time, this phenomenon is expressed to a greater extent in the physically non-linear part of the calculation. This is due to the significant increase in the equilibrium line of the rotating flexible thread (thread dislocation from the initial position) depending on the angular rotor velocity from the initial equilibrium state. Calculation by linear features compared to physical nonlinearity undervalues the value of the inertia moment of the body, combined with the thread from 0.02109 to 0.01314 kg\(\cdot\)m\(^2\) of sagging, i.e. an increase of 1.6 times is required. The non-linearity coefficient is \(k_{inel} = 1/1.6 = 0.623\).

Figure 6. Graphs of changing the maximum sag of the flexible thread.

Figure 7. Graphs of changing the moment of inertia of the elastic body.

The loading option number 3. The elastic body with density \(\rho = 350\) kg/m\(^3\), length \(d = 0.415\) m, width \(b = 0.055\) m and thickness \(h = 0.025\) m is firmly fixed along its entire length to the flexible thread going inside the body through its mass center. At the same time, the flexible thread with supports installed in the vertical surface span \(l = 0.415\) m, section \(A = 12.56 \times 10^{-8}\) m\(^2\), is located parallel to the rotation axis at the distance \(R = 0.18\) m. The initial sagging arrow and the load defining the initial shape are absent. The characteristics of the flexible thread material defined: temporary resistance – \(\sigma_2 = 1270\) MPa; conditional fluidity limit – \(\sigma_1 = 950\) MPa, relative elongation after breakage \(\varepsilon_2 = -0.15\), elasticity module \(E = 2.1 \times 10^8\) MPa. Hardness of resilient supports \(u = 36.63\) kN/m.
In turn, the graphs of changes of the cross equally spread load (depending on rotation frequency), which is the equivalent of the centrifugal force of inertia, are presented in figure 8. Analysis of these graphs shows a decrease in the centrifugal force of inertia on the attached elastic body, which emerges due to reduction of the flexible thread firmness caused by the development of plastic deformations. At the same time, it should be noted that the moment of inertia of the attached elastic body increases. This means that the process of changing the firmness of the system emerges faster than the change in the moment of inertia of the attached elastic body caused by its deformation. Calculating by linear properties compared to physical nonlinearity overstates the value of the equivalent body load, combined with the thread from 390 N/m to 650 N/m, i.e. the reduction in the calculated value of 0.643 times is required. That is, the non-linearity ratio is \( k_{non} = 1/0.643 \approx 1.56 \).

Given that interested readers cannot always implement thread modeling, and modeling results can help them design similar devices, the results of the calculations of some specific cases of thread loading are given as reference data. The number 1 loading option is described above.

The loading option number 2. The elastic body with density \( \rho = 500 \text{ kg/m}^3 \), length of \( d = 0.415 \text{ m} \), width \( b = 0.055 \text{ m} \) and thickness \( h = 0.025 \text{ m} \) is firmly fixed along its entire length to the flexible thread going inside the body through its mass center. In its turn, the flexible thread with supports installed in the vertical surface with the span \( l = 0.415 \text{ m} \), cross-section \( A = 12.56 \times 10^{-8} \text{ m}^2 \), is located parallel to the rotation axis at the distance \( R = 0.18 \text{ m} \). The initial sagging arrow and the load causing the initial shape are absent. The characteristics of the flexible thread material are defined: temporary resistance – \( \sigma_2 = 1270 \text{ MPa} \); conditional fluidity limit – \( \sigma_1 = 950 \text{ MPa} \), relative elongation after breakage \( \varepsilon_2 = 0.15 \), elasticity module \( E = 2.1 \times 10^5 \text{ MPa} \). Hardness of resilient supports \( u = 36.63 \text{ kN/m} \).

The results received. At the critical rotation frequency of 280 min\(^{-1}\), the maximum deflection in the middle of the span is 101.2 mm; maximum expansion stresses are 1270 MPa; equivalent even-distributed load is 540.06 N/m; the moment of inertia of the attached elastic body – 0.01759 kg\(\cdot\)m\(^2\).

At the critical rotation frequency of 350 min\(^{-1}\), the maximum deflection in the middle of the span – 101.2 mm; maximum expansion stresses are 1270 MPa; equivalent even-distributed load is 562 N/m; the moment of inertia of the attached elastic body – 0.01759 kg\(\cdot\)m\(^2\).

![Figure 8. Graphs of changing of the equivalent load.](image)

At the critical rotation frequency of 350 min\(^{-1}\), the maximum deflection in the middle of the span – 101.2 mm; maximum expansion stresses are 1270 MPa; equivalent even-distributed load is 562 N/m; the moment of inertia of the attached elastic body – 0.01287 kg\(\cdot\)m\(^2\). Displacement of supports – 14.5 mm.

In the experimental studies of the authors, the rotation rotor frequency of 400 min\(^{-1}\) was marked as the critical frequency leading to the first single cases of breakage in the reinforced wire [6].

The break of the thread at the rotor rotation frequency of 400 min\(^{-1}\) provides an elastic body with density of 270 kg/m\(^3\), along with this, the results of the abovementioned thread were received. The results received. At the critical rotation frequency 400 min\(^{-1}\), the maximum deflection in the middle of
the span – 111.87 mm; maximum expansion stresses – 1270 MPa; equivalent even-distributed load – 565 N/m; moment of inertia of the attached elastic body – 0.00997 kg·m². Displacement of the supports – 14.5 mm.

Thus, when performing engineering calculations of the centrifuge for honey or waxing mass extraction at the time of the rotor setting its critical rotation frequency, the degree of filling bee bread cells should correspond to about 0.54. ≈ 0.5.

4. Conclusion
The results of the numerical studies based on the previously developed methods and tested computer programs based on MathCAD showed that the calculated values of the emerged stresses and deformations when rotating the flexible thread are significantly impacted by the development of plastic deformations. As a result, modeling of the stress-strain state of the rotating flexible thread should be implemented with the consideration of emerging plastic deformations.

When modeling objects with a rotating flexible thread, it is possible to use methods of both physically non-linear calculation and simpler and more simple and accessible linear calculation. In the latter case, the result of the calculation should be divided by nonlinearity coefficient.

The non-linearity coefficient is: for expansion stresses – 2.68; for elongation of the thread – 0.41; for the value of mid-span sagging – 0.41; the value of the moment of inertia of the body combined with the thread is 0.623; the equivalent load of the body combined with the thread – 1.56. The degree of filling the volume of honeycombs (cells), when conducting strength calculations of the flexible thread of the centrifuge for honey and waxy mass extraction, should correspond to about 0.5.

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