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# ВИБІР КОНСТРУКТИВНОГО РІШЕННЯ ВИСОКОЧАСТОТНОГО СТАБІЛІЗАТОРА ДИНАМІЧНОГО НАТЯГУ НИТОК ОСНОВИ ШЛЯХОМ МОДАЛЬНОГО АНАЛІЗУ В ПАКЕТІ ANSYS

***У роботі розглядається конструкція високочастотного стабілізатора динамічного натягу ниток основи у формі тонкої трикутної пластини з вирізом при змінній кількості ребер жорсткості. Розв'язок завдання про вільні коливання такої системи можна одержати тільки чисельними методами. Завдання вирішується методом скінченних елементів із застосуванням програми ANSYS. Досліджено шість різних конструктивних варіантів високочастотного стабілізатора. У вихідній моделі варіювалася кількість ребер жорсткості і їх довжина. Моделювання виконане таким чином, що довжину кожного ребра можна змінювати від нуля, що відповідає відсутності відповідного ребра, до граничного значення. Визначені перші п'ять власних частот і форм коливань. Аналіз результатів показує, що власна частота коливань досягає свого максимального значення при наявності трьох підкріплювальних ребер максимальної довжини.***

***Ключові слова:*** *високочастотний стабілізатор, трикутна пластина, метод скінченних елементів, власна частота, форма коливань, ANSYS.*

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# ВЫБОР КОНСТРУКТИВНОГО РЕШЕНИЯ ВЫСОКОЧАСТОТНОГО СТАБИЛИЗАТОРА ДИНАМИЧЕСКОГО НАТЯЖЕНИЯ НИТОК ОСНОВЫ ПУТЕМ МОДАЛЬНОГО АНАЛИЗА В ПАКЕТЕ ANSYS

***В работе рассматривается конструкция высокочастотного стабилизатора динамического натяжения ниток основы в форме тонкой треугольной пластины с вырезом при переменном количестве ребер жесткости. Решение задачи о свободных колебаниях такой системы можно получить только численными методами. Задача решается методом конечных элементов с применением программы ANSYS. Исследованы шесть различных конструктивных вариантов высокочастотного стабилизатора. В исходной модели варьировалось количество ребер жесткости и их длина. Моделирование выполнено таким образом, что длину каждого ребра можно изменять от нуля, что соответствует отсутствию соответствующего ребра, до предельного значения. Определены первые пять собственных частот и форм колебаний. Анализ результатов показывает, что собственная частота колебаний достигает своего максимального значения при наличии трех подкрепляющих ребер максимальной длины.***

***Ключевые слова:*** *высокочастотный стабилизатор, треугольная пластина, метод конечных элементов, собственная частота, форма колебаний, ANSYS.*

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# CHOICE OF A CONSTRUCTIVE SOLUTION OF A HIGH-FREQUENCY STABILIZER OF DYNAMIC TENSION OF A WARP THREADS BY MEANS OF A MODAL ANALYSIS IN ANSYS PACKAGE

***The paper discusses the design of a high-frequency stabilizer of dynamic tension of warp threads in the form of a thin triangular plate with a notch with a variable number of stiffeners. The solution of the problem of free oscillations of such a system can be obtained only by numerical methods. The problem is solved by the finite element method using the ANSYS program. Studied six different design options for high-frequency stabilizer. In the initial model, the number of stiffeners and their length varied. The simulation is performed in such a way that the length of each edge can be changed from zero, which corresponds to the absence of the corresponding edge, to the limit value. The first five natural frequencies and oscillation modes are determined. Analysis of the results shows that the natural frequency of oscillation reaches its maximum value in the presence of three reinforcing ribs of maximum length.***

***Keywords:*** *high-frequency stabilizer, triangular plate, finite element method, natural frequency, vibration shape, ANSYS.*

**Introduction.** An analysis of the designs of the warp knitting and sewing machines showed that, as a stabilizer of dynamic tension of the warp threads (SDTWT), mainly passive stabilizers are used, the structures of which are based on an elastic element. Analytical studies of passive SDTWT confirmed the validity of the requirements for the operating parameters of passive SDTWT - to have the maximum possible natural frequency of oscillations with a relatively (comparable with the stiffness of the warp threads in an elastic refueling system) low stiffness [1 - 4]. These requirements identified the main directions of improvement of passive SDTWT - creation of high-frequency structures.

**Problem formulation.** The basis of the existing design solutions used to increase the natural frequency, is to reduce the mass of moving elements of the stabilizer. At the same time, a way of increasing the rigidity of structures, which directly follows from the well-known formula seems to be promising - [5 – 7]

$$\omega = \sqrt{\frac{c}{m}},$$

where $\omega$ — natural frequency; $c$ — stiffness.

Moreover, it is proposed to increase the rigidity of the stabilizer by the arrangement of stamped stiffeners, because this approach does not increase the mass of the system.

The paper considers the design of a high-frequency stabilizer of dynamic tension of warp threads (SDTWT) in the form of a thin triangular plate with a notch with a variable number of stiffeners (Fig. 1).

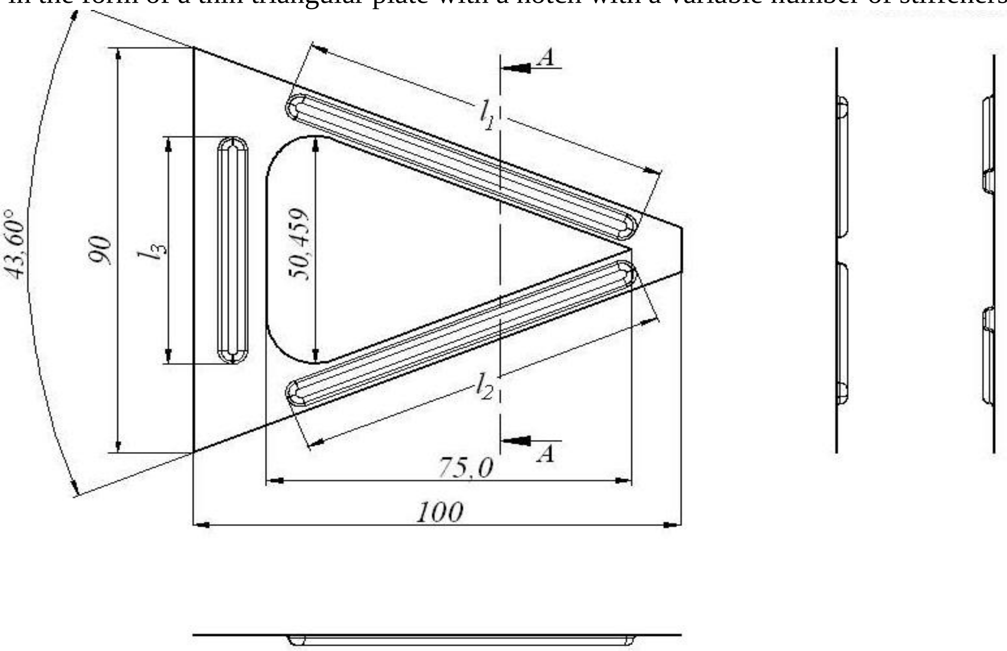



*Fig. 1.* **Model of high-frequency SDTWT**

**Research results.** The solution of the problem of free oscillations of such a system can be obtained only by numerical methods, and the most effective, in our opinion, is the finite element method. Currently, there are a large number of computer programs that implement this method. Among them, the authors selected the ANSYS package [8–9], the capabilities of which allow the following types of dynamic calculations to be performed: transient dynamics, modal analysis, response to harmonic influence, spectral analysis, and response to random vibration.

Modal analysis allows you to determine the natural frequencies and mode shapes. In addition, it is used as a reference for other, more detailed dynamic calculations, such as non-stationary dynamic analysis or the response of the system to harmonic effects.

In ANSYS, modal analysis is a linear procedure. Any non-linearities like plasticity or gap-contact elements are ignored, even if they are specified. Four methods are available to detect natural modes of vibration, including damping.

When performing a modal analysis, the Young's modulus and the density of the material, which is assumed to be linear, isotropic, or orthotropic, with properties depending or independent of temperature are set.

When defining eigenmodes of vibration, four methods can be used: short-cut, subspace, asymmetric (for problems with an asymmetric matrix, for example, when the fluid interacts with the structure) and decrement (when friction cannot be neglected, for example, when analyzing movement on a supporting surface).

For most applications, you need to choose between two methods: short-cut and subspace. The first of them works faster, since it uses a shortened (condensed) matrix system to obtain a solution. However, compared with the second method, it is less accurate.

The file of calculation results contains natural frequencies and shapes forms, as well as corresponding stresses and forces.

When determining the natural frequencies and modes of oscillations of a structure, it is assumed that free continuous oscillations occur [10]:

$$\overline{M}\vec{u}'' + \overline{K}\vec{u} = 0 \quad (1)$$

Note that the stiffness matrix of the structure $\overline{K}$ may include the effect of preloading.

For a linear system, the free oscillations will be harmonic:

$$\vec{u} = \vec{\varphi}_i \cos\omega_i t, \quad (2)$$

where $\vec{\varphi}_i$ – eigenvector representing the *i*-th form of oscillations;

$\omega_i$ – *i*-th natural circular eigenfrequency (radians per unit time);

$t$ – time.

Thus, the matrix equation (1) takes the following form:

$$\left(-\omega_i^2\overline{M} + \overline{K}\right)\vec{\varphi}_i = 0. \quad (3)$$

This equation has a solution, in addition to the trivial $\vec{\varphi}_i = 0$, only when the determinant of this system $\left(-\omega_i^2\overline{M} + \overline{K}\right)$ is zero, that is:

$$\left|-\omega_i^2\overline{M} + \overline{K}\right| = 0. \quad (4)$$

The last equation is the eigenvalue problem [11 - 12]. The solution of equation (4), if *n* is the order of the matrix, is the characteristic polynomial of the *n*-th order, which has *n* roots: $\omega_1^2$, $\omega_2^2 \ldots \omega_n^2$, where *n* is the number of degrees of freedom. These roots are eigenvalues of the equation. The eigenvectors $\vec{\varphi}_i$ are obtained by substituting the obtained roots $\omega_i^2$ into equation (3). The eigenvalue $\omega_i^2$ determines the eigenfrequency of the system $\sqrt{\omega_i^2}$, and the eigenvector $\vec{\varphi}_i$ defines the corresponding form of oscillations (displacement of the system).

The values of the natural cyclic frequencies $\omega$ and the natural technical frequencies f are related by the following relationship:

$$f_i = \frac{\omega_i}{2\pi}, \quad (5)$$

where $f_i$ – *i*-th technical eigenfrequency (cycles per unit time).

Usually, the eigenvector $\vec{\varphi}_i$ is called normalized if the following equality holds (reflecting the orthogonality property of the forms of natural oscillations):

$$\vec{\varphi}_i^T\,\overline{M}\vec{\varphi}_i = 1. \quad (6)$$

In another case, the eigenvector $\vec{\varphi}_i$ is normalized from the condition that its largest components are equal to one. The condition of the orthogonality of the forms of oscillations can be explained as the equality to zero of the forces of inertia of the *i*-th form of oscillations on the displacements of the *k*-th form of oscillations.

When using the method of frequency condensation (reduction of degrees of freedom), the *n* eigenvectors can then be expanded at the “expansion” stage to the full set of modal degrees of freedom of the structure:

$$\vec{\varphi}_{si} = -\left[\overline{K}_{ss}\right]^{-1}\left[\overline{K}_{sm}\right]\vec{\varphi}_i, \quad (7)$$

where $\vec{\varphi}_{si}$ — the vector of excluded (auxiliary) degrees of freedom of the *i*-th mode (the auxiliary degrees of freedom are those degrees of freedom that will be condensed to reduce the dimension of the system);

$\left[\overline{K}_{ss}\right]$, $\left[\overline{K}_{sm}\right]$ — submatrices of stiffness with respect to auxiliary degrees of freedom and the connection of auxiliary degrees of freedom with those held respectively;

$\vec{\varphi}_i$ — vector of the held (basic) degrees of freedom of the *i*-th mode.

The model is approximated by the standard Shell63 finite element. The element is defined by four nodes, four values of thickness (in this case it is a constant value), the stiffness of the elastic base and the properties of an orthotropic material. The direction of orientation of an orthotropic (in general) material is related to the coordinate system of the element. The x-axis of the coordinate system can be rotated through a certain angle.

The work of the element is based on the Kirchhoff – Love theory.

6 different design variants of high-frequency SDTWT were investigated (Fig. 1).

In the initial model, the number of stiffeners and their length ($l_1, l_2, l_3$) varied. The simulation is performed in such a way that the length of each edge can be changed from zero (which corresponds to the absence of the corresponding edge) to the limiting value:

$l_1^{max} = 77мм,\ l_2^{max} = 77мм,\ l_3^{max} = 60,28мм$.

SDTWT options investigated:

- option 1: $l_1 = l_2 = 77мм,\ l_3 = 0$;
- option 2: $l_1 = l_2 = 38,5мм,\ l_3 = 0$;
- option 3: $l_1 = l_2 = 77мм,\ l_3 = 60,28мм$;
- option 4: $l_1 = l_2 = 38,5мм,\ l_3 = 30,14мм$;
- option 5: $l_1 = l_2 = l_3 = 0$ (triangle plate without ribs);
- option 6: triangle plate without ribs and notch.

The number of frequencies and vibration forms that can be obtained as a result of the calculation in the ANSYS program is practically unlimited, however, it is obvious that higher frequencies can be of only theoretical interest. Therefore, Table 1 shows the values of only the first five natural frequencies (technical and cyclic) for each of the calculation options.

*Table 1*

**Eigen frequencies**

| №№ opt. | Eigen frequency | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Technical, Hz | | | | | Cyclic, $s^{-1}$ | | | | |
| | $f_1$ | $f_2$ | $f_3$ | $f_4$ | $f_5$ | $\omega_1$ | $\omega_2$ | $\omega_3$ | $\omega_4$ | $\omega_5$ |
| 1 | 66,04 | 502,08 | 779,65 | 1133,7 | 1676,0 | 414,9 | 3154,7 | 4898,7 | 7123,2 | 10530,6 |
| 2 | 46,44 | 271,58 | 393,40 | 621,47 | 712,54 | 291,8 | 1706,4 | 2471,8 | 3904,8 | 4477,0 |
| 3 | 72,21 | 548,7 | 827,56 | 1185,7 | 1835,6 | 453,7 | 3447,6 | 5199,7 | 7450,0 | 11533,4 |
| 4 | 46,77 | 275,52 | 397,55 | 642,86 | 714,16 | 293,9 | 1731,1 | 2497,9 | 4039,2 | 4487,2 |
| 5 | 42,67 | 199,86 | 243,18 | 536,98 | 585,61 | 268,1 | 1255,8 | 1527,9 | 3373,9 | 3679,5 |
| 6 | 55,60 | 250,91 | 265,25 | 624,63 | 682,75 | 349,3 | 1576,5 | 1666,6 | 3924,7 | 4289,8 |

Table 1 analysis shows that the natural frequency of oscillation reaches its maximum value ($\omega_1 = 453,7c^{-1}$) in the presence of three reinforcing ribs of maximum length ($l_1 = l_2 = 77мм,\ l_3 = 60,28мм$). In this case, the mass of the plate (on the basis of the value $\rho = 7800кг/м^3$) is 10.362846 grams (Table 2).

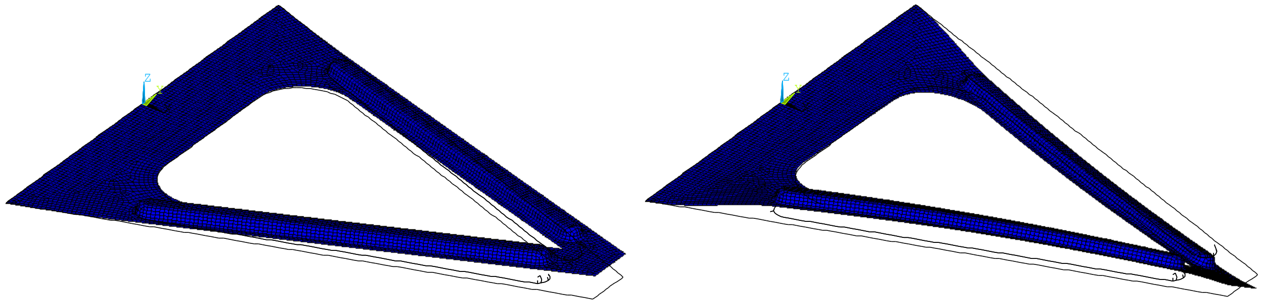
Note that the ANSYS program automatically calculates the mass of the plate for each of the options considered, and minor differences in the numerical values of the mass are due to the fact that the program perceives the edges as structural elements, not taking into account that these edges are obtained by stamping, without the use of additional material.

*Table 2*

**Plate volume and mass**

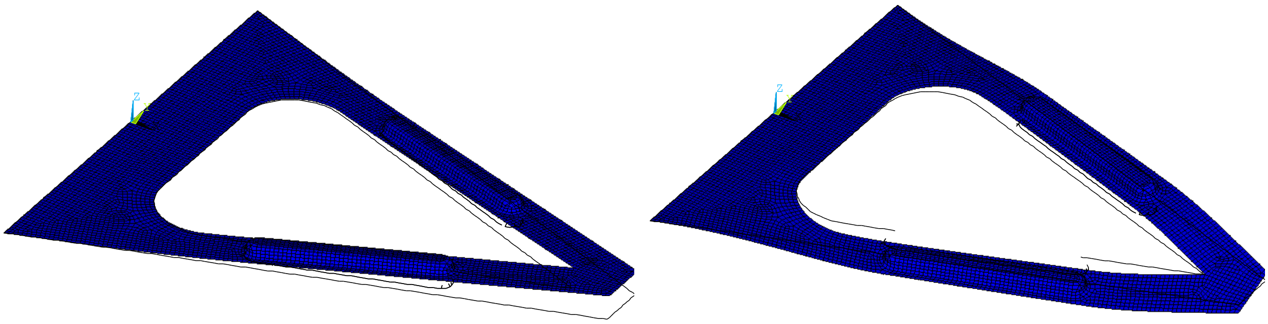
| №№ of option | Plate volume, $mm^3$ | Plate mass, g |
|---|---|---|
| 1 | 1275.50 | 9,94890 |
| 2 | 1208.30 | 9,42474 |
| 3 | 1328.57 | 10,362846 |
| 4 | 1232.53 | 9,613734 |
| 5 | 1140.17 | 8,893326 |
| 6 | 2000.00 | 15,6 |

Forms of natural oscillations corresponding to the first two eigenfrequencies given in Table 1 are shown in Fig. 2-7.



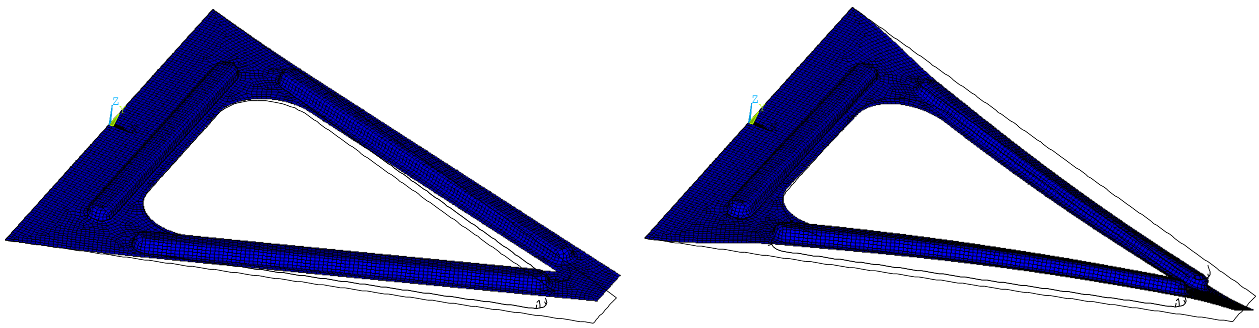
1 form 2 form

*Fig. 2.* **Oscillation forms of the first calculation option**



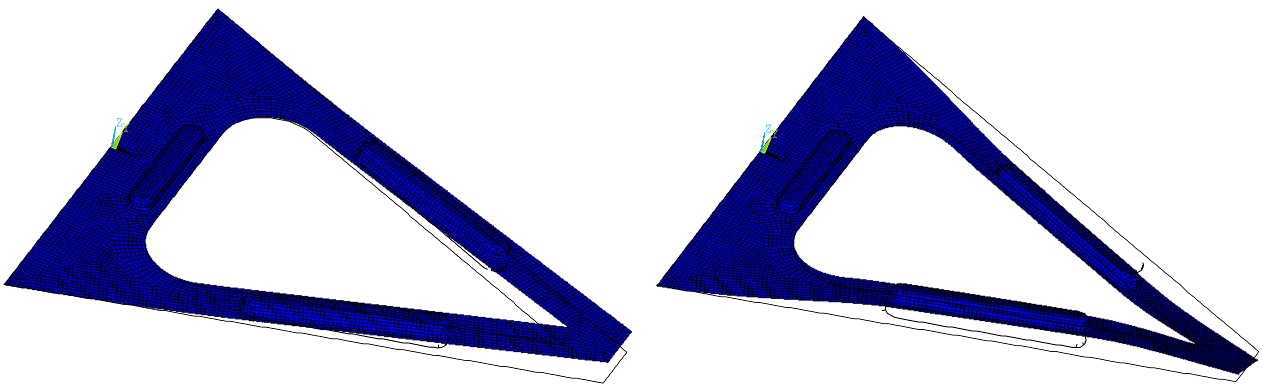
1 form 2 form

*Fig. 3.* **Oscillation forms of the second calculation option**



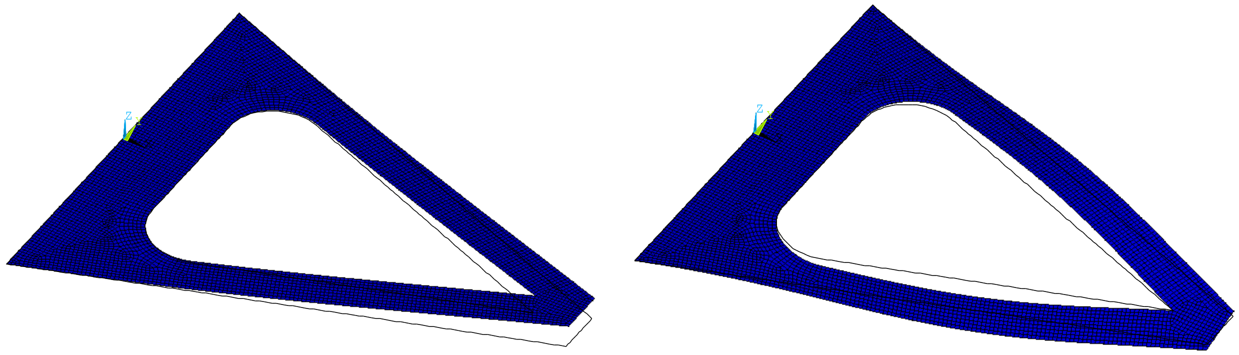
1 form 2 form

*Fig. 4.* **Oscillation forms of the third calculation option**



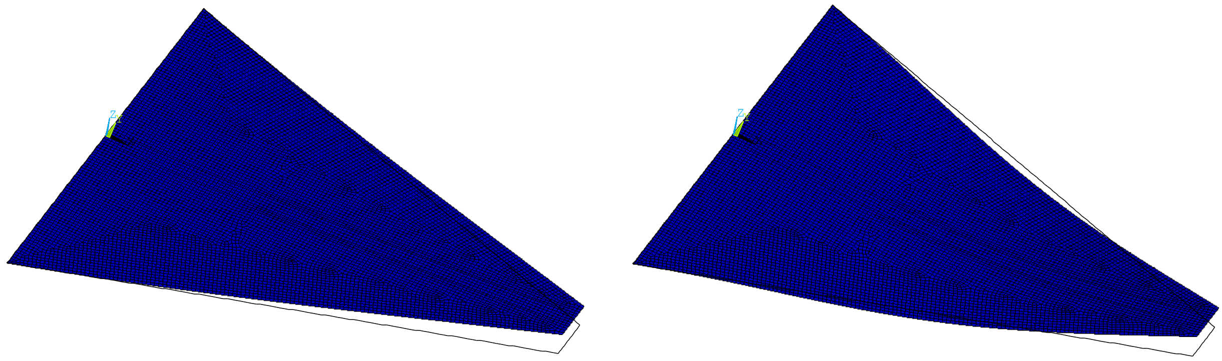
1 form 2 form

*Fig. 5.* **Oscillation forms of the fourth calculation option**



1 form 2 form

*Fig. 6.* **Oscillation forms of the fifth calculation option**



1 form 2 form

*Fig. 7.* **Oscillation forms of the sixth calculation option**

**Conclusions.** In all the considered variants of calculation, the first form of oscillations is flexural, and the other four forms are flexural-twisting.

Table 1 analysis shows that the natural frequency of oscillation reaches its maximum value ($\omega_1 = 453,7c^{-1}$) in the presence of three reinforcing ribs of maximum length ($l_1 = l_2 = 77мм$, $l_3 = 60,28мм$).

**References**

1. CHaban V.V. Dinamika osnovov’yazal'nih mashin /V.V. CHaban, L.A. Bakan, B.F. Pipa – K.: KNUTD, 2012. – 287 s.

2. Pipa B.F. Dinamika privodu osnovov’yazal'nih mashin /B.F. Pipa, V.V. CHaban //Visnik HNU. – 2007. – № 6. – S.11-14.

3. CHaban V.V. Analiz podhodov k raschetu vysokochastotnogo stabilizatora dinamicheskogo natyazheniya nitej osnovy //Vestnik SPGUTD. Dizajn. Materialy. Tekhnologii (S-Peterburg). – 2013. – № 1(26). – S. 45-49.

4. Paraska G.B. Stabilizaciya natyagu nitok osnovi na v’yazal'nih mashinah: monografiya / G.B. Paraska. — Hmel'nic'kij: HNU, 2012. — 275 s.

5. Sur’yaninov M. G. Teoretichni osnovi dinamiki mashin: Uchb. posibnik / M. G. Sur’yaninov, O. F. Dashchenko, P. O. Bilous, G. O. Obors'kij // Pid red. M.G. Sur’yaninova. — 2-e vid., perer. i dop. — Odessa: Astroprint, 2008. — 412 s.

6. Babakov I. M. Teoriya kolebanij / I. M. Babakov — M.: Nauka, 1968. — 560 s.

7. Biderman V.L. Teoriya mekhanicheskih kolebanij / Biderman V.L. – M.: Vysshaya shkola, 1980. – 408 s.

8. Dashchenko A. F. ANSYS v zadachah inzhenernoj mekhaniki / A. F. Dashchenko, D. V. Lazareva, N. G. Sur'yaninov / Izd. 2-e, pererab. i dop. Pod red. N. G. Sur'yaninova. — Odessa. — Pal'mira, 2011. — 505 s.

9. Fedorova N.N. Osnovy raboty v ANSYS 17 / N.N. Fedorova, S.A. Val'ger, M.N. Danilov, YU.V. Zaharova — M.: DMK Press, 2017. — 210 s.

10. Vasilenko M.V. Teoriya kolivan' i stijkosti ruhu / M.V. Vasilenko, O.M. Aleksejchuk — K.: Vishcha shk., 2004. — 525 s.

11. Myshkis A.D. Matematika dlya VTUZov. Special'nye kursy. – M.: Nauka, 1971. – 632 s.

12. Rajs Dzh. Matrichnye vychisleniya i matematicheskoe obespechenie. – M.: Mir, 1984. – 264 s.

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