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THE IMPACT OF ELASTIC WEDGES ON THE EFFICIENCY OF KNITTING MACHINES

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In this paper, we present the results of the study of the influence of the wedge design of a knitting machine on dynamic loads that arise in the needle-wedge interaction zone. A design for an elastic wedge, capable of effectively reducing dynamic loads in the knitting mechanism of the knitting machine is hereby proposed.

Keywords: knitting machine, wedge, dynamic loads, work efficiency.

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Reducing dynamic loads that occur in the impact interaction zone of the knitting machine needles with the wedge is one of the pressing issues in the knitting machinery engineering. One of the promising approaches to addressing this issue is replacing traditional wedges with rigid working surfaces (rigid wedges) with wedges containing elastic working surface (elastic wedges). Assessing the feasibility and efficiency of using such wedges is an integral part of the development of new designs for elastic wedges.

Previous theoretical and experimental studies show that the reduction of dynamic loads in the knitting mechanisms of knitting machines can be achieved by using wedges with elastic working surfaces (the surface that interacts with the heels of the needles). In some works, a number of designs of elastic wedges are proposed, which can increase the efficiency of knitting machines. However, despite the accumulated experience in the practice of research on improving the designs of wedges of knitting machines, solving the problem of reducing dynamic loads in knitting systems by improving the wedges still remains relevant.

This paper is aimed at developing a new design for an elastic wedge and evaluating its efficiency in the knitting mechanisms of knitting machines.

The obtained results prove the feasibility and efficiency of using the proposed elastic wedge design in the knitting mechanism of knitting machines.

The use of the proposed wedge design allows:

- improving the durability of the wedge operation by reducing dynamic loads in the area of its interaction with needles;
- enhancing the quality of knitted fabric by increasing the stability of the needle-wedge pair performance;
- increasing the productivity of the knitting machine by improving the wedge durability.

In this paper, we present the results of the study of the influence of the wedge design of a knitting machine on dynamic loads that arise in the needle-wedge interaction zone, as well as the design for elastic wedge, capable of effectively reducing dynamic loads in the knitting mechanism of the knitting machine.

General articulation of the issue and its relationto important scientific or practical tasks

Reducing dynamic loads that occur in the impact interaction zone of the knitting machine needles with the wedge is one of the pressing issues in the knitting machinery engineering [1, 2]. One of the promising approaches to addressing this issue is replacing traditional wedges with rigid working surfaces (rigid wedges) with wedges containing elastic working surface (elastic wedges) [3-6]. Assessing the feasibility and efficiency of using such wedges is an integral part of the development of new designs for elastic wedges.

Studies and publications analysis

Numerous theoretical and experimental studies [3-5] conducted by Prof. V.M. Harbaruk, O.M. Khomiak et all. demonstrate that reducing dynamic loads in the knitting mechanisms of knitting machines can be achieved by using wedges with elastic working surfaces (i.e., surfaces interacting with the needle heels). In this paper [6], we present a series of designs for elastic wedges, which are capable of improving the efficiency of knitting machines. However, despite the accumulated experience in research practice aimed at enhancing the designs of knitting machine wedges, solving the problem of reducing dynamic loads in knitting systems through wedge improvement remains a pressing issue.

Paper target

This paper is aimed at developing a new design for an elastic wedge and evaluating its efficiency in the knitting mechanisms of knitting machines.

Presentation of basic material

According to the studies [2], the efficiency of the knitting machine wedge is evaluated using the coefficient of dynamic load reduction in the knitting systems *n*, which is determined as follows:

$$n = \sqrt{\frac{P_1}{P_2}} = \sqrt{\frac{\delta_2'}{\delta_1'}} = \sqrt{\frac{C_1}{C_2}}, \tag{1}$$

Where: P_1, P_2 is the maximum impact force of the needle on the wedge according to the traditional (rigid) design and according to the new wedge design, particularly the elastic wedge,

$$P_1 = V\sqrt{\frac{m}{K\delta_1'}}tg\alpha, \quad P_2 = V\sqrt{\frac{m}{K\delta_2'}}tg\alpha; \tag{2}$$

δ_1', δ_2' is the compliance of the needle-wedge pair when using the traditional (rigid) wedge design and the elastic wedge;

C_1, C_2 is the stiffness (inverse of compliance) of the needle-wedge pair when using the rigid wedge and elastic wedge, respectively;

V is the velocity of the impact interaction between the needle and the wedge;

m is the needle mass;

K is a structural parameter of the knitting system [2];

α is the angle of the wedge profile in the needle impact zone (i.e., the angle of contact between the needle heel and the wedge).

As the analysis of needle-wedge interaction during the impact period [2] shows, the needle-wedge system deformation is determined as follows:

$$\delta = \delta_x tg\alpha + \delta_y, \tag{3}$$

Where: δ is the needle-wedge system total deformation;

δ_x, δ_y is the needle-wedge system deformation along the X and Y axes.

The analysis of the operating conditions of the needle in the knitting machine showed that the compliance of the needle-wedge pair is primarily determined by: the bending and twisting of the needle heel; the bending of the stem (needle guide); and deformation of the needle, wedge and stem materials in the zone of their contact interaction.

Given the foregoing, the static deformation of the needle-wedge system along the X-axis can be determined as follows:

$$\delta_x = \delta_{1x} + \delta_{2x} + \delta_{3x} + \delta_{4x} + \delta_{5x}, \tag{4}$$

Where: δ_{1x} is the needle heel and wedge compression deformation;

δ_{2x} is the needle heel bending deformation;

δ_{3x} is the needle body and the stem compression deformation;

δ_{4x} is the stem bending deformation;

δ_{5x} is the needle heel twisting deformation.

Using the theory of contact strength and bending [7], it is possible to derive the following:

$$\delta_{1x} = 1{,}16\frac{P}{l_1 E(1+\mu ctg\alpha)sin^2\alpha}; \tag{5}$$

$$\delta_{2x} = P\frac{a^3}{3E_1 J_{1x}}; \tag{6}$$

$$\delta_{3x} = 1{,}16\frac{P}{b^2 E'}\left[\frac{a^2}{l_2}+\frac{(a+b)^2}{l_3}\right]; \tag{7}$$

$$\delta_{4x} = P\frac{(a+b)l^3}{3bE_2 J_{2x}}; \tag{8}$$

Where: P is the component of the impact force acting along the X-axis (horizontal component);

l_1, l_2, l_3 is the length of the contact lines corresponding to the needle heel with the wedge, the back of the needle with the stem, and the needle with protruding edge of the stem;

E, E' are reduced elastic moduli corresponding to the needle-wedge and needle-stem pairs materials;

E_1, E_2 are elastic moduli of the needle and stem materials;

μ is the needle-wedge pair friction coefficient;

a is the lever arm of the impact force P;

J_{1x}, J_{2x} are inertia moments of the cross-sections of the needle heel and the stem.

The magnitude of the needle heel twisting deformation is determined as follows [4]:

$$\delta_{5x} = 0{,}5h \cdot tg\varphi, \tag{9}$$

Where: h is the heel width;

φ is the angle of the needle heel twisting at the moment of impact.

Since φ is small, we can assume that: $\delta_{5x} = 0{,}5h\varphi$. (10)

As is known [7]: $\varphi = \frac{M_k \alpha}{GJ_p}$, (11)

Where: M_k is the torque that arises, when the needle heel impacts on the wedge;

G is the shear modulus of the needle material;

J_p is the polar moment of inertia of the needle heel cross-section.

Based on the needle equilibrium condition, it is possible to derive the following: $M_k = 0{,}5Nh\left(1-\frac{\Delta}{h}ctg\alpha\right)sin\alpha$,

or, given that $P = N\,sin\alpha$: $M_k = 0{,}5P(h-\Delta ctg\alpha)$, (12)

Where: Δ is the heel width.

Since for stamped needles, it is usually assumed that $h/\Delta \geq 4$, as based on [7]: $J_p = \frac{(h/\Delta - 0{,}63)\Delta^4}{3}$.

Given (11), (12), equation (10) takes the following appearance:

$$\delta_{5x} = 0{,}25P\frac{(h - \Delta ctg\alpha)h\alpha}{GJ_p}. \tag{13}$$

The deformation of the needle-wedge pair along the Y-axis can be determined as follows:

$$\delta_y = \delta_{1y} + \delta_{2y} + \delta_{3y}, \tag{14}$$

Where: δ_{1y} is the deformation along the Y-axis caused by the compression of the needle and the wedge;

δ_{2y} is the deformation due to the bending of the needle heel along the Y-axis;

δ_{3y} is the deformation due to the twisting of the needle heel.

Similarly to the foregoing: $$\delta_{1y} = 1{,}16\frac{P}{l_1E(tg\alpha + \mu)cos^2\alpha}; \tag{15}$$

$$\delta_{2y} = P\frac{a^3ctg(\alpha + \rho)}{3E_1J_y}, \tag{16}$$

Where: ρ is the needle-wedge pair friction angle;

J_y is the moment of inertia of the cross-section of the needle heel relative to the Y-axis;

With a sufficient degree of accuracy, it can be written as: $\delta_{3y} = 0{,}5\delta_{5x}\varphi$. (17)

Given (11), (12) i (13), we derive the following:

$$\delta_{3y} = 0{,}25p^2h\left[\frac{0{,}5(h - \Delta ctg\alpha)a}{GJ_p}\right]^2. \tag{18}$$

For KO-type single-bed circular knitting machines with needles positioned 0–388, 0–384 [8] ($l_1 = 2{,}5$; $l_2 = 81$; $l_3 = 25{,}5$; $a = 1{,}5$; $b = 3{,}8$; $h = 3$; $h_1 = 73$; $\Delta = 0{,}5$; $\Delta_1 = 0{,}6$ mm; $E_1 = E_2 = E = E' = 2{,}2\cdot10^5$ MPa; $G = 8{,}1\cdot10^4$ MPa; $\alpha = 56°$; $\mu = 0{,}17$; $\rho = 9°40'$), using obtained dependencies (5) - (8), (13), (15), (16) i (18), we derive the following

$\delta_{1x} = 2{,}75\cdot10^{-9}\,P$; $\delta_{2x} = 163{,}6\cdot10^{-9}\,P$; $\delta_{3x} = 0{,}412\cdot10^{-9}\,P$; $\delta_{4x} = 0{,}0062\cdot10^{-9}\,P$;

$\delta_{5x} = 330{,}2\cdot10^{-9}\,P$; $\delta_{1б} = 4{,}08\cdot10^{-9}\,P$; $\delta_{2б} = 2{,}05\cdot10^{-9}\,P$; $\delta_{3y} = 0{,}036\cdot10^{-9}\,P^2$ m.

The analysis of obtained results showed that in engineering calculations, the compliance of the needle-wedge system in KO-type single-bed circular knitting machines can be determined with sufficient accuracy using the following equation:

$$\delta' = (\delta'_{2x} + \delta'_{5x})\,tg\alpha, \tag{19}$$

Where: δ'_{5x} is the system compliance due to deformations δ_{2x}, δ_{5x}:

$$\delta'_{2x} = \frac{\delta_{2x}}{P};\quad \delta'_{5x} = \frac{\delta_{5x}}{P}. \tag{20}$$

Substituting the values from (20) into (19), and considering (6) and (13), we derive the following:

$$\delta' = \frac{a^3tg\alpha}{3E_1J_{1x}} + 0{,}25\frac{(htg\alpha - \Delta)ha}{GJ_p}. \tag{21}$$

Using (21) for KO-type circular knitting machines, we derive the following:

$$\delta' = 732 \cdot 10^{-9} \text{ м/Н.}$$

To enhance the compliance of the needle-wedge pair, which ensures the reduction of dynamic loads in the knitting system, the authors proposed a wedge design, the schematic of which is presented in Fig. 1. Unlike the previously known wedges with enhanced compliance of the working surface proposed by the authors earlier [6], this wedge is characterized by simplicity of design, manufacturability and material efficiency.

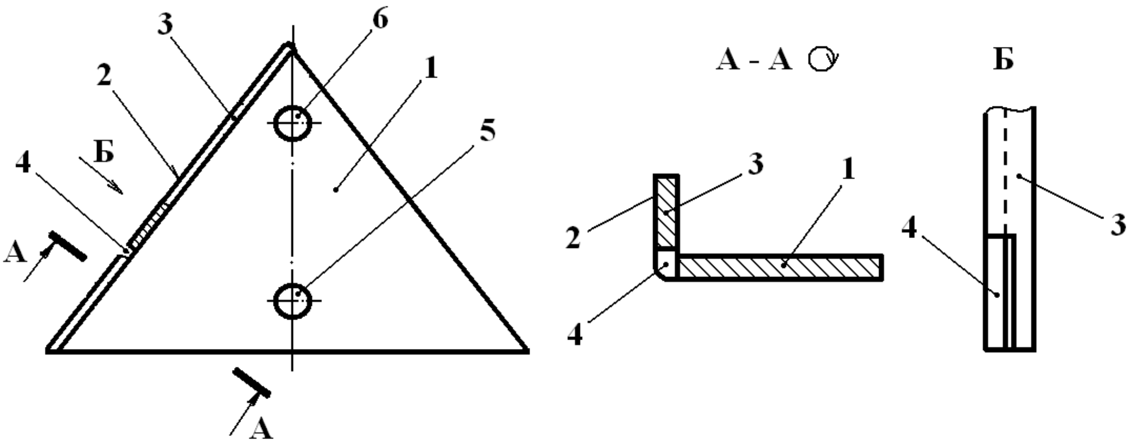

Fig .1. A knitting machine wedge with a bend and an elastic working surface.

The wedge of the knitting machine consists of a body (1), a working surface (2), a bend (3), and a through slot (4). One of the surfaces of the bend (3) serves as the working surface (2), and the through slot (4) is located at the lower part of the bend, parallel to the working surface. The wedge also includes two holes (5, 6) for attaching the body (1) to the locking blocks of the knitting mechanism (not shown in Fig. 1). The body (1) is made of sheet material through stamping.

The wedge operates as follows: When the circular knitting machine is turned on, the needles, installed in the needle cylinder of the knitting mechanism (not shown in Fig. 1), begin to rotate. As the needles encounter the working surface (2) along their path, they move upward (according to Fig. 1) to carry out the technological process of forming loops for the knitted fabric (not shown in Fig. 1). The presence of the through slot (4) reduces the stiffness of the bend (3) in the zone of impact interaction between the needles and the wedge, which results in a decrease in dynamic loads in the needle-wedge pair [2].

Let's evaluate the efficiency of using the proposed wedge design.

In this case, the needle-wedge pair compliance δ' is determined as follows:

$$\delta' = \delta_1' + \delta / P, \tag{22}$$

Where: δ_1' is the compliance of the needle-wedge pair with a rigid wedge;

δ is the deformation of the bend with a slot in the new wedge in the needle impact zone;

P is the maximum impact force of the needle on the wedge (horizontal component).

Given that [2] $\delta_y = \delta_x ctg\alpha$, equation (3) takes the following appearance:

$$\delta = \delta_x (tg\alpha + ctg\alpha) = \frac{2\delta_x}{sin 2\alpha} \cdot \tag{23}$$

Substituting (22) into (23), neglecting technological loads and considering the results of studies [2], we derive the following:

$$\delta' = \frac{2\delta_1' + A^2}{2} \pm \sqrt{\left(\frac{2\delta_1' + A^2}{2}\right)^2 - (\delta_1')^2} \, , \tag{24}$$

Where:

$$A = \frac{\delta_x}{V\sqrt{\frac{m}{K}} sin^2 \alpha} \cdot \tag{25}$$

Using the parameters of the KO-type circular knitting machine [2, 8]: $\delta_1' = 0{,}725 \cdot 10^{-3}$ mm/N; $V = 0{,}71$m/s; $m = 0{,}713 \cdot 10^{-3}$kg; $K = 0{,}418$; $\alpha = 56^0$ and assuming, to avoid disruption of the loop formation process [1], that $\delta_x = 0{,}2$ mm, as based on [2], we derive the following: $\delta' = 36{,}2 \cdot 10^{-3}$mm/N. The maximum impact force of the needle on the bend of the proposed wedge design is 12.2 N, which is approximately 7 times less as compared to the use of existing rigid wedges [8] in KO-type circular knitting machines.

The conclusions of the study and the prospects for further research in this field.

The obtained results prove the feasibility and efficiency of using the proposed elastic wedge design in the knitting mechanism of knitting machines.

The use of the proposed wedge design allows:

- improving the durability of the wedge operation by reducing dynamic loads in the area of its interaction with needles;
- enhancing the quality of knitted fabric by increasing the stability of the needle-wedge pair performance;
- increasing the productivity of the knitting machine by improving the wedge durability.

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