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EXTENDING THE SERVICE LIFE OF CONVEYOR BELTS USED IN FOOD PRODUCTION ENTERPRISES

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Abstract
The article discusses the improvement of technological equipment used in food enterprises, the widespread use of conveyor transport, the study of fatigue wear properties of rubber-fabric belts, and the development of practical recommendations to extend their service life.

Keywords: Drum, belt, frame, semi-finished product, rubber, conveyor, fatigue wear, durability, elongation, layer, tension, diameter, nomogram, roller.

Introduction

The growing global population is leading to an increasing demand for food. This necessitates the expansion of food production enterprises, the creation of new ones, and the improvement of technological equipment used in existing enterprises, making it a pressing issue today.
The launch of new food production enterprises, the widespread use and improvement of conveyor transport, and the enhancement of conveyor belt durability highlight the urgency of solving several technical challenges. One of the most important challenges is studying the fatigue wear properties of rubber-fabric belts and developing practical recommendations to extend their service life.
This is because materials used as raw materials for various types of food products, semi-finished products, and packaged ready-to-eat food must be delivered to consumers intact and without defects. The productivity of the production line depends on how well the transport machines are designed, their durability, and reliability.
In the food industry, belt conveyors, roller conveyors, and overhead conveyors are widely used. Among them, belt conveyors stand out due to their operational efficiency, simplicity of design, and versatility.
Based on previous scientific work, it can be concluded that most studies on belts focus on identifying the maximum stresses in flat sections and additional stresses caused by belt bending over drums and support rollers.
The purpose of the presented study, based on an analysis of previous research, is to improve the methods for calculating the resistance of conveyor belts to variable stresses caused by stretching and bending over drums during the transmission of tensile forces.
The research shows that the compression stresses in the lower layers of the belts completely disappear due to the stresses caused by bending and tensile forces, and the layers only work under tension. There is an optimal number of layers, and increasing the number of layers does not further reduce the stresses.
Based on the analysis of the given relationships and mathematical descriptions, the following formula was proposed to determine the stresses during bending:

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$$\sigma_{и} = \frac{Eh\delta(i-2k+1)}{D}\left(\frac{3G}{E}\right)^{0,01} \quad (1)$$

Where:
E – Elastic modulus of the layer, kgf/cm²
h – Thickness of the rubber-coated layer, cm
δ – Thickness of the layer, cm
i – Number of layers in the belt
k – Layer number, K = 1, 2, 3, ..., n
G – Shear modulus of the rubber, kgf/cm²
D – Diameter of the drum, cm
To determine the tension during the rotation...

$$\sigma_{Po} = \frac{\Delta S}{2iB}\left(\frac{E}{3G}\right)^{0,02} \quad \text{kgf/cm.layer} \quad (2)$$

Where:
ΔS – Tensile force, kgf
B – Width of the belt, cm

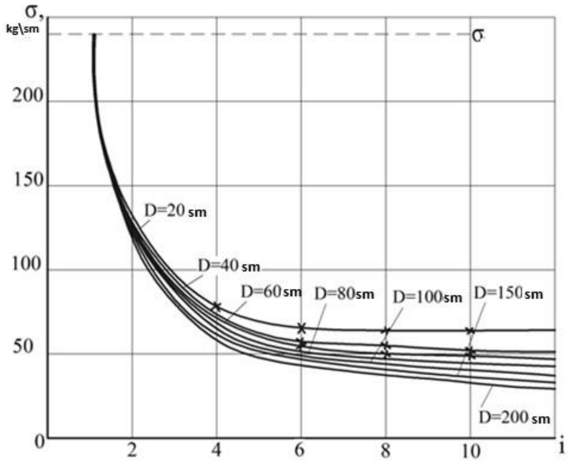

Figure 1. Graph of the stresses in the layers of a nylon-based conveyor belt, related to the number of layers under tension, bending, and tensile forces.

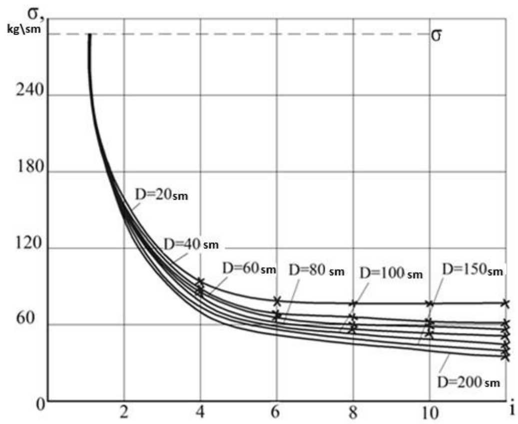

Figure 2. Graph of the stresses in the layers of a nylon-based conveyor belt, related to the number of layers under tension, bending, and tensile forces.

The total stresses from tension, bending, and rotational forces:

$$\sigma_{\Sigma} = \frac{S_P}{Bi} + \frac{Eh\delta(i-2k+1)}{D}\left(\frac{3G}{E}\right)^{0{,}01} + \frac{\Delta S}{2iB}\left(\frac{E}{3G}\right)^{0{,}02} \quad \text{kgf/sm.layer} \quad (3)$$

Where **SP** – Belt tension, kgf.

Optimal number of layers in the belt:

$$i_{opt} = \frac{\sigma_{qo'sh} + \dfrac{Eh\delta(2k-1)\left(\frac{3G}{E}\right)^{0{,}01}}{D} - \sqrt{\left[\sigma_{qo'sh} + \dfrac{Eh\delta(2k-1)\left(\frac{3G}{E}\right)}{D}\right]^2 - 4\left[S_P + \dfrac{\Delta S}{2}\left(\frac{E}{3G}\right)^{0{,}02}\right]}}{\dfrac{2Eh\delta\left(\frac{3G}{E}\right)^{0{,}01}}{D}} \quad (4)$$

The formula applies to the positive value of the expression under the root. The permissible stresses in the nylon layers of conveyor belts can be considered equal to 0.2-0.35 Бr.

Thus, the recommended calculation method allows determining the required number of layers in the belt based on the given permissible stress and a specified tension, as well as the belt width and drum diameter.

Figures 1 and 2 show the changes in stresses in the belt based on the number of layers. The graphs indicate that the lower stress area corresponds to a static strength reserve NnN_nNn in the range from 7 to 9. However, it is generally accepted that the strength reserve of belts is much higher. This, in turn, leads to an unreasonable increase in the number of layers and, consequently, an increase in the cost of the belt.

According to the strength of the layers, it is possible to use a six-layer belt instead of an eight-layer one. However, it should be noted that in the conditions of transporting large-sized materials, the criterion for selecting the thickness of the belt may be the stresses that occur when large pieces come into contact with the loading section.

Calculation of the Maximum Tension of Conveyor Belts Based on Fatigue Strength Conditions

A method for calculating the durability of conveyor belts has been proposed, taking into account the fatigue phenomena that arise from the interaction between the driving and tensioning drums, as well as the support rollers [1].

The variable stresses resulting from the transmission of bending, tensile, and pulling forces can serve as criteria for selecting the diameters of the driving and tensioning drums and the tension in the belt throughout all parts of the conveyor. If the belt operates under variable stresses that do not exceed the fatigue limit, layer separation of the belt will not occur during operation.

Taking into account the conditions of tension, bending, rotational forces, and temperature, the initial tension in the belt can be determined using the following formula:

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$$S_{max} = \left[\sigma_r\ K_t\ K_{Po} - (\sigma_{Po} + \sigma_u)\right]\frac{B\ i}{n_\delta} \quad (5)$$

Here:
Ϭr – Fatigue limit of the belt under tension, bending, and rotational forces, kgf/cm.layer;
Kt – Coefficient accounting for temperature conditions;
KPo – Coefficient accounting for rotational forces;
ϬPo – Stress resulting from rotational forces, kgf/cm.layer;
Ϭu – Stress resulting from bending, kgf/cm.layer;
i – Number of layers;
nδ – Safety limit for resistance to bending and rotational forces in the drums;
The coefficients **Kt** and **KPo** are determined from the stresses **Ϭu** and **ϬPo** using formulas (1) and (2).
The permissible stress of the belt under the combined influence of pulling force, bending, and tension on the driving drum is proposed to be determined using the following formula:

$$S^1_{max} = (\sigma_r k_t k_{Po} - \sigma_u)\frac{Bi}{n_\delta}$$ **kgf.** \quad (6)

If the stresses in the belt exceed the fatigue limit, the calculations should be carried out based on the relationship (7) for limited durability [2]:

$$\sigma_\Sigma^m N = C \quad (7)$$

where
NNN – the number of loading cycles until the belt failure;
σΣ\sigma_\Sigmaσ Σ – the total stress in the belt resulting from bending, stretching, and centrifugal forces;
mmm and CCC – coefficients describing the fatigue properties of the belt.
Calculating durability allows for the prevention of premature separation under operating conditions, the diameter of the drums, and the unreasonable increase in the strength limit of the belt.

Calculation of the stress in the leading part of the conveyor belt under the condition that it does not exceed the allowable bending on the supporting rollers.

During the movement of the belt under load on the supporting rollers, shear stresses arise in the rubber coverings. The maximum shear stresses can be found using formula (5). However, it is necessary to first determine the bending radius on the supporting rollers and the wrap angle.
By analyzing the relationships presented by O.G. Karbasov and G.T. Kodik (6) and (7), and taking into account the stiffness of the belt, the bending between and around the supporting rollers under distributed and directed load effects is determined. In the research works of V.S. Bondarev, V.I. Kovalenko, and V.P. Golovan, the stiffness of the conveyor belt is not considered. Using relationships (7), taking into account the stiffness of the belt, the maximum values of the wrap angles of the supporting rollers are determined: [3,4,5]

"For flat belts -" $$\cos\alpha = 1 - \frac{Y}{R} = 1 - \frac{\frac{qt^2}{8S} + \frac{Pt}{4S} - \frac{n_4\left(qt + P\right)\sqrt{EJ}}{2S\sqrt{S}}}{R} \quad (8)$$

"For crowned belts -" $$\cos\alpha = 1 - \frac{Y}{R} = 1 - \frac{\frac{qt^2}{8S} + \frac{Pt}{4S} - \frac{n_4 n_5\left(qt + P\right)\sqrt{EJ}}{2S\sqrt{S}}}{R} \quad (9)$$

After determining the wrap angles, the slip tensions arising in the belt during bending can be calculated using formula (7).

The tension in the belt on both sides of the supporting roller can be assumed to be approximately the same. The values of the resistance coefficients for the rotation and rolling of the roller's bearing units vary between 0.02 and 0.05, so the change in belt tension around one supporting roller is negligible and can be disregarded.

Based on expressions (8) and (9), we determine the allowable minimum tension of the working part of the belt on the last drum under the allowable bending conditions of the supporting rollers. Let SSS be considered an unknown function of K, and we expand it into a Maclaurin series based on the powers of K:

$$S_{\min} \geq \left\{\frac{t\left(qt + 2P\right)}{8Y} + \frac{K}{2Y}\left[\frac{8Y}{t\left(qt + 2P\right)}\right]^{\frac{1}{2}}\left\{1 - \frac{K}{4Y}\left[\frac{8Y}{t\left(qt + 2P\right)}\right]^{\frac{1}{2}}\right\}\right\}\frac{Bn_p}{B_1} \quad (10)$$

We will determine:

- For a smooth belt;
- For a corrugated belt;
- The distance between the supporting rollers, cm;
- q – uniformly distributed load along the distance between the supporting rollers, kgf/m;
- P – directed static load equal to the weight of large pieces, kgf;
- B– width of the belt, cm;
- R – radius of the supporting rollers, cm;
- [α] – allowable maximum angle of contact between the supporting rollers and the belt, degrees;
- npnpnp – safety factor for bending on the supporting rollers, np=1.5
- B1 – width of the cargo piece, cm.

The angle value can be obtained using formula (5), based on the results of fatigue tests for stretching and bending of the belt.

Selection of the Diameters of Driving and Tensioning Drums and the Optimal Tensioning of the Belt

Based on research, graphs of longitudinal stresses in the fabric covering of belts and slip stresses in rubber coatings have been obtained for various values of drum diameters, where the initial tensions of the belts and the number of layers are different. From the graphs, it can be seen that as the diameters of the drums change, the distribution of normal stresses between the layers

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varies along the height of the belt cross-section. Similar changes also occur in the slip stresses in the rubber coatings.

When comparing graphs obtained for belts of the same type but with different drum diameters, it can be noted that due to the uneven distribution of tension stresses at certain diameters, there is a decrease in the associated slip stresses.

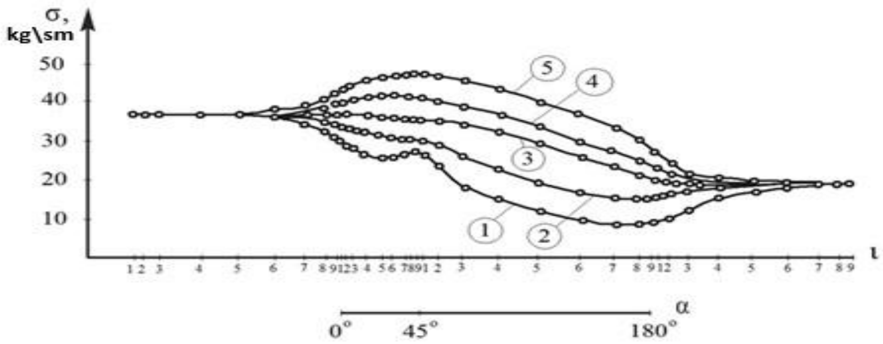

Figure 3. Graph of the Variation of Tensile Stresses Along the Coverage of a Five-Layer Conveyor Belt (Ϭ=15 kg/cm², E=2500 kg/cm², H=0.19 cm, D=20 cm, δ=0.15 cm², i=5, α=1800, β=450, So=140 kg, Fтр=0.3)

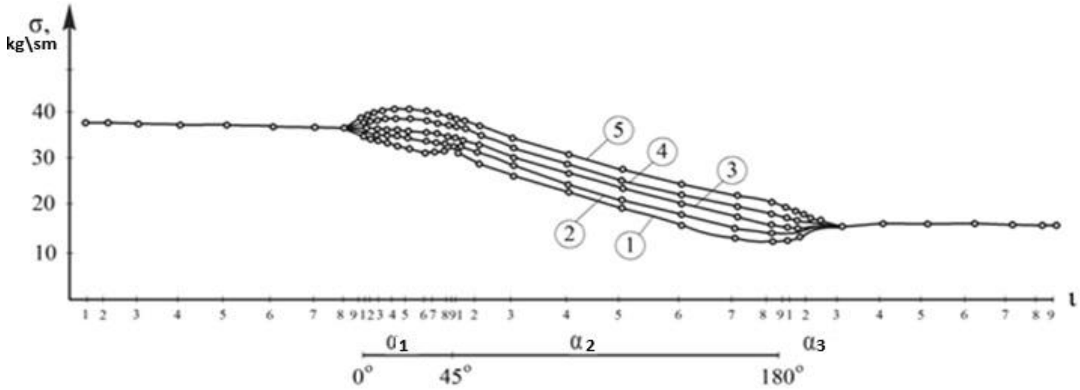

Figure 4. Graph of the Variation of Tensile Stresses Along the Coverage of a Five-Layer Conveyor Belt (Ϭ=15 kg/cm², E=2500 kg/cm², H=0.19 cm, D=60 cm, δ=0.15 cm², i=5, α=1800, β=450, So=140 kg, Fтр=0.3)

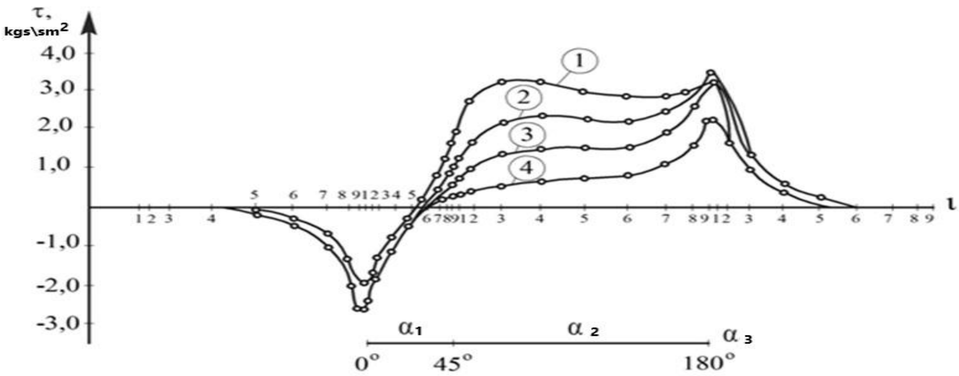

Figure 5. Graph of the Variation of Shear Stresses Along the Coverage of a Five-Layer Conveyor Belt Under the Combined Influence of Tensile, Bending, and Tension Forces (Ϭ=15 kg/cm², E=2500 kg/cm², H=0.19 cm, D=20 cm, δ=0.15 cm², i=5, α=1800, β=450, So=140 kg, Fтр=0.3)

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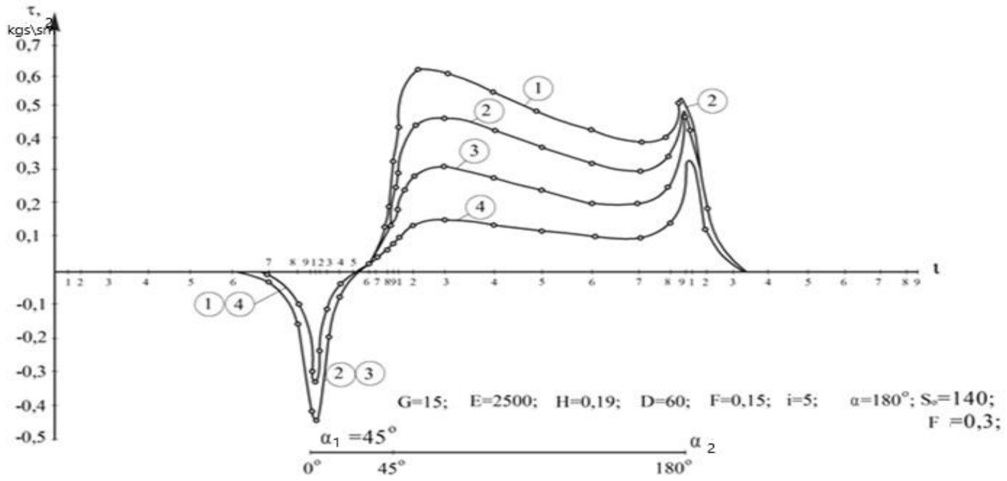

Figure 6. Graph of the Variation of Shear Stresses Along the Coverage of a Five-Layer Conveyor Belt Under the Combined Influence of Tensile, Bending, and Tension Forces (Ϭ=15 kg/cm², E=2500 kg/cm², H=0.19 cm, D=60 cm, δ=0.15 cm², i=5, α=1800, β=450, So=140 kg, Fтр=0.3)

The main reason for the failure of conveyor belts due to fatigue is the excessive loading of individual layers or rubber coatings resulting from the uneven distribution of loads between the layers. Therefore, the information regarding the variation of stresses (Figures 7 and 8) can serve as a criterion for selecting the diameters of the drums.

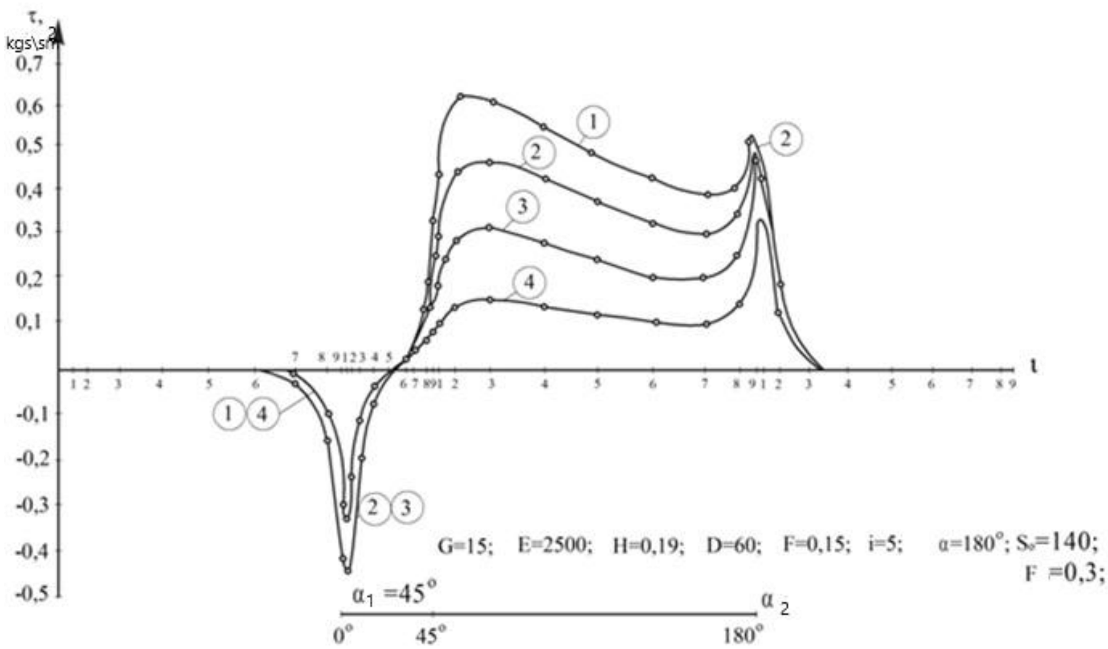

Figure 7. Graph of the Variation of Shear Stresses Along the Coverage of a Five-Layer Belt Under the Combined Influence of Tensile, Bending, and Rotational Forces (Ϭ=15 kg/cm², E=2500 kg/cm², H=0.19 cm, D=160 cm, δ=0.15 cm², i=5, α=1800, β=450, So=140 kg, Fтр=0.3)

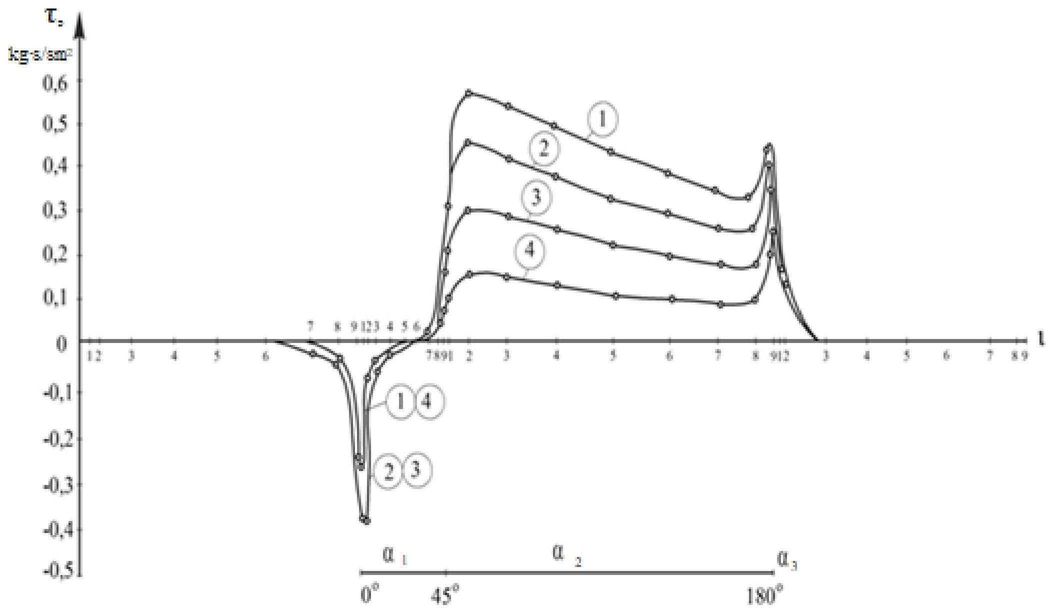

Figure 8. Graph of the Variation of Shear Stresses Along the Coverage of a Five-Layer Belt Under the Combined Influence of Tensile, Bending, and Rotational Forces (Ϭ=15 kg/cm², E=2500 kg/cm², H=0.19 cm, D=140 cm, δ=0.15 cm², i=5, α=1800, β=450, So=140 kg, Fтр=0.3)

For instance, from the graphs in Figures 9 and 10, it can be observed that the stresses between layers are most evenly distributed when the drum diameter is 1400 mm. In these conditions, the shear stresses in the rubber coating are minimal due to the reduction of differences in loading between the fabric layers. Therefore, a drum diameter of 1400 mm is considered optimal for this loading mode.

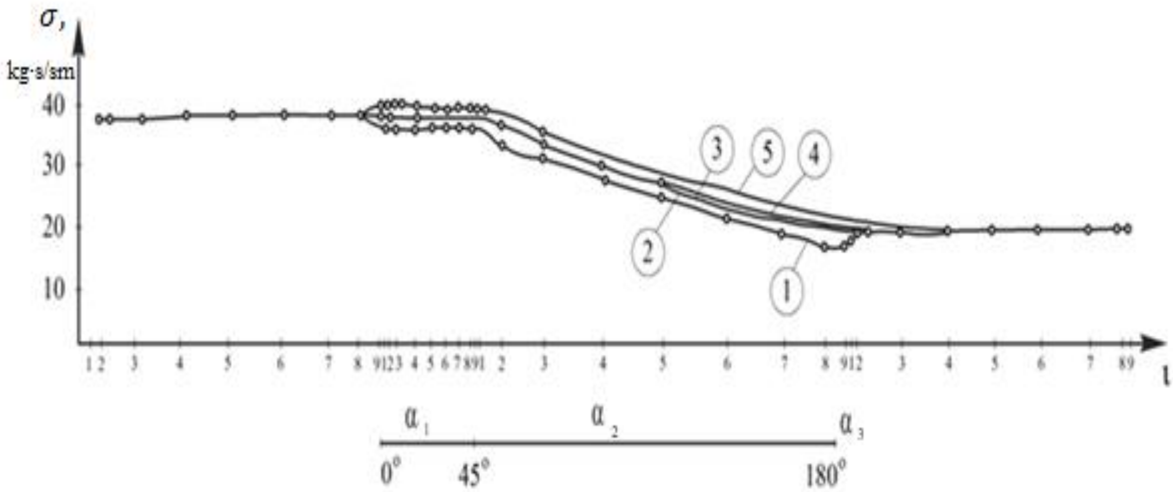

Figure 9. Graph of the Variation of Tension Stresses Along the Coverage of a Five-Layer Conveyor Belt (Ϭ=15 kg/cm², E=2500 kg/cm², H=0.19 cm, D=160 cm, δ=0.15 cm², i=5, α=1800, β=450, So=140 kg, Fтр=0.3)

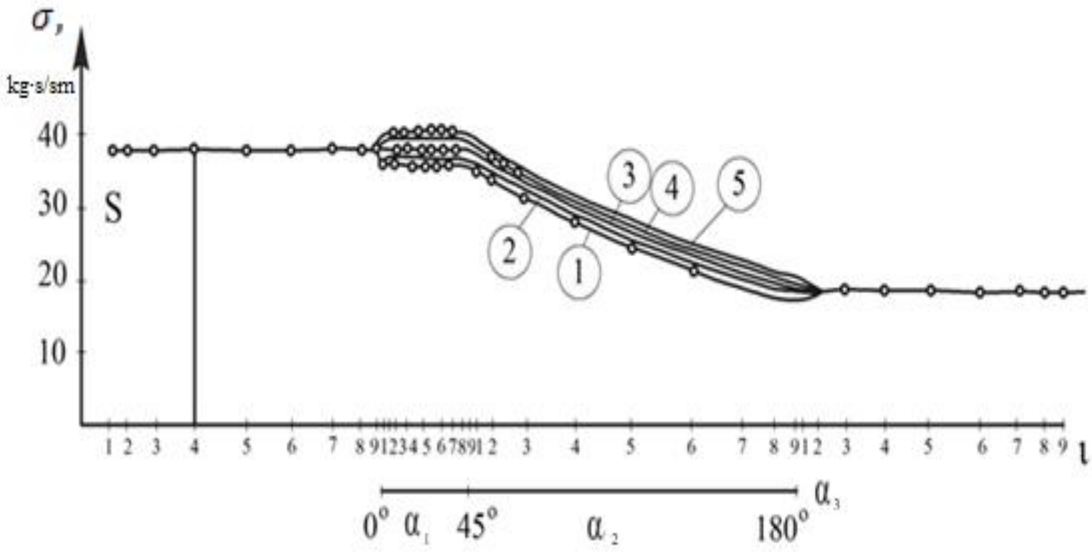

Figure 10. Graph of the Variation of Tension Stresses Along the Coverage of a Five-Layer Conveyor Belt (Ϭ=15 kg/cm², E=2500 kg/cm², H=0.19 cm, D=140 cm, δ=0.15 cm², i=5, α=1800, β=450, So=140 kg, Fтр=0.3)

Using the graphs, it is possible to determine the optimal values of drum diameters under different loading conditions.

As a result of analyzing the data on stresses in multilayer conveyor belts calculated with the help of electronic digital computing machines (EDCM), a relationship was established between the optimal values of drum diameters and the number of layers in the belt. For practical use, the graph in Figure 4 is proposed.

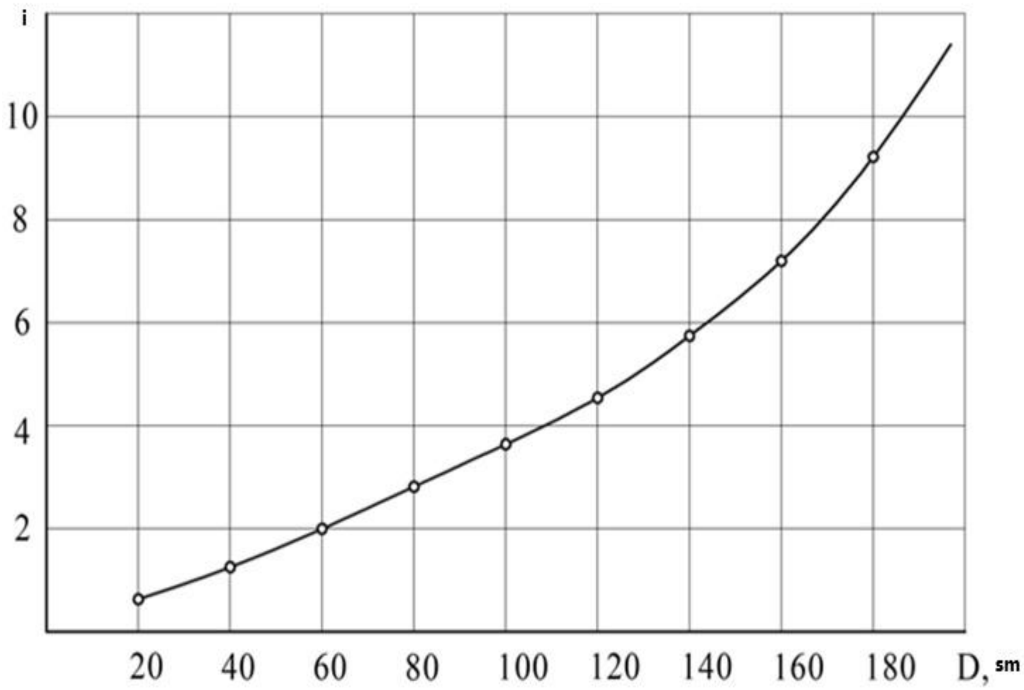

Figure 11. Graph of the Relationship Between the Number of Belt Layers and Drum Diameter

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The uniform distribution of tension in the layers serves as only one criterion for selecting drum diameters.

For conveyors, it is necessary to establish the tension of the belt, taking into account bending and rotational force transmission.

To achieve this, nomograms shown in Figures 12 and 13 were created. The relationship between the maximum tensile stresses and slip stresses for five-layer belts based on nylon is defined by the diameters of the drums, rotational forces, and the initial tension values of the belt.

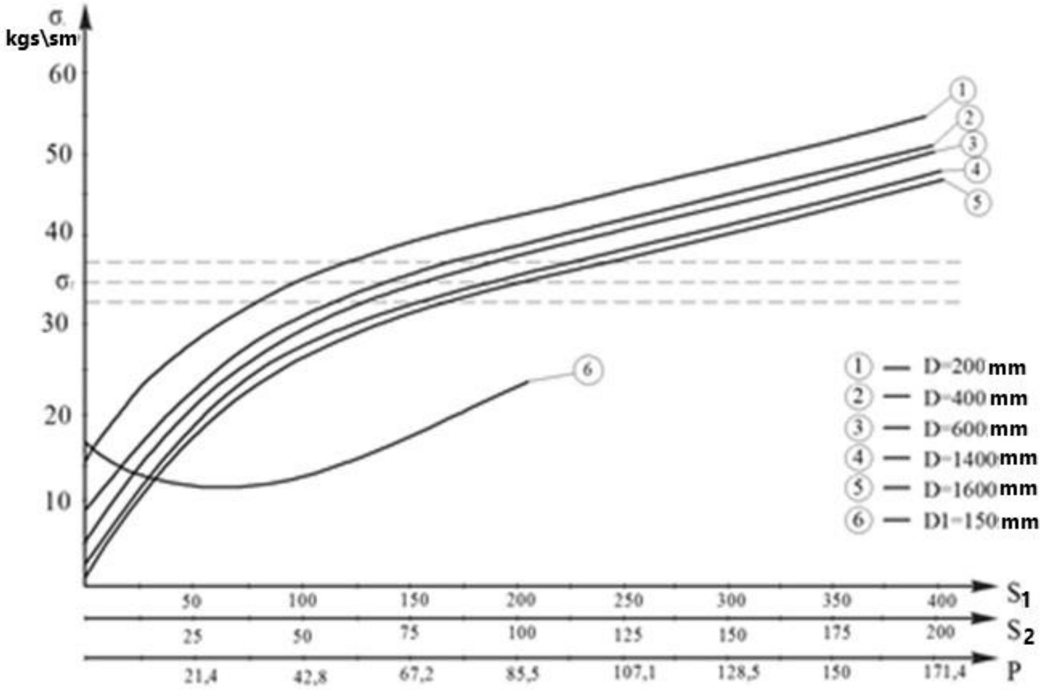

Figure 12. Nomogram of the Relationship Between Drum Diameters, Initial Belt Tension, and Tensioning Force.

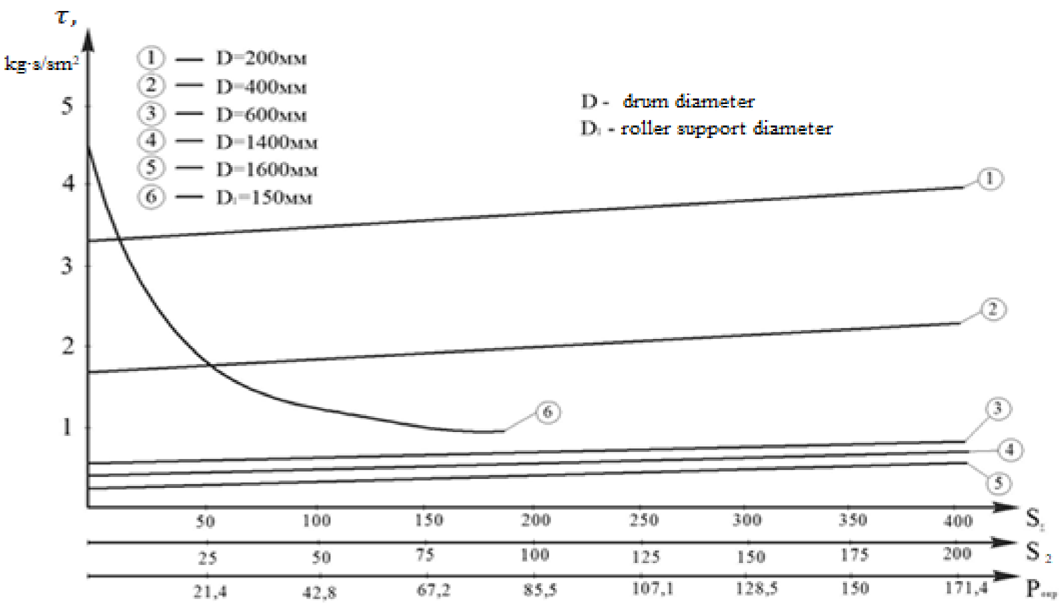

Figure 13. Nomogram of the Relationship Between Drum Diameter, Initial Belt Tension, and Tensioning Force.

The relationship graphs of the tensions generated by the tension of the belt during its movement on the supporting rollers are also provided. The tension value in the tensioning section of the belt can be determined through calculations or experimentally.
Using the nomogram, knowing the required tensioning force PailP_{\text{ail}}Pail, one can select the initial tension of the belt SΣS_{\Sigma}SΣ and the minimum diameter of the driving drum.
From the graph of the tensions in the fabric layers on the supporting rollers, it can be seen that they are insignificant. However, it is necessary to consider that the belt may wear out due to local buckling when transporting large-sized materials with sharp edges. In such cases, it is advisable to use pre-crushed material or special belts with breaker fabric layers.
From the nomogram, it can be seen that when the tension of the belt is low, the sliding tensions in the rubber layer of the supporting rollers significantly increase. Therefore, it is necessary to check the belts under sliding tension conditions.
Next, we consider an example of calculating the durability of a conveyor belt for a complex operating in the Kursk magnetic anomaly quarries. The conveyor length L=1000L = 1000L=1000 m; drum diameters: driving (2 drums) Db=1600D_b = 1600Db=1600 mm, bending drums – 1400 mm and 1000 mm, tensioning drum – 1250 mm, last drum – 1500 mm; diameters of supporting rollers d6=159d_6 = 159d6=159 mm, spacing distance l=1l = 1l=1 m; two-drum drive power 2520 kW; conveyor productivity Qtech=3÷4Q_{\text{tech}} = 3 \div 4Qtech=3÷4 thousand m³/hour; belt speed V=5.24V = 5.24V=5.24 m/sec. The conveyor is equipped with a five-layer PAS 500 belt based on nylon, with a sliding module G=15G = 15G=15 kgf/cm²; tension module E=2500E = 2500E=2500 kgf/cm², layer thickness 2 mm, width 2000 mm.
Calculating the tension in the belt yields the following tension values:
Smax=38000S_{\text{max}} = 38000Smax=38000 kgf, S2=31200S_2 = 31200S2=31200 kgf, S3=27200S_3 = 27200S3=27200 kgf, S4=22800S_4 = 22800S4=22800 kgf, S5=22400S_5 = 22400S5=22400 kgf, S6=18000S_6 = 18000S6=18000 kgf, and the required tension amounts P1=16280P_1 = 16280P1=16280 kgf, P2=13400P_2 = 13400P2=13400 kgf.
To refine the obtained parameters based on fatigue resistance conditions, the tension values are recalculated with respect to the width of 1 cm of the belt.
The relative tension of the belt...

$$S^1_{max1}=(38000\backslash 200)=190 \text{ kgk}\backslash\text{sm}$$
$$S^1_2=(31200\backslash 200)=156\text{kgk}\backslash\text{sm}$$
$$S^1_3=(27200\backslash 200)=136 \text{ kgk}\backslash\text{sm}$$
$$S^1_4=(22800\backslash 200)=114 \text{ kgk}\backslash\text{sm}$$
$$S^1_5=(22400\backslash 200)=112 \text{ kgk}\backslash\text{sm}$$
$$S^1_6=(1800\backslash 200)=90 \text{ kgk}\backslash\text{sm}$$

$$P^1_1=81{,}4 \text{ kgk}\backslash\text{sm}$$
$$P^1_2=67 \text{ kgk}\backslash\text{sm}$$

Relative rotational force The correctness of selecting drum diameters under the condition of uniform distribution of stress between the layers is provided by the following. According to the nomogram in Figure 12, the relative tension S'{max} and relative rotational force P'{1} correspond to the diameters of the drive drums D'{1} = 1600 mm. The values S'{2} and P'{2} correspond to D'{2} = 1600 mm; S'{3} = 136 kgf/cm corresponds to D'{3} = 1200 mm; S'{4} = 114 kgf/cm corresponds to D'{4} = 900 mm; S'{5} = 112 kgf/cm corresponds to D'{5} = 850 mm; and S'{6} = 90 kgf/cm corresponds to D'{6} = 600 mm.

The maximum values of the initial tension of the belt can also be determined using formulas 5 and 6. It has been established that it is possible to determine and define the optimal values of the belt tension that do not exceed the fatigue limit during the bending stresses in the drums and supporting rollers.

A method for determining the optimal parameters of tension, the number of layers, and drum diameters, which minimizes the stresses occurring in the belt, has been proposed. A method for calculating the durability of conveyor belts has been suggested, which allows for preventing their premature failure due to wear caused by bending in the drums and rollers. The proposed calculation method, in addition to determining the necessary dimensions of the belt drums and supporting rollers, allows for estimating the service life of the belt in cases where the actual stresses exceed those corresponding to the fatigue limit.

The diameter of the drive drum and the initial tension of the belt at a given rotational force are determined according to the nomogram, which allows for achieving uniform stress distribution between the layers and limiting these stresses to the fatigue limit.

Evaluating the fatigue resistance of conveyor belts solely under stretching and bending conditions in drums does not sufficiently describe the operational performance of the belt. The transmission of the pulling force from the driving drum must also be taken into account.

A complex stress state arises in a conveyor belt operating under conditions of bending, stretching, and the transmission of rotational force from the drum. An increase in the sliding stresses in the layers of the belt close to the drum surface is observed. The stretching stresses in the belt layers also increase.

The resultant stresses caused by the transmission of the pulling force significantly affect the wear resistance of the belt. Therefore, it is essential to consider the rotational force when assessing the bending stresses of the belt.

During the operation of the conveyor belt on the drum surface, compressive stresses arise between the layers. It can be assumed that these stresses, combined with the sliding stresses, will lead to the fatigue and failure of the rubber layers of the belt.

The stress state in the belt frame was studied using the method of integral equations, based on which the stretching and sliding stresses in the fabric layers for various drum diameters, the initial tension of the belt, and the rubber layers were obtained. These can serve as criteria for calculating the optimal values of the belt tension, rotational forces, and the diameters of the drive drums.

A nomogram of the maximum tensile and sliding stresses, graphically connected to the drum diameters, rotational forces, belt tension, and fatigue limit values, has been constructed. A formula has been proposed for determining the initial tension of the belt during the operation of the conveyor, ensuring that the variable stresses in the belt do not exceed its fatigue limit.

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References

1. Тожиев, Р. (2024). Схема экспериментов по умерщвлению куколок тутового шелкопряда. Новый Узбекистан: наука, образование и инновации, 1(1), 711-714.
2. Monastyrskiy, D.Sh. Investigation of the Uneven Distribution of Tensions Between the Layers of Conveyor Belts Used in the Coal and Mining Industries. Author's abstract of the dissertation submitted for the degree of Candidate of Technical Sciences. Moscow, 1967.
3. Golovany, V.P. Investigation of Fatigue Wear of Conveyor Belts Under Bending on Drums and Supporting Rollers. Dissertation, Kyiv, 1970.
4. Chentsov, V.F., Golovany, V.P., Tojiyev, R.J. Calculation of Conveyor Belts for Endurance. Progressive Designs, Research and Calculation of Conveyor Belts. Abstracts of Reports at the Scientific and Technical Conference, Sverdlovsk, 1972.
5. Golovany, V.P., Tojiyev, R.J. Analysis of Stresses in Conveyor Belts on Drums During the Transmission of Traction Force. In: Mining, Construction and Road Machines, Issue 19, Kyiv, "Tekhnika", 1975.
6. Tojiyev, R., Isomidinov, A., Alizafarov, B. Strength and Fatigue of Multilayer Conveyor Belts Under Cyclic Loads. Turkish Journal of Computer and Mathematics Education, 2021, Vol. 12, No. 7, pp. 2050-2068.
7. Rasuljon, Tojiyev, and Alizafarov Bekzod. Theoretical Research of Stress in Rubber-Fabric Conveyor Belts. Universum: Technical Sciences, 4-12 (97) (2022), pp. 5-16.
8. Tojiyev, R., Alizafarov, B., & Muydinov, A. Theoretical Analysis of Increasing Conveyor Tape Endurance. Innovative Technologica: Methodical Research Journal, 3(06), 167-171, 2022.
9. Ergashev, M., Tojiev, R. J., Komilov, N. M., & Ishmuradov, S. U. (2024). Results of Multifactor Experiments Conducted on Based Parameters of Combined Machine Disc Softeners. Naturalista campano, 28(2), 271-276.
10. Rasuljon, T., Azizbek, I., & Nargiza, R. (2024). Study of the effect of dryer hydraulic resistance on material temperature. Universum: технические науки, 6(6 (123)), 54-60.
11. Rasuljon, T., Abdurakhmon, S., & Ulugbek, U. (2024). Mathematical modeling of the acceptable parameters of the plate scrubber. Universum: технические науки, 4(1 (118)), 12-18.
12. Tojiev, R. J., & Isomidinov, A. S. (2024). Application of the principle of gas dynamics to agricultural technologies: monograph. Publishing house «UKRLOGOS Group», 72-72.
13. Rajabova, N., & Rasuljon, T. (2023). Impact of disperse materials internal structure to drying process. In НАУКА И ТЕХНОЛОГИИ-2023 (pp. 10-21).
14. Tojiyev, R., Nargizakhon, R., & Abdusamad, M. (2023). Influence of the internal structure of the building material on the drying process. American Journal of Technology and Applied Sciences, 13, 8-13.
15. Rasuljon, T., & Nargizaxon, R. (2022). Impact on the internal structure of materials to drying process. Universum: технические науки, (10-6 (103)), 10-18.
16. Tojiyev, R. J., Yusupov, A. R., & Rajabova, N. R. (2022). Qurilishda metrologiya, standartlash va sertifikatlashtirish [Matn]: darslik. Toshkent:«Yosh avlod matbaa, 464.
17. Jumaboevich, T. R., & Rakhmonalievna, R. N. (2022). Installation for drying materials in a fluidized bed. Innovative Technologica: Methodical Research Journal, 3(11), 28-36.

18. Тожиев, Р. Ж., Миршарипов, Р. Х., Ражабова, Н. Р., & Муллажонова, М. М. (2022). Оптимизация существующей конструкции сушильного барабана.
19. Tojiyev, R., & Rajabova, N. (2021). Experimental study of the soil crust destruction mechanism. Scientific progress, 2(8), 153-163.
20. Тожиев, Р. Ж., & Эркабоев, Х. Ж. (2023). Технология глубокого рыхления подпахотного слоя почвы с использованием энергии детонационной волны. In наука, общество, технологии: проблемы и перспективы взаимодействия в современном мире (pp. 58-70).
21. Rasuljon, T., Sulaymanov, A., Madaminova, G., & Agzamov, S. U. (2022). Grinding of materials: main characteristics. International Journal of Advance Scientific Research, 2(11), 25-34.
22. Rasuljon, T., Isomiddinov, A., Ortiqaliyev, B., & Khursanov, B. Z. (2022). Influence of previous mechanical treatments on material grinding. International Journal of Advance Scientific Research, 2(11), 35-43.
23. Тожиев, Р. Ж., Юнусалиев, Э. М., & Абдуллаев, И. Н. (2022). Газодетонационный метод мониторинга сейсмостойкости эксплуатируемых зданий и сооружений.
24. Jumabayevich, T. R., & Maxamadovich, S. A. (2022). Hydrodynamics of a scrubber for wet cleaning of powder gas generations. Механика и технология, 2(7), 120-127.
25. Тожиев, Р. Ж., Ахунбаев, А. А., & Миршарипов, Р. Х. (2022). Барабанли қуритгичда иссиқлик агенти тезлиги ва маҳаллий қаршилик коэффициентларини тажрибавий тадқиқ этиш. Механика и технология, 2(7), 159-165.
26. Тожиев, Р. Ж., Ахунбаев, А. А., & Миршарипов, Р. Х. (2022). Минерал ўғитларни конвектив қуритиш жараёнини оптималлаштириш. Механика и технология, 2(7), 127-135.
27. Ахунбаев, А. А., Миршарипов, Р. Х., Йигиталиев, М., & Тожиев, Р. Ж. (2022). Высушивание дисперсных материалов в кондуктивном аппарате.
28. Rasuljon, T., Voxidova, N., & Khalilov, I. (2022). Activation of the Grinding Process by Using the Adsorption Effect When Grinding Materials. Eurasian Research Bulletin, 14, 157-167.
29. Karimov, I., Tojiyev, R., Madaminova, G., Ibroximov, Q., & Xamdamov, O. T. (2021). Hydrodynamics of wet dush powder black drum equipment. Barqarorlik va yetakchi tadqiqotlar onlayn ilmiy jurnali, 1(5), 49-56.
30. Sadullaev, X., Tojiyev, R., & Mamarizaev, I. (2021). Experience of training bachelor-specialist mechanics. Barqarorlik va yetakchi tadqiqotlar onlayn ilmiy jurnali, 1(5), 116-121.