**Analysis of the Influence of the Tension Roller Radius on the Force of Interaction with the Belt with Its Variable Tension**

Baxtiyordjan Davidbayev, Yunus Mirzakhanova), Nargizakhon Davidboeva, Khojiakbar Ruzaliev

*Fergana State Technical University, Fergana, Uzbekistan*

*a)Corresponding author:* [*yunus1965@gmail.ru*](mailto:yunus1965@gmail.ru)

**Abstract.** The article considers a belt transmission with a composite tension roller, where the forces of interaction between the roller and the belt are determined, as well as specific recommended transmission parameters. The article presents a structural diagram and operating principles of the centering tension device of a belt transporter. Based on theoretical studies, the parameters of the centering device of a belt transporter are substantiated. Production tests substantiate the technical and economic indicators of a belt transporter with a recommended centering tension device.

**Keywords:** driver, driven, pulley, branches, connection, flexible, transmission, force, rigid, angle, velocity, tape, technological, acceleration, tension roller, analysis.

**INTRODUCTION**

Usually in belt drives with constant belt tension, the gear ratio is constant. In this case, the tension of the driver and driven branches is also constant. However, in technological machines the load in the belt drive will be variable [1]. In this case, the tension of the belt branches will also change. In order to maintain the belt tension within certain limits and increase its service life, we recommend a belt drive, the tension roller of which is made composite with an elastic element.

The tremendous growth of scientific knowledge over the past 50 years has resulted in an intense pressure on the engineering curricula of many universities to substitute “modern” subjects in place of subjects perceived as weaker or outdated. The result is that, for some, the kinematics and dynamics of machines has remained a critical component of the curriculum and a requirement for all mechanical engineering students, while at others, a course on these subjects is only made available as an elective topic for specialized study by a small number of engineering students. Some schools, depending largely on the faculty, require a greater emphasis on mechanical design at the expense of depth of knowledge in analytical techniques. Rapid advances in technology, however, have produced a need for a textbook that satisfies the requirement of new and changing course structures [2-5].

**METHODS**

This article is intended to cover that field of engineering theory, analysis, design, and practice that is generally described as mechanisms or as kinematics and dynamics of machines. Although this text is written primarily for students of mechanical engineering, the content can also be of considerable value to practicing engineers throughout their professional careers. To develop a broad and basic comprehension, the text presents numerous methods of analysis and synthesis that are common to the literature of the field. The authors have included graphic methods of analysis and synthesis extensively throughout the book, because they are firmly of the opinion that graphic methods provide visual feedback that enhances the student’s understanding of the basic nature of, and interplay between, the underlying equations [6-7]. Therefore, graphic methods are presented as one possible solution technique, but are always accompanied by vector equations defined by the fundamental laws of mechanics, rather than as graphic “tricks” to be learned by rote and applied blindly. In addition, although graphic techniques, performed by hand, may lack accuracy, they can be performed quickly, and even inaccurate sketches can often provide reasonable estimates of a solution and can be used to check the results of analytic or The authors also use conventional methods of vector analysis throughout the book, both in deriving and presenting the governing equations and in their solution.

In mechanical engineering, belt transporters are used to transport various goods. At the same time, depending on the loading of the transporter belt, in particular, the change in the frequency and amplitude of the change in the transporting load, the parameters of the belt are selected.

Due to the uneven load on the belt in transporters, the belt jumps out of the pulley and guide rollers. Therefore, it is considered expedient to center the tape to eliminate the sideslip of the tape.

The well-known design of the centering lever-hinged tension roller contains automatic adjusting devices with asymmetric configurations to eliminate the lateral exit of the tape or belt.

Another design of the centering lever-hinged tension roller is equipped with an additional pair of intersecting bracket-shaped connecting rods, each of which is hingly connected by one end to the other end of the corresponding mentioned connecting rod with an attached spring holder, and the other end to the shackle of the other part of the tension roller, and the axes of the hinged connections of the connecting rods with each other and the connecting rods with shackles are located parallel to each other[7]. The disadvantage of these structures is the limited functionally opportunity and low reliability: due to the large pulling force at high speed conditions, automatic control devices and limitness rotation speed of the tension roller; reducing the accuracy of transferring movement between the rollers due to high values of reaction forces in the kinematic pairs of connecting rods and in the supports of the tension roller, and others.

The proposed design with a tension roller lies in the fact that with an increase in the shear force, one or another part of the roller moves axially in the direction of the shear force, thereby increasing the working contact area of the roller surface with the transporter belt and, due to the concave curved surface of the roller parts, the belt descent is eliminated from the drums. At the same time, due to the selection of the required stiffness of the tension and compression springs, the roller automatically adapts to changes in the asymmetric forces of the belt coming off the belt transporter.

In another known design of the centring tension roller in order to increase the functionality by increasing the reliability of operation, it is provided with a spring attachment with at least one elastic element connected to the links of the intermediate device or to the corresponding shackle of one of the half of the tension roller, and the driven roller is installed with the possibility axial displacements. The disadvantage of this design is the low reliability of the system due to large deformation of the springs and a decrease in elasticity effects at high rates of withdrawal forces. This leads to instability and a decrease in the reliability of the system.

In the centring lever-hinge roller, the connecting rods are additionally connected to each other by telescopic connecting rods and spring attachments. The disadvantage of this mechanism is the complexity of the design and the limited speed modes of operation.

The task of the new design is to increase the reliability of the transmission of the elimination of lateral mixing of the transporter belts from the central axis.

In belt transporters with a tension roller, containing a driving and driven drums, a tension roller with a symmetrical curved profile and a transporter belt covering them, the roller is made of two symmetrical parts with concave curved surfaces connected by a tension spring. In addition, the half-shafts of the roller parts can perform rotational and reciprocating movements along the axis and, accordingly, have compression springs between the roller parts and the body (worn on the half-shafts of the roller parts). Plastic bushings are fixed at the inner ends of the roller parts.

**RESULTS AND DISCUSSION**

In this work, a new design of a centring tension roller has been developed, with a new adjusting device with a flexible element. This recommended new design serves to prevent the belt from coming off the pulley sideways or for production to prevent the belt from coming off the side of the drum. A new design has been developed and the operating principle of the design with a description has been proposed.

Knowing that the power and rotational speed of the electric motor are known, we can determine the calculated speed of movement for different diameters of the drive pulley:

,  (1)

where D1, D2 are the diameters of the driving and driven pulleys. [3].

The circumferential force Ft and the resultant FB force in the belt drive according to the kinematic diagram shown in Fig. 1 are determined by the formulas:

 (2)

where P – is the power consumption of the electric motor;

K – is the coefficient taking into account the type of belt.

According to the kinematic diagram in Fig. 1, the shear force arising from the non-parallelism of the axes of rotation of the pulleys is determined

 (3)

where P is the angle of deviation of the axis of rotation of the driven pulley relative to the axis of the driving pulley.

In the future, we will consider the derivation of the formula for determining the shear force for the raw cotton loader when the drum axes are not parallel. Shear force in the transporter of the raw cotton loader.

 (4)

where U.W.C. transporter drive;

V – linear speed of the transporter belt;

PT - required power on the drum shaft:

 (5)

where λ is a coefficient taking into account the nature of the transporter operation .

The torque is determined according to:

 (6)

where cd, ld - tension forces in the incoming and outgoing branches of the transporter. Table 1 shows similar calculations for determining the shear force in a composite tension roller with an automatic regulating device for a flat-belt transmission according to (3) [4].

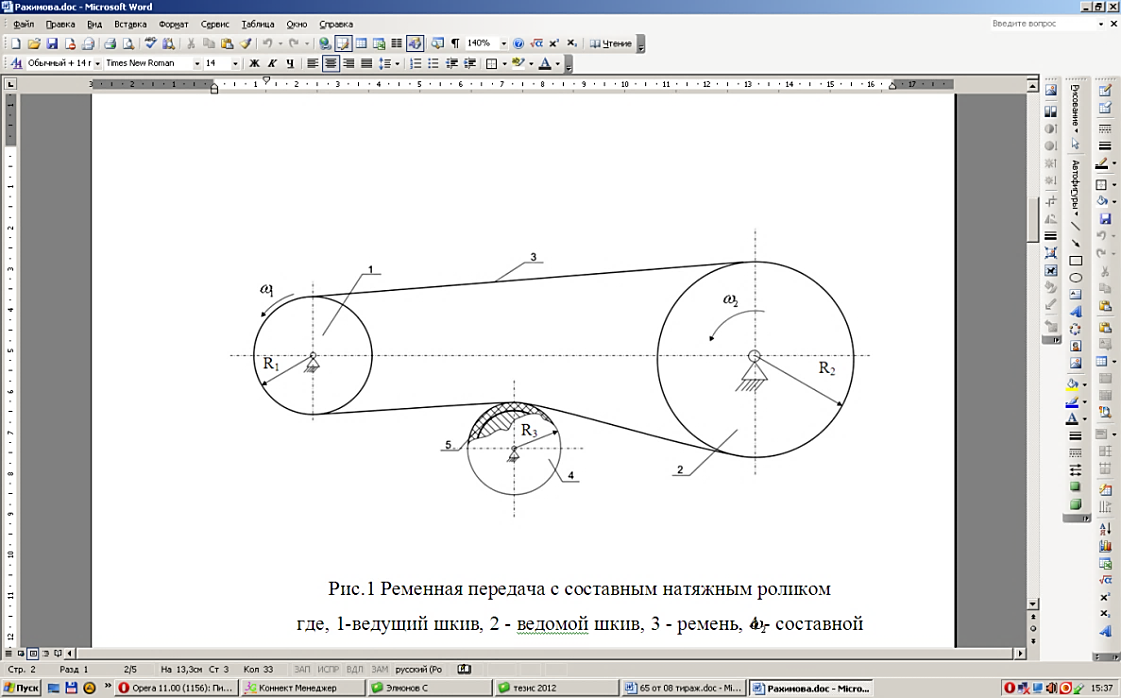
Fig. 1a shows a diagram of the recommended belt drive, from which it is evident that the elastic bushing 5 dampens the vibrations of the driven branch of the belt to some extent. The degree of interaction of the belt 3 with the bushing 5 depends on the transmission parameters, especially on the rigidity of the bushing 5 of the composite tension roller 4 [5].

From work [1] it is known that the initial belt tension is determined from the expression:

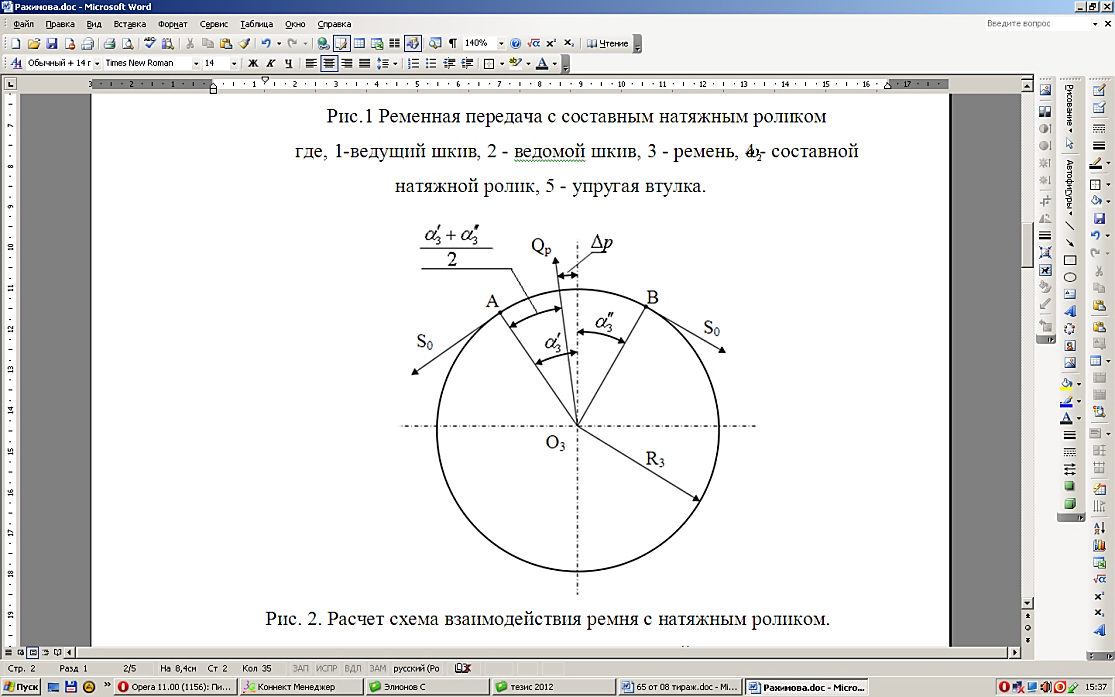
 (7)

where, - specific gravity of the belt,  - belt width,  - belt thickness,  - peripheral speed,  - acceleration of gravity.

The resulting system of second-order differential equations (7) describes the motion of the considered machine unit. Its solution mainly depends on the mathematical description of the motor characteristics and resistances from raw cotton. Analysis of the system of differential equations (7) shows that to simplify the problem, it is advisable to solve it analytically without taking into account the viscous resistance in the elastic transmission. In this case, according to the results of the solution, it is possible to obtain the maximum values of the amplitude of the load fluctuations in the elastic transmission and the coefficients of unevenness of the angular velocities of the driving and driven drums.



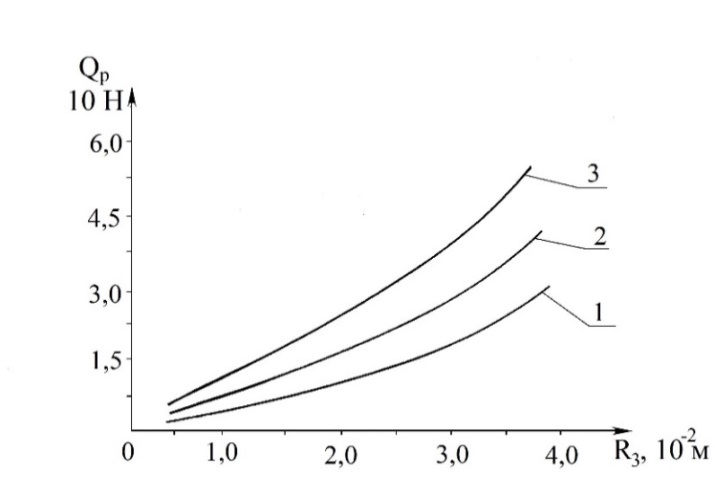
a)



**FIGURE 1.** a) belt drive with a composite tension roller; b) Calculation diagram of the interaction of the belt with a composite tension roller: 1-driver pulley, 2 - driven pulley, 3 - belt, 4 - compound tension roller, 5 - elastic bushing:  - components of the angle of the belt wrap around the elastic sleeve of the compound tension roller;  - angle between force  and vertical axis of the belt.

Let's consider the calculation scheme presented in Fig. 1.b. According to this scheme, we determine the force of interaction of the tension composite roller with the belt, taking into account (1):

 (8)



**FIGURE 2.** Patterns of change in the force of interaction of the tension roller with the belt when varying the radius of the tension roller: 1- ; 2-; 3-

When the belt interacts with the elastic bushing of the composite tension roller, the elastic bushing is deformed in the vertical direction [6].

Fig. 2 shows the graphical dependencies of the change in the force of interaction of the belt when interacting with the tension composite roller, taking into account the angle.

It is evident from the graphs that an increase in the radius of the tension roller due to an increase in the area of contact with the belt increases the interaction force according to a nonlinear pattern. So, with the radius of the tension roller 1,0∙10-2m the force of interaction with the belt reaches 13,8 N, and when the radius of the tension roller increases to 3,5∙10-2 m this force increases to 53,7 N. Increase in force Qр provides the necessary change in the angular velocity of the driven pulley. Therefore, by selecting the radius of the tension composite roller, it is possible to ensure the necessary changes in the angular velocity of the driven pulley and the associated working element of the technological machine [7].

Production test results for transporter belt with recommended centering idler roller. The technical and economic efficiency of the proposed belt transporter with a tensioning device is to increase the reliability and efficiency of the transporter by eliminating the sideslip of the belt.



**FIGURE 3.** General view of the tensioner prototype.

To study the efficiency of the effective design of the centring tensioner, a prototype was made from the Kaprolon -V material and installed on a belt transporter of the TLH-18 type.

Figure 3 shows a general view of the installation of a prototype of the recommended design on a transporter belt of the TLH-18 type.

Comparative tests of the modernized belt transporter with the recommended centring tension roller show high performance and reliability in operation. At the same time, there is no actual slipping of the tape from the drums, even if the axis of the drums is not aligned (7-8 degrees), the slaughter of raw cotton during transportation is eliminated. Productivity has increased by 12% but in relation to the existing structure. The service life of the recommended structure has increased (10-15) percentage.

In addition, the choice of values for the rigidity coefficient of the elastic bushing at small values of the radius of the tension composite roller and the largest values of the wrap angle are important. In this case, for the belt drive under consideration, the recommended parameters are: ()=1,11,3 rad, R3=(2,53,5)•102 m; С=(4,15,3)•102 N/m.

**CONCLUSION**

This study has proposed and substantiated a new design of a centering tension roller with an elastic element intended for belt drives and belt transporters operating under variable load conditions. Unlike conventional belt drives with constant tension, technological and transporting machines are characterized by fluctuating loads, which inevitably lead to changes in belt tension, increased wear, lateral belt displacement, and a reduction in service life. The developed composite tension roller effectively addresses these problems by automatically adapting to asymmetric and variable forces acting on the belt.

The theoretical analysis, supported by kinematic and force models, has shown that the interaction force between the belt and the composite tension roller depends nonlinearly on the roller radius and the belt wrap angle. An increase in the roller radius leads to a larger contact area and, consequently, to a controlled increase in the interaction force, which ensures the necessary regulation of the angular velocity of the driven pulley. The inclusion of an elastic bushing and compression springs allows the system to dampen load fluctuations, reduce dynamic stresses, and maintain belt alignment even when pulley axes are misaligned.

Experimental and production tests carried out on a TLH-18 belt transporter confirmed the theoretical results. The prototype centering tension roller demonstrated stable operation, complete elimination of belt side slip, and reliable performance even at pulley axis deviations of up to 7–8 degrees. As a result, the productivity of the transporter increased by approximately 12%, while the service life of the belt drive system improved by 10–15% compared to the conventional design.

**REFERENCES**

1. Vorobyev, I. I. (2021). *Belt drives*. Moscow, Russia: Mashinostroenie.
2. Uicker, J. J., Jr. (2020). *Theory of machines and mechanisms.* New York, NY: Oxford University Press.
3. Kamke, E. (2022). *Handbook of ordinary differential equations*. Moscow, Russia: Nauka.
4. Djuraev, B. N., Davidbaev, B. N., Zhalyaev, A. A., Melemedov, R. Y., & Mirzakhanov, Y. U. (2021). *Tension roller for flat-belt transmission* (Uzbek Patent No. 50 FVP Uz, RA No. 6). Agency of Intellectual Property of the Republic of Uzbekistan.
5. Mirzakhanov, Y. U., et al. (2000). *Calculation of flat-belt transmissions with centering tension devices*. Bishkek, Kyrgyzstan: Technology Publishing House.
6. Davidboev, B., Mirzakhanov, Y., Makhmudov, I., & Davidboeva, N. (2023). Investigation of lateral belt assembly in flat-belt transmissions and transport mechanisms. *International Journal of Scientific and Technological Research, 9*, p. 3666–3669.
7. Khusanov, Y., Alimjonova, G., Usmonov, M., Nazarova, G., Gapparov, Q., Mirzamaxmudova, N., & Mamayusupov, J. (2024b). Prospective methods of improving productivity in mechanical processing of agricultural parts. *BIO Web of Conferences, 141*, 04002. <https://doi.org/10.1051/bioconf/202414104002>