https://doi.org/10.24412/2617-149X-2026-1-24-31

УДК 621.833.3

# Оценка увеличения нагрузочной способности планетарного роликовинтового механизма в зависимости от конструктивных параметров

**Я. П. Зенкина** **Московский государственный технический университет им. Н. Э. Баумана, г. Москва, Российская Федерация**

**Аннотация.** Работа посвящена разработке нового подхода к оценке статической и динамической грузоподъемности планетарных роликовинтовых механизмов (ПРВМ) для совершенствования их конструктивных параметров и ускорения процесса создания новых изделий. Существующие классические расчетные модели опираются на эмпирические исследования и обладают серьезным ограничением: они не учитывают влияние геометрических параметров резьбы и статичны по своей сути. Это не позволяет оценить динамику изменения конструктивных характеристик механизма, среди которых определяющую роль играет угол профиля резьбы, непосредственно влияющий на распределение напряжений и общую надежность. В качестве методов исследования в работе применено математическое моделирование, позволившее отойти от сугубо эмпирических расчетов. Автором создана оригинальная расчетная методика, учитывающая геометрию резьбовых деталей и изменчивость угла профиля витков, от которого зависит концентрация напряжений в зацеплении. Эффективность предложенного подхода подтверждена сравнительным анализом стандартной конструкции ПРВМ с углом профиля 90° и модернизированного варианта с углом профиля 70°. Расчеты доказали, что уменьшение угла профиля витков обеспечивает превосходство в нагрузочной способности по сравнению с традиционным исполнением. Область применения результатов охватывает инженерно-технические центры и конструкторские бюро. Разработанный инструментарий может эффективно использоваться при создании и модернизации силовых линейных приводов с повышенным ресурсом работы для нужд машиностроения, авиастроения и робототехники.

*Ключевые слова: планетарный роликовинтовой механизм, нагрузочная способность, статическая грузоподъемность, динамическая грузоподъемность, неравномерность нагружения витков, угол профиля резьбы.*

# Evaluation of the Increase in the Load Capacity of the Planetary Roller Screw Mechanism Depending on the Design Parameters

**Yana P. Zenkina** **Moscow State Technical University N. E. Bauman, Moscow, Russian Federation**

**Abstract.** The paper is devoted to a new approach for evaluating the static and dynamic load capacity of planetary roller screw mechanisms (PRSM) to improve design parameters and accelerate product development. Existing classical models rely on empirical studies and lack consideration of thread geometry. This limitation prevents assessing the dynamics of design characteristics, specifically the thread profile angle, which critically influences stress distribution and overall mechanism reliability. As research methods, the paper utilizes mathematical modeling to move away from purely empirical calculations. The author developed an original method accounting for threaded component geometry and profile angle variability, which dictates stress concentration in the mesh. The approach's effectiveness is validated via a comparative analysis of a standard PRSM (90° profile angle) and a modernized version (70° profile angle). Calculations proved that reducing the profile angle ensures superior load capacity over traditional designs. The scope of application covers engineering centers and design bureaus. The developed tools

can be effectively used to design and modernize heavy-duty linear actuators with an increased operating life for mechanical engineering, aerospace, and robotics.

*Keywords: planetary roller screw mechanism, load capacity, static load capacity, dynamic load capacity, uneven loading of threads, thread profile angle.*

**INTRODUCTION**

The development of new mechanical engineering products and the modernization of existing ones must be carried out in accordance with the main trends in mechanical engineering.

In aircraft and rocket technology, ball screw mechanisms (BSM) are used to convert rotary motion into linear motion; they have a lower load capacity than planetary roller screw mechanisms (PRSM). To improve the quality of such mechanisms, there is a trend to replace hydraulic cylinders with electromechanical drives, the power mechanism of which must have a high load capacity. PRSM is such a mechanism [1, 2, 3].

One of the trends in mechanical engineering development is increasing the load capacity of mechanical engineering products, which for the aerospace industry translates into increasing load capacity without increasing weight.

This means that devices must be modernized to transition to a lighter-weight design, which must maintain the load-carrying capacity of the previous, heavier device. Load capacity is the maximum load (force) a device (mechanism) can withstand without parametric failure, functional damage, or loss of functionality under specified conditions.

A review of foreign literature [4–7] showed that the determining factors for static strength are uneven load distribution and manufacturing errors, while durability under extreme conditions is limited by contact fatigue wear mechanisms under linear coupling conditions.

Along with theoretical models, there are industry design standards implemented in the designs of manufacturers. Using reference data, such as static and dynamic load capacities, provided in manufacturers' catalogs to calculate such transmissions presents a number of challenges for the following reasons.

1. In the PRSM catalogs, for each standard size of the mechanism, a set of numerical values of its main parameters is given. These characteristics are given without a methodology for their assessment, which makes it impossible to apply them. The connection and overall parameters of the mechanism are given structurally. Data is provided only for the assembly unit, without information about specific details, including limiting loads.
2. When synthesizing a machine unit, it may turn out that the required transmission or actuator is not presented in the catalog.
3. The lack of data on the methodology for determining dynamic characteristics precludes design modifications by changing roller lengths.
4. The lack of data on methods does not allow us to evaluate new constructive designs of mechanisms that are not presented in the catalog, and at least approximately estimate their resource.

Taking into account the arguments presented above, the objective of this work is to develop a method for predicting the static and dynamic load capacity of the PRSM, depending on the mechanism structure, the dimensions of its components, the acting loads, and the mechanical characteristics of the materials from which these components are made. Previously, such a technique was developed in [8], but this technique does not provide for the possibility of modifying the angle α of the profile of threaded parts of PRSM. To achieve this goal, the task is to modernize the method proposed in [9], i.e., it is necessary to take into account the possibility of changing the profile angle of the threaded components of the PRSM.

**MAIN PART.** General Calculation Assumptions

To develop the methodology, the general calculation assumptions discussed in [10, 11] were adopted:

1. All threaded components of the PRSM are manufactured with absolute precision.
2. The uneven distribution of the axial force $F_A$ of the mating PRSM components in the direction of tension-compression displacement is not taken into account; i.e., all rollers are equally loaded with the axial force $F_A$.
3. From a strength perspective, the worst-case scenario is considered when there are depressions in the mating surface; i.e., the effective length of the roller $L_r$ is assumed to be equal to the length of the nut $L_n$.
4. The replacement of the roller and nut threads with annular projections is considered.
5. The initial contact points of the mating threads of the nut and roller and the mating threads of the roller and screw are assumed to be located at the mid-diameters of the mating PRSM components.
6. Taking into account all the assumptions made, we have identical nominal axial forces $F$ in the mating surfaces of the roller threads with the nut threads and with the screw threads.

Development of a methodology for assessing the static and dynamic load capacity of the PRSM

Let us consider the case when the working axial force $F_{n\Sigma}$ is applied to the nut. Taking into account the nominal axial force $F = F_{A\Sigma} / N$, the following dependence is obtained

$$F = \frac{F_{\mathrm{n}\Sigma}}{N \cdot m_p} = \frac{F_{\mathrm{n}\Sigma} \cdot P}{N \cdot L_{\mathrm{p}}}, \quad (1)$$

where $F$ is the axial force; $N$ is the number of rollers; $m_p = L_p / P$ is the number of roller turns in the two specified mating positions; $L_p$ is the length of the threaded portion of the roller; $P$ is the thread pitch of the PRSM components.

In the work [10, 11] the overturning moment of the roller was derived, which takes the form taking into account the working axial force $F_{n\Sigma}$:

$$M_x = (F_{n\Sigma} \cdot d_{2p}) / N. \quad (2)$$

Then we obtain a dependence taking into account the profile angle of the threaded parts of the PRSM:

$$M_x \approx \frac{4 \cdot F^*_{MAX} \cdot tg(\alpha/2) \cdot P}{k} \cdot \sum_{J=1}^{k} J^2, \quad (3)$$

where $\sum_{J=1}^{k} J^2$ – sum of the number of turns on the roller's generatrix; $k = L_p / (2 \cdot P)$ – number of working turns (number of radial forces).

Using equation (3), we find the maximum force:

$$F^*_{MAX} \approx \frac{F_{\Gamma\Sigma} \cdot d_{2\mathrm{p}} \cdot L_{\mathrm{p}}}{4 \cdot 2 \cdot N \cdot P^2 \cdot tg(\alpha/2) \cdot \sum_{J=1}^{k} J^2}. \quad (4)$$

The resulting formula (4) takes into account the change in the angle α of the PRSM threaded parts profile. And the maximum force value $F^*_{MAX}$ can be used to determine the load distribution between the threads of the PRSM threaded parts.

Then, taking into account the obtained formulas, we can clarify the coefficient of uneven loading of the threaded parts of the PRSM:

$$K_H = \frac{F + F^*_{MAX}}{F} =$$

$$= \left( \frac{F_{\mathrm{n}\Sigma} \cdot P}{N \cdot L_{\mathrm{p}}} + \frac{F_{\mathrm{n}\Sigma} \cdot d_{2\mathrm{p}} \cdot L_{\mathrm{p}}}{4 \cdot 2 \cdot N \cdot P^2 \cdot tg(\alpha/2) \cdot \sum_{J=1}^{k} J^2} \right) \cdot \frac{N \cdot L_{\mathrm{p}}}{F_{\mathrm{n}\Sigma} \cdot P}.$$

Therefore:

$$K_H = \left( 1 + \frac{d_{2\mathrm{p}} \cdot {L_{\mathrm{p}}}^2}{8 \cdot P^3 \cdot tg(\alpha/2) \cdot \sum_{J=1}^{k} J^2} \right). \quad (5)$$

From equation (5) it is evident that the obtained coefficient of uneven loading of the threaded parts of the PRSM does not depend on the working axial force $F_{n\Sigma}$ acting on the output link of the PRSM.

To determine the static lifting capacity of a planetary roller screw mechanism, it is necessary to consider the maximum compressive force $F_{MAX}$ acting on the mechanism's output link. If the maximum equivalent stresses $\sigma_{MAX}$ in the most heavily loaded roller turn are equal to the permissible stresses $[\sigma]$, then the working axial force $F_{n\Sigma}$ will be equal to the static lifting capacity of the PRSM $C_{a0}$. From this, we obtain the condition for the static lifting capacity:

$$F_{MAX} \leq C_{a0}.$$

In accordance with GOST 18854-2013, the axial static load capacity of the PRSM is determined by the initial plastic deformation condition. To simplify calculations, it is assumed that the permissible axial force on a static transmission is determined by the condition of initial plastic deformation. The critical threshold is the ultimate strength $\sigma_u$ in the most heavily loaded roller thread of the PRSM. With this $\sigma_u$ value, the residual deformation in the contact zones of the rollers with the screw and nut threads is approximately equal to 0.0001 times the double radius $R$ of the roller profile.

To calculate the axial static load capacity, it is necessary to determine the permissible design stresses. Due to the high cost of full-scale testing of actual structures, it is most likely that the manufacturer SKF reduces the determination of the PRSM passport parameters to an analytical method, based on analogs−rolling bearings, as noted in [11]. The nature of the contact interaction in PRSM is closest to the nature of the contact interaction in thrust-radial bearings [11].

For thrust or thrust-radial bearings, the value of $C_{a0}$ permissible contact stress is $[\sigma]$ = 4200 MPa [11]. Hence, $\sigma_{MAX}$ = 4200 MPa, and the force $F_{n\Sigma}$ corresponding to these stresses is $C_{a0}$ static load capacity of the PRSM.

In order to calculate $\sigma_{MAX}$ from the action of the working force $F_{n\Sigma}$, for a PRSM with arbitrary geometry, it is necessary to use the Hertz solution [12, 13] for the initial point contact, the contact area (Fig. 1) is considered to be flat elliptical.

The theory of elastic deformations of complex-shaped bodies at their points of contact allows us to determine the shape and dimensions of the contact area, the magnitude and nature of the pressure distribution over the contact area, and the degree of convergence of the bodies. This requires knowledge of the principal curvatures of the contacting surfaces at the point of initial contact, the elastic constants of the material, and the magnitude of the applied load. This problem is examined in detail in [12] in a general form. The data presented allow us to easily determine the shape

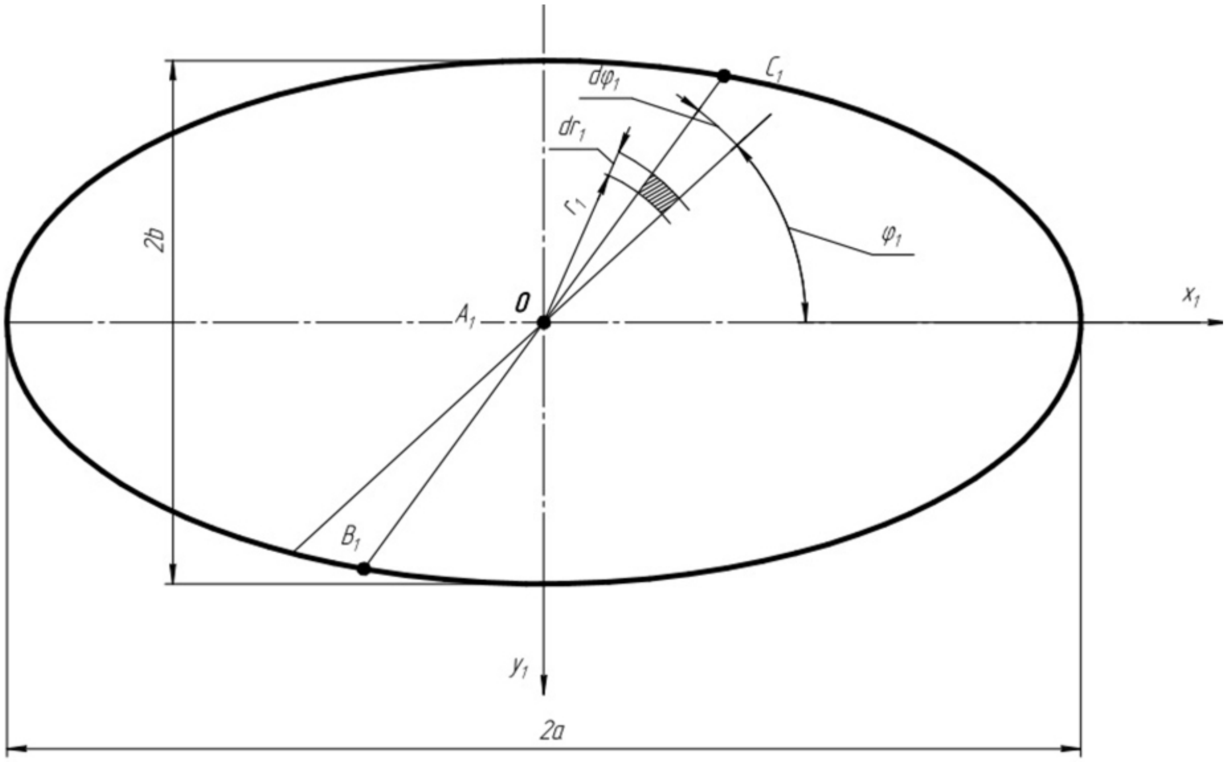


**Figure 1 − Contact area at initial point contact, where a is the major semi-axis and b is the minor semi-axis**

of the contact area and the maximum stresses in the contact zone of two elastic bodies for various cases encountered in practice.

For our case, the contact pressure $q$ is distributed on a flat elliptical area over half of the ellipsoid, where the greatest contact pressure $q_{MAX}$ will be located in the center of the ellipsoid (Formula 7), and the area of the elliptical contact area with the major $a$ and minor $b$ semi-axes (Formula 6); a detailed calculation of the contact pressure for the PRSM was given in [10, 11].

$$a = n_a \cdot \sqrt[3]{\frac{3}{2}\frac{E^* \cdot F_{n,MAX}}{\sum \varkappa}},$$

$$b = n_b \cdot \sqrt[3]{\frac{3}{2}\frac{E^* \cdot F_{n,MAX}}{\sum \varkappa}}. \quad (6)$$

And the greatest contact pressure occurs at the center of the contact area (the solution to the Hertz problem is given):

$$q_{MAX} = \frac{n_p}{\pi} \cdot \sqrt[3]{\frac{3}{2}\left(\frac{\sum \varkappa}{E^*}\right)^2 \cdot F_{n,MAX}}, \quad (7)$$

where $E^*$ − is the elastic constant of the PRSM component materials; $\sum \varkappa$ − is the sum of the principal curvatures of the mating turns at the point of their initial contact; $n_a$, $n_b$ and $n_p$ − are coefficients determining the dimensions of the contact area and the magnitude of the maximum contact pressure.

The value of the elastic constant $E^*$ for a screw and roller made of steel ШХ15 (ShKh15) or ХВГ (KhVG), 20Х3МВФ (20Kh3MVF), etc.:

$$E^* = 0{,}867 \cdot 10^{-5} \text{ 1/MPa}.$$

The sum of the principal curvatures at the point of initial contact of the screw and roller turns is given in the formula:

$$\sum \varkappa = \frac{\sqrt{2}}{d_{2\text{в}}} + \frac{4 \cdot sin(\alpha/2)}{d_{2\text{р}}}. \quad (8)$$

In the work [10, 11] the dependence of the argument is given as $\Omega$:

$$\Omega = \sqrt{2} \cdot d_{2\text{р}} / \left(\sqrt{2} \cdot d_{2\text{р}} + 4 \cdot d_{2\text{в}} \cdot sin(\alpha/2)\right). \quad (9)$$

The regression dependences of the coefficients $n_a$, $n_b$ and $n_p$ on the argument $\Omega$ take the form

$$n_a = 1{,}4077 \cdot \Omega^{0{,}11664},$$

$$n_b = 0{,}73537 \cdot \Omega^{-0{,}10288},$$

$$n_p = 0{,}96626 \cdot \Omega^{-0{,}013655}. \qquad (10)$$

Using the Hertz contact problem for initial point contact, the greatest contact pressure, we obtain:

$$q_{MAX} = \frac{n_p}{\pi} \cdot \sqrt[3]{\frac{3}{2}\left(\frac{\sum \varkappa}{E^*}\right)^2 \cdot \frac{F_{n\Sigma} \cdot P \cdot K_H}{N \cdot L_p \cdot cos(\alpha/2)}} \qquad (11)$$

We determine the permissible static axial force ($C_{a0} = F_{n\Sigma}$ static load capacity). We calculate the maximum axial force $F_{BMAX}$ acting between the pair of most heavily loaded screw and roller threads based on formula (1), taking into account the coefficient of uneven loading of the threaded parts of the PRSM.

$$F_{B\,\max} = \frac{F_{n\Sigma} \cdot P \cdot K_H}{N \cdot L_p} \qquad (12)$$

To determine the contact stresses acting between the pair of most loaded turns of the screw and roller, the maximum normal force in the contact zone takes the form:

$$F_{n\,\max} = \frac{F_{B\,\max}}{cos(\alpha/2)} = \frac{F_{n\Sigma} \cdot P \cdot K_H}{N \cdot L_p \cdot cos(\alpha/2)} \qquad (13)$$

In accordance with GOST 18854-2013, the basic static axial load capacity is $C_{a0} = 4200\ N$. Then the force $F_{\Gamma\Sigma}$ corresponding to these stresses is the static load capacity of the PRSM $C_{a0}$.

By solving the given problem, we obtain the dependence for the static load-bearing capacity:

$$C_{a0} = \frac{2 \cdot N \cdot L_p \cdot cos(\alpha/2) \cdot \left(\frac{4200 \cdot \pi}{n_p}\right)^3}{3 \cdot P \cdot K_H} \cdot \left(\frac{E^*}{\sum \varkappa}\right)^2 \qquad (14)$$

Roller screw drives are also calculated for dynamic load capacity $C_a$.

Based on the results of work [11], it is possible to obtain the dependence of the static $C_{a0}$ and dynamic $C_a$ lifting capacity, which are in correlation with the average diameter of the screw thread $d_{2B}$ and the displacement of the output link of the PRSM in one revolution of its input link $P_h$, which determine the size of the PRSM.

As a result, the above correlation takes the form of a power-law dependence of the coefficient $k_C$:

$$k_C = 0{,}3638 \cdot d_{2B}^{0{,}5229} \cdot P_h^{-0{,}2027} \qquad (15)$$

The coefficient $k_C$ evaluates the ratio of static and dynamic load capacities

$$k_C = C_{a0} / C_a$$

From here:

$$C_a = C_{a0} / k_C \qquad (16)$$

<u>Methodology for predicting the static and dynamic load capacity of the PRSM</u>

<u>Initial data.</u> Calculation for a thread profile angle of $\alpha = 90°$.

We use the vector of geometric characteristics and the vector of strength characteristics obtained from the proposed methods for determining geometric and strength parameters outlined in [11, 14]. We set the basic requirements for ensuring the total operating time.

<u>Calculation:</u>

1. Using formula (14), we determine the static load capacity for the PRSM;
2. We determine the coefficient $k_C$ using formula (15);
3. We determine the dynamic load capacity $C_a$ using formula (16).

<u>Analysis of calculation results using the modernized methodology</u>

Using this methodology, a Delphi computer program was developed that allows one to determine the values of static $C_{a0}$ and dynamic $C_a$ load capacities depending on the $\alpha$ profile angle of the threaded parts of the PRSM. The obtained $C_{a0}$ value coincided with the value obtained in [11] with an error of 2 %. This does not exceed the permissible value of 10 %, which confirms the accuracy of the conducted research.

Let us consider the graphs (Fig. 2) of the dependence of the static $C_{a0}$ and dynamic $C_a$ lifting capacities on the profile angle of the threaded parts of the PRSM using the example of the PRSM of size 39×15.

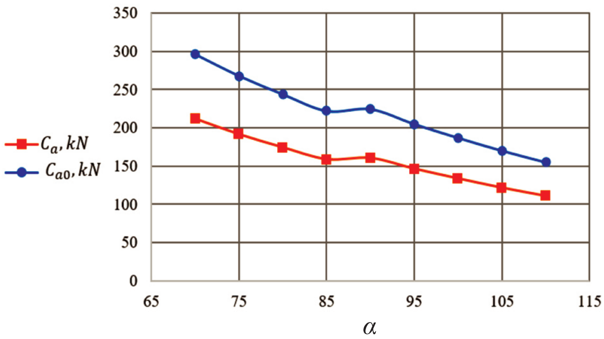


**Figure 2 – Dependences of static $C_{a0}$ and dynamic $C_a$ lifting capacities for size 39×15 on the angle $\alpha$ of the profile of the threaded parts of the PRSM**

From Figure 2 it can be seen that the value of the maximum load capacity for this size 39×15 is at an angle of $\alpha$ = 70°, and this value is 32 % greater compared to the traditional profile angle $\alpha$ = 90°.

From this, we can conclude that the load-bearing capacity of the PRSM does increase with changes in the $\alpha$ angle of the PRSM threaded parts. Therefore, the proposed prediction method describes the obtained values of static $C_{a0}$ and dynamic $C_a$ load-bearing capacities with sufficient accuracy and can be used.

It is worth noting that the data presented in Figure 2 correspond to the elastic constant $E^* = 0{,}867 \cdot 10^{-5}$ 1/MPa, which depends on the material of the manufactured PRSM parts. This means that as the material changes, the data presented will change.

SKF and Ewellix catalogs use X5CrNiCuNb16-4 steel, which has a first-order elastic modulus of $E_B$ = 1,96 МПа and a Poisson's ratio of $\mu_B$ = 0,3. The elastic constant $E^*$ will change accordingly. This suggests that the static and dynamic load capacities of PRSMs are directly dependent on the structural materials used.

Let's consider the example of a 39×15-size PRSM, taking into account the material change. For a thread profile angle of $\alpha$ = 90°, the PRSM parts are shown in Table 1. The calculation results are summarized in Table 1.

**Table 1 – Load capacities $C_{a0}$ and $C_a$ of the designed PRSM of size 39×15 in comparison with other catalogues**

| Parameter | $C_{a0}$, kN | Error in comparison with the method, % | $C_a$, kN | Error in comparison with the method, % |
|---|---|---|---|---|
| SKF Catalog | 272,89 | 5,30 | 167,64 | 8,00 |
| Ewellix Catalog | 273,00 | 5,40 | 168,00 | 7,80 |
| According to the methodology | 258,40 | 0,00 | 181,10 | 0,00 |

A comparative analysis of data from catalogues of leading manufacturers and the results obtained using the method of predicting dynamic C_a and C_a0 static load capacity confirms the reliability of the calculations: the deviation of the values does not exceed 10%.

Thus, the proposed forecasting method describes the obtained values of static and dynamic load capacities with sufficient accuracy and can be used.

## CONCLUSION

This article examines the potential for increasing the load capacity of a planetary roller screw mechanism depending on its design parameters. Changing the profile angles increased the load capacity of the modernized PRSM ($\alpha$ = 70°), necessitating the development of a calculation methodology and algorithm that takes into account arbitrary values of this angle. The following key results and conclusions were obtained by solving the problem posed in this article.

1. A method for predicting static and dynamic load capacities is proposed, taking into account changes in the profile angle of PRSM threaded components.
2. Application of the developed load capacity prediction method showed that switching from a traditional profile angle to a modernized one provides a 32 % increase in static and dynamic load capacities.
3. The developed algorithm for calculating the service life and dynamic load capacities is an effective tool for designing new PRSM models. Using these solutions allows for the creation of transmissions with improved technical characteristics.

## СПИСОК ЛИТЕРАТУРЫ

1. Дикун, Е. В. Надежность и долговечность роликовинтовых механизмов для изделий аэрокосмической техники / Е. В. Дикун, Д. С. Блинов, А. Ю. Колобов // XLVI Академические чтения по космонавтике, посвященные памяти академика С. П. Королёва и других выдающихся отечественных ученых – пионеров освоения космического пространства. Королёвские чтения 2022: XLVI Академические чтения по космонавтике, Москва, 25–28 января 2022 года. Том 4. – Москва: Издательство МГТУ им. Н. Э. Баумана, 2022. – С. 17–20.

2. Блинов, Д. С. Проектирование планетарных роликовинтовых механизмов для электромеханических приводов летательных аппаратов (теория) / Д. С. Блинов, А. Ю. Колобов, Е. В. Дикун // *Справочник. Инженерный журнал.* – 2018. – № 8. – С. 14–23. – DOI: 10.14489/hb.2018.08.pp.014-023.

3. Блинов, Д. С., Леонтьев, С. К., Морозов, М. И. Перспективные роликовинтовые механизмы электромеханических приводов для авиационной и ракетной техники / Д. С. Блинов, С. К. Леонтьев, М. И. Морозов // *Вестник воздушно-космической*

*обороны*. – 2017. – № 1 (13). – С. 66–71. http://www.raspletin.com/NOC/vko/VKO_1(13).pdf.

4. Miao, J. Performance and challenges of planetary roller screw mechanism in complex conditions: *A comprehensive review of recent progress / J. Miao, R. Tang, X. Du, X. Shan // Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science.* – 2025. – Vol. 239(11). – P. 4298–4322. – DOI: 0.1177/09544062251318929.

5. Xing, M. Multi-rollers statics analysis of planetary roller screw mechanism based on explicit expression / M. Xing, S. Liu, Z. Cheng, Y. Cui, L. Zu, J. Xu // *Tribology International.* – 2024. – Vol. 197. – P. 109818. – DOI: 10.1016/j.triboint.2024.109818.

6. Fu, X. A multi-roller static model of the planetary roller screw mechanism considering load sharing / X. Fu, X. Li, Sh. Ma, D. Gerada, G. Liu, Ch. Gerada // *Tribology International.* – 2022. – Vol. 173. – P. 107648.

7. Li, X. Review on Motion and Load-Bearing Characteristics of the Planetary Roller Screw Mechanism / X. Li, G. Liu, X. Fu, S. Ma // *Machines.* – 2022. – № 10(5). – P. 317. – DOI: 10.3390/machines10050317.

8. Блинов, Д. С., Морозов, М. И. Прогнозирование нагрузочной способности роликовинтовых механизмов / Д. С. Блинов, М. И. Морозов // *Приводы и компоненты машин.* – 2014. – № 1. – С. 12–14.

9. Блинов, Д. С. Прогнозирование ресурса винтовых механизмов качения / Д. С. Блинов, М. И. Морозов // *Наука и образование: научное издание МГТУ им. Н.Э. Баумана*. – 2015. – № 2. – С. 24–41.

10. Блинов, Д. С., Зенкина, Я. П., Нахатакян, Ф. Г. Контактное взаимодействие ролика в планетарной роликовинтовой передаче и его влияние на нагрузочную способность механизма / Д. С. Блинов, Я. П. Зенкина, Ф. Г. Нахатакян // Вестник машиностроения. – 2022. – № 6. – С. 30–37. – DOI: 10.36652/0042-4633-2022-6-30-37.

11. Блинов, Д. С. Винтовые передачи линейных приводов / Д. С. Блинов. – Москва: Издательство МГТУ им. Н. Э. Баумана, 2021. – 200 с.

12. Пономарев, С. Д., Бидерман, В. Л., Лихарев, К. К. Расчеты на прочность в машиностроении / С. Д. Пономарев, В. Л. Бидерман, К. К. Лихарев. – Москва: Машгиз, 1958. – Т. II. – 974 с.

13. Феодосьев, В. И. Сопротивление материалов. – Москва: Изд-во МГТУ им. Н.Э. Баумана, 2016. – 543 с.

14. Блинов, Д. С. Общий случай определения размеров и полей допусков на них для резьбовых деталей планетарных роликовинтовых механизмов – Текст : электронный / Д. С. Блинов, Я. П. Зенкина // *Наука и образование: научное издание МГТУ им. Н.Э. Баумана.* – 2017. – № 4. – С. 16–32. – URL: https://scispace.com/journals/science-and-education-of-the-bauman-mstu-3ggm6d3d (дата обращения: 15.05.2026). – DOI: 10.7463/0417.0001123.

**REFERENCES**

1. Dikun EV, Blinov DS, Kolobov AYu. Reliability and Durability of Roller Screw Mechanisms for Aerospace Products. In: XLVI Akademicheskie chteniya po kosmonavtike, posvyashchennye pamyati akademika S.P. Korolyova i drugikh vydayushchikhsya otechestvennykh uchenykh – pionerov osvoeniya kosmicheskogo prostranstva. Korolyovskie chteniya 2022: XLVI Akademicheskie chteniya po kosmonavtike = XLVI Academic Readings on Cosmonautics, dedicated to the memory of Academician SP Korolev and other outstanding Russian scientists – pioneers of space exploration. Korolev Readings 2022: XLVI Academic Readingson Cosmonautics, 25–28 January, 2022, Moscow. Moscow: Publishing House of Bauman Moscow State Technical University; 2022;4:17–20. (In Russ.).

2. Blinov DS, KolobovAYu, Dikun EV. Designing of planetary roller-screw mechanisms for electromechanical aircraft actuators (theory). *Spravochnik. Inzhenernyi zhurnal = Handbook. An Engineering Journal.* 2018;8:14-23. – DOI: 10.14489/hb.2018.08.pp.014-023. (In Russ.).

3. Blinov DS, Leontyev SK, Morozov MI. Promising roller-screw mechanisms of electromechanical drives for aviation and rocket technology. *Vestnik vozdushno-kosmicheskoy oborony = Journal «Aerospace Defense Herald».* 2017;1(13):66-71. http://www.raspletin.com/NOC/vko/VKO_1(13).pdf (In Russ.).

4. Miao J, Tang R, Du X, Shan X. Performance and challenges of planetary roller screw mechanism in complex conditions: *A comprehensive review of recent progress. Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science.* 2025;239(11):4298-4322. – DOI: 0.1177/09544062251318929.

5. Xing M, Liu S, Cheng Z, Cui Y, Zu L, Xu J. Multi-rollers statics analysis of planetary roller screw mechanism based on explicit expression. *Tribology International.* 2024;197:109818. – DOI: 10.1016/j.triboint.2024.109818.

6. Fu X, Li X, Ma Sh, Gerada D, Liu G, Gerada Ch. A multi-roller static model of the planetary roller screw mechanism considering load sharing. *Tribology International.* 2022;173:107648. –

DOI: 10.1016/j.triboint.2022.107648.

7. Li X, Liu G, Fu X, Ma S. Review on Motion and Load-Bearing Characteristics of the Planetary Roller Screw Mechanism. *Machines.* 2022;10(5):317. – DOI: 10.3390/machines10050317.

8. Blinov DS, Morozov MI. Forecasting the load capacity of roller screw mechanisms. *Privody i komponenty mashin = Machine drives and parts*. 2014;1:12-14. (In Russ.).

9. Blinov DS, Morozov MI. Predicting rolling screw mechanisms service life. *Nauka i obrazovanie: nauchnoe izdanie MGTU im. N.E. Baumana = Science and Education of the Bauman MSTU (MGTU im. N.È. Baumana).* 2015;2:24-41. – DOI: 10.7463/0215.0756784 (In Russ.)

10. Blinov DS, ZenkinaYaP, Nakhatakyan FG. Contact interactions of the roller in a planetary roller screw and their influence on the load capacity. Vestnik mashinostroeniya = Russian Engineering Research. 2022;9(42):884-890. – DOI: 10.36652/0042-4633-2022-6-30-37 (In Russ.).

11. Blinov DS. Vintovye peredachi lineynykh privodov = Screw transmissions of linear drives. Moscow: Izd-vo MGTU im. N.E. Baumana; 2021:200. (In Russ.).

12. Ponomarev SD, Biderman VL, Likharev KK. Raschety na prochnost' v mashinostroenii = Strength Calculations in Mechanical Engineering. Moscow: Mashgiz; 1958;II:974. (In Russ.) .

13. Feodosyev VI. Soprotivlenie materialov = Strength of Materials. Moscow: Izd-vo MGTU im. N.E. Baumana; 2016:543. (In Russ.).

14. Blinov DS, ZenkinaYaP. A General Case to Determine Dimensions and Their Tolerance Zones for the Thread Parts of Epicyclic Roller-Screw Mechanisms. *Nauka i obrazovanie: nauchnoe izdanie MGTU im. N.E. Baumana = Science and Education of the Bauman MSTU (MGTU im. N.È. Baumana).* 2017;4:16-32. Available from: https://scispace.com/journals/science-and-education-of-the-bauman-mstu-3ggm6d3d. [Accessed 15 May 2026]. – DOI: 10.7463/0417.0001123 (In Russ.).

**Сведения об авторах**

**Зенкина Яна Павловна**

Старший преподаватель кафедры «Теория механизмов и машин» Московского государственного технического университета им. Н.Э. Баумана, г. Москва, Российская Федерация

*E-mail: zenkina.y@bmstu.ru*

ORCID: http://orcid.org/ 0009-0006-7796-4539

**Information about the authors**

**Yana P. Zenkina**

Senior lecturer of the Department Theory of Mechanisms and Machines, «Moscow State Technical University N.E. Bauman», Moscow, Russia

*E-mail: zenkina.y@bmstu.ru*

ORCID: http://orcid.org/ 0009-0006-7796-4539

Статья поступила в редакцию 27.05.2026.