Development and studies of new rolling screw-type mechanisms

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Abstract. One of the tendencies in mechanical engineering is the transition from sliding friction to rolling friction in part joints moving under load, motor assemblies and machinery. For the present time, planetary roller screws (PRS), which are rolling mechanisms, possess the biggest potential as converters of rotational motion to linear motion. However production of high-precision parts for PRS requires special expensive equipment. The emerging world tendency for reduction of technological costs is to develop nutless roller screws (NRS), which is the most complex PRS part to manufacture. Its production requires an additional machine tool or a universal grinding machine for external and internal grinding. The new rolling NRS has been developed in line with this tendency. This paper focuses on the design and operation of the mechanism and on theoretical studies. A test model and a testing rig have been developed to confirm the results of the theoretical studies and to determine operation parameters of the new NRS. The experiments showed that the newly developed NRS loses to PRS in some parameters, while outperforming it in other. It is also simpler and cheaper to produce and after additional studies and tests rational applications for the newly developed NRS can be found.

1. Introduction
The role of machine engineering has been considerably growing recently and one of its main problems is related to friction and wear. By different estimates, about 70% to 85% of machine engineering products break down due to wear. The losses due to friction in developed countries account for 5 … 6% of the national income, 20 … 25% of energy produced globally in a year is used to overcome friction. With understanding of the importance of this problem for the humanity, the science of friction, wear and greasing, tribology [1], has been intensively developing starting from the middle of the last century, together with its application for engineers, triboengineering [2].

One of the radical means of reducing the negative consequences of friction and wear is development of assemblies and mechanisms employing rolling friction.

The objects of research are roller screws, which have not been widely used in machine engineering in Russia yet, but the need for them is constantly growing. Screw-based rolling mechanisms can be of two basic designs: ball screws and roller screws [3]. One of the roller screw subclasses are planetary roller screws (PRS), figure 1, which, for the present day, are the most prospective screw mechanisms. The screw, rollers and the nut have a special thread with the angle of \( \alpha = 90^\circ \), figure 2.

PRS in figure 1 consists of multi-start screw 1 and nut 2, \( n \) threaded rollers 3, separators 4, and casing parts and sealings, which are not shown in figure 1. In order to synchronize the operation, each roller is additionally connected with the nut by cogged engagement. The profile of the roller thread
turns is convex enabling an initial point contact of the roller thread turns with the interfacing thread turns of the screw and the nut and excluding unwanted edge contacts.

![Figure 1. Planetary roller screw.](image1)

In order to implement rolling friction in PRS, the pitch diameters of the screw, rollers and nut threads are related to each other and to other thread parameters by mathematically-defined relations [4].

PRS are high precision mechanisms with power flows, closed dimension chains and numerous redundant constraints. Launching of PRS serial manufacturing requires huge material and time input with an unclear prospect. Only expensive specialized machine tools, well-adjusted technology process and machine-tool operator's experience can ensure precision in manufacturing of the main PRS threaded parts. Manufacturing of high-precision multiple threads on the nut, tempered to a high hardness, represents a particular challenge.

The desire to do without the most complex in manufacturing PRS part, the nut, has led to development of nutless roller screws (NRS). The number of NRS patents sold in the advanced industrial countries exceeds several hundred [5-8], that means a tendency for creation of such mechanisms has emerged and we can talk about a new subclass of roller screws – NRS. This confirms the relevancy of their development. It should be noted that NRS should retain one of the PRS most important features that is the rolling friction.

Many of the developed NRS have considerable shortcomings, due to which they are not applicable in practice. In many of the mechanisms rolling friction is not carried out. But new NRS are being developed so that the subclass of these mechanisms is constantly growing.

2. Purpose and objectives of the research
The purpose and objectives of the research are to develop a new operable NRS design, to prove that rolling friction is effected in it, as well as to demonstrate by calculations and tests that the main parameters of the new NRS are similar to the parameters of traditional PRS. Furthermore, NRS should be cheaper and easier than the latter to manufacture, and launching their production, both serial and piecework, should be affordable for many of the Russian enterprises.

3. Studies of PRS friction zones
In order to compare NRS with the prototype (PRS) let's review the main sources of friction losses in PRS under load. These include numerous (hundreds) contact spots of interfacing nut and roller threads, figure 3, and interfacing roller and screw threads, figure 4.

The roller, figure 3, is rotating in planetary movement around the screw axis (point O) with angular velocity of translation motion $\omega_e$, while all the points of the roller are rotating around its axis with angular velocity of translation motion $\omega_r$. When summing up the linear velocities of these two motions only in the initial contact point A, the absolute velocity will be zero, since the roller is engaged by a
gear ring with the nut, which does not rotate. When under load, this will cause spin friction around A point in the contact pattern [9].

Figure 3. Contact pattern S between thread turns of a roller 2 and a nut 3.

Figure 4. Contact pattern S between thread turns of screw 1 and roller 2.

One of PRS features is the unequal screw and nut thread angles. Due to this, if the PRS center distance is nominal, roller threads don't fit into the cavities between the adjacent screw threads [4, 10]. In order to fit the roller thread into the cavity between the adjacent screw thread turns, it is necessary to increase the nominal value of the center distance by $\Delta a_n$, figure 4 [10]. This causes the initial contact point to move to point C relatively to the line that connects the axes of the screw and the roller, figure 4. This line includes point B, in which the velocities of the screw and the roller are equal.

When under load, a contact pattern (area) S will emerge there with point C inside, figure 4. Depending on the load and screw and roller thread parameters, point C may fall either within or outside of area S. In any case there shall be spin friction around point B [9].

4. Development of a new NRS design

PRS nut has two the main functions, figure 1: the nut takes up the load from the screw and transfers it to the casing; the nut keeps the rollers from radial movement. Taking in account these functions and NRS development experience, several NRS have been developed and patented in Russia successively. As a result, two most successful designs were developed [11, 12], one of which (see patent [11]) is shown in figure 5.

Figure 5. The developed NRS structure with fixed roller axes.
The NRS consists of a screw 1 with \( z \) thread starts and thread pitch \( P \), threaded rollers 2 with \( z_r \) thread starts, installed inside casing 3 with covers 4. The covers are fixed on the casing by the means of screws 5 and lock washers 6. The screw and rollers are engaged by their threads along the screw generators. Adjusting shims 7 are placed between casing 3 and covers 4. Ball seats \( A \) with radius \( R_k \) are made on the inner face surface of each cover. The number of seats on each cover is equal to the number of rollers. Casing 3 has basic elements C, designed to fix NRS with actuating device. Oil slinger 9 is fixed on the outer face surface of each cover 4 by the means of bushing 8. Each roller not far from its ends has two ring grooves \( B \) with installed rings 10. End faces of each roller have spherical seats \( D \) with radius \( R_p = R_k \). Balls 11 with radius \( R_S < R_k \) are installed in the spherical seats \( A \) of covers and \( D \) of rollers at the same time. To make the axis of each roller parallel with the screw axis, spherical seats \( A \) on the covers are made with high precision of angular position, and an angular orientation of covers relative to the casing is ensured by the means of pins.

When NRS is in operation, the screw rotates around its axis, the rollers rotate around their axes and together with the casing make forward motion along the crew axis. The working axial force is transferred from the screw to the rollers via the interfacing thread turns of these parts, then from the rollers to the casing via the contact of spherical seats \( D \) on the rollers with the balls and the balls with spherical seats \( A \) on one of the covers. Rings 10, which keep rollers from radial movement, rotate.

One important difference of NRS from PRS is the lack of mathematical dependency, which make the basic parameters of the mechanism interrelated, that allows for higher flexibility in designing of NRS. For example, the screw can be single- or double-threaded.

5. Kinematical calculations of NRS

Let's review the NRS velocity diagram, figure 6, with consideration of slipping. Let's introduce a relative slip factor \( \zeta = (V_v - V_r) / V_v \), where \( V_v \) and \( V_r \) are linear velocities of the initial contact point of the interfacing screw and roller thread turns.

Taking the screw angular velocity \( \omega_v \) as initial, let's define the angular velocities of the roller \( \omega_r \) and the ring \( \omega_k \).

\[
\omega_r = \omega_v \cdot \left( d_{2a} / d_{2p} \right) \cdot (1 - \zeta)
\]

\[
\omega_k = \omega_v \cdot d_{2a} \cdot d_S \cdot (1 - \zeta) / \left( d_{2p} \cdot (d_{2a} + d_{2p} + d_S) \right)
\]

Figure 6. NRS velocity diagram with consideration of slipping. Where: \( d_S \) and \( d_k \) is the roller pin diameter and the ring hole diameter; \( V_S \) and \( V_k \) are linear velocities of the roller pin and the ring; \( V_{vp} \) and \( V_{pk} \) are relative linear velocities.

NRS gear ratio, mm/revolution

\[
H = P \cdot z_n \pm \left( d_{2a} / d_{2p} \right) \cdot P \cdot z_p \cdot (1 - \zeta)
\]

In equation (3): «+» is used for unidirectional screw and roller threads; «−» is used for screw and roller threads of different directions.

The effects in the interface of the screw and the roller threads will be the same as in PRS, figure 4. The rings, driven by the roller pins, will be rolling over them with \( \omega_v \), while each ball in the spherical
seats of the rollers and the cover will orbit around the roller axis, so that spin friction will be carried out in the power contacts. Therefore the developed NRS belongs to screw rolling mechanisms.

6. Theoretical studies of NRS under load

The purpose of the studies is defining the permitted axial force $F_{\Sigma A}$ acting on the mechanism, figure 7, and its operational life. Load capacity and operation life of ball screws and roller screws is accessed by the means of static $C_{0A}$ and dynamic $C_{\lambda}$ load carrying capacities [13, 14].

Let’s assume that all rollers are loaded equally. Figure 7 shows: screw 1, roller 2, two balls 3, ball supports 4 and two rings 5. The following is indicated in figure 7: $F_{AP}$ is axial force acting upon the roller; $F_{K1}$ and $F_{K2}$ are radial forces acting on the 1st (left) and 2nd (right) rings; $F_{N}[J]$ is normal force, acting on $J$-th turn of the roller thread. The number of thread turns on roller generator $m$, where $m$ is of several dozen.

The specific feature of roller loading is generation of an overturning moment from a pair of forces $F_{AP}$ with the arm $(d_2r/2)$. A mathematical NRS model [15] is developed to eliminate redundancy, which considers overturning moment, acting upon the roller, contact flexibility of the interfacing screw and roller thread turns and the ring tension and bending flexibility.

In order to implement the mathematical model a numerical method with forward and backward elimination and a special computer program [15] were developed.

These allows defining the most heavily loaded thread turns of NRS parts, the most heavily loaded junction of a ring with a roller pin, figure 8, and contact force in the ball - spherical seat junction, figure 9.
$E$ and $\mu$ shown in figure 8 and figure 9 are elastic modulus and Poisson’s ratio for the correspondent part.

After defining a pair of the most heavily loaded screw and roller thread turns and the normal force, acting in their junction [15], let's calculate the stress-strain behavior of the thread turns using the Hertz method for an initial point contact [16]. Let's define the stress-strain behavior for the heaviest loaded ring, figure 7, [15] with consideration of tensioning and bending [17] and calculate the stress-strain behavior for the ring, figure 8, using the Hertz method for an initial point contact [16]. Let's define the stress-strain behavior, figure 9, for the initial point contact of the ball and the spherical seat [16].

The biggest stresses generated in these joints or in the most heavily loaded ring define the permitted static strength of the mechanism or its static load capacity $C_{0A}$. By varying dimensions it is possible to achieve static uniform strength of the parts in the joints.

By using the found correlation between the roller screw load carrying capacities $C_{0A}$ and $C_A$ and the number of screw threads $z$, its pitch diameter $d_{2n}$ and thread pitch $P$, a method for forecasting of dynamic load carrying capacity (life time) of these mechanisms based on the known static load carrying capacity $C_{0A}$ [15]

$$C_A = C_{0A} / (0.3638 \cdot (d_{2n})^{0.5729} \cdot (P \cdot z)_{0.2027}$$

7. Experimental analysis of NRS

An electromechanical drive, an NRS test model, figure 10, and a test rig, figure 11, were developed and manufactured for the tests. All parts of the test model are manufactured on home-produced equipment [15].

The test NRS model assembly, figure 10, consists of a screw and a set of rollers, which are fixed to the screw by two rings.

The power part of the rig, figure 11, consists of an electromechanical drive based on NRS 2 and a pneumatic cylinder 6, fixed to the frame 1 and connected to each other via a force sensor 3, a coupling 4 and a flexible coupling 5 (MK–PC080).

The results of theoretical studies on measuring the movement of the ring outer surface points, located approximately at the same radius with the roller axis [15], were experimentally proven.

The assembled and adjusted NRS without greasing and loading was studied in order to define kinematic parameters [15]. The difference between the theoretical and experimental values of the gear ratio is 3%.

A tendency for increasing the operation speed of electromechanical drives, and, consequently, their actuators emerged in the recent years. During NRS tests, the value of the output element forward speed of 379.4 mm·s\(^{-1}\) under the maximum possible motor shaft rotation rate of 2036.00 rev·min\(^{-1}\) (the nominal motor shaft rotation rate is 2000.0 rev·min\(^{-1}\)) was achieved [15]. The received rate value
is not the limiting value for this mechanism. Defining the maximum possible linear velocity of NRS output element requires continuing the tests with a more high-speed motor.

The maximum value of efficiency factor in the experiments is 81.4% under the load of 2.0 kN and with the screw rotation rate of 50.0 rev·min⁻¹, the mean value of efficiency factor is about 76%. It should be noted, that these efficiency factor values include losses in the whole drive, consisting of: a roller screw, bearing units, guides, sealings, junction couplings, anti-rotation device etc. [15].

In order to define the rotating resistance moment of the NRS test model [15] a cold chamber NZ-250/70 and a warm chamber KT-0.05/315M were used. The drive was kept in one of the specified chambers with the specified temperature within 2 hours, after that it was taken from the chamber for testing, figure 12. The tests showed that NRS is operable under the temperatures in the range of from –60°C to +80°C.

![Figure 12. Drive with NRS under the temperature of –60°C.](image)

Life time tests for NRS drive have been conducted in laboratory conditions at a test rig, controlled by LabVIEW and a computer, during one and a half years [15]. During this time the test model was in operation for more than three thousand hours. This corresponds to approximately 1.8 mil. cycles of back and forth movement of the NRS output element. The load to the output element was about 2000 N ± 200 N. The operation (no-load) stroke was 70 mm. The load and the procedure were controlled by LabVIEW and a computer.

Grease «Plasgir» as per STO 07548712-008-2010 was used for the tests. The grease has not been changed during the tests. A range of parameters was measured during the life time tests. For example, the steady-state temperature of the NRS test model depended from the season and temperature of the environment and in different days was from +32°C to +47°C. A bellows coupling failed and was changed during the tests. By the end of the tests the temperature in the place of double-sided angular-contact ball bearing started to grow and a knocking sound appeared, that means the support devices, as in PRS case, make problems for the drive. Inspection of the NRS test model parts showed, that there were no serious wear of threaded parts. NRS stayed operable.

The produced research proved the operability of NRS test model under a wide range of loads, linear velocities of output element and temperatures.

The paper [15] contains comparison of the developed NRS parameters with the parameters of the traditional roller screws is given in the paper. The values of some of the compared parameters are similar, for some parameters NRS is better than a roller screw, while for the others it is not. There are some parameters, which can't be compared due to lack of information, since there are no data either for a roller screw, or, in the most cases, for NRS. Therefore the further studies of this new mechanism are required.

8. Conclusion
Production of threaded parts for the traditional roller screws requires special expensive high-precision equipment. Threaded parts for NRS can be manufactured on correspondent high-precision equipment,
including the universal one. There is no need for a machine tool for grinding a thread on the inner surface of the nut. This is the main advantage of these mechanisms.

For the present time the developed NRS can be used in nonvital products under low and medium loads. Defining the rational fields of application of the developed NRS in vital products of machine engineering requires additional research.

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