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Research paper

Analytical prediction of the geometry of contact ellipses and kinematics in a roller screw versus experimental results

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ABSTRACT

This paper brings together the geometry of threaded contact areas and roller screw kinematics in order to shine some light on the local phenomena which dissipate power in the mechanism. The subject has received little attention in current literature, often because only the global kinematics is studied. Moreover, principal directions of curvature are usually assumed from the start. In this paper, differential geometry is used to calculate the result, which proves to be different from published research. Next, the classic Hertzian theory is adapted for slightly conforming contacts, which can be encountered in common roller screw designs. The generalized equations are used to deduce the shape, size and orientation of the contact ellipses for both the roller-screw and the roller-nut contacts. The two appear to be very different in terms of local kinematics. A new stationary model is introduced, which can calculate the sliding velocity field at any point within the contact area. The model has only one degree of freedom in the form of a slip ratio, which depends on lubricant properties and dynamics. An experimental setup was designed to measure this ratio and thus allow comparison with analytical models available in the literature.

1. Introduction

The roller screw mechanism is a powerful rotation-translation converter used in a variety of industries. Two main types exist today: standard and inverted. The mechanism's main advantage in comparison with classic friction screws is the introduction of rolling elements (in this case, threaded cylinders), which greatly reduce friction and increase efficiency, while sustaining high loads and precision. Its qualities have allowed the roller screw to be considered as a replacement for hydraulic cylinders [2].

The three main components (screw shaft, rollers and nut) are all threaded and might have a different number of starts. Several geometric and kinematic conditions have to be imposed for the mechanism to function correctly. These conditions have been extensively studied before [3–6]. In practice, however, it is virtually impossible to satisfy all of them simultaneously, mainly due to elastic deformations, wear and manufacturing imperfections. For this reason, a planet carrier and a system of gears are added, such that the mechanism works similarly to an epicyclic gear train, with the addition of axial movement and some amount of circumferential slip. The gears were already present in the initial patent [7].

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Table 1
Notation.

a	[m]	Major Hertz ellipse half-diameter
b	[m]	Minor Hertz ellipse half-diameter
c	[m]	(>0) radial distance between screw (or nut) and roller axes
E'	[Pa]	Relative Young modulus of elasticity, see Eq. (40)
$F_{/n}$	[N]	Resistive force on the nut
$h(r_m)$	[m]	Shape function, depending on the thread profile
h', h''	[,1/m]	First and second derivatives of $h(r_m)$
k	[ ]	(≥ 1) ellipticity ratio, $k = a/b$, $k \geq 1$
l	[m]	Lead; if right-handed: $l > 0$, left-handed: $l < 0$; in general: $\|l\| = np$
n	[ ]	Number of thread starts
$\overrightarrow{n}$	[, ,]	Surface/contact normal vector at point M
N	[N]	(≥ 0) contact normal load
N_C	[ ]	Number of R/S or R/N contacts per roller
p	[m]	(>0) axial pitch of the threaded profile
r_B	[m]	Profile curvature radius
r_m,θ_m, z_m	[m,rad,m]	Cylindrical coordinates for a point on the threaded surface
$\overrightarrow{t_{1,2}}$	[, ,]	Principal directions of curvature
r_{IS},θ_{IS}; r_{IR},θ_{IR}	[m,rad]	Polar coordinates of I from the screw (S) and roller (R) axes, see ref[1].
r_{JN},θ_{JN}; r_{JR},θ_{JR}	[m,rad]	Polar coordinates of J from the nut (N) and roller (R) axes, see ref[1].
r	[m]	Nominal (pitch) radius
$\overrightarrow{v_{t/p}}(M)$	[m/s,m/s,m/s]	Relative (sliding) velocity of (t) with respect to (p) at point M
x, y, z	[m,m,m]	Cartesian coordinates
$\dot{z}_{n/s}$	[m/s]	Lead speed of the nut along z with respect to the screw
α_n	[rad]	Normal pressure angle, usually around $\pi/4$
β	[rad]	Thread helix angle; if right-handed: $\beta > 0$, left-handed: $\beta < 0$, $\tan\beta = l/2\pi r$
γ	[ ]	Function that specifies the top face of the thread (-1) or the bottom face (+1)
Γ	[ ]	Gear (overdrive) ratio, see Eq. (45)
λ	[ ]	Non-dimensional ratio related to ϵ by Eq. (48)
ϵ	[ ]	Non-dimensional slip ratio, $\epsilon = \omega_{p/n}/\omega_{s/n}$
$\rho_{1,\,2}$	[1/m]	Principal curvatures, $\|\rho_1\| \leq \|\rho_2\|$
χ	[ ]	Boolean indicator of the PRS type: 1 for standard and 0 for inverted
$\omega_{s/n}$	[rad/s]	Rotation speed around z of the screw (s) with respect to the nut (n)
I	Roller-screw contact point	
J	Roller-nut contact point	
PRS	abbreviation for Planetary Roller Screw	

The introduction of gears to constantly correct the position of the rollers only works if the pitch diameter of the gear and thread are identical, otherwise a phenomenon known as roller migration occurs [8]. As shown by Zhang and Zhao [9,10], it is possible to machine the two features at the same time and thus ensure a good synchronization. This reduces the axial displacement error to a very small percentage ($<0.01\%$ in [11]), making the roller screw one of the most precise rotation-translation converters there is. Further studies on the transmission accuracy have been performed by Ma et al. [12] under MSC Adams and experimentally by Mamaev et al. [13].

Some of the current literature focuses on the contact point locations on the threads. Analytical formulas were proposed by Jones and Velinsky [5], but the model does not take backlash into account and profile shapes cannot be modified. A more recent analytical study performed by Liu et al. [14] shows that elliptical profiles provide better performance than circular or parabolic ones. Backlash along all three directions (radial, transversal and axial) is eventually considered in a numerical model proposed by Fu et al. [15].

Other authors studied the load distribution among different thread pairs, which depends on the way the mechanism is loaded [16,17]. In the static case, finite elements models have been published by Abevi et al. [18,19]. The transient, dynamic case was investigated by Fu et al. [20] using a linear distribution for normal contact forces. Models for the axial stiffness of standard roller screws have been proposed by Ma et al. [4], Jones and Velinsky [21] and Zhang et al. [22].

Wear and tribology have also been subjects of interest for authors like Aurgan et al. [23,24], who studied the stick/slip distribution within the contact area and the damage modes with a specific coating. Sokolov et al. [25] developed principles for evaluating wear resistance. Finally, the thermal aspect was analyzed using a 2D FEM model in ANSYS [26] and the frictional moment deduced in [27].

The current paper provides detailed information on how the shape, size and orientation of the contact areas between threads can be obtained. The subject has received little attention, yet is of significant importance in choosing the optimal roller screw dimensions for a given application. We adapt the classic Hertzian theory for slightly conforming contacts, in order to make the equations more robust and ensure that all possible cases are covered. In the context of roller screws, other authors [20] have also used the Hertz ellipse to model contact areas, but without explaining how to obtain it. While principal surface curvatures and directions of curvature are important model inputs, current literature [4,5] often presumes the results without a clear justification. In this work, differential geometry is used to obtain values which prove to be different.

One of the hypotheses we use is that all roller-screw contacts are identical and all roller-nut contacts are also identical. This assumption does not take the load distribution among threads into account, but greatly simplifies the problem. We