Research Article

Kinematics of Planetary Roller Screw Mechanism considering Helical Directions of Screw and Roller Threads

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Based on the differential principle of thread transmission, an analytical model considering helical directions between screw and roller threads in planetary roller screw mechanism (PRSM) is presented in this work. The model is critical for the design of PRSM with a smaller lead and a bigger pitch to realize a higher transmission accuracy. The kinematic principle of planetary transmission is employed to analyze the PRSM with different screw thread and roller thread directions. In order to investigate the differences with different screw thread and roller thread directions, the numerical model is developed by using the software Adams to validate the analytical solutions calculated by the presented model. The results indicate, when the helical direction of screw thread is identical with the direction of roller thread, that the lead of PRSM is unaffected regardless of whether sliding between screw and rollers occurs or not. Only when the direction of screw thread is reverse to the direction of roller thread, the design of PRSM with a smaller lead can be realized under a bigger pitch. The presented models and numerical simulation method can be used to research the transmission accuracy of PRSM.

1. Introduction

Planetary roller screw mechanism (PRSM) is used in various motion-delivery devices where power is transmitted by converting rotary motion to linear motion. The main components of PRSM are the nut, the screw, and the rollers, and the key components for transmission are the rollers. Compared to ball screw mechanism, PRSM has higher precision, higher speed, heavier load, and longer life, though the manufacture cost of PRSM is relatively high due to its complex structure. As such, the PRSM finds its applications as an actuator device in various machineries, such as machine tool [1], medical equipment [2], port equipment of ship [3], and flight control equipment of more-electric aircraft [4, 5].

The published research on the PRSM has been mainly dedicated in the following areas: efficiency and failure modes study [6]; dynamic load testing [7]; wet and dry lubrications under oscillatory motion [8]. Besides, Hojjat and Mahdi [9] analyzed the capabilities and limitations of PRSM and proved that large leads and extremely small leads can be easily obtained in PRSM. The forces acting on the rollers during

the rotation of screw have been analyzed for investigating the slip phenomenon. Velinsky et al. [10] applied the concept of orbital mechanics to study the kinematics and the efficiency of PRSM. They have found that although slip has to occur between the rollers and the screw in the PRSM, the overall lead of the mechanism is independent of such slip. Sokolov et al. [11] developed the principles to evaluate the wear resistance of the PRSM. Jones and Velinsky [12] built a kinematic model to predict the axial migration of the rollers with respect to the nut in the PRSM. It is announced that roller migration is due to slip at the nut side, which is caused by a pitch mismatching between the spur-ring gear and effective nut-roller helical gear pair. The results indicate that the roller migration does not affect the overall lead of the PRSM. Furthermore, they have applied the principle of conjugate surfaces to the contact kinematical modeling at the screw-roller and nut-roller interfaces [13]. It was shown that the contact point cannot locate on the bodies' line of centers at the screw side. Considering the curved profile of roller thread, the equations for calculating contact radii of the roller, screw, and nut bodies have also been derived.

Recently, Jones et al. [14] constructed a stiffness model of the roller screw mechanism with the direct stiffness method. In addition to the prediction of the overall stiffness of the mechanism, the load distributions across the threads of individual bodies were also calculated. Ryś and Lisowski [15] presented the computational model for predicting the load distribution between the elements in PRSM based on the deformations of rolling elements as the deformations of rectangular volumes subjected to shear stresses. The contact deformations of threads and the deformations of screw and nut cores were considered by introducing a properly chosen shear modulus. However, there is no comprehensive kinematics model that can apply differential principles to the design or estimate the lead of PRSM, when the helical directions between the screw thread and the roller thread are considered.

To address the aforementioned problems and facilitate the PRSM design, a kinematic model and a comprehensive study on the helical directions between screw thread and roller thread based on differential principle of thread transmission are developed in this work. The mechanical structure of PRSM is introduced, followed by the motion analysis and computational modeling of the lead. Then, kinematics simulations of the PRSM are performed to validate the motion analysis which considers the helical directions between the screw thread and the roller screw. Finally, the models are examined in detail to explicitly show the relationships of the helical directions and the lead of the PRSM in the force analysis.

2. Analytical Modeling

2.1. Mechanical Structure. Figure 1 shows a transmission mechanism of PRSM. The rotary motion of the screw shaft is converted into an axial force through the operation of the rollers and the nut on the shaft. The efficiency of such process is much greater than that of a conventional threaded shaft-nut mechanism because of the introduction of the rollers which are in rolling contacts with both the screw and the nut instead of sliding contacts in the conventional mechanism. There is no relative axial movement between the nut and the rollers since the helix angle of the roller is the same as that of the nut. Therefore, the rollers and the nut are able to move axially with respect to the rotating screw shaft. At the same time, the rollers and the nut remain in constant rolling contact.

The roller possesses a single start thread and two spur gears at each end of the roller and ring gears fixed at each end of the nut. The rollers roll around the inside of the nut as the screw turns, and one revolution of the screw causes the nut to advance one lead regardless of rolling or slipping between the components. The grooves on the nut and screw are helical with multistart. In order to ensure that the rollers do not become skewed or driven out of axial alignment between the screw and the nut, a planetary carrier and a gear pair are provided. The planetary carrier is located under each pivot end of the rollers and keeps the rollers spaced apart circumferentially around the screw. A gear pair includes a ring gear and spur gear on the roller. The ring gears time the spinning and orbit of the rollers about the screw axis by meshing gear teeth near the ends of the rollers. At the same, the planetary

FIGURE 1: Configuration of PRSM.

FIGURE 2: Kinematic principle of PRSM.

carriers float relative to the nut, axially secured by the spring rings which aligned with an axial groove in the wall of the nut.

2.2. Motion Analysis

2.2.1. Angular Motion Analysis of Components. Based on the principle of a planetary gear train, the kinematic principle of PRSM is shown in Figure 2, and the angular motions of components in PRSM are described in Figure 3 which represents the axial view of the PRSM.

According to the relationships of the movements and assuming no slip between the screw and the rollers, the linear velocity of contact point B between the roller and the screw is defined as v_B in Figure 3. Since the rotational direction of nut is prohibited, the linear velocity of contact point A between the roller and the nut is zero. The linear and orbital speeds of the roller center point O are $v_o = v_B/2$ and ω_m , respectively. The linear speed of the roller center can be further expressed as $v_o = v_B/2 = \omega_s d_s/4$, where ω_s and d_s are the angular velocity and effective diameter of the screw, respectively. On the other hand, v_o can also be shown as $v_o = \omega_m d_m/2$,

FIGURE 3: Axial view of PRSM.

where d_m denotes orbital diameter of roller. Accordingly, the relationship between the orbital speed of center point O, ω_m , and angular velocity of screw ω_s can be written as

$$\omega_m = \frac{d_s}{2d_m} \omega_s = \frac{d_s}{2(d_s + d_r)} \omega_s = \frac{k}{2(k+1)} \omega_s, \quad (1)$$

where $k = d_s/d_r$.

Then the nut is fixed in the rotational direction, assuming that a roller travels from an initial point A to a final point F with one revolution of the screw. To analyze the angular motion of the components, φ_m and φ_r (not shown in Figure 3) are defined as orbital angle and rotational angle of the roller and φ_{nm} denotes the angular arc of contact of screw with roller. The pure sliding angle φ_{slide} must be zero; therefore, the angular arc of contact of roller equals the angular arc of contact of nut, $\widehat{AE}$; that is,

$$\frac{\varphi_r d_r}{2} = \frac{\varphi_m d_n}{2}, \quad (2)$$

where d_r and d_n are the effective diameter of the roller and the nut, respectively.

Considering the relationships between the effective diameters of the screw and the roller and between the roller and the nut, (2) can be rewritten as

$$\frac{\varphi_r}{\varphi_m} = \frac{d_n}{d_r} = \frac{d_s + 2d_r}{d_r} = k + 2. \quad (3)$$

Combining (1) and (3) and using the relationship $\varphi_r/\varphi_m = \omega_r/\omega_m$, the rotational speed of the roller ω_r can be given as

$$\omega_r = \frac{k(k+2)}{2(k+1)} \omega_s. \quad (4)$$

The relationships of helix angles of the screw, the roller, and the nut in terms of pitch, starts, and effective diameters are given in the following [10]:

$$\begin{aligned} \tan \lambda_s &= \frac{n_s p}{\pi d_s}, \\ \tan \lambda_r &= \frac{n_r p}{\pi d_r}, \\ \tan \lambda_n &= \frac{n_n p}{\pi d_n}, \end{aligned} \quad (5)$$

where λ_s , λ_r , and λ_n are the helix angles of the screw, the roller, and the nut, respectively. n_s , n_r , and n_n are the starts of the screw, the roller, and the nut, respectively. p is the pitch.

The helix angles of the roller and the nut are equal, that is, $\lambda_r = \lambda_n$ and combining $n_s = \omega_r/\omega_m = d_n/d_r$ and $d_s/d_r = z_n/z_r - 2 = n_s - 2 = n_n - 2$, where z_n is the tooth number of ring gears and z_r is the tooth number of gears near the ends of rollers, respectively. The relationship can be obtained as follows:

$$d_s = (n_s - 2) d_r. \quad (6)$$

Equation (6) is utilized to calculate the starts if d_s and d_r are given. On the other hand, the starts of the screw have to be more than or equal to 3; that is, $n_s = n_n \geq 3$. Considering the angular motion of the PRSM is quite similar to the motion of a planetary gear train, the relationships between the angular velocities can be shown as follows:

$$\begin{aligned} \frac{\omega_s - \omega_{p1}}{\omega_r - \omega_{p1}} &= -\frac{d_r}{d_s}, \\ \frac{\omega_s - \omega_{p2}}{\omega_n - \omega_{p2}} &= -\frac{d_r z_n}{d_s z_r}, \end{aligned} \quad (7)$$

where ω_{p1} , ω_{p2} are the angular velocities of the planetary carriers on the left side and right side, respectively. The transmission ratio between the screw and the roller is defined as i_{sr} and that between the roller and the nut is i_{nr} .

In order to ensure pure rolling of the rollers inside the nut, the angular velocity of the planetary carrier on the left side must be equal to that on the right side, which can be written as

$$\omega_{p1} = \omega_{p2} = \omega_H. \quad (8)$$

The transmission ratios of components are shown as

$$\begin{aligned} i_{sr} &= \frac{\omega_r - \omega_H}{\omega_s - \omega_H} = -\frac{d_s}{d_r}, \\ i_{nr} &= \frac{\omega_n - \omega_H}{\omega_s - \omega_H} = -\frac{d_s}{d_n}. \end{aligned} \quad (9)$$

Utilizing the relationship for the starts of nut, that is, $n_n = (d_s + 2d_r)/d_r = d_n/d_r$, and (9), it yields

$$\omega_H = \frac{d_s}{d_s + d_n} \omega_s = \frac{n_s - 2}{2n_s - 2} \omega_s. \quad (10)$$

The relationships of angular velocities between the screw, the roller, and the planetary carrier are described as

$$\omega_s - \omega_H = \frac{d_n}{d_s} \omega_H = \frac{n_s \omega_s}{2n_s - 2}, \quad (11)$$

$$\omega_r - \omega_H = -\frac{d_s}{d_r} (\omega_s - \omega_H) = \frac{-n_s^2 + 2n_s}{2n_s - 2} \omega_s. \quad (12)$$

2.2.2. Helical Direction and Parameter Relationships on Nut Side. As aforementioned, the roller rolls on the inner surface of the nut. The helix angles of the two components are identical. The roller gear meshes with the ring gear. No slip between the roller and the nut is allowed; however, there is always slip between the screw and the roller in the axial direction [10]. Accordingly, for the case with roller-screw slip, the angular motion of the components can be decomposed into two components, that is, the relative motion without rotational slip and the relative motion with pure rotational slip. As shown in Figure 3, $\widehat{AF}$ denotes the angular arc on the surface of the nut that is in contact with the roller within one revolution of the screw. $\widehat{BD}$ denotes the angular arc on the surface of the screw that is in contact with the roller assuming that no slip occurs between the screw and the roller. $\widehat{DC}$ denotes the angular arc of pure sliding motion between the screw and the roller.

Because the roller and the nut have different leads and effective diameters, we assume that the axial displacement of roller relative to nut, L_1 , can be decomposed into two components: the axial displacement of a rotationally constrained roller relative to a rotating screw L_{1r} and the axial movement of a rotating roller relative to a fixed nut L_{1c} . The simple relationship yields

$$L_1 = L_{1r} + L_{1c}. \quad (13)$$

Based on the relative movement of the components, L_{1c} is equal to axial displacement of the nut L_{1n} in which it is hypothesized that the nut rotates an angle φ_m with angular velocity ω_n , that is, $\widehat{AE}$, but the direction of L_{1c} is reversed to L_{1n} ; that is,

$$L_{1c} = -L_{1n}. \quad (14)$$

As discussed above, the L_{1n} is influenced by the angle φ_m and the lead of the nut $n_n p$, where n_n and p are the starts and pitch of the nut, respectively. Similarly, the axial displacement of the roller relative to the nut, L_{1r} , is influenced by the rotational angle of the roller φ_r and the lead of the roller $n_r p$, where n_r is equal to one. Therefore, axial displacements L_{1r} and L_{1n} can be written as

$$\begin{aligned} L_{1r} &= \frac{\varphi_r}{2\pi} p \\ L_{1n} &= \pm \frac{\varphi_m}{2\pi} n_n p, \end{aligned} \quad (15)$$

where the negative sign indicates that the helical directions are identical between the roller and the nut and positive sign denotes that the helical directions are reversed between the roller and the nut.

Substituting (14) and (15) into (13), it can be rearranged as

$$L_1 = \frac{\varphi_r}{2\pi} p \mp \frac{\varphi_m}{2\pi} n_n p. \quad (16)$$

It is well known, in the PRSM, that there is no relative axial displacement between the nut and the roller; that is, $L_1 = 0$. Therefore, symbol “−” should be chosen in (16); on the other hand, the helical directions should be identical between the roller and the nut. Substituting (3) into (16) yields

$$n_n = k + 2. \quad (17)$$

2.2.3. Helical Direction and Parameter Relationships on Screw Side. Considering the slip between the screw and the roller, similarly, the angular motions of the PRSM can be decomposed into two components which are the motion without slip and motion with pure sliding [10]. Therefore, the axial displacement of the roller relative to the screw, L_2 , is the sum of two components, that is, pure rolling which is generated on arc $\widehat{BD}$ and pure sliding on arc $\widehat{DC}$. Define L_{2r} and L_{2s}^* as the axial displacement components of the roller mentioned above, respectively. That is,

$$L_2 = L_{2r} + L_{2s}^*, \quad (18)$$

where

$$L_{2r} = L_{1r} = \frac{\varphi_r}{2\pi} p, \quad (19)$$

$$L_{2s}^* = -L_{2s}, \quad (20)$$

where L_{2s} is the axial displacement of screw relative to roller. L_{2s} is influenced by angular arc of contact of screw with roller φ_{nm} , pure sliding angle φ_{slide} , and lead of the screw $n_s p$, where n_s denotes the starts of the screw. The axial displacement of screw relative to the roller is expressed as

$$L_{2s} = \mp \frac{\varphi_{nm} + \varphi_{slide}}{2\pi} n_s p, \quad (21)$$

where the negative sign denotes that the helical directions of the screw and the roller are identical and positive sign indicates that the helical directions of the screw and the roller are reversed.

Based on the geometry relationship, as shown in Figure 3, one may get

$$\varphi_{nm} + \varphi_{slide} + \varphi_m = 2\pi. \quad (22)$$

Substituting (19), (21), and (22) into (18), and assuming that the pure sliding angle is equal to zero, the axial displacement of the roller relative to the screw can be represented as

$$L_2 = \frac{\varphi_r}{2\pi} p \mp \frac{\varphi_m}{2\pi} n_s p \pm n_s p. \quad (23)$$

While the relative sliding occurs between the roller and the screw for one revolution of the screw, the pure sliding angle, φ_{slide} , is a variable, and φ_c and φ_r are also the variables in (3) and (23). Therefore, the axial displacement of the roller

relative to the screw will generate a higher fluctuation during the screw rotation. Assuming there is no slip between the screw and the rollers in the rotational direction or in order to avoid pure sliding phenomenon, the first two terms in (23) have to be zeroes, so the negative sign should be used; that is,

$$\begin{aligned} \frac{\varphi_r}{2\pi}p - \frac{\varphi_m}{2\pi}n_s p &= 0, \\ L_2 &= \pm n_s p, \end{aligned} \quad (24)$$

where the positive sign denotes that the helical directions of the screw and the roller are identical and the negative sign indicates that the helical directions of the screw and the roller are reversed.

Because there is no relative axial movement between the nut and the rollers, the axial displacement of the roller relative to the screw is equal to the axial displacement of the nut, L_n . In other words, the lead $n_s p$ can be expressed as a function of the axial displacement of the nut. Thus, it is stated as

$$L_n = \frac{\omega_s t}{2\pi} n_s p, \quad (25)$$

where t denotes the operating time of the screw.

Furthermore, the axial speed of the nut is calculated by differentiating the displacement of the nut with respect to time, as shown in the following:

$$v_n = \frac{dL_n}{dt} = \frac{\omega_s}{2\pi} n_s p. \quad (26)$$

2.3. Lead of PRSM considering Helical Directions of the Screw Thread and the Roller Thread. Based on the analyses of Section 2.2, it is known that the helical direction of the roller thread must be identical with the helical direction of the nut thread, and the helical direction of the screw thread is identical with or reversed to the helical direction of the roller thread. Therefore, the following research is focused on lead of PRSM considering helical directions of the screw thread and the roller thread.

2.3.1. Identical Helical Directions of the Screw Thread and Roller Thread. For the case in which the helical direction of screw thread is identical with that of roller thread, the lead of the PRSM can be written as

$$L_0 = p \left[n_s - \frac{\omega_r}{\omega_s} - \frac{\omega_H}{\omega_s} (n_s - 1) \right], \quad (27)$$

where L_0 denotes the lead of the PRSM.

Substituting (4), (6), and (10) into (27), the lead of PRSM, L_0 , can be represented as

$$L_0 = n_s p. \quad (28)$$

Equation (28) indicates that the lead of PRSM is equal to the lead of the screw or the lead of the nut, which means that the lead of PRSM is only determined by the starts and the pitch of the screw or the nut.

2.3.2. Reverse Helical Directions of the Screw Thread and the Roller Thread. For the case in which the helical direction of the screw thread is reversed to that of the roller thread, the lead of the PRSM can be expressed as

$$L_0 = p \left[n_s - \frac{\omega_r}{\omega_s} - \frac{\omega_H}{\omega_s} (n_s + 1) \right]. \quad (29)$$

Similarly, substituting (4), (6), and (10) into (29), the lead of PRSM, L_0 , can be represented as

$$L_0 = \frac{n_s p}{n_s - 1}. \quad (30)$$

Also, the lead of PRSM is only determined by the starts and the pitch of the screw. Only when the helical direction of screw thread is reversed to the helical direction of the roller thread can the design of a bigger pitch and a smaller lead be realized with the same parameters of the starts and the pitch.

As indicated by (28) and (30), regardless of whether slip occurs between the screw and the roller or not, the lead of the PRSM is a constant due to the fact that the lead of the screw is not changed. If the slip occurs, however, the sliding of rollers can cause undesirable moments and heat due to friction. The frictional heat is directly related to efficiency and energy loss in the PRSM, and the high temperature from the heat will cause deterioration of lubrication and eventually leads to mechanical failure of the PRSM.

The leads of the PRSM can be calculated by using (28) and (30). For example, when the starts and the pitch of the screw are 5 and 0.5 mm, the leads of identical helical directions of the screw thread and the roller thread case and that of the reverse case are 2.5 mm and 0.625 mm, respectively. The former is four times the latter. Obviously, in order to obtain higher transmission accuracy, the reverse helical direction can be applied in the practical PRSM structure. If a smaller pitch especially can be obtained in machining, then the smallest lead of the PRSM can be further realized. Therefore, a higher transmission accuracy can be obtained if the reverse helical directions of the screw thread and the roller thread are applied to the PRSM.

Furthermore, the PRSM is an accuracy transmission which achieves the smallest lead by introduction of thread directions; however, compared to the conventional ball screw, the small lead is extremely difficult to reach due to the requirements of carrying capacity and transmission accuracy and the design difficulty of the return tube.

3. Numerical Modeling of PRSM

3.1. Kinematics Model of PRSM. A model of kinematics simulation (as shown in Figure 4) of PRSM is developed with a software MSC.Adams with an original CAD geometry of PRSM converted from Solidworks software. The parameters of thread pair and gear pair are shown in Tables 1 and 2.

3.2. Constraints

3.2.1. Displacement Constraints. Assume that all rollers have identical movements in the PRSM, and the steady state motion of the screw is considered in this paper.

FIGURE 4: The numerical model of PRSM.

TABLE 1: Parameters of thread pair.

Parameter name	Symbol	Unit	Value
Effective diameter of screw	d_s	mm	39
Starts	$n_s = n_n$		5
Pitch	p	mm	5
Effective diameter of roller	d_r	mm	13
Effective diameter of nut	d_n	mm	65

TABLE 2: Parameters of gear pair.

Parameter name	Symbol	Unit	Value
Module	m	mm	1
Tooth number of roller gear	z_r		13
Tooth number of ring gear	z_n		65
Pressure angle	α_n	°	20
Addendum coefficient	h_a^*		0.8
Clearance coefficient	c^*		0.3
Modification coefficients	x_n		0
Tooth width of roller gear	b_1	mm	10
Tooth width of ring gear	b_2	mm	10

Based on the relative movement shown in Figure 1, the displacement constraints applied to the PRSM are as follows: (1) the moving joint is enforced between the nut and the frame, which means only axial translation of the nut is reserved; (2) rotating joint is imposed between the screw and the frame, which only allows rotation of the screw; (3) the rollers may spin and revolute; therefore, rotating joints are applied between planet carriers and rollers; (4) column joint is introduced between the planet carriers and the frame, because the planet carriers have both revolution and axial translation. The connection relationships of components are shown in Figure 5 in detail.

3.2.2. Load Constraints. In order to realize the kinematic transmission, the load constraints in the kinematics model are as follows: contact interactions are applied at the interfaces between the screw and the rollers and those between the rollers and the nut. Similarly, the contact interactions are also applied at the interfaces between the spur gear of the rollers and the ring gears.

The stiffness coefficient is set as $K = 1.0 \times 10^5$ N/mm, rigidity index is 1.5, damping coefficient is 50 N·s/mm, and depth of penetration is 0.1 mm. The coulomb friction force is considered in this model for describing the real contact state, where the static friction coefficient is $\mu_s = 0.3$; the dynamic friction coefficient is $\mu_d = 0.25$; the static slip velocity is $v_s = 0.1$ mm/s; the dynamic slip velocity is $v_d = 10$ mm/s; elastic modulus is 2.1×10^9 Pa; Poisson's ratio is 0.3; density is 7.8×10^3 kg/m³; and axial force applied on the nut center of mass is 10 kN in a direction reversed to its movement. The constant revolution speed of screw is 720°/s; that is, $\omega_s = 12.5664$ rad/s; simulation time is set to 1.0 s.

4. Results and Discussions

4.1. Identical Helical Directions of Screw Thread and Roller Thread. When the helical direction is identical between the screw thread and the roller thread, the relationships of helical direction in PRSM are as follows: screw is right-hand, roller is right-hand, and nut is right-hand. The simulation results are shown in Figures 6–9.

As Figure 6 shows, the displacement curve of the nut is nearly a straight line due to the synchronized roller rotations since all rollers are connected together by the planetary carrier and the precise mesh between the roller gear and the ring gear.

Figure 7 exhibits the averaged angular velocity of the roller which is 1037.9664°/s, that is, 18.1159 rad/s, which is the resultant of spinning angular velocity and revolution angular velocity.

The averaged angular velocity of the planet carrier is 260.6253°/s, that is, 4.5488 rad/s, as is demonstrated in Figure 8. Based on the relative movement between the roller and the planet carrier, spinning angular velocity of the roller can be approached; that is, $\omega_r = 18.1159$ rad/s + 4.5488 rad/s = 22.6647 rad/s.

As shown in Figure 9, the averaged axial speed of the nut is 49.7138 mm/s. Comparisons of analytical solutions with simulation results are shown in Table 3.

4.2. Reversed Helical Directions of Screw Thread and Roller Thread. When the helical direction of the screw thread is reversed to that of the roller thread, the relationships of helical direction in PRSM are as follows: screw is right-hand, roller is left-hand, and nut is left-hand. The simulation results are shown in Figures 10–13.

Compared to Figure 6, a very analogous trend of axial displacement of the nut can be obtained, as shown in Figure 10. Figures 11 and 12 showed that the averaged angular velocity of roller is 1090.6219°/s, that is, 19.0349 rad/s, and the averaged angular velocity of planet carrier is 272.8972°/s, that is, 4.7630 rad/s, respectively. Based on the relative movement between the roller and the planet carrier, spinning angular velocity of the roller can be calculated as $\omega_r = 19.0349$ rad/s + 4.7630 rad/s = 23.7979 rad/s.

As shown in Figure 13, the averaged axial speed of the nut is 11.8169 mm/s. Comparisons of analytical solutions with simulation results are shown in Table 4.

TABLE 3: Comparison of the analytical solutions with simulation results.

	Displacement of nut at point A	Displacement of nut at point B	Angular velocity of roller (ω_r)	Angular velocity of planet carrier (ω_m)	Axial speed of nut (v_n)
Simulation results	24.8601 mm	49.8571 mm	22.6647 rad/s	4.5488 rad/s	49.7138 mm/s
Analytical solutions	25 mm	50 mm	23.5620 rad/s	4.7124 rad/s	50 mm/s
Relative error	0.5596%	0.2858%	3.8083%	3.4632%	0.5724%

FIGURE 5: The connection relationships of components in PRSM.

FIGURE 6: Axial displacement curve of the nut.

FIGURE 7: Angular velocity curve of the roller.

The analytical solutions of angular velocity of planet carrier, angular velocity of roller, axial speed of nut, and displacement of nut can be obtained by (10), (12), (26), (28), and (30), respectively. The comparisons of analytical solutions with simulation results are shown in Tables 3 and 4.

As shown in Tables 3 and 4, the analytical solutions are very close to the simulation results with errors less than 4% for identical helical direction (screw thread direction and roller thread direction) case and errors less than 6% for reverse helical direction (screw thread direction and

TABLE 4: Comparisons of analytical solutions with simulation results.

	Displacement of nut at point A	Displacement of nut at point B	Angular velocity of roller (ω_r)	Angular velocity of planet carrier (ω_m)	Axial speed of nut (v_n)
Simulation results	5.9200 mm	11.9506 mm	23.7979 rad/s	4.7630 rad/s	11.8169 mm/s
Analytical solutions	6.25 mm	12.5 mm	23.5620 rad/s	4.7124 rad/s	12.5 mm/s
Relative error	5.2800%	4.3952%	1.0012%	1.0738%	5.4648%

FIGURE 8: Angular velocity curve of planet carrier.

FIGURE 10: Axial displacement curve of nut.

FIGURE 9: Axial speed curve of nut.

FIGURE 11: Angular speed curve of roller.

roller thread direction) case, respectively. The relative errors may originate from the following. (1) The form of roller thread is designed with rounded half-section to enhance the carrying capacity and improve the contact characteristics. However, the radius of the rounded half-section (the radius can be denoted as $R = d_r/2 \sin \beta$, where β is contact angle of the roller thread) is decimal fraction in the numerical model, which leads to error of meshing position between the analytical model and the numerical model. (2) The slip ratio is a nonconstant, which leads to slipping between the screw and the roller and between the roller and the nut. Furthermore, the slip ratio cannot be ascertained in the numerical model. (3) The meshing clearance and impact

(the contact of components is defined by using impact function in the numerical model and the impact is correlative to meshing clearance) are considered in the numerical model, which lead to fluctuation of simulation results.

According to the results of numerical simulation, the angular velocity and axial speed curves of the components generate a higher fluctuation. In addition to the influence of impact and clearance, the sliding is another important factor. Therefore, the analysis of the forces has been performed.

When the helical direction of screw thread is identical with that of the roller thread, as shown in Figure 14, the friction force (equal to $\mu F_N \cos \lambda_r$, where μ is coefficient of friction) applied on the roller thread is in the helical direction

FIGURE 12: Angular velocity curve of planet carrier.

FIGURE 13: Axial speed curve of nut.

of roller movement and the tangent force component is opposite to the movement direction. Such configuration tends to slip and requires sufficient friction force to work properly [9]. On the other hand, the roller rotates due to friction force and the lack of friction force (compared with tangent force) causes slipping.

In Figure 14, F_a is axial force, F_t is tangential force, μF_N is friction force, and F_N is resultant force of F_a and F_t .

When the helical direction of screw thread is reversed to that of the roller thread, the force analysis is shown in Figure 15 [9].

It is similar to Figure 14; the directions of friction force (also equal to $\mu F_N \cos \lambda_r$) and the tangent force component are reversed. If there is not enough friction force between the screw and the rollers, the roller has tendency to slip.

Furthermore, the relative displacement errors shown in Table 4 are 9.44 times (point A) and 15.38 times (point B) the corresponding values in Table 3. In other words, the reversed helical directions of screw thread and roller thread have higher slipping tendency than the identical helical directions of screw thread and roller thread under the same constraint conditions.

FIGURE 14: Force analysis when thread direction is identical between the screw thread and the roller thread.

FIGURE 15: Force analysis when thread direction is reversed between the screw thread and the roller thread.

Besides, the results of Table 4 also indicate that some slip always occurs between the roller and the screw as a result of reversed relative movement direction between the screw and the roller. The slip is closely impacted by the rotational speed of the screw, axial load applied on the nut, lubrication conditions, and so on. Generally, the position accuracy of the PRSM is secured by applying a higher preload on the nut and operating at high axial load and low rotational speeds.

5. Conclusions

This paper develops the kinematics by analytical modeling and numerical modeling of the PRSM considering helical directions between screw thread and roller thread to provide a method to support its design and application. The major findings are as follows:

- (1) The analytical modeling considering helical directions between the screw and the roller threads in PRSM is presented to realize the design of PRSM with a smaller lead under a bigger pitch based on the differential principle of thread transmission. Numerical modeling is developed by using Adams to validate the proposed analytical solutions. Besides, the kinematic models and simulation method considering helical directions of screw and roller threads are available to PRSM, which are beneficial to the further research of the PRSM.

- (2) The analytical solutions are close to the numerical results with errors less than 4% and 6% when the direction of screw thread is identical with or reversed to the direction of roller thread, respectively.
- (3) When the helical direction is identical between the screw thread and the roller thread, the friction force applied on the roller thread is in the helical direction of roller movement. However, the tangential force component is opposite to the movement direction. Therefore, such case has slip tendency and requires sufficient friction force to work properly.
- (4) When the helical direction of the screw thread is reversed to that of the roller thread, the PRSM is an accuracy transmission which achieves the smallest lead by introduction of a bigger pitch and a smaller lead as compared to the conventional ball screw where the small lead is extremely difficult to reach due to design difficulty of the return tube.

Notations

b_1 :	Tooth width of roller gear
b_2 :	Tooth width of ring gear
c^* :	Clearance coefficient
d_s :	Effective diameter of the screw
d_m :	Denotes orbital diameter of roller
d_r :	Effective diameter of the roller
d_n :	Effective diameter of the nut
F_a :	Axial force
F_t :	Tangential force
F_N :	Resultant force of F_a and F_t
h_a^* :	Addendum coefficient
i_{sr} :	Transmission ratio between the screw and the roller
i_{nr} :	Transmission ratio between the roller and the nut
K :	Stiffness coefficient
L_1 :	Axial displacement of roller relative to nut
L_{1r} :	Axial displacement of roller relative to a rotating screw
L_{1c} :	Axial displacement of a rotating roller relative to a fixed nut
L_{1n} :	Axial displacement of the nut relative to roller
L_2 :	Axial displacement of the roller relative to the screw
L_{2r} :	Axial displacement component of the roller
L_{2s}^* :	Axial displacement component of the roller relative to the screw
L_{2s} :	Axial displacement of screw relative to roller
L_n :	Axial displacement of the nut
L_0 :	Lead of the PRSM
m :	Module of gear pair
n_s :	Start of the screw
n_r :	Start of the roller

n_n :	Start of the nut
p :	Pitch
R :	Radius of rounded half-section of roller thread
t :	Operating time of the screw
v_s :	Static slip velocity
v_d :	Dynamic slip velocity
v_B :	Linear velocity of the contact point
v_o :	Linear speed of the roller center point
v_n :	Axial speed of the nut
x_n :	Modification coefficient
z_n :	Tooth number of ring gears
z_r :	Tooth number of gears near the ends of rollers
α_n :	Pressure angle of gear pair
β :	Contact angle
λ_s :	Helix angles of the screw
λ_r :	Helix angles of the roller
λ_n :	Helix angles of the nut
μ_s :	Static friction coefficient
μ_d :	The dynamic friction coefficient
φ_m :	Orbital angle of the roller
φ_r :	Rotational angle of the roller
φ_{nm} :	Angular arc of contact of screw with roller
φ_{slide} :	Pure sliding angle
ω_m :	Orbital speeds of the roller center point
ω_s :	Angular velocity of the screw
ω_r :	Rotational speed of the roller
ω_{p1} :	Angular velocities of the planetary carrier on the left side
ω_{p2} :	Angular velocities of the planetary carrier on the right side
ω_H :	Angular velocities of the planetary carriers
ω_n :	Angular velocity of the nut.

Conflict of Interests

The authors declare that there is no known conflict of interests associated with this publication and there has been no significant financial support for this work that could have influenced its outcome.

Authors' Contribution

The authors confirm that the paper has been read and approved by all named authors and that there are no other persons who satisfied the criteria for authorship but are not listed. The authors further confirm that the order of authors listed in the paper has been approved by all of them.

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