Research on the friction pair of the space high-power and long-life rolling ring

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Abstract. With the development of spacecraft, the requirement for high-power, long-life and high-reliability electrical transmission technology of the rolling ring is increasingly urgent. Aiming at the friction pair’s structure of the rolling ring, relevant theoretical and experimental research was carried out. The kinematic model is established to determine the friction pair’s structure of the rolling ring, and the zero sliding of friction pair is verified by ADAMS simulation. The electrical contact resistance and fatigue life of the friction pair are theoretically analyzed, and the cycling life of the flexure was analyzed. The stress distribution of flexure during rolling was simulated by FEM. The lifetime experiment of 51.5 10^5 r that the prototype transmitted the 6.6 kW of single loop was verified. The experimental result shows that the electrical contact resistance and the fluctuation value of the single loop are less than 1 mΩ during the rolling, although the transmission power increased to 10 times as compared with the traditional slip ring. The kinematic model can provide a certain theoretical reference for design of the space high-power and long-life rolling ring.

1. Introduction
The collector ring is a device for electrical transmission between two continuous relative rotated equipment, which generally adopt the slip ring or the rolling ring [1]. The collector ring is widely used in various solar array drive mechanism (SADM), scanning drive mechanism (SDM) and other drive mechanisms of spacecraft whose operational reliability is directly related to the energy supply of spacecraft, that is, the collector ring is the life line of the spacecraft [2]. With the rapid development of aerospace engineering technology, the spacecraft has the long-time requirement of more than 15 years on low earth orbit (LEO), high-power and high-reliability electrical transmission. In contrast to the traditional slip ring, the rolling ring substitutes the rolling friction pair for the sliding friction pair of the traditional slip ring, which reduces the friction and wear of the friction pair and remarkably improves the operating life, transmission power and the electrical contact property (ECP) [3]. Hence, the rolling ring become a preferred solution to achieve the high-power and long-life electrical transmission for SADM [4]. Rolling ring had applied for spacecraft after several decades of development since its birth [5]. Power supplies of the rolling ring in international space station (ISS) has attained 65.5kW, which consists 24 power loops. To study the electrical transmission characteristics of the rolling ring, the various experimental studies on rolling ring applied for ISS has been performed by NASA [6-7]. Zheng et al. [8] built a model to obtain proper operating stress and structure parameters in flexure design to ensure its reliable electrical contact between the inner and outer conducting ring. Hou et al [9] proposed the primary schematic of an innovative Multi-Rotary joints concept by using various power rolling ring to solve the single point failure of the space station. Liu et al [10] established the model of contact resistance to investigate the rolling ring’s key parameters and finished the steady-state thermal analysis of different current transferring.

However, the above theory of the rolling friction pair’s design is still not perfect and requires to be improved. The objective of this work was to establish the kinematic model of the rolling friction
and verified its reasonability. Verification was conducted by means of both numerical simulation (FEM and ADAMS) and experimental tests. The motivation for this work was to produce a general model, obtained the key structural parameters achieving the rolling ring’s high-power and long-life electrical transmission for SADM, and provided a theoretical reference for the design of the rolling ring.

2. Structural design and analysis
2.1. Friction pair structure design
In this paper, the space high-power rolling ring is composed of the inner and outer conducting rings, flexures, idlers and other parts and its configuration is shown in Fig. 1. The rolling friction pair is the core component of the rolling ring, including the inner and outer conducting rings, the flexure, the idler and the guide rail. Multiple pre-deformation flexures are embedded between the inner and outer conducting rings, which is used to obtain a greater transfer current capacity. The outer conducting ring fixed, the inner conducting ring rotate clockwise, and multiple flexures rotate counterclockwise and revolve clockwise, which acquires high-power electrical transmission between the inner and outer conducting rings.

![Figure 1. Structure of the high-power rolling ring](image1.jpg)

To avoid sliding friction at rolling interface, the structural size of various rolling components must be ensured to realize the zero sliding of the friction pair. The inner edge of idler is used to provides rolling interface at every flexure and idler contact interface, which prohibit sliding between adjacent flexure. During the rolling, the outer edges of the idlers are piloted by a guide rail that is fixed to and rotate with the inner conducting ring, which averts motion interference between the flexures or influence to electrical transmission performance of the rolling friction pair. The schematic of the rolling friction pair is shown in Fig. 2, and the zero sliding analysis is performed in the next.

![Figure 2. Schematic of the friction pair](image2.jpg)

The simplified schematic of kinematic model is shown in Fig. 3. There are two basic drive chains in the rolling friction pair from Fig. 3(a), which respectively are recorded as $L_1$ (inner conducting ring—flexure—outer conducting ring) and $L_2$ ($OO_A$—idler—guide rail).
Figure 3. The kinematic model of rolling friction pair

Assuming that the various elements are zero sliding at the rolling interface, the angular velocity’s relationship for two drive chains expressed as

\[
\begin{align*}
L_1: \quad \frac{\alpha_1 - \alpha_{h1}}{\alpha_{h1}} &= \frac{\alpha_1 - \alpha_{h1}}{\omega_{h}} = \frac{R_{o}}{R_{i}} \\
L_2: \quad \frac{\alpha_1 - \alpha_{h1}}{\alpha_{h1}} &= \frac{r}{R_{i}} \\
\end{align*}
\]

Where \(R_{i}\) and \(R_{o}\) respectively represent the radius of the inner and outer conducting ring; \(R_{b}\) represents the radius of the guide rail; \(r\) represents the radius of the flexure; \(R_{r}\) represents the radius of the inner edge of the idler; \(R_{\alpha}\) represents the radius of the outer edge of the idler; \(\omega_{1}\) represents the angular velocity of the inner conductive ring and the guide rail; \(\omega_{h1}\) represents the revolution angular velocity of the idler; \(\omega_{2}\) represents the rotation angular velocity of the flexure; \(\omega_{3}\) represents the rotation angular velocity of the idler.

Since the drive chains \(L_1\) and \(L_2\) have an equivalent angular velocity, the velocity matching condition at their engagement point is given as

\[
R_{p} \omega_{h} = -r \omega_{2} \tag{2}
\]

The instantaneous centers and velocity vectors involved of every element are shown in Fig. 3(b). The velocity vectors of \(P_2\) and \(P_3\) are respectively recorded as \(v_1\) and \(v_2\). Because the instantaneous center of the flexure is \(P_1\), the velocity direction at \(P_2\) of the flexure is along the line \(P_2P_1\). Therefore, the velocity direction of the idler at \(P_2\) must also follow the line \(P_3P_4\). According to the velocities of the idler at \(P_1\) and \(P_3\), the instantaneous center of the idler is \(P_3\) and the following relationship is obtained, expressed as

\[
\begin{align*}
\frac{v_2}{v_1} &= \frac{P_2P_3}{OP_2\omega_{h}} \\
\frac{OP_2}{OP_3} &= \frac{R_{o}}{R_{r}} \\
\end{align*}
\]

Considering the geometric relationship in Fig. 3(b), the \(\alpha\) and \(\beta\) are deduced

\[
\begin{align*}
\beta &= 45^\circ + \frac{\theta - \alpha}{2} \\
\alpha &= \arccos \left( \frac{(R_1 + r)\sin \theta}{r + R_r} \right) \tag{4}
\end{align*}
\]
Where $\alpha$ is the acute angle between the lines $P_2P_3$ and $P_2P_5$; $\beta$ is the acute angle between the lines $P_2P_5$ and $P_2P_9$; $\theta$ is the acute angle between the lines $OP_2$ and $OP_5$. Similarly, the analytical expressions of lines $P_2P_3$, $OP_2$ and $P_4P_6$ are given as
\[
\begin{align*}
\frac{y-(R_t+2r)}{\tan \beta x} &= -\tan \beta x \\
\frac{y-\cot \theta x}{OP_2} &= -\tan \theta x \\
\frac{y+\tan(90^\circ+\alpha)x+(R_t+r)}{P_4P_6} &= -\tan(90^\circ+\alpha)x+(R_t+r)
\end{align*}
\]

According to Eqs. (1), (2), (4) and (5), the $x$ and $y$ coordinates of $P_4$ and $P_5$ can be obtained, which is substituted into Eq. (6) and leads to
\[
R_\alpha - R_n = \frac{2(R_t+r)^2 R_n}{(R_t+r)R_t + R_n(R_t+2r)\tan\frac{90^\circ-\alpha+\theta}{2} + \cos \theta}
\]

The Eq. (6) implies the variables $R_\alpha$ and $R_n$. Given that the design parameters $r$ and $R_t$ are confirmed, it can be considered to solve Eq. (6) by the Quasi-Newton Methods.

The size design of the rolling friction pair that is composed of the inner and outer conducting rings, the flexures, the idlers and the guide rail is completed according to the above analysis. The size design ensures zero sliding at rolling interface of the rolling friction pair, which minimizes the sliding friction to reduce wear debris.

In space high-power rolling ring, contact resistance and fatigue life are the two most important indicators. The two parameters are analyzed in the next.

2.2. Fatigue life

During the rolling of the flexure, the instantaneous force analysis is shown in Fig. 4. Assuming that the flexure remains circular during the rolling, it is according to the analysis of the statically indeterminate structure in mechanics of material that the bending moment and the maximum stress of the flexure’s arbitrary section are obtained, which is given by

![Figure 4. The force analysis of the flexure](image)

\[
M(\gamma) = Fr\left[\frac{1}{\pi} \cos \frac{\gamma}{2}\right]
\]

\[
\sigma_{\text{max}} = \frac{Mt}{I}
\]

Where $\gamma$ denotes the angle between a section of the flexure and the positive direction of the $x$-axis.

Based on the material mechanics, the relationship between the deformation of the flexure and the radial bending force is given as
\[
\Delta = \frac{Fr^3}{EI} \left[\frac{\pi^2 - 2}{4}\pi\right]
\]
According to Eqs. (7), (8) and (9), the maximum stress of arbitrary section is obtained as

\[ \sigma_{\text{max}}(\gamma) = 2 \frac{Et \Delta}{r^2 \left( \pi^2 - 8 \right)} (2 - \pi \cos \gamma) \]  
(10)

During the clockwise rotation of the flexure, the bending moment and the maximum stress of section change periodically. When the flexure rotates 0°, the stress reaches the minimum; when the flexure rotates 90°, the stress reaches the maximum. According to Eq. (10), the theoretical maximum stress of the flexure is 87.1 MPa in this design.

According to the S-N curve of the QBe 2.0 [11], the fatigue life of the flexure was analyzed, as shown in Fig. 5.

![Figure 5](image-url)  
Figure 5. The fatigue life of beryllium bronze

During the rolling of the flexure, the maximum of theoretical stress is 87.1 MPa, which is less than the corresponding stress of 10^5 r for QBe2.0 (>200 MPa). Therefore, according to the theoretical results, the flexure can achieve the lifetime of the 10^5 r, which meets the requirements of the long-life electrical transmission of spacecraft.

3. Simulation analysis
3.1. Zero sliding analysis

The rolling friction pair was designed by the establishment of the kinematic model. It is using the ADAMS that analyzes the linear velocity of every element at the rolling interface to verify the zero sliding of designed friction pair.

At the point of the flexure contacting with idler, the linear velocities of the flexure and the idler was comparatively simulated. The result is shown in Fig. 6. It can be seen in Fig. 6 that the velocity ratio of the flexure to the idler is approximately 1 at the contacting point, that is, the flexure and the idler are zero sliding at the rolling interface.

![Figure 6](image-url)  
Figure 6. The velocities of contacting point between flexure and idler
At the contacting point between the flexure and the fixed outer conducting ring, the linear velocity of the flexure was analyzed, which is shown in Fig. 7. It can be obtained from the simulation analysis that the velocity of the flexure is 0 at the rolling interface, and the velocity of the outer conducting ring is 0. Therefore, it is not possible that the flexure and the outer conducting ring slide at the contacting point. In other words, the flexure and the outer conducting ring are zero sliding at the rolling interface.

![Figure 7. The velocity of contacting point between flexure and outer ring](image1)

According to the simulation analysis of the velocity, the flexure is zero sliding at the point of contacting with the idler and the outer conducting ring. Hence, it can be deduced that the flexure is impossibly sliding at the rolling interface with the inner conducting ring from theoretical analysis based on the simulation results in above.

3.2. Fatigue life

During the rolling of the flexure, the maximum stress of the flexure was verified by FEM simulation. Since each flexure has the same forced state, it can be simplified as a single flexure to cooperate with the inner and outer conducting rings. The small displacement collision model was used to analyze the stress distribution of the flexure in simulation at the rolling interface, the inner and outer ring. The material properties during the FEM simulation of cyclic stress are shown in Table 1.

![Figure 8. The stress distribution of the flexure](image2)

| Description       | Material | Density (kg/m³) | Young's modulus (GPa) |
|-------------------|----------|-----------------|-----------------------|
| Inner and outer   | Copper   | 8900            | 123.5                 |
| Flexure           | QBe 2.0  | 8300            | 137.0                 |

According to the asymmetric deformation of the flexure, the stress exists a slight difference at the area where the flexure respectively contacts with the inner and outer ring. The overall stress distribution of the flexure is shown in Fig. 8. From Fig. 8, the stress near the outer conducting ring is slightly larger with the maximum of 82.46 MPa. The stress near the inner conducting ring is slightly smaller, and the maximum is 81.98 MPa.
Considering the results of the theoretical calculation and simulation, the maximum stress of the flexure is less than 87.1 MPa during the rolling. When the cycle life of the QBe 2.0 reach $10^6$ r, the stress of the material is larger 200MPa. Hence, based on the analysis of theory and FEM simulation, the flexure could achieve the lifetime of $10^7$ r.

The lifetime of the rolling ring refers to the rotations of the inner conducting ring. According to the Eqs. (1) and (2), the relationship of the angular velocity between the flexure and the inner ring can be obtained. When the flexure rotates $10^6$ r, the rotations of the inner conducting ring is $1.4 \times 10^7$ r, which is the lifetime of the rolling ring.

The speed of the SADM is less than 0.1 revolutions per minute (rpm) on LEO. Hence, the rotations of the SADM is less than $10^4$ r during one year. The long-life requirement of the SADM is generally 15 years on LEO, that is, the lifetime of the SADM is required to be $1.5 \times 10^5$ r. According to the above analysis of the theory and FEM simulation, the lifetime of the designed friction pair of the rolling ring is much longer than $1.5 \times 10^5$ r, which can meet the long-life electrical transmission requirements of the SADM.

4. Experimental verification
On the basis of theoretical calculation and simulation analysis of the friction pair, a space high-power rolling ring prototype was developed, as shown in Fig. 9. The prototype consists of four loops that the assembly shown in Fig. 1 is a loop. Each loop is applied with the nominal 60A current and 110V voltage and the speed of inner ring is 20 rpm, which is used to verify the transmission capacity and the ECP of the prototype.

![Prototype of high-power rolling ring](image)

**Figure 9. The prototype of high-power rolling ring**

The contact resistance is composed of the constriction resistance and the film resistance. Since the area of contact spots is smaller than the nominal contact area, the current is shrunk when the current is transmitted through the "α spot" between the contact interfaces, so that the resistance is increased. This resistance is constriction resistance. In addition, the contaminated film of the contact surfaces increases the resistance of the contact spot, and the resistance is the film resistance. The factors affecting the contact resistance are complex, such as contact material, contact form, surface roughness, contact load, surface film state, etc [12-13].

As shown in Fig. 10, the ECP of each loop was tested in the experiment campaign whose the contact resistance represent a increasing trend during the rolling of the prototype. Due to the rising of the temperature and the wear in the power track, the contact resistance approximately increased from 0.1mΩ in average to 0.8mΩ in average in the vacuum environment. Each loop can transmit the power of 6.6kW, which meet the high-power transmission requirement of the SADM on LEO. When the lifetime of the prototype reached $1.5 \times 10^5$ r, the maximum and the fluctuation of the contact resistance were less than 1mΩ, which presents the excellent electrical performance during the test. Hence, though the qualification of test, it was proved that the designed prototype is able to realize the long-life electrical transmission.
Figure 10. The contact resistance in prototype at 60A during the rolling

Through the experimental verification, the prototype of the rolling ring was proved to be able to achieve the high-power and long-life electric transmission and was robust during the long-life test of $1.5 \times 10^5$ r.

5. Conclusion
In this paper, an innovative analytical kinematic model is proposed to provide a theoretical reference for the friction pair’s design of the space high-power and long-life rolling ring for the SADM on LEO. The following conclusions are drawn:

(1) Based on the designed parameters, the prototype of the rolling ring has been proved to have the excellent ECP that the contact resistance is less than 0.8 mΩ during the experiment verification of $1.5 \times 10^5$ r.

(2) In the test, transmitting the 6.6kW, the single loop of the prototype presents remarkable improvement at power transmission, which is approximately 10 times larger than the traditional slip ring.

(3) The experimental result shows that the design can meet the high-power and long-life requirement of the electrical transmission for the SADM. The research results can provide a theoretical reference for the engineering development of the high-power and long-life rolling ring.

6. Reference

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