TECHNICAL RESEARCH REPORT

The Development of Flex-gears for the Conduction of Electricity Across a Continuously Rotating Joint

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ABSTRACT

The transfer of electricity across a continuously rotating joint is especially important in robot joints and brush-type motors. In this paper, the shortfalls of present technologies, such as electric brushes, are discussed. Flex-gears are presented as an alternative method of transferring electricity across a continuously rotating joint. A new type of gear, called a pitch-rolling-gear, is developed from involute gear technology for use as a flex-gear. Finally, the design of planetary flex-gear devices using pitch-rolling-gears is studied.

BACKGROUND

Traditionally, wires or brushes transfer electricity across a rotating joint. However, wires that commonly span robot joints restrict their motion and add significant weight. Brushes, which are used in brush-type motors, incur considerable wear especially in the vacuum of space. This wear generates debris which promotes arcing and could lead to the short circuit of sensitive spacecraft instrumentation. For these reasons brushless motors are predominately used in space applications, however, fail to deliver the necessary stall torque to tighten bolts, a primary requirement for the construction of a space station.

Roll-rings were developed by the Sperry Corporation to conduct electrical power across a continuously rotating joint with little wear [Sperry, 1981]. The roll-ring device shown in Figure 1 employs one or more flexible hollow *planets* to conduct electricity between two concentrically rotating conductors called the *sun* and the *ring* [Porter, 1985]. Since only rolling contact occurs, the roll-ring device incurs much less wear than brushes. However, the flexible planets of a roll-ring device will eventually run into one another by *walking* (a result of micro slip) or *jerking* out of position, resulting in extreme wear between the planets.

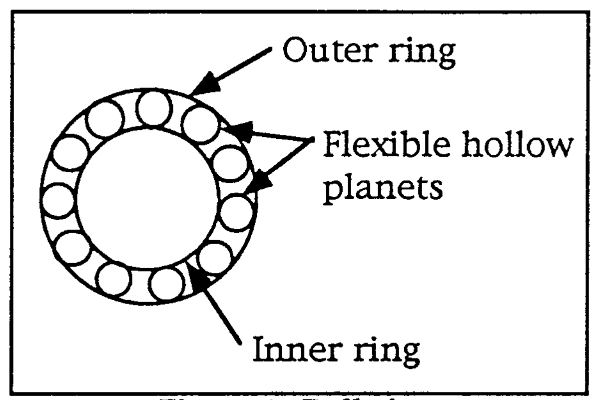

Figure 1: Roll-rings

A *flex-gear* device differs from a roll-ring device by employing flexible gears as planets to maintain their own position in the annulus of the sun and ring gears by gear meshing [Vranish, 19xx]. Hollowed for additional flexibility, the planet gears are compressed in the annulus of their sun and ring gears similarly to the planets of the roll-ring device to maintain robust electrical contact.

Standard gears do not make suitable flex-gears. Upon compression, contact and severe wear would occur at both points Q and R in Figure 2. Hence, the search for a new type of gear was warranted.

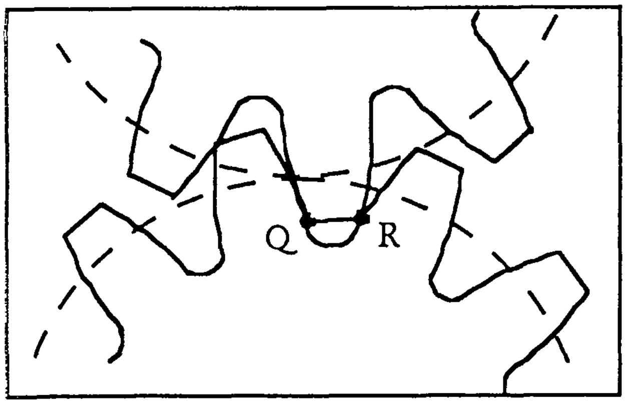

Figure 2: Standard Gear Meshing

ROLLING-GEARS

Combining the advantages of both the roll-ring and the standard gear flex-gear, the *rolling-gear* shown in Figure 3 was conceived. A rolling-gear is simply a roll-ring with gear teeth added to prevent uncontrolled sliding. A roll-ring and a standard gear flex-gear each have only one type of contact surface. A roll-ring only has a surface for electrical contact. A standard gear flex-gear only has surfaces for gearing that are also used for electrical contact. Rolling-gears, however, have two entirely different types of contact surfaces: one to transfer electricity and the other to maintain the position of the gear, as shown in Figure 3.

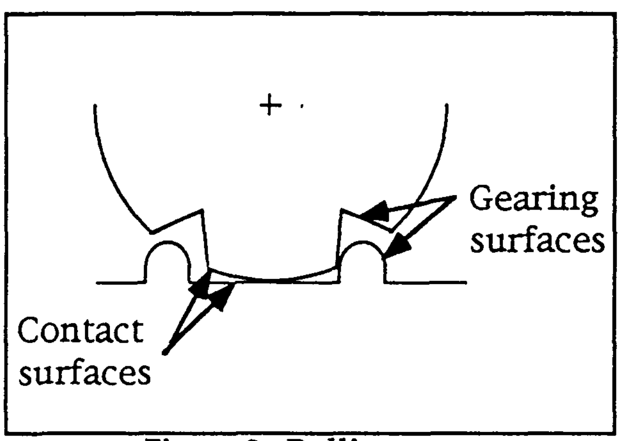

Figure 3: Rolling-gears

The first type of surface on a rolling-gear is called a *contact surface* which is designed to transfer electricity. At least one of the two mating contact surfaces is flexible and forced against its mating surface to produce sufficient contact area and pressure for electrical flow. To minimize wear, the contact surfaces are designed to roll rather than slip. The second type of surface is called a *gearing surface* which maintains the position of the rolling-gear in the annulus. Gearing surfaces are nothing more than gear teeth and corresponding valleys.

If gear teeth are put on the planets, corresponding valleys must be put on the ring and sun, and visa versa. Figure 3 shows a rolling-gear with arbitrarily chosen valley and gear tooth shapes. The rolling-gear can slide somewhat freely between the gear teeth. Like a roll-ring, the rolling-gear is held in position mostly by the friction between the contact surfaces, rolling over the gear teeth without making contact. Should slipping occur between the contact surfaces by the rolling-gear walking or jerking out of position, the gear teeth reposition the rolling-gear.

Upon repositioning, some sliding occurs between both the contact and gearing surfaces. Sliding between the contact surfaces contributes to significant wear and mechanical power loss due to high contact pressure and area. To diminish wear and mechanical power loss, a search began for gear tooth and valley shapes that would minimize sliding on the contact surfaces. Candidate gear tooth curves included simple shapes like those shown in Figure 3 and those common to gear technology, such as cycloids and involutes. A special application of involute curves (or any other conjugate curves) was found to reduce the sliding on the contact surfaces to zero. This special configuration of a rolling-gear is called a *pitch-rolling-gear*.

PITCH-ROLLING-GEARS

A brief review of involute gears is given below. For a more thorough discussion of involute gears, the reader is directed to [Shigley, 1989].

Consider the meshing between two standard involute gears as shown in Figure 4a. Initial gear contact is made at point *A*. A combination of sliding and rolling occurs between the mating teeth as the point of instantaneous contact follows the *pressure* *line* until separation at point *B*. The *dedendum circle* marks the roots of the gear teeth, while the *addendum circle* marks their tips. The *pitch circle* is defined as the circle that passes through the pitch point *P*.

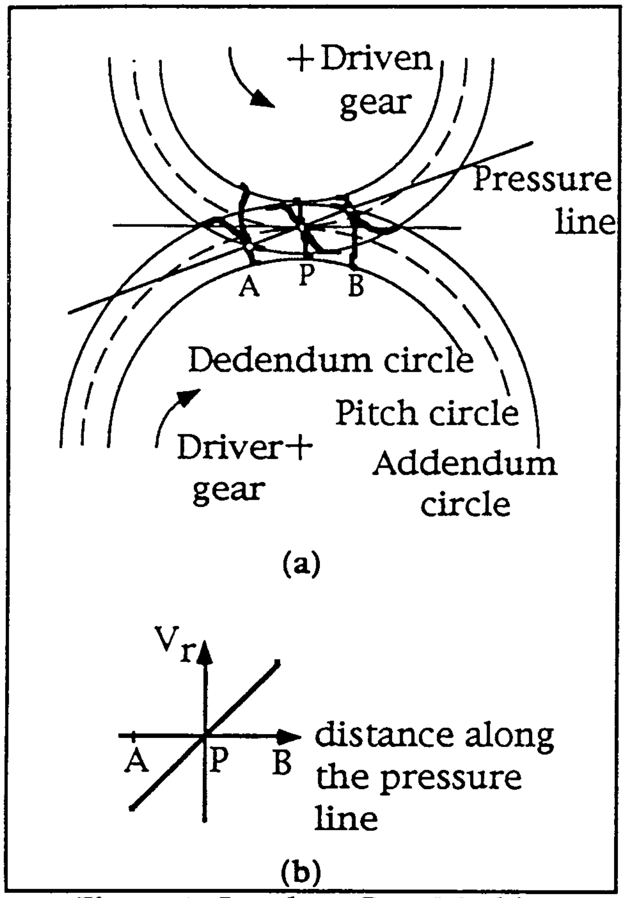

Figure 4: Involute Gear Meshing

The relative velocity between mating gear teeth, at their point of instantaneous contact, is a measure of sliding. In Figure 4b, the relative velocity V_r is zero at the pitch point *P*, indicating pure rolling at that point. This means that an involute gear rotates as if it were rolling on the imaginary pitch circle of its mating gear, thereby transmitting a constant angular velocity. The transmission of constant angular velocity is characteristic of involute gears that is referred to as *conjugate action*. By exploiting conjugate action, special involute gear teeth can be formed on a rolling gear such that the sliding is eliminated on the contact surfaces.

The exploitation of conjugate action is implemented by modifying involute gears as shown in Figure 5. *Dedendum gear teeth* are formed on the dedendum gear by cutting off the addendum of a standard involute gear. *Addendum gear teeth* are formed on the addendum gear by filling in the dedendum of a standard involute gear. Upon meshing, the cut-off portions of a dedendum gear roll on the filled-in portions of an addendum gear to produce rolling contact*. These surfaces are the contact surfaces like those of a rolling-gear. Since these contact surfaces lay on the pitch circles of their respective gears, these types of gears are called pitch-rolling-gears. Meanwhile, the contact between the gearing surfaces maintains conjugate action to ensure pure rolling on the contact surfaces.

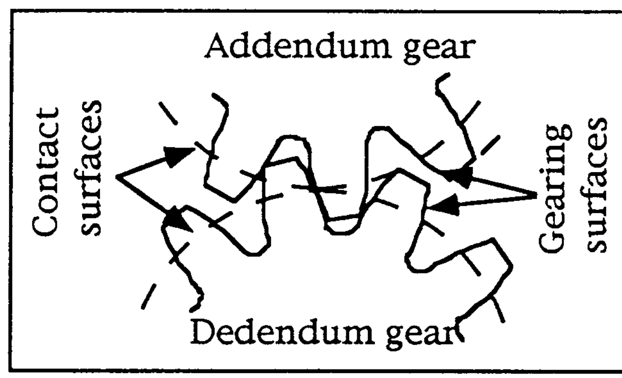

Figure 5: Pitch-rolling-gears

In Figure 6a, the dedendum gear teeth have been widened and the addendum gear teeth slimmed to maximize the duration of contact between the contact surfaces. The leftmost addendum gear tooth makes initial contact with the dedendum gear at point *A*. Sliding and rolling occurs between the mating teeth along the line of action until separation at the pitch point *P*, similar to standard gear meshing. Figure 6b shows the sliding, or relative velocity V_r, between the gearing surfaces of mating pitch-rolling-gears. Notice how this compares to the sliding or

* The phenomenon of microslip is neglected in this paper.

relative velocity of standard gears in Figure 4b.

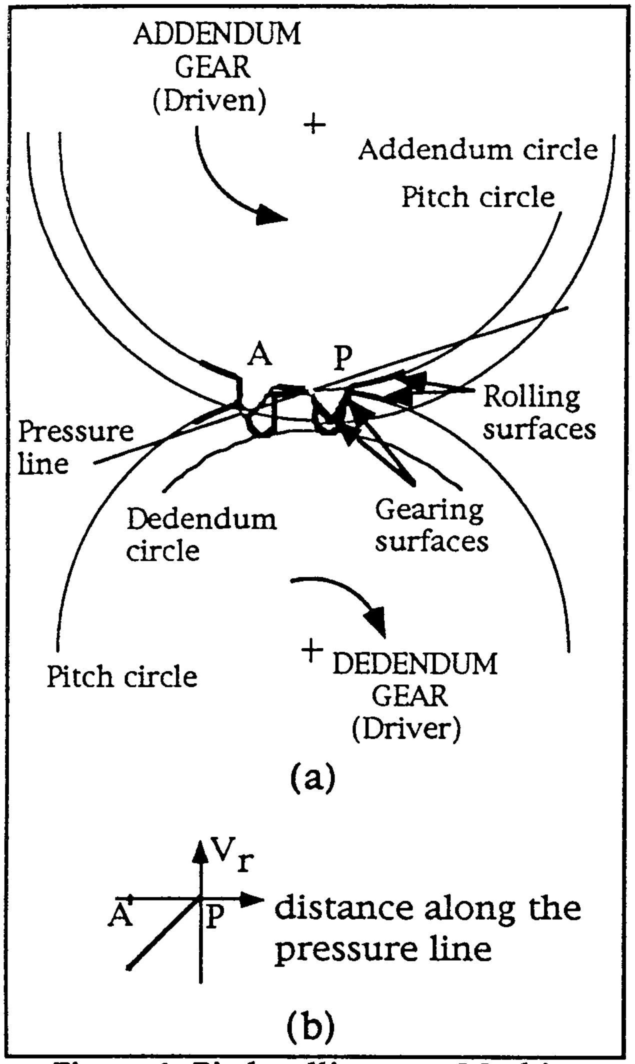

Figure 6: Pitch-rolling-gear Meshing

The relationship between the pitch circle and the contact surface is the key to pitch-rolling-gears. Referring to Figure 7, the pitch circle has a *pitch radius* r, where the *gear pitch* p is defined as the distance between teeth along the pitch circle:

$$p = \frac{2\pi r}{N} \tag{1}$$

The contact circle has a *contact radius* R, where the *contact pitch* is defined as the distance between teeth along the contact circle:

$$p_c = \frac{2\pi R}{N} \tag{2}$$

Only when the pitch and contact radii are equal does pure rolling exist between the contact surfaces of meshing pitch-rolling-gears. By judicious placement of the contact circle with respect to the pitch circle, the sliding between the contact surfaces of meshing pitch-rolling-gears can be either eliminated or controlled.

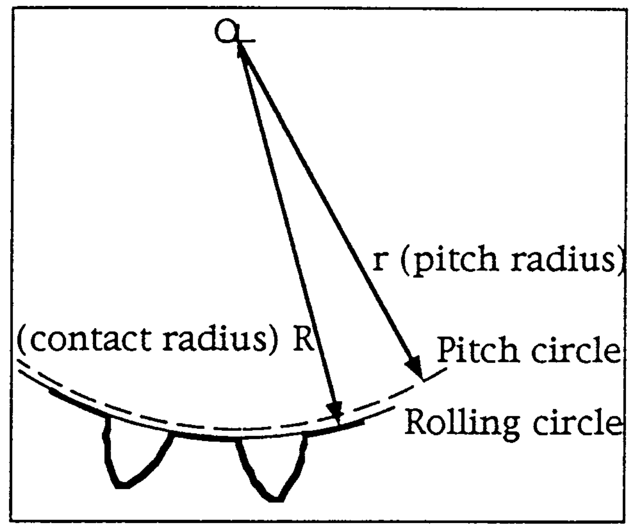

Figure 7: Pitch and Contact Circles

CHANGING THE CONTACT RADIUS OF A PITCH-ROLLING-GEAR

To incorporate a predetermined amount of sliding between the contact surfaces of meshing pitch-rolling-gears, consider oversizing the contact radius of a pitch-rolling-gear. Figure 8a shows a pitch-rolling-gear pair, whose pitch point and the contact point coincide. Figure 8b shows the gear pair after oversizing the contact circle of gear *1* by ΔC. The contact surface of gear *2* is pushed up away from gear *1*, and the entire gear is allowed to move up so that no compression occurs (compression of pitch-rolling-gears will be discussed in the next section). Because the pitch circles which govern gear meshing are no longer in contact, they no longer act as the pitch circles. New pitch circles, called *working pitch circles*, pass through the *working pitch point P*, as shown in Figure 8b. The effect of oversizing the contact radius R is simply to increase the center distance of the gear pair.

Since pitch-rolling-gears are special forms of standard involute gears, the effect of their change in center distance is the same as that of standard involute gears, whose change in center distance has been investigated to understand the effect of imperfectly mounted gears, from errors in machining and assembly. Since the number of teeth remains the same, the relative angular velocity between the gear pair remains the same. From this, the location of the working pitch point *P* is found [Kimbrell, 1991]. The corresponding working pitch radii r_1' and r_2' are given by

$$r_1' = \frac{N_1}{N_1 + N_2}(C + \Delta C) \qquad (3)$$

$$r_2' = \frac{N_2}{N_1 + N_2}(C + \Delta C) \qquad (4)$$

where ΔC is the change in center distance C between gears.

Gear meshing enforces pure rolling at the working pitch point *P*, so that,

$$\omega_2 = -\frac{r_1'}{r_2'}\omega_1 = -\frac{r_1}{r_2}\omega_1 \qquad (5)$$

where ω_1 and ω_2 are the angular velocities of gear *1* and 2, respectively.

Knowing the location of the contact point with respect to the working pitch point, sliding between the contact surfaces can be calculated. The relative velocity of the contact surfaces of gear *1* and 2, referred to as *sliding velocity*, is given by

$$\Delta \overline{V}_A = \overline{\omega}_1 \times \overline{R}_1 - \overline{\omega}_2 \times \overline{R}_2 \qquad (6a)$$

or

$$\Delta V_A = \omega_1\left(R_1 - \frac{r_1}{r_2}R_2\right) \qquad (6b)$$

Notice that if $R_1 = r_1$ and $R_2 = r_2$, then sliding between the contact surfaces of gears *1* and 2 is zero.

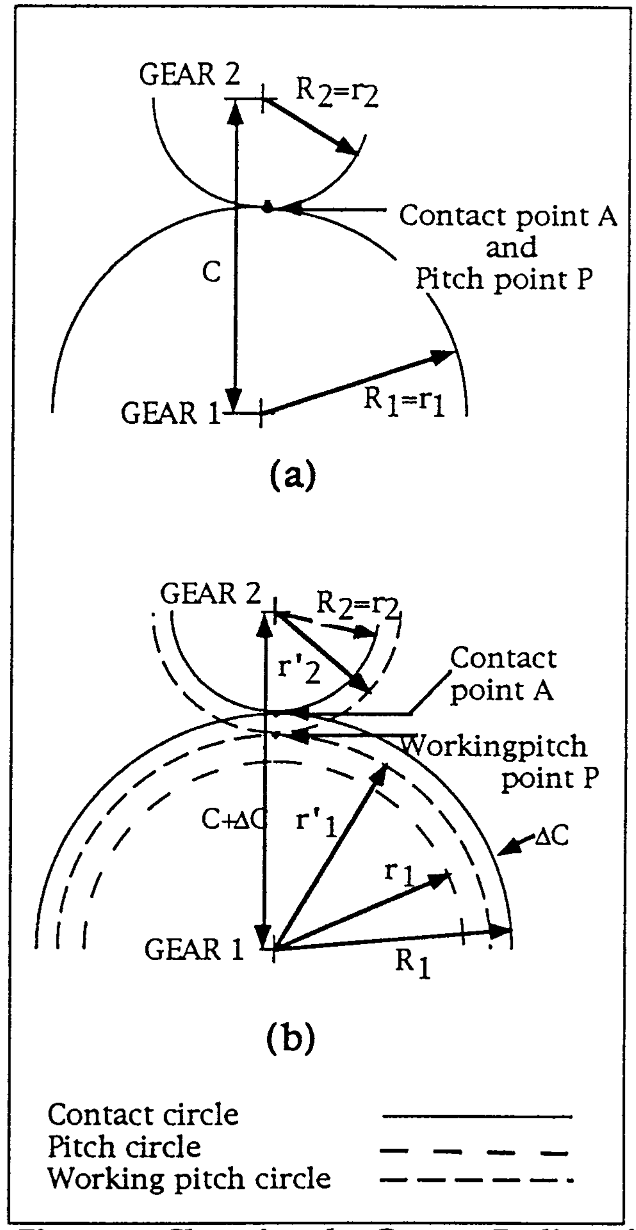

Figure 8: Changing the Contact Radius of Pitch-rolling-gears

With the freedom of controlling or eliminating sliding by judicious placement of the contact circle, pitch-rolling-gears are ideal rolling-gears and therefore make excellent flex-gears. The differentiation between the pitch and contact circles is especially important in the design of planetary flex-gear devices.

PITCH-ROLLING-GEAR COMPRESSION

As a flex-gear in a planetary flex-gear device, a pitch-rolling-gear must be compressed in the annulus of the sun and ring gears for robust electrical contact. The compression of a pitch-rolling-gear is shown in Figure 9.

Through gear compression, pure conjugate action is lost, since involute gear meshing was developed for circular gears, not deformed gears. Without conjugate action, the velocity of the contact surfaces of mating gears at the point of contact is not exactly the same at all times. However, the *average* velocity of the gears at the contact point is approximately the same, since the contact pitch of both gears remains approximately the same throughout compression. The contact pitch of the compressed gear remains approximately the same throughout compression because the circumference of the contact circle does not change significantly. This is supported by Timoshenko [1936], who used the inextensibility of thin rings to analyze their behavior under compression.

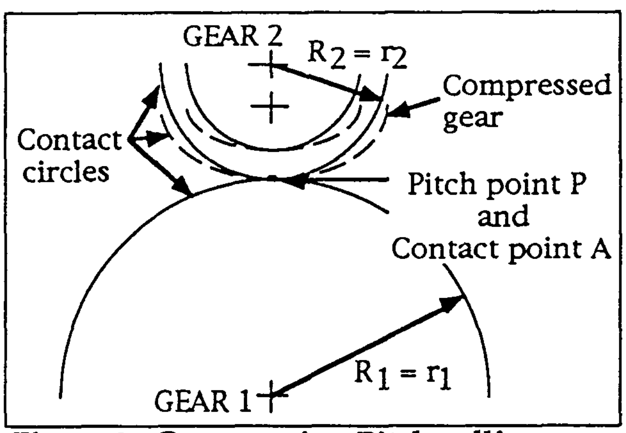

Figure 9: Compressing Pitch-rolling-gears

Defining the pitch point as the point of *average* rolling between meshing gears, the pitch point remains coincident with the contact point. Then, the predetermined or zero sliding is maintained throughout compression as the average sliding. The variation of sliding due to gear deformation is thought to be negligible, since gear compression for robust electrical contact can be 100 times smaller than the diameter of the gear itself.

PLANETARY FLEX-GEAR DEVICES

A planetary flex-gear device that uses pitch-rolling-gears, as shown in Figure 10, is much like a standard involute planetary gear train. However, a planetary flex-gear device differs from a standard planetary gear device in that a flex-gear device has a compressed planet for robust electrical contact.

A standard planetary gear train abides by four constraints. The first is that the pitch p of each gear is the same:

$$p_1 = p_2 = p_3 \tag{7}$$

where pitch is defined as the distance between teeth along the pitch circle (second constraint):

$$p_i = \frac{2\pi r_i}{N_i} \tag{8}$$

where r_i is the pitch radius and the number of teeth N_i is an integer (third constraint). The last constraint of a standard planetary gear train is that

$$r_1 + 2r_2 = r_3 \tag{9a}$$

where the subscripts *1*,*2*, and *3* respectively refer to the sun, planet, and ring gears, as labelled in Figure 10.

The design of a planetary flex-gear device differs from that of a standard planetary gear device in the fourth constraint, which is

$$R_1 + 2R_2 - \delta = R_3 \tag{9b}$$

where R_i are the contact radii and δ is the compression of the planet.

For pure rolling to exist in a planetary flex-gear device (i.e., $r_i = R_i$), δ must be divisible by R_i / N_i (where i=*1*,*2*, or *3*), according to the four constraints above.

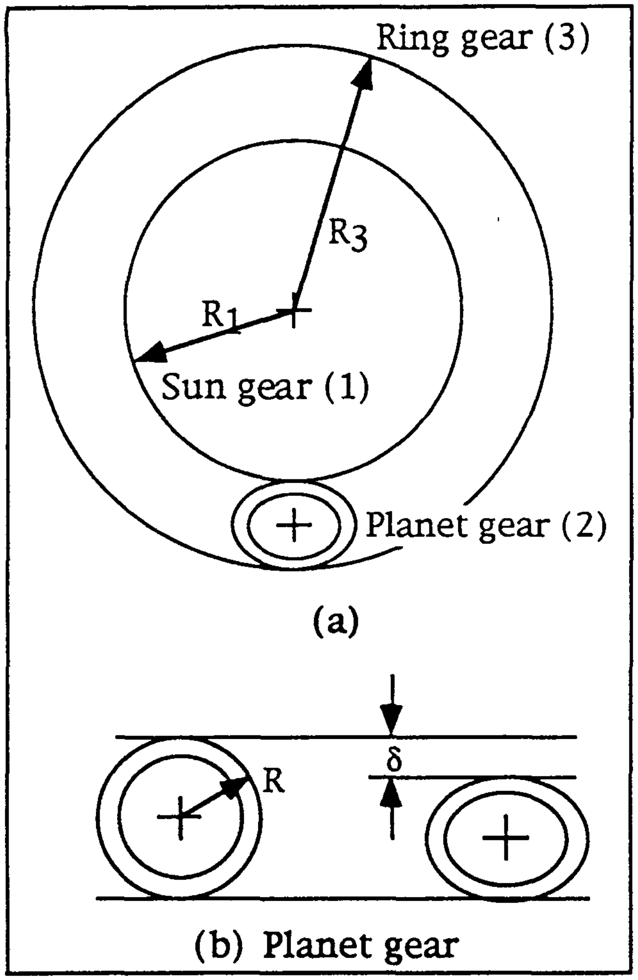

Figure 10: Planetary Flex-gear Device

SUMMARY AND FINAL REMARKS

Flex-gears are a viable alternative to standard methods of transferring electricity across a continuously rotating joint. Because flex-gears maintain their own position in the annulus of the sun and ring gear, multiple planets can be used to transfer more electricity than roll-rings. Furthermore, flex-gears incur much less wear on their surfaces of electrical contact than brushes, because sliding can be controlled or eliminated.

In this study, a new class of gears called pitch-rolling-gears is developed for use in flex-gear devices. These are designed to transfer electrical power, whereas standard gears are designed to transfer mechanical power.

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