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Technical Brief

Design of a Friction Clutch Using Dual Belleville Structures

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This paper presents a simplified friction clutch design, which consists of the dual Belleville spring and the friction plates. This design exploits the preset angle on the Belleville spring to increase the friction area during operation; thus, the load on spring is reduced at a given transmitted torque. Due to the increasing of friction area, the Belleville spring can also act as a friction plate, and the components required for the clutch can be reduced. The maximum transmittable torque of the clutch is easily adjusted by varying the preload on the Belleville spring. Moreover, it is very easy to assemble the components of the present clutch. To demonstrate the present design, a prototype friction clutch with a Belleville spring has been fabricated and tested. We showed that the transmitted torque remained constant for different operating speeds. [DOI: 10.1115/1.2748454]

Keywords: friction clutch, Belleville spring, coefficient of friction

Introduction

The overload protection devices are designed to prevent the damage of machinery or drive line in the event of an overload or jamming. The overload protection devices have been extensively employed in various applications such as automatic screwdrivers, printers, conveyor belts, and automobile power windows, etc. The most common overload protection devices are the friction clutches that adapted to the smooth engagement of shafts with different angular velocities. The basic operation concept is to force two opposing surfaces into frictional contact by means of a preset load. The torque, if it is not overloaded, will be fully transmitted by the clutch under the assistance of friction force. When the transmitted torque exceeds the design limit (i.e., overload), the slip between the two friction surfaces will occur. The clutch will no longer fully transmit this overload torque, and will act as an overload protection device.

In general, the friction clutches have normally engaged and normally disengaged designs. Figure 1 shows a typical normally disengaged radial friction clutch that consists of the drive shaft, the friction plate (lining), the helical spring, and the driven shaft. The torque capacity of friction clutch is the function of clutch size, friction coefficient μ, number of friction faces, and axial force [1]. When the drive shaft applies a load to the helical spring, so as to lead to the contact of friction plates, the power will transmit from the drive shaft to the driven shaft. The friction force between the friction plates can be adjusted by varying the restoring force of the helical spring. Thus, the torque allowed to transmit by this friction clutch is determined. As a second example, the drive members and the driven members are normal engaged. The friction plates normally make contact with each other except for the existence of overload. In [2,3], the Belleville spring (also referred to as the conical disk spring) is used to apply an axial load to push the friction faces.

The disadvantage of the existing friction clutch is the requirement of various components. In addition, it is time consuming to assemble these components. In this study, a normally engaged friction clutch has been designed and analyzed, as illustrated in Fig. 2. The friction clutch mainly consists of the dual slit Belleville spring and lining. These Belleville springs are used to generate the friction force of the clutch, so as to further determine the maximum torque transmitted by the clutch. Since the Belleville spring is stiffer than the helical compression spring, it can provide huge force with a small deflection. Moreover, the assembly of the present clutch is simple and fast; and its cost can be significantly reduced.

Design Concept

The present clutch shown in Fig. 2 employs the restoring force of Belleville springs to provide an axial load. Figure 3 shows the exploded drawing of the clutch in Fig. 2. The power inputs into the mechanism through drive gear (#9), outputs from the mechanism through the driven gear (#1). The shaft #8 is fixed to gear #9, and the friction plates (lining) #6 with four inner anchors are locked to the key ways of shaft (#8). Thus, the friction plates and shaft are driven by gear #9, and all of these components (named the drive set) have the same angular velocity. The Belleville springs (#5) and intermediate plate (#7) with four outer anchors are fixed to the slots of the driven gear (#1), and all of these three components (named the driven set) have the same angular velocity. The drive set and the driven set have the same angular velocity in normal operation circumstances, whereas these two sets have different angular velocities during the overload situation. The Belleville springs will apply an adjustable load P on the friction plate #6 by varying the position of the nut (#2), so as to further induce a friction force between the friction plates and the intermediate plate. After that, the nut is fastened to the shaft (#8) using the setscrew (#3). Thus, the power can transmit from the drive gear #9 to the driven gear #1. Moreover, the transmitted power

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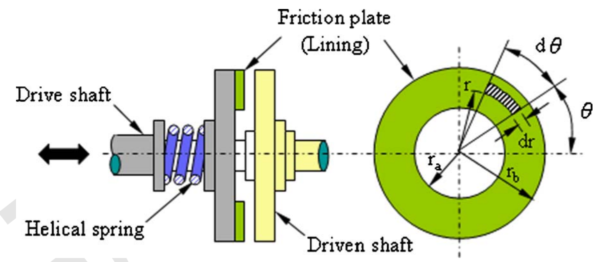

Fig. 1 Schematic of the traditional radian friction clutch

can be tuned by varying the load P. When the friction torque T induced by the transmitted power exceeds the design limit $T_{\max}$, the friction plates will slip and the input power from gear #9 will no longer transmit to gear #1. Thus, the whole device performs as an overload clutch. Equation (A1) in Appendix A expresses the relation of load P and friction torque T [1].

Figure 4(*a*) shows the cross section of the present Belleville spring, which consists of A (conical disk) and B (lever arm) segments. As discussed in [4], the load-deflection relation of the Belleville spring is highly nonlinear for a large deformation. The buckling of the shell structure, which has similar mechanics behavior, has been discussed in [5,6]. The influence of structure thickness on the stiffness of the Belleville spring has been studied

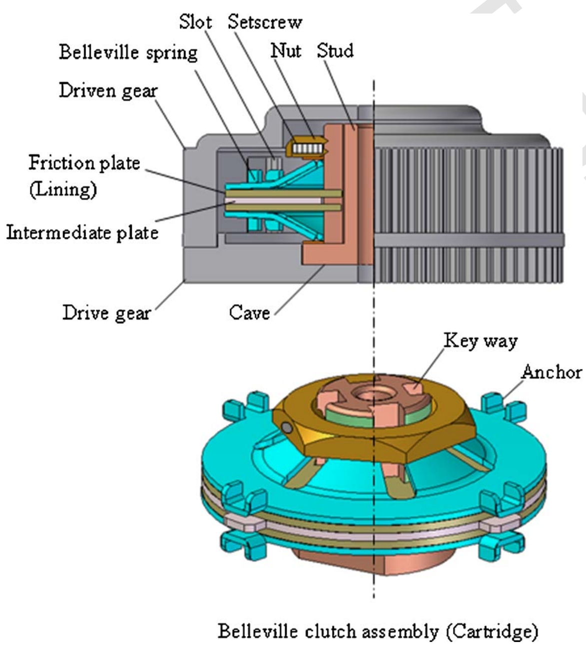

Fig. 2 Schematic of the friction clutch with dual Belleville springs

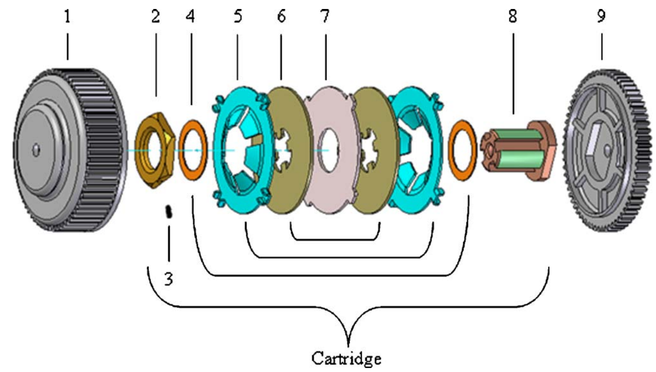

Fig. 3 Key components of the present friction clutch: driven gear (#1), nut (#2), setscrew (#3), washer (#4), Belleville springs (#5), friction plate (lining, #6), intermediate plates (#7), stud (#8), and drive gear (#9)

in [7], and the material nonlinearity effect of conical disk springs has been analyzed in [8]. In order to design and predict the characteristics of the Belleville spring, a small deformation is required to assure operation at the linear range. The load-deflection characteristics of the Belleville spring are superposed by both of these two segments. A more detailed description of the Belleville spring with a two-segment design is available in Appendix B [9]. The deformation behaviors of the Belleville spring during loading are indicated in Fig. 4. The case (I) represents the configuration of spring with no loading. As the load applied on the spring is increased, the spring will be deformed and the center hole will be decreased. As depicted in case (II), the angle β will become zero when a critical load P_c is applied to the spring. Thus, the contact area of the spring will reach a maximum. As shown in case (III), the edge of region A could be warped if the applied load is greater than P_c; meanwhile, the contact area is decreased. The design and optimization of Belleville springs have been discussed in [10,11].

In summary, the merit of this design is to employ the preset angle β on the Belleville spring to increase the contact area for friction at a given preload. According to Eq. (A1), the load P on the spring can be reduced for a designated transmitted torque. Moreover, the Belleville structure can act not only as a spring, but also as a friction plate, and the components required for the clutch can be reduced. It is possible to tune the mechanical characteristics of the Belleville spring by varying various geometry parameters, such as thickness, spring height, angle β, etc. This study aims to demonstrate the feasibility of applying the Belleville spring for friction clutch, but not to optimize the design of the spring.

Typical Simulation and Experimental Results

This study employed the commercial finite element software ANSYS to predict the influence of thickness under the similar shape

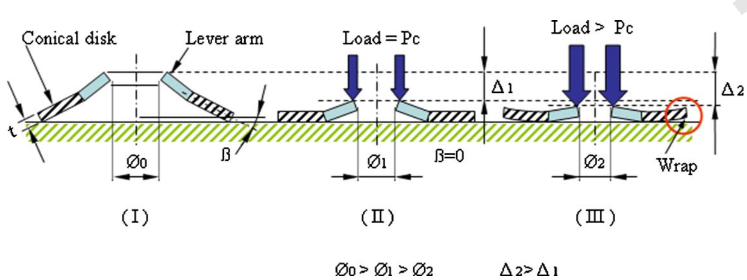

Fig. 4 The characteristics of the Belleville spring under different loads, (I) no loading, (II) maximum contact area under critical loading P_c until β=0 deg, and (III) the loading over P_c

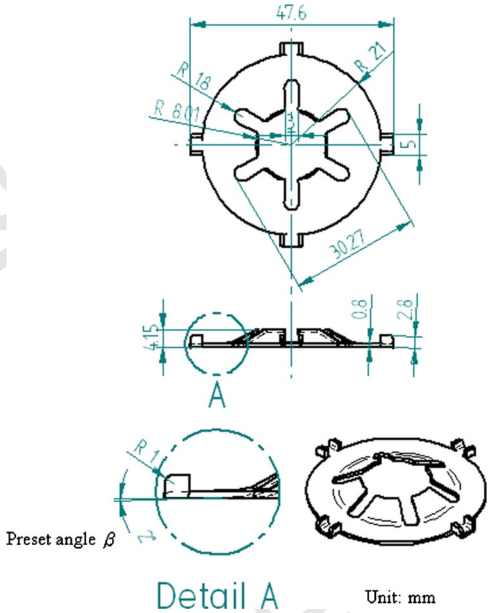

Fig. 5 Dimensions of the prototype Belleville spring for experiment

of the Belleville spring. The dimensions of the Belleville spring established in the finite element model (FEM) are indicated in Fig. 5. In this case, the preset angle β and thickness t of the Belleville spring were in the range $\beta=2-10$ deg, and $t=0.6-1.5$ mm, respectively. Moreover, the element type was SHELL93, and the spring had material constants of Young's modulus $E=200$ GPa and Poisson's ratio $\nu=0.29$. The typical finite element results in Fig. 6(*a*) show the deflection of the spring with $\beta=2$ deg and thickness $t=0.8$ mm after applying a 185 N load. The slashed line region in Fig. 6(*b*) summarizes the loads to cause the angle β to become 0 deg for various spring thicknesses ($t=0.6$, 0.8, 1.0, 1.2, and 1.5 mm) and two preset angles ($\beta=2$ deg and $\beta=10$ deg). Thus, the load can be tuned in the range of 85–4837 N by varying these spring thicknesses and preset angles. These simulation results also point out that the relation between the load and the thickness of spring is nonlinear.

In order to demonstrate the present concept, a prototype friction clutch with a Belleville spring was implemented. Figure 7 demonstrates various key components of the fabricated clutch. As shown in Fig. 7(*a*), this study selected the SUS304 sheet (surface roughness PRa$=0.346$ μm measured by Talysurf Series 2) to make the intermediate plate #7 and friction plate #6 using wire electrical discharge machining. The SUS304 material is most widely used for industrial products to offer excellent corrosion resistance [12]. The Belleville spring #5, which demands good formability and strong stiffness, was made of SUP 9 (surface

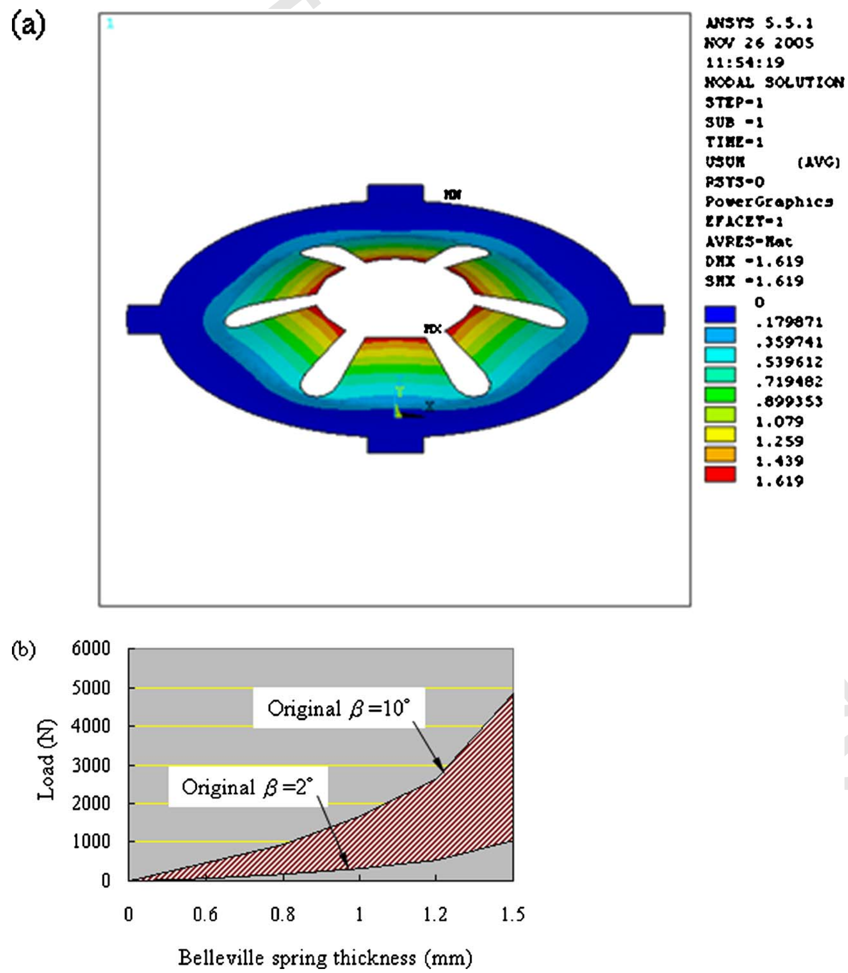

Fig. 6 FEM simulation results (*a*) typical FEM results when β=2 deg, t=0.8 mm, and preset load of 185 N, and (*b*) the preset loads required to reach the maximum contact area status in Fig. 4 (II) for different thickness t (t=0.6, 0.8, 1.0, 1.2, and 1.5 mm) and angle β (β=2 deg and 10 deg)

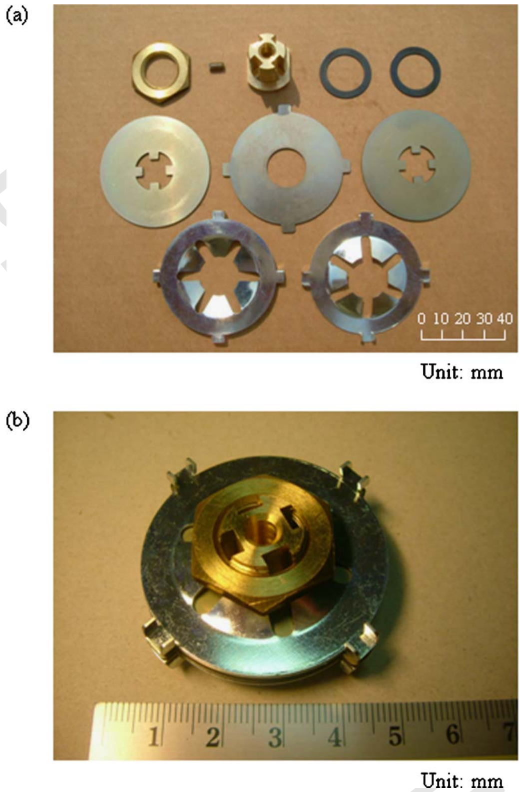

Fig. 7 (*a*) The fabricated key components of the clutch, and (*b*) the assembly of the components

roughness PRa=0.5472 μm). Moreover, the gears #1 and #9 are made of acetal copolymer by means of the injection-molding process. All specimens were inspected and ultrasonically cleaned in acetone for 10 min after fabrication. The surfaces for friction contact were coated with grease before assembly. Figure 7(*b*) shows the assembly of these components.

The experimental setup in Fig. 8 was established to characterize the performance of the clutch. A DC motor (Mabuchi RS-540) was employed to move the drive gear (#1); however, the driven gear (#9) was connected and held by a torque meter (Torqueleader 36/4). Thus, this experiment acted as an overload protection condition, and the relative angular velocity between the gears #1 and #9 was determined by the DC motor. Thus, the output torque applied on the shaft was determined. A current/voltage (50 A/50 mV) transducer (Lutron ST-50) was used to measure the motor armature current. A low-pass filter and an amplifier (amplification gain=1000) were used to reduce the noise during measurement. The resistance and capacitance were $R=28$ ohm and $C=47$ μf, respectively, and the cut-off frequency was f_c≒120 Hz. During the test, the nut #2 was adjusted to give the $T_{\max}$ of 2 N m. The DC motor had three different angular speeds (20, 40, and 60 RPM) to demonstrate the performance of clutch at different relative angular velocity of gear #1 and gear #9. The measurement results in Fig. 9 shows that the torque was close to the preset value of 2 N m for these three angular velocities (20, 40, and 60 RPM), even though the original torque remained the same after 1000 cycles of operation.

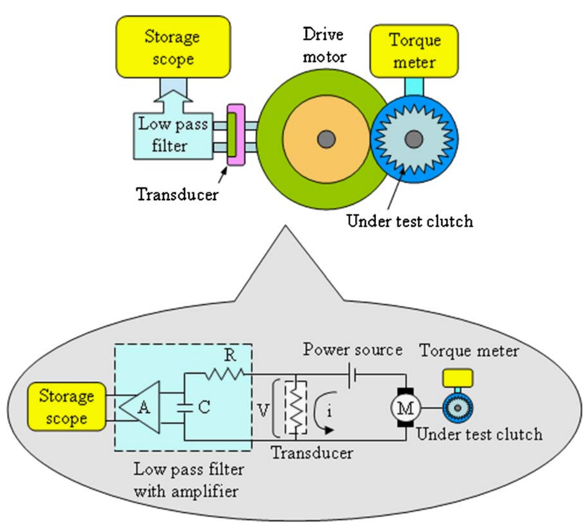

Fig. 8 The experimental setup for the dynamic spinning test of the clutch

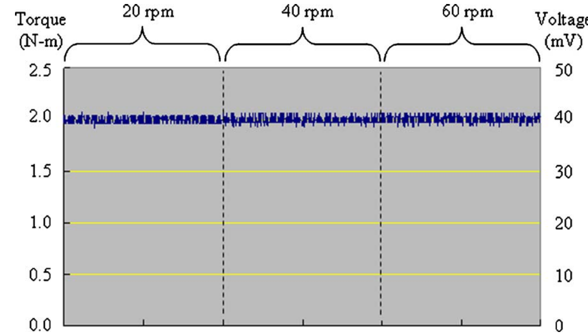

Fig. 9 The torque was close to the preset value of 2 N m for three different operating velocities (20, 40, and 60 RPM)

Conclusions

The study has presented a novel friction clutch using the Belleville spring as the key component. This design exploits the preset angle β on the Belleville spring to increase the friction area during operation; thus, the load P on spring is reduced at a given transmitted torque. Due to the increasing of friction area, the Belleville spring can also act as a friction plate, and the components required for the clutch can be reduced. The allowable transmitted torque is adjusted by tuning the predeformation of the Belleville spring. Once the transmitted torque exceeds the friction torque preset by the restoring force of the Belleville spring, the friction plates and the intermediate plate will slip, so as to perform as an overload clutch. In application, a friction clutch with a Belleville spring has been fabricated and tested. The measurement results demonstrated that the mechanism did act as an overload friction clutch; the transmitted torque was close to a preset value of 2 N m for three different angular velocities, including 20, 40, and 60 rpm. Thus, various torque requirements for different applications can easily be satisfied. To enhance the reliability of the clutch, the problem relating to the temperature rise and wear can be improved by adding the lubrication grease. Moreover, it is easy to assemble the present clutch so as to minimize required space as well as the cost. The basic ideas of this study are applicable to any situation while overload exists, such as tension on paper, wire film, and a power window on a vehicle.

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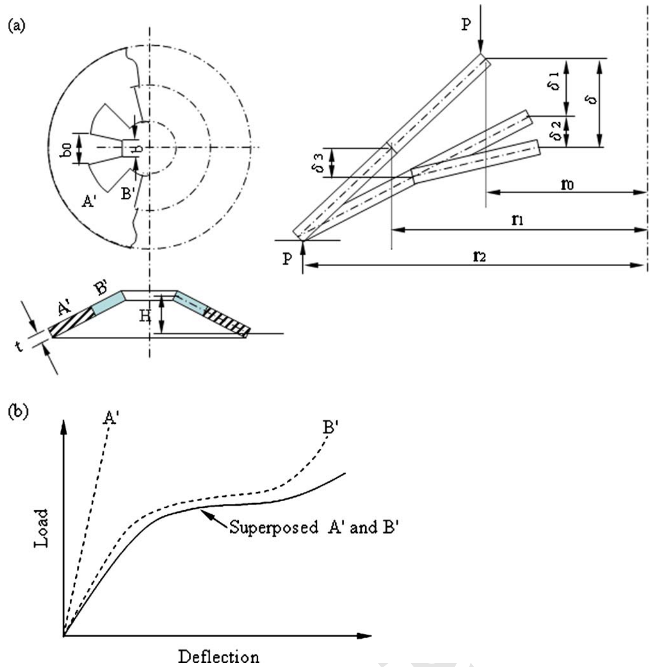

Fig. 10 (***a***) **Schematic of the Belleville spring model, and** (***b***) **variation of the load and the deflection** δ

Appendix A: Disk Clutch Torque

The slip friction type clutches consist of one or more friction disks. As shown in Fig. 1, the torque capacity T at an axial clamping force P is expressed as [1],

$$T = NP\mu R_f, \tag{A1}$$

where μ is the constant friction coefficient, and N is the number of contact surface. In addition, the friction radius R_f is

$$R_f = \frac{2}{3}\left(\frac{r_b^3 - r_a^3}{r_b^2 - r_a^2}\right) \tag{A2}$$

where r_a and r_b are the inner radii and outer radii, respectively, of the friction plate shown in Fig. 1. Equation (A2) is appropriate for a brand new clutch. However, after the uniform wear takes place, the friction radius R_f becomes

$$R_f = \frac{r_b + r_a}{2} \tag{A3}$$

According to Eq. (A1), the torque capacity of a clutch increases with the area of friction surface.

Appendix B: Belleville Spring Structure Load-Deflection Characteristics

The slit Belleville spring design formula has been studied in [9] to show the relation of spring features and stiffness. As shown in Fig. 10(a), the relationship between the load P and the spring is

$$P = \frac{r_2 - r_1}{r_2 - r_0}\,\frac{C_1 C_2 E t^4}{r_2^2} \tag{B1}$$

where r_0=inner radii of Belleville spring, r_1=radii of slit outer Belleville spring edge, r_2=radii of Belleville spring outer edge, E=Young's module of materials, t=thickness, and the constants C_1 and C_2 are expressed as

$$C_1 = \left(\frac{\alpha + 1}{\alpha - 1} - \frac{2}{\log \alpha}\right)\pi\left(\frac{\alpha}{\alpha - 1}\right)^2 \tag{B2}$$

$$C_2 = \frac{\delta_3}{(1 - \nu^2)t}\left[\left(\frac{H}{t} - \frac{\delta_3}{t}\right)\left(\frac{H}{t} - \frac{\delta_3}{2t}\right) + 1\right] \tag{B3}$$

where H=free height minus thickness, ν=Poisson's ratio, α=radius ratio, r_2/r_1, and $\delta 1 - \delta 3$ are deflections of the Belleville spring, which can be further expressed as

$$\delta_1 = \delta_3 \frac{r_2 - r_0}{r_2 - r_1} \tag{B4}$$

$$\delta_2 = K\frac{4P(r_1 - r_0)^3}{E b_0 t^3 m}(1 - \nu^2) \tag{B5}$$

where b=width of slit tip, b_0=width of slit datum, m=number of slits, and the constant K is

$$K = \frac{3}{(1 - b/b_0)^3}\left[\frac{1}{2} - 2\left(\frac{b}{b_0}\right) + \left(\frac{b}{b_0}\right)^2\left(\frac{3}{2} - \log_{10}\frac{b}{b_0}\right)\right] \tag{B6}$$

According to Eqs. (B1)–(B6) the nonlinear load-deflection curves shown in Fig. 10(b) are obtained.

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#1 AQ: Pls check edits.
#2 AQ: If PRa is an acronym, please define. If there are separate variables, make them italic.