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Single piece compliant spatial slider–crank mechanism

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ABSTRACT

This paper presents a novel study on the analysis of the "single piece" compliant spatial slider–crank mechanism. In the previous study performed by the authors, a "partially compliant" version of the spatial slider–crank mechanism was introduced. Since there was a rigid ball joint in that case, there was no torsional loading at flexural hinges. For the single piece case, there is no ball joint in the structure of the mechanism. Therefore, there is also torsion available at multiple axis flexural hinges. Analysis and design of such a compliant mechanism is different from the partially compliant case. In this study, deflections of the multiple axis flexural hinges are determined as bending and twist separately. Fundamental angles for manufacturing a mechanism in a particular position are obtained. A real model is built and results of the mathematical model are verified with experiments. A fatigue test is performed for flexural hinges of a similar mechanism. After one and a half million cycles it is monitored that there is no indication of failure.

1. Introduction

Spatial linkages are used to obtain mechanical input–output relationship between spatially oriented axes of movement. A RSSP slider–crank mechanism has a revolute joint between the crank and the ground, two spherical joints connecting the crank and the slider to the coupler link, and a prismatic joint of arbitrary orientation. RSSP mechanisms are commonly used in pumps and compressors. The analysis and synthesis of spatial slider–crank linkages are attractive and challenging problem. Several authors have made significant contributions in the area of rigid link RSSP mechanisms [1–7]. Mechanisms that gain some or all of their motion through the deflection of flexible members are classified as compliant mechanisms [8]. Planar compliant mechanisms are extensively studied in the literature [9–11]. There are also papers about planar compliant slider–crank mechanisms [12,13]. Compliant mechanisms can be classified as fully compliant and partially compliant mechanisms [14]. PRBM (pseudo-rigid body model) technique replaces the flexible segments with an equivalent system of rigid links, joints and torsional springs [15,16]. PRBM can be used for the design of compliant mechanisms when the compliant mechanism behavior is such that links can be assumed to be rigid and the flexural pivots can be assumed to behave as torsional springs [17]. This technique applies the body of knowledge that has been developed for rigid body techniques to compliant mechanisms. The purpose of the PRBM is to provide a simple method of analyzing systems that undergo large, nonlinear deflections. PRBM is accurate even for large deflections for many cases. If bending is the dominant loading in a flexural hinge, PRBM is more accurate. The nature of small-length flexural pivots ensures that the assumption that bending is the predominant loading is accurate in most applications. PRBM is useful in the early design phases where many design iterations are required to design a mechanism that fulfills the design objectives specified. It is usually preferred to employ PRBM to make initial approximations in the design to obtain a general understanding of mechanism characteristics and then use other methods to improve the design [8].

Studies on spatial compliant mechanisms are limited in the literature [18,19]. There are studies about multiple axis flexural hinges in the literature [20,21]. In the previous studies, partially compliant versions of the spatial slider–crank mechanism were introduced

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[22,23]. There was a spherical joint in those cases, so there was no torsional loading at flexural hinges. A single piece compliant version of the spatial slider–crank mechanism with small length flexural pivots is considered in this study. There is no spherical joint in this mechanism thus; there is a torsional loading available at multiple axis flexural hinges. This mechanism has an advantage of manufacturing when compared with the partially compliant case [23]. Analysis and design of such a compliant mechanism is extremely challenging to perform theoretically. In this paper, deflections of the multiple axis flexural hinges are obtained as bending and twist separately. Essential angles for manufacturing the mechanism at the undeflected position are determined. In the design example a prototype is built and it is shown that the results of the theoretical model, FEA and the real model are consistent. Finally, a fatigue test is performed for flexural hinges of a similar mechanism.

2. Kinematic analysis of RSSP linkage

The kinematic analysis for rigid RSSP is required in order to employ the PRBM. Disregarding the rotation of the coupler about its axis (which is also constrained in the compliant RSSP), the mechanism possesses one degree-of-freedom. A RSSP converts a rotation about one axis into a translation in another axis in space or vice versa.

The relationship between θ and s can be obtained by substituting trigonometric identities into general displacement equation [24].

$$\begin{aligned} s &= B \pm \sqrt{\Delta} \\ \Delta &= B^2 + 2a_2 p\cos\theta - p^2 - {a_2}^2 - f^2 + {a_3}^2 \\ B &= a_2 \sin\xi \sin\theta + f\cos\xi \end{aligned} \tag{1}$$

The unit vector along link 2 can be determined as:

$$\overrightarrow{e_{a2}} = \cos\theta\,\overrightarrow{i} + \sin\theta\,\overrightarrow{j} \tag{2}$$

The unit vector along link 3 can be determined as:

$$\overrightarrow{e_{a3}} = \frac{p - a_2\cos\theta}{a_3}\overrightarrow{i} + \frac{s\sin\xi - a_2\sin\theta}{a_3}\overrightarrow{j} + \frac{s\cos\xi - f}{a_3}\overrightarrow{k} \tag{3}$$

The unit vector along the slider axis can be determined as:

$$\overrightarrow{e_s} = \sin\xi\,\overrightarrow{j} + \cos\xi\,\overrightarrow{k} \tag{4}$$

3. Single piece compliant spatial slider–crank mechanism

There is inherently a prismatic pair in slider–crank mechanisms. If all of the pairs except prismatic pair are replaced by compliant hinges, the mechanism can be produced from one compliant link, thus it can be called as single piece compliant mechanism. The single piece compliant spatial slider–crank mechanism in the most general form is presented in Fig. 1a. The PRBM of this compliant mechanism is shown in Fig. 1b.

For the compliant mechanism in Fig. 1a, deflections of the multiple-axis flexural hinges contain both twist and bending. However, if height of cross section is large enough, then the torsion at the single-axis flexural hinge can be neglected. This situation was verified in the previous studies [18,19,22,23]. Stresses occurred at flexural hinges are proportional to deflections. Hence, deflection analysis is one of the most important design criteria in compliant mechanisms. However, for the spatial flexural hinges determination of deflection is rather complex. Deflections of the multiple-axis flexural hinges are determined separately as bending and twist in the next sections.

4. Determination of bending of the flexural hinges

In this section, bending of the multiple-axis flexural hinges will be determined by using the method which was proposed in the previous work [23]. It was verified that [18,19,22,23] if the instantaneous compliant axis is determined from the undeflected and the current positions, then the case can be considered as an initially straight flexural beam subjected to a single plane bending along its new compliant axis. At the end of this section, deflection of the single-axis flexural hinge will be determined with respect to crank position.

In Fig. 2 the mechanism is shown at the current and undeflected positions. Here, the current position is for an arbitrary crank position and the mechanism is assembled to the ground. The undeflected position occurs when the single-axis flexural hinge 1–2 is disassembled from the ground. The unit vectors on the compliant axis of multiple axis flexural hinge 3–4, on the undeflected and current positions of rigid segment 3 are $\overrightarrow{e_{\alpha 1}}$, $\overrightarrow{e_{u1}}$ and $\overrightarrow{e_{a3}}$ respectively as shown in Fig. 2. α_1 is the angle between the undeflected and current positions. The essential angles for manufacturing are μ and φ. All of these parameters were previously determined in [23].

Now, bending of the multiple-axis flexural hinge 2–3 is determined similarly. In Fig. 3 the mechanism is shown at the current and undeflected positions. Here, the current position is for arbitrary crank position and the mechanism is assembled to the ground. The undeflected position occurs when the prismatic pair is disassembled from ground. The unit vectors on the compliant axis of the