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Design and Implementation of XY-Planner Flexure Mechanism

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Abstract: Flexure joints are widely used in precision-motion stages and micro robotic mechanisms due to their monolithic construction. Flexures are compliant structures that depend on material elasticity for their functionality. Motion is generated due to deformation at the molecular level, which results in two primary characteristics of flexures smooth motion with precision and high speed application. The dry friction has been neglected in macro motion, but it is one of the most important factors limiting the performance in the precision positioning application. Therefore, to prevent the problem of such dry friction and backlash, regardless of the size of the traveling flexural mechanism widely used. In this paper, we consider the analysis of XY planner mechanism.

Keywords: Backlash, Flexure joints, monolithic construction, Precision-motion stages, XY planner mechanism.

I. Introduction

Flexure mechanisms are a designer's delight. Flexure joints offer significant advantages over conventional joints in terms of both ease of manufacturing and operational characteristics. Flexure joints are typically manufactured monolithically, and therefore, avoid assembly errors. One may think of flexures as being means for providing constraints [1]. It is this capability of providing constraints that make flexures a specific subset of springs. The importance of properly constrained design is well known to the engineering community. The objective of an ideal constraining element, mechanism, or device is to provide infinite stiffness and zero displacements along certain directions, and allow infinite motion and zero stiffness along all other directions.

The directions that are constrained are known as Degrees of Constraint (DOC), whereas the directions that are unconstrained are referred to as Degrees of Freedom (DOF) [1]. The behavior of a flexure joint is sensitive to its geometry, therefore high dimensional accuracy during fabrication is required. Motion of flexure joints can be quite complex. For example, a common type of flexure joint, the circular notch hinge, is frequently approximated by a pure rotational joint, but the actual deformation also involves translation. A mechanism is kinematically stable if it is fully constrained when all the active joints are locked, i.e., there is no uncontrolled motion under external load. However, kinematic stability should be replaced by a stiffness criterion, since the joint stiffness may be used to prevent excessive undesired motion [1], [3].

Compact XY flexure stages that provide large range of motion are desirable in several applications such as high-density memory storage, scanning interferometry and atomic force microscopy, micromanipulation and micro assembly, and MEMS sensors and actuators, semiconductor mask and wafer alignments[3]. Flexure stages generally lack in motion range. Challenges in the design of large range mechanisms arise from the basic tradeoff between Degrees of Freedom (DOF) and Degrees of Constraint (DOC) in flexures. As constraint elements, flexures pose a compromise between the motion range along DOF and the stiffness and error motions along DOC.

A realistic performance prediction, not possible using a linear analysis, is thus obtained without the need for iterative or numerical methods. The parametric nature of these results offers several insights into XY flexure mechanism design, particularly in terms of performance characteristics and compromises there in [1]. The presented analyses also provide a mathematical verification of the design axiom that geometric symmetry in mechanism design yields improved performance. To verify the analytical predictions, an experimental setup has been designed that accommodates multiple sensors and actuators to reliably characterize one of the proposed XY flexure mechanism designs.

II. Basic Theory

Given the wide applicability and advantages of flexures, there exists a considerable amount of design knowledge on these devices. While flexure design has been traditionally based on creative thinking and engineering intuition, analytical tools can aid the design conception, evaluation and optimization process. Consequently, a systematic study and modeling of these devices has been an active area of research. Some of the existing literature deals with precision mechanisms that use flexures as replacements for conventional hinges, thus eliminating friction and backlash.

III. Design

The key difference between flexure mechanism and parallel mechanisms with conventional joints is that kinematic stability is no longer a design consideration. Instead, task space stiffness needs to be carefully designed to avoid undesired motion in the presence of external loads [4]. We pose the design problem as a multi-objective optimization with manipulability and stiffness as performance measures and maximum joint stress and design parameter bounds as constraints.

There are two kinds of design configurations for multi-DOF mechanisms – serial and parallel. Serial designs present a stacked assembly of several single-DOF stages and incorporate moving actuators and cables, which can be detrimental for precision and dynamic performance. Parallel designs, which are considered here, are usually compact and allow ground mounting of actuators. While the generic performance characteristics of flexures such as mobility, error motions and stiffness variations have been defined in the prior literature, the specific desirable attributes of a parallel kinematic XY flexure mechanism and associated challenges are listed here [2].

1) The primary objective of the design is to achieve large ranges of motion along the desired directions X and Y, and an obvious limitation comes from material failure criteria. For a given maximum stress level, high compliance in the directions of primary motion or DOF increases the range of motion. However, in designs with out-of-plane constraints, this conflicts with the need to maintain high stiffness and small error motions in the out-of-plane directions, thus making planer designs preferable.

2) In an XY mechanism, the motion stage yaw is often undesirable. Given this requirement, the motion stage yaw may be rejected passively or actively. While both options have respective advantages, fewer actuators make the former favorable due to reduced design complexity and potentially better motion range. Thus, the mechanism has to be designed such that the rotation of the motion stage, being a parasitic error motion, is inherently constrained.

This error motion may be further attenuated by exploiting the Center of Stiffness (COS) concept to appropriately place the actuators, as explained in the following section.

3) Minimal cross-axis coupling between the X and Y degrees of freedom is an important performance requirement, especially in applications where end-point feedback is not feasible or the two axes are not actively controlled. Cross-axis coupling refers to any motion along the Y direction in response to an actuation along the X direction, and vice versa. In the absence of endpoint feedback, it necessitates an additional calibration step to determine the transformation matrix between the actuator coordinates and the motion stage coordinates. In unactuated or under-actuated systems, cross-axis coupling can lead to undesirable internal resonances.

4) An important challenge in parallel mechanism design for positioning is that of integrating the ground-mounted actuators with the motion stage [5]. Linear displacement or force source actuators typically do not tolerate transverse loads and displacements. Therefore, the point of actuation on the flexure mechanism must be such that it only moves along the direction of actuation and has minimal transverse motions in response to any actuator in the system. Eliminating transverse motion at the point of actuation, is termed as actuator isolation, and is generally difficult to achieve due to the parallel geometry.

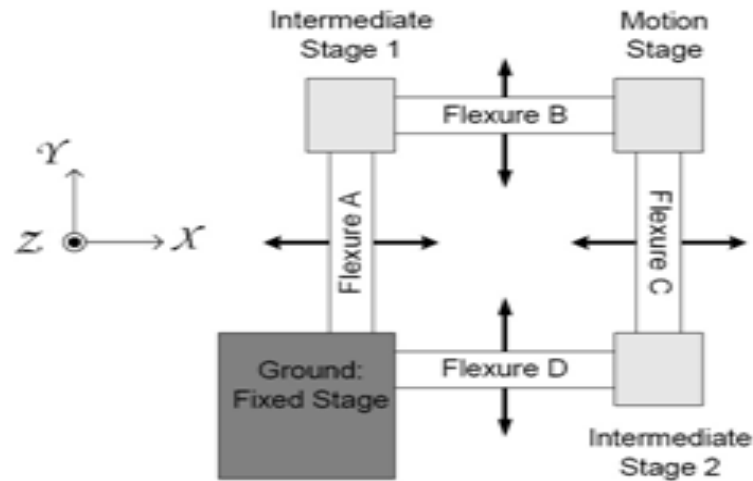
5) In the absence of adequate actuator isolation, the actuators have to be connected to the point of actuation by means of a decoupler, which ideally transmits axial force without any loss in motion and absorbs any transverse motions without generating transverse loads. However, a flexure-based decoupler, which is desirable to maintain precision, is subject to its own tradeoffs. Increasing its motion range and compliance in the transverse direction results in lost motion between the actuator and motion stage, affecting precision, and loss in stiffness along its axial direction [5]. The latter contributes to drive stiffness, which is the overall stiffness between the point of actuation and the motion stage, and influences the dynamic performance of the motion system.

6) Low thermal and manufacturing sensitivities are important performance parameters for precision flexure mechanisms in general. Both these factors, being strongly dependent on the mechanism's geometry, may be improved by careful use of reversal and symmetry. Because of the tradeoff between quality of DOF and DOC, all these performance measures including parasitic errors, cross-axis coupling, actuator isolation, lost motion and drive stiffness, deteriorate with increasing range of motion, as shall be shown quantitatively in the subsequent sections [5]. Depending upon the application, these collectively restrict the range of a parallel kinematic mechanism to a level much smaller than what is allowed by material limits. While geometric symmetry plays an important role in improving performance, it can over constrain the primary directions resulting in a significantly reduced range, if implemented inappropriately.

The constraint arrangement includes four basic rigid stages: Ground, Motion Stage, and intermediate Stages 1 and 2. Stage 1 is connected to ground by means of flexure module A, which only allows relative X translation; the Motion Stage is connected to Stage 1 via flexure module B, which only allows relative Y translation; the Motion Stage is connected to Stage 2 via flexure module C, which only allows a relative X translation; and finally, Stage 2 is connected to Ground by means of flexure module D, which only allows relative Y translation. Thus, in any deformed configuration of the mechanism, Stage 1 will always have only an X displacement with respect to Ground while Stage 2 will have only a Y displacement. Furthermore, the Motion Stage inherits the X displacement of Stage 1 and the Y displacement of Stage 2, thus acquiring two translational degrees of freedom that are mutually independent. Since the Y and X displacements of the Motion Stage do not influence Stage 1 and Stage 2, respectively, these are ideal locations for applying the actuation loads, obviating the need for decouplers. In an ideal scenario where the flexure modules A, B, C and D are perfect constraints, this arrangement would

yield a flawless design. However, due to the inherent imperfection in flexures, the actual resulting designs are expected to deviate slightly from ideal behavior. Any approximately linear motion flexure module can be used as the building-blocks A, B, C and D.

Figure 1 Illustrates the proposed constraint arrangement that can help achieve the above-listed desirable attributes in XY mechanisms.



IV. Application

The proposed handheld instrument is a 3-DOF manipulator that is driven by three piezoelectric actuators, because of their good resolution, rapid response, and high bandwidth. The three desired degree of freedom of this manipulator is the position of the needle tip. Fig. 2(a) shows the schematic diagram of the mechanism while Fig. 2(b) shows a photo of the manipulator. There are two portions. The upper portion is the mechanism while the lower part houses the piezoelectric actuators. The two parts are connected via screws. Fig. 2(c) shows the mechanism. The mechanism used is a parallel mechanism.

There are three limbs connecting the moving platform to the base. Each limb consists of a prismatic joint in series with a pin joint and a ball joint. The main function of the prismatic joint is to provide a spring-like effect to keep the ball joints in contact with the actuators, and the actuators in contact with the base. The lengths of the actuator slots are shorter than the length of the actuators by 0.1 mm, thus, providing a pre-stress of 3 N. An ideal prismatic joint is defined as one that is able to generate pure linear motion, without cross-axis effect like generating motion or rotation on other axes. The most conventional prismatic joint uses a parallelogram see Fig 3(a).

Figure 2 Handheld mechanism for micromanipulation. (a) Schematic diagram of the mechanism. (b) Photograph of the mechanism with actuator holder. (c) Exploded view of the mechanism

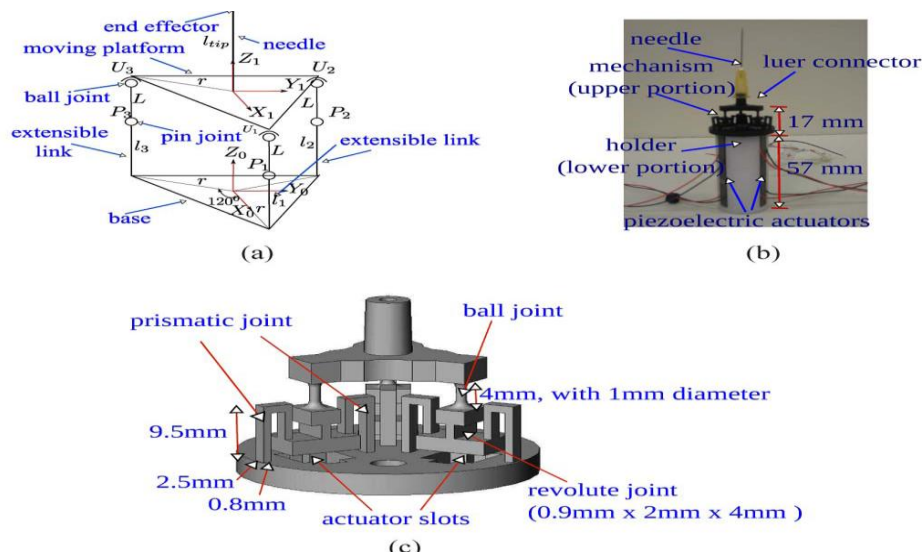
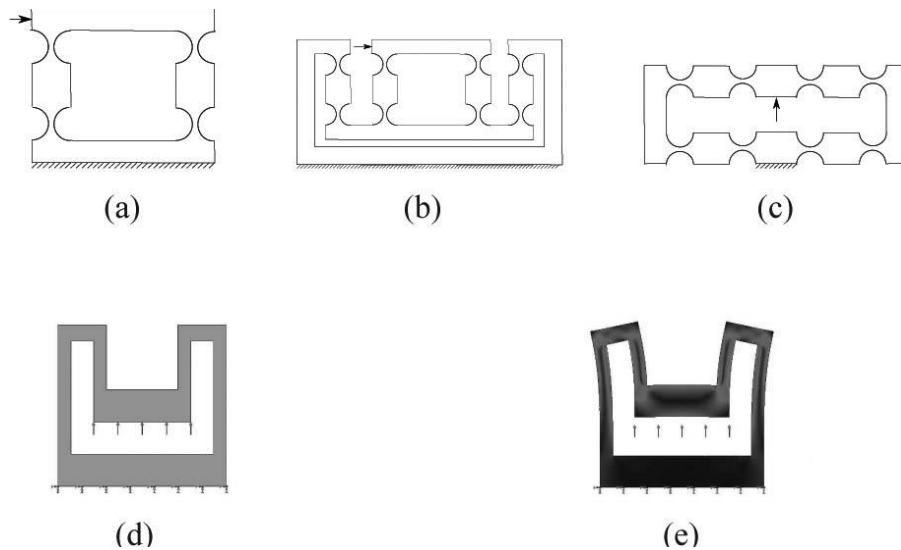


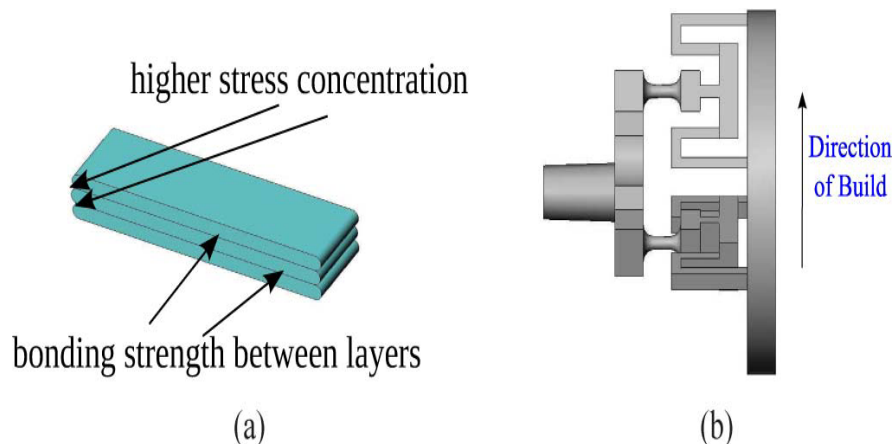
Figure 3 Flexure-based prismatic joints. (a) Conventional prismatic joint. (b) Prismatic joint using double parallelograms. (c) Prismatic joint from. (d) Prismatic joint making use of symmetrical deformation. (e) Deformation of prismatic joint



This conventional parallelogram prismatic joint is simple, but generates a linear motion with parasitic motion in the other axis. To overcome that, the idea of using double parallelograms see Fig. 3(b) was used [6]. The main disadvantage of Fig. 3(b) is that it is not symmetrical and changes due to temperature will not be uniform. As space constraint is a major consideration, the mentioned designs were not suitable. Hence, the idea of symmetry for symmetrical deformation is used and the prismatic joint is shown in Fig. 3(d). As the length of the ball joint in the axial direction is not relatively long enough, it is not an ideal ball joint as the bending stiffness along the axial direction is greater than along the two radial directions.

However, this design can still be used because the workspace is small and the axial direction of the joints is along the global z-axis when at zero position. Hence, very little bending is needed along each axial direction. The size of the manipulator is small enough for handheld purpose. The diameter of the mechanism (excluding the holder) is 27mm and the height from the mechanism's base to the moving platform is 17 mm.

Fig. 4. Fabrication of the handheld mechanism. (a) Part built layer by layer. (b) Orientation of the mechanism in the RP machine.



The diameter of the holder is 22 mm and the height of the holder actually depends on the length of the actuator, which is dependent upon the stroke length required for the application. In this paper, the height of the holder is 57 mm. A male connector is also built on the moving platform. A needle is attached to the moving platform via tight fit with the male connector. For a wider range of usage, the design of the connector can be changed accordingly to suit the required tool to be used.

V. Conclusion

In this paper, Key performance attributes and challenges in XY mechanism design have been explained, and new parallel kinematic XY flexure mechanism designs based on systematic and symmetric constraint

arrangements are proposed. These constraint arrangements allow large primary motions and small error motions without running into over-constraint problems.

Analyze the forces transmitted through the flexures in the four bar and design the flexures to always point in the direction of the transmitted forces. For flexural mechanisms, it is also very important to design the flexures to never buckle under typical operating conditions. For 3D mechanisms where mobility arises from geometric constraints, it is important to analyze for the effect of misalignment and design the structure in a manner which ensures proper alignment.

As a future work, heating elements will be embedded into or close to the flexures to decrease power consumption and to decrease the stiffness change response time for higher bandwidth motion control. Such active stiffness tunable flexure joints could be applied to any flexural miniature mobile robot and device mechanisms.

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