TOPOLOGY EVOLUTION OF HIGH PERFORMANCE XY FLEXURE STAGES

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Abstract

This paper presents a systematic constraint-based approach to the topology design of large motion range XY flexure stages. A generic parallel-kinematic constraint pattern is proposed with consideration to important performance metrics such as error motions, actuator isolation, and stiffness variation. Single DOF flexure modules are employed as constraint elements to construct XY flexure designs based on the proposed constraint pattern. Variations in the choice of constraint modules and the degree of geometric symmetry result in the evolution of high-performance flexure stages. Tradeoffs in performance are highlighted as these designs are discussed and compared. Analytical prediction and experimental measurement of the performance of a selected design are presented.

Keywords: XY Flexure, Constraint-based Design, Topology Synthesis

Introduction and Background

Despite the simplicity of serial stages, compactness, precision and manufacturability concerns make parallel stages preferable at the meso and micro scales. However, large range of motion has traditionally been a challenge in parallel kinematic designs due to problems associated with overconstraint and error motions [1-2]. This is a direct consequence of the fundamental tradeoff between the quality of Degrees of Freedom (DOF) and Degrees of Constraint (DOC) in flexures [3-4], which results in a deterioration of other performance metrics with increasing range of motion.

When designed for a motion system, an XY flexure needs to have low cross-axes errors to reduce dependence on calibration and end-point feedback. In the absence of an additional yaw control axis, it is also important that the parasitic yaw rotation of the motion stage is minimal. A key limitation of traditional parallel designs is that actuators cannot be directly connected to the motion stage, and intermediate decouplers that transmit linear motion but absorb transverse motions are needed. The motion range of the XY stage thus gets limited by the maximum transverse motion that the decoupler can provide without causing unacceptable errors, overconstraint, and loss in stiffness between the actuator and motion stage. It is therefore desirable that the point of actuation on the flexure stage moves only along the direction of actuation and does not respond to any other actuators in the system, an attribute termed as actuator isolation. Furthermore, XY stages suffer from undesirable stiffness variations since the stiffness of the constituent flexure modules change with loads and displacements. These factors collectively restrict the range of a parallel kinematic stage to a level much smaller than what is allowed by material limits [3,5]. Therefore, the XY stage topology needs to be generated by carefully considering the constraint behavior of flexures so that these performance metrics can be met with minimal compromises. Deviating from the traditional designs, a generic yet systematic constraint arrangement that can be implemented using common flexure modules is presented in this paper.

XY Flexure Topology Generation

The proposed parallel kinematic 2-DOF constraint arrangement, which utilizes multiple single DOF constraints, is illustrated in Fig.1. This arrangement includes four basic rigid stages: ground, motion stage, and intermediate stages 1 and 2. Stage 1 is connected to ground by means of a constraint that only allows relative X translation, and the motion stage is connected to stage 1 such that only a relative Y translation is allowed. Similarly, the constraint element connecting Stage 2 to ground only allows relative Y translation, and that connecting the motion stage to stage 2 only allows a relative X translation. Thus, in any displaced 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 DOF that are mutually independent. Since the Y and X displacements of the motion stage do not influence stages 1 and 2, respectively, the latter provide ideal locations for actuation.

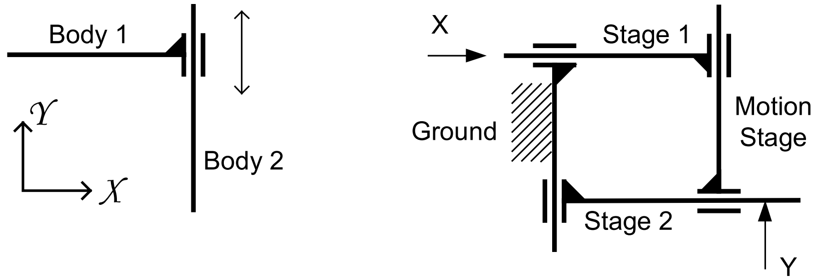

Fig.1 a) Ideal Single DOF Constraint b) Ideal Two-DOF Parallel kinematic constraint

Also, since all the connecting constraint elements inhibit relative planer rotation, the stage rotations are constrained. This arrangement would yield an XY stage with flawless performance if ideal constraints and assembly were possible. However, to maintain high precision, flexure modules that approximate single linear DOF constraints are used. Obviously, the imperfections of a given module affect the performance of the resulting XY stage. Using the common parallelogram flexure module, we obtain the XY design illustrated in Fig. 2A. Because of the parallelogram flexure's constraint properties, the planer rotation of the motion stage is constrained quite well. Furthermore, since this parasitic yaw has an elastic component, it may be further attenuated by an appropriate placement of actuation forces. Cross-axis coupling clearly exists due to the axial kinematic and elastokinematic errors. An X actuation force produces a small displacement of the motion stage in the Y direction. The point of application of X actuation force also moves slightly in the Y direction, which compromises actuator isolation. The overall performance of this design may be improved by making an insightful use of geometric symmetry where the design is mirrored about a diagonal axis without causing overconstraint, as shown in Fig.2B. The motion stage rotation in this case is further reduced due to the additional rotational constraints. On careful inspection, it may be observed that on the application of an X actuation force, the two sides of the mechanism tend to produce displacement of the motion stage in Y direction that oppose each other, and therefore cancel out. However, perfect actuator isolation is still not achieved due to the inherent properties of the parallelogram flexure. The latter design also provides an improved out-of-plane stiffness.

Another set of designs may be generated using the double parallelogram module, which also provides an approximate linear DOF. Rotations of the stages are constrained as earlier. Axial parasitic errors in this flexure module affect the cross-axis coupling, inline stiffness and actuator isolation in the resulting XY design of Fig.2C. Because of their nonlinear elastokinematic nature, these performance metrics are excellent for small displacements but deteriorate with increasing loads. Improvement in performance may be obtained by mirroring the design as shown in Fig. 2D. The out-of-plane stiffness is enhanced due to additional support, and space utilization is improved. Furthermore, the double parallelogram module is relatively insensitive to uniform thermal variations because any change in the beam lengths is readily absorbed by its secondary stage. However, this free secondary stage is also a drawback, since it leads to a significant loss in the axial stiffness of the module with increasing motion. A double tilted-beam flexure has been shown to be significantly better in this regard but exhibits higher error motions [4]. A combination of these two modules, as shown in Fig 2E, may be used to achieve best of both attributes.

It may be noticed that while the XY stage designs in Fig. 2D and 2E are rotationally symmetric, they lack a mirror-symmetry. In an obvious topological evolution, a design with a higher degree of symmetry is illustrated in Fig. 2F. With shorter beams, this mechanism has a smaller range of motion, but a larger motion stage. Performance attributes like cross-axis coupling, parasitic rotations, out-of-plane stiffness, and actuator isolation, appear to be better than the previous cases. In addition, because of the increased number of flexures, elastic averaging plays an increasingly important role in making the design more robust to manufacturing and assembly variations. This design may also be constructed using the double tilted beam flexure module to overcome the shortcomings discussed earlier.

Analytical Predictions and Experimental Results

A non-linear analysis, which incorporates elastokinematic and stress-stiffening effects in the building blocks, is necessary to capture the performance tradeoffs in the proposed designs [5]. Limited closed-form analytical results, corroborated by FEA, are presented here for the XY stage design 2D.

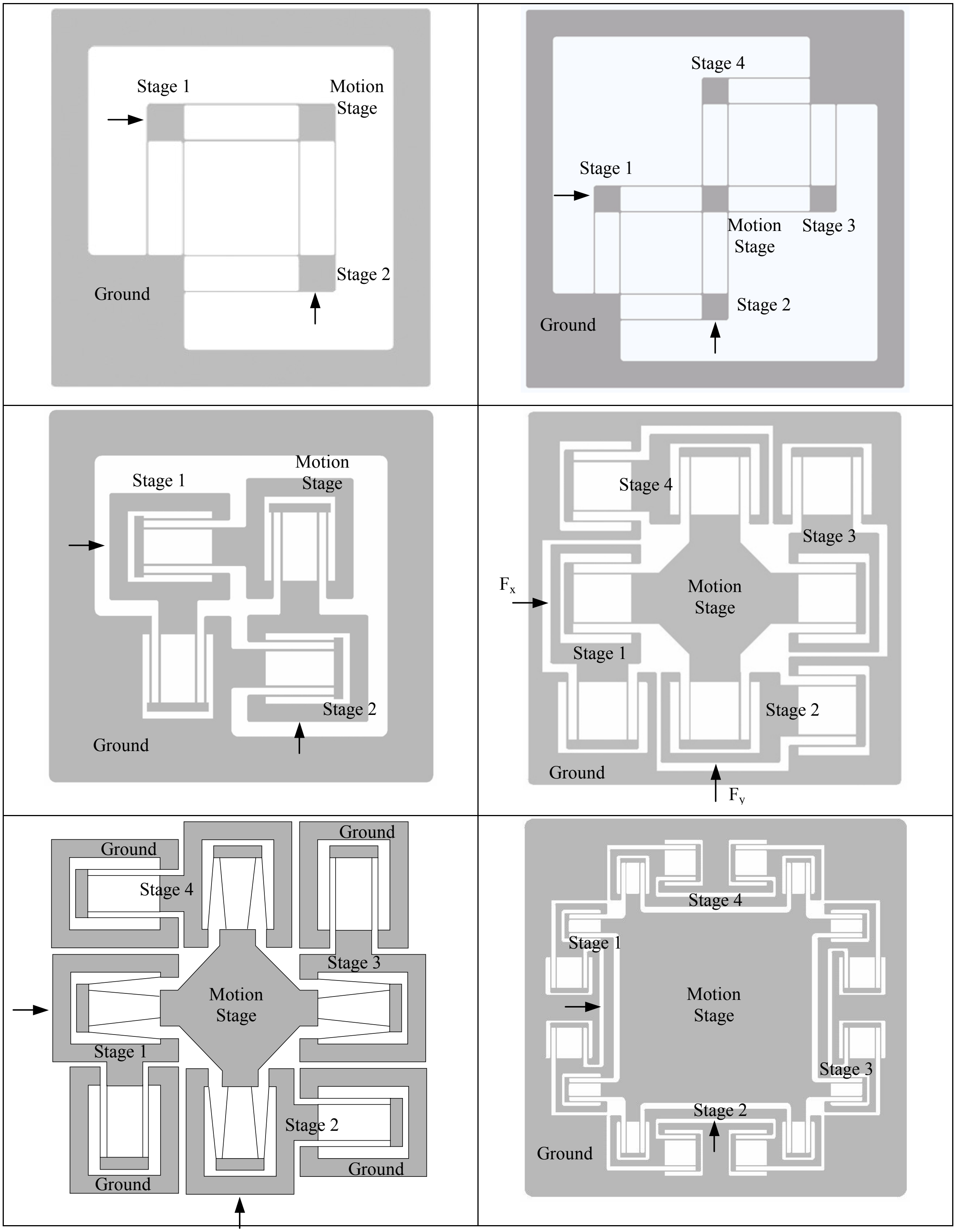

Fig.2 XY Flexure Stage Designs: A, B, C, D, E and F

Forces are applied along the center of stiffness axes of the mechanism to minimize stage rotations. All forces and displacements are normalized with respect to the beam properties. *a=12, e=1.2 and i = -0.6* are non-dimensional numbers characteristic of the simple beam, while *d* is the normalized axial stiffness.

$$x_{ms} = \frac{f_x}{4a}\frac{64a^2}{\left(64a^2 - 3e^2 f_y^{\,2}\right)} \quad ; \quad x_1 = \frac{f_x}{4a}\frac{64a^2}{\left(64a^2 - 3e^2 f_y^{\,2}\right)} + \frac{3f_x}{4}\left(-\frac{y_2^{\,2}ei}{2a} + \frac{1}{d}\right) \quad ; \quad y_1 = \frac{f_y}{4}\left(-\frac{x_1^{\,2}ei}{2a} + \frac{1}{d}\right)$$

$$x_2 = \frac{f_x}{4}\left(-\frac{y_2^{\,2}ei}{2a} + \frac{1}{d}\right) \quad ; \quad y_2 = \frac{f_y}{4a}\frac{64a^2}{\left(64a^2 - 3e^2 f_x^{\,2}\right)} + \frac{3f_y}{4}\left(-\frac{x_1^{\,2}ei}{2a} + \frac{1}{d}\right) \quad ; \quad y_{ms} = \frac{f_y}{4a}\frac{64a^2}{\left(64a^2 - 3e^2 f_x^{\,2}\right)}$$

$$x_3 = \frac{f_x}{4a}\frac{64a^2}{\left(64a^2 - 3e^2 f_y^{\,2}\right)} - \frac{f_x}{4}\left(-\frac{y_2^{\,2}ei}{2a} + \frac{1}{d}\right) \quad ; \quad y_3 = \frac{f_y}{4}\left(-\frac{x_1^{\,2}ei}{2a} + \frac{1}{d}\right)$$

$$x_4 = \frac{f_x}{4}\left(-\frac{y_2^{\,2}ei}{2a} + \frac{1}{d}\right) \quad ; \quad y_4 = \frac{f_y}{4a}\frac{64a^2}{\left(64a^2 - 3e^2 f_x^{\,2}\right)} - \frac{f_y}{4}\left(-\frac{x_1^{\,2}ei}{2a} + \frac{1}{d}\right)$$

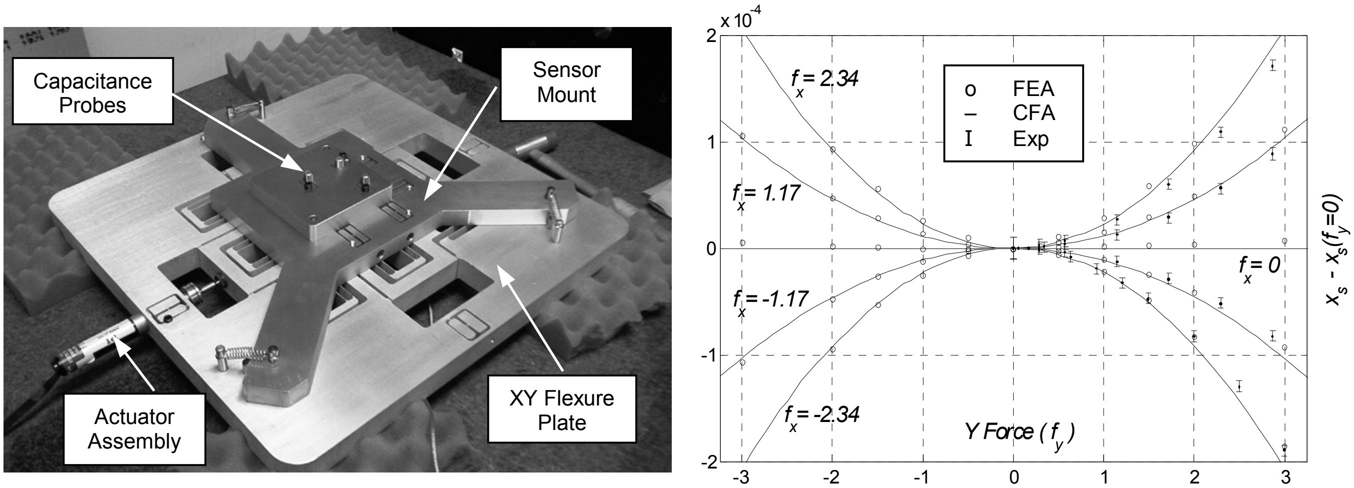

Fig. 3 a) Experimental Set-up b) Analytical vs. Experimental Results

A 300mm x 300mm AL6061-T651 prototype was precisely machined using wire-EDM for the purpose of experimental validation (Fig.3a). The experimental set-up was designed so that the stage can be actuated along its analytically determined center of stiffness axes using free weights, motorized precision micrometers, and piezoelectric stacks, and measured using plane mirror interferometry, autocollimation and capacitance gages. The tested prototype exhibits a 5mm x 5mm range with cross-axis errors less than 10μm, and motion stage yaw errors within 5 μradians. Experimental measurements are shown to be in excellent agreement with the closed-form analysis (CFA) and FEA in Fig.3b.

Conclusions

Several other XY stage designs may be conceived and tailored for specific applications using the topology generation method proposed in this paper. Obviously, the performance of a given mechanism is the direct consequence of the imperfections of its constituent modules, but their influence may be reduced by means of geometric symmetry. XY stages D, E and F are currently being developed for applications in nanopositioning for molecular experiments, meso-machining center, and MEMS devices. Ongoing analytical research includes dynamic, thermal and sensitivity analyses.

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