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Finite Element Analysis Of Planar Flexural Mechanisms For High Precision, High Speed Scanning Applications.

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Abstract

The beam flexure is an important constraint element in flexure mechanism design. Planar flexural mechanisms are best suitable candidate for high precision, high speed scanning applications. Parallel kinematic planar flexure mechanism design based on systematic constraint pattern that allow large range of motion without causing over-constraint or significant error motion are discussed in this paper. The standard parallelogram and double parallelogram flexure module are used as a constraint building block and its force-displacement characteristics are employed mathematically for predicting the performance characteristics of planar flexure mechanism design. Mathematical predictions are validated by means of Finite Element Analysis.

1. Introduction

Micro and nano-positioning stages play a very important role in modern technology. It finds applications in many fields, such as micromachining and scanning probe (such as scanning tunnelling, atomic force, etc.) microscopy. Various XY scanning mechanisms are developed which ranges from screw type to high precision recirculation ball mechanisms. XY scanners developed up till now have many limitations such as limited scanning range, limited performance characteristics, accuracy, backlash and many more. Also it is difficult to develop appropriate control system to achieve desired performance. Hence new era of mechanisms called compliant/flexural mechanisms are developed for high speed precision applications. Flexures are compliant structures that rely 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. There are numerous advantages of these types of mechanisms such as smooth motion with zero friction, zero backlash, high speed scanning etc.

2. Literature Review

Shorya Awtar, Alexander H. Slocum [1] presents parallel kinematic XY flexure mechanism designs based on systematic constraint patterns that allow large ranges of motion without causing over-constraint or significant error motions. Comparisons between closed-form linear and non-linear analyses are presented. It is shown that geometric symmetry in the constraint arrangement relaxes some of the design tradeoffs, resulting in improved performance. The non-linear analytical predictions are validated by means of computational FEA and experimental measurements.

Shorya Awtar, Shiladitya Sen [2] provides a highly generalized yet accurate closed-form parametric load-displacement model for two-dimensional beam flexures, taking into account the nonlinearities arising from load equilibrium applied in the deformed configuration. In particular, stiffness and error motions are parametrically quantified in terms of elastic, load-stiffening, kinematic, and elastokinematic effects. The proposed beam constraint model incorporates a wide range of loading conditions, boundary conditions, and beam shape. The accuracy and effectiveness of the proposed beam constraint model is verified by nonlinear finite elements analysis.

Yangmin Li et al. [3] presents the modeling and evaluation of a nearly uncoupled XY micromanipulator designed for micro-positioning uses. The manipulator is featured with monolithic parallel-kinematic architecture, flexure hinge-based joints, and piezoelectric actuation. Based on pseudo-rigid-body simplification and lumped model methods, the mathematical models for the kinematics and dynamics of the XY stage have been derived in closed-forms, which are verified by finite element analysis (FEA).

3. Building Blocks and Related Flexural Mechanism

In designing high performance planer two-axis error-free flexural mechanisms any linear motion flexure unit, which comes close to the stated idealizations, can be used as a building block to produce a two DOF planer mechanism. This sections present designs generated with three types of building blocks: the simple beam flexure, the parallelogram flexure and the compound (or double) parallelogram flexure.

3.1 The Simple Beam Flexure as a Building Block

A simple beam is not a very good single degree of freedom flexure unit, but due to its simplicity may use it as a building block in the mechanism. From the figure the beam tip translates (δ) as well as rotates (θ) when it experiences a force (F). Furthermore, it also exhibits a parasitic error in the X direction (ε).

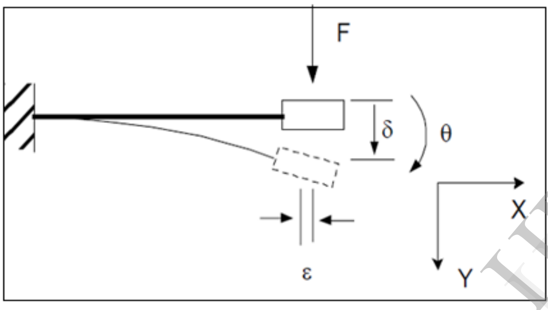

Figure 1 Simple beam flexure

Utilizing the above building block that single beam flexure the two axes planer flexure mechanism can be developed as represented in following figure 2.

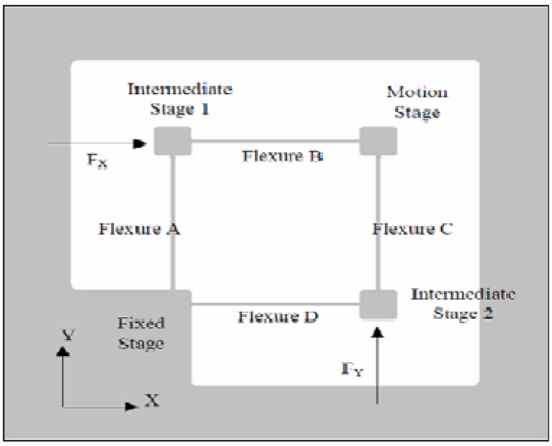

Figure 2 Two-axes planer flexure mechanism1

3.2 The Parallelogram Flexure as a Building Block

The parallelogram flexure unit is can be used in various flexural mechanisms. Figure 3 provides a schematic of the flexure in its deformed and undeformed configurations. Beam bending analysis can be used to predict the force-deformation characteristics of this flexure. It can be analytically shown that parallelogram flexure offers little resistance to relative motion in Y direction but is very stiff with respect to relative motion in X and rotation. Hence, it a much better approximation for a single DOF flexure as compared to the single beam used in the previous case.

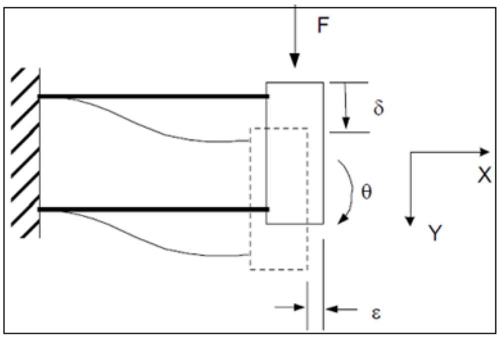

Figure 3 Parallelogram Flexure as a Building Block

Utilizing the above building block the two axes planer flexure mechanism can be developed as represented in following figure 4.

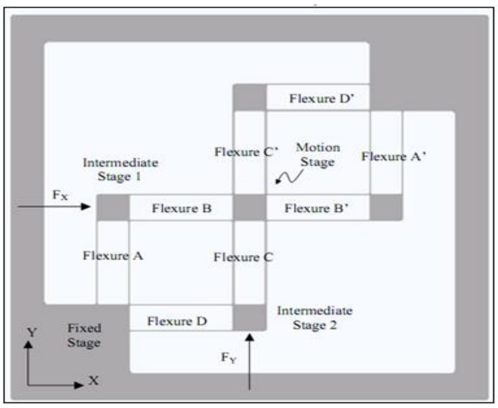

Figure 4 Two-axis planer flexure mechanism2

3.3 The Double Parallelogram Flexure as a Building Block

The simple parallelogram flexure suffers from inherent motion errors due to its geometry. If used as a building block, these errors also appear in the resulting flexural mechanism, thus leading to inadequate performance measures. Instead of using a simple parallelogram flexure, if use the double parallelogram flexure unit, shown in Figure 5, as the building block for two-axis planer flexural mechanisms, better overall performance can be achieved.

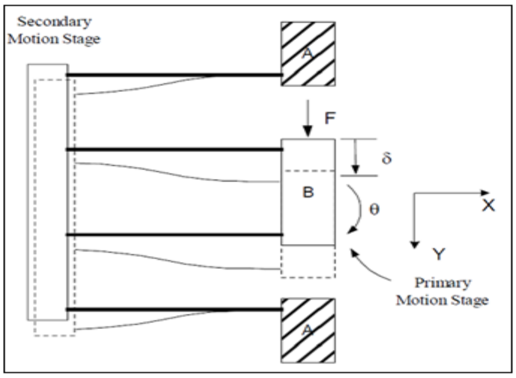

Figure 5 Double Parallelogram Flexure as a Building Block

Utilizing the above building block the two axes planer flexure mechanism can be developed as represented in following figure 6.

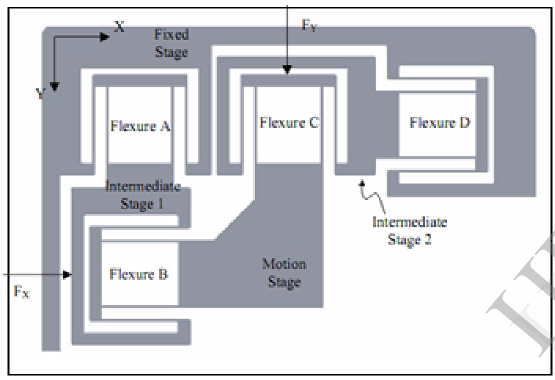

Figure 6 Two-axes planer flexure mechanism3

4. Static Analysis Of Flexural Building Block

4.1 Beam Constraint Model

A brief overview of the BCM for a simple beam flexure uniform thickness and initially straight is provided below.

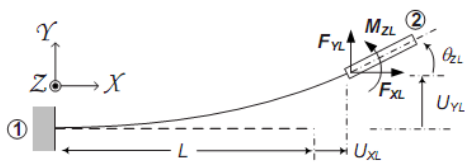

Figure 7 Simple Beam Flexure

Figure 7 illustrates a simple beam length: L, thickness: T, and depth: H, interconnecting rigid bodies1 and 2, subjected to generalized end-loads F_{XL}, F_{YL}, and M_{ZL}, resulting in end-displacements U_{XL} (DOC), U_{YL} (DOF), and θ_{ZL} (DOF) with respect to the coordinate frame X-Y-Z. In these, all loads, displacements, and stiffness terms are naturally normalized with respect to the beam parameters: displacements and lengths are normalized by the beam length L, forces by EI_{zz}/L^2 and moments by EI_{zz}/L. Thus we can define,

$$\frac{F_{XL}L^2}{EI_{ZZ}} = p; \quad \frac{F_{YL}L^2}{EI_{ZZ}} = f; \quad \frac{M_{ZL}L}{EI_{ZZ}} = m;$$

$$\frac{U_{XL}}{L} = x; \quad \frac{U_{YL}}{L} = y; \ \theta_{ZL} = \theta; \quad \frac{T}{L} = t$$

With these notations mathematical model for simple beam flexure which is basic of further analysis is given as:

$$\begin{bmatrix} f \\ m \end{bmatrix} = \begin{bmatrix} a & c \\ c & b \end{bmatrix}\begin{bmatrix} y \\ \theta \end{bmatrix} + p\begin{bmatrix} e & h \\ h & g \end{bmatrix}\begin{bmatrix} y \\ \theta \end{bmatrix}$$

$$x = \frac{1}{d}p + [y \quad \theta]\begin{bmatrix} i & k \\ k & j \end{bmatrix} + p[y \quad \theta]\begin{bmatrix} r & q \\ q & s \end{bmatrix}\begin{bmatrix} y \\ \theta \end{bmatrix}$$

The stiffness coefficients(a,b,c,e,h,g) and constraint coefficients(i,j,k,r,q,s) in general, are non-dimensional beam characteristic coefficients that are solely dependent on the beam shape and not its actual size.

4.2 Analysis of Parallelogram Flexure Module

$$y = \frac{f - (2c + ph)\theta}{(2a + pe)} \approx \frac{f}{(2a + pe)}$$

$$x \approx \frac{p}{2d} + y^2 i + \frac{p}{2}y^2 r$$

$$\theta = \frac{(4a^2 + 4pea + p^2e^2 + f^2dr)\,[(2ma - 2fc + p(me - fh)]}{2w^2 d(2a + ep)^3}$$

These equations give transverse force-displacement variation, transverse stiffness variation, axial stiffness variation and stage rotation prediction for parallelogram flexure module.

4.3 Analysis of Double Parallelogram Flexure Module

$$y = \frac{4af}{(2a)^2 - (ep)^2}$$

$$x = \frac{p}{d} + py^2 \frac{r[(2a)^2 + (ep)^2] - 8aei}{(4a)^2}$$

$$\theta = \frac{1}{2w_1^2}\left(\frac{1}{d} + \frac{f^2}{(2a-pe)^2}r\right)\left[m - \frac{f}{(2a-pe)}\left(1 - \frac{p}{(2a+ep)} + (2c-ph)\right)\right] + \frac{1}{2w_2^2}\left(\frac{1}{d} + \frac{f^2}{(2a+pe)^2}r\right)\left[m - \frac{f}{(2a+pe)}(2c+ph)\right]$$

5. Finite Element Method

The finite element method is a numerical technique. In this method all the complexities of the problems, like varying shape, boundary conditions and loads are maintained as they are but the solutions obtained are approximate. Because of its diversity and flexibility as an analysis tool, it is receiving much attention in engineering. The fast improvements in computer hardware technology and slashing of cost of computers have boosted this method, since the computer is the basic need for the application of this method. A number of popular brand of finite element analysis packages are now available commercially. Some of the popular packages are STAAD-PRO, GT-STRUDEL, NASTRAN, NISA and ANSYS. Using these packages one can analyze several complex structures. The Finite Element Method (FEM) was developed more by engineers than mathematicians using abstract methods. The FE method is the way of getting a numerical solution to specified problem. The FE analysis does not produce formula as a solution, nor does it solve the class of problems. Also the solution is approximate unless the problem is so simple that convenient exact formula is available.

6. Finite Element Analysis

Basic flexural module and related mechanisms are analysed in ANSYS V11 workbench. Engineering data used for the analysis are: Material-structural steel, Young's modulus (E)-2e11Pa, Poisson's ratio-0.3, Beam material density-7860 kg per cubic meter.

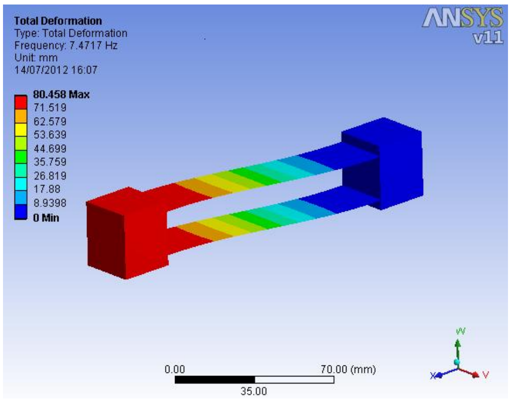

Fig.8 ANSY'S Result of total deformation for parallelogram flexure module.

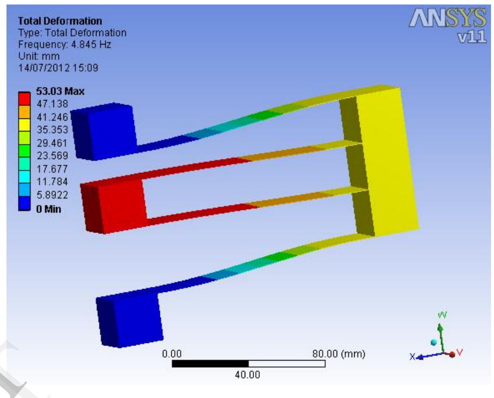

Fig.9 ANSY'S Result of total deformation for double parallelogram flexure module.

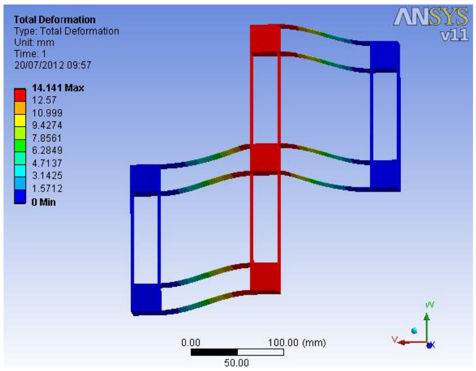

Fig. 10 ANSY'S Result of total deformation for parallelogram flexural mechanism

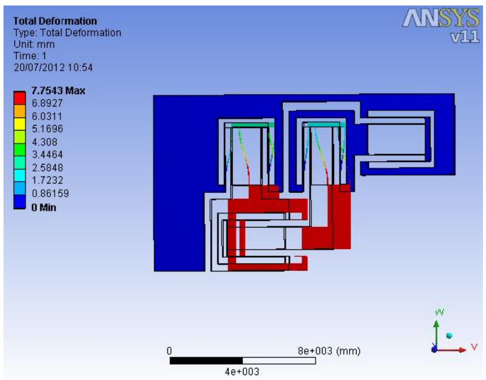

Fig.11 ANSY'S Result of total deformation for double parallelogram flexural mechanism

7. Result and Discussion

A thorough Finite Element Analysis has been performed in ANSYS for Parallelogram flexure module and Double parallelogram flexure module to validate the mathematical analysis (CFA) results. The closed-form analysis (CFA) and FEA predictions are found to be in close agreement. In both the cases, these two analysis methods match to within 10% in terms of numerical value. Solid 186 elements have been used with the large displacement option turned on. The geometric parameters for the two flexure module are kept identical or as close as possible to provide an even comparison. The result of Force-displacement behaviour of flexural module and its mechanism are produced below.

7.1 Parallelogram Flexure Module: (PFM)

Force	1	2	3	4	5	6
y (FEA)	0.04	0.08	0.11	0.15	0.2	0.2
y (CFA)	0.04	0.08	0.12	0.16	0.20	0.25

Table.1 Normalized Force-Displacement Result for PFM

7.2 Double Parallelogram Flexure Module: (DPFM)

Force	1	2	3	4	5	6
y (FEA)	0.05	0.11	0.16	0.21	0.26	0.31
y (CFA)	0.08	0.16	0.25	0.33	0.42	0.50

Table.2 Normalized Force-Displacement Result for DPFM

Force(F)	0	20	40	60	80	100	120
Deflection-X in mm	0	1.57	3.15	4.74	6.31	7.89	9.47
Deflection-Y in mm	0	2.02	4.04	6.06	8.08	10.1	12.12

7.3 Parallelogram Flexural Mechanism

Table.3 Force-Displacement Result for Parallelogram Flexural Mechanism

7.4 Double Parallelogram Flexural Mechanism

Force (F)	0	50	100	150	200	250	300
Deflection-X in mm	0	1.29	2.58	3.87	5.17	6.46	7.75
Deflection-Y in mm	0	1.1	2.18	3.27	4.36	5.4	6.53

Table.4 Force-Displacement Result for Double Parallelogram Flexural Mechanism

8. Conclusion

Parallel elasto-kinematics is used for planer flexure mechanism design that results in simple and compact embodiments. Two-axis (2 DOF) planer mechanism designs based on the simple beam flexure, the parallelogram flexure and the double-parallelogram flexure as building blocks, are presented. Novel ideas for using symmetry to increase out of plane stiffness and improve performance and robustness in design are presented. Errors due to imperfect building blocks are corrected by use of geometric symmetry.

9. Future Scope

The designs presented in this paper are very fundamental and can be used over a wide range of macro, meso or micro scale precision machines where decoupled multiple degrees of freedom are required. Potential applications can be found in optical instruments, Micro and Nano Electro Mechanical Systems, precision metrology, etc.

10. References

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