www.ijecs.in
International Journal Of Engineering And Computer Science ISSN:2319-7242
Volume – 4 Issue - 12 December, 2015 Page No. 15376-15380

Design And Analysis Of Xy Positioning Table

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Abstract: Flexure mechanisms have massive range in various industrial application required for high precision and frictionless motion. There are many study on concept to make precision manipulators, but only some of them can achieved to satisfy the high speed with precision. Pro-E software is used for parametric modeling of XY positioning table ANSYS is used for Static analysis and dynamic analysis . Deflection of motion is concluded by static analysis with force. The Deformation of XY mechanism is equivalent to S-shaped cantilever beam deformation. Force and deformation curve is linear. There results get compare with mathematical calculation with FEA results.

Keywords: Flexure Mechanism, FEA, XY positioning.

1. Introduction

When Flexure mechanisms uses as bearing to provide smoothen motion. A flexure mechanism is a single-piece mechanism that transfers movement without any relative motion between joints or linkages, thus motion is wear free, energy efficient, higher resolution, and high speed device. Flexures are structure that depends on Material elasticity for their functionality. Motion is generated due to deformation at molecular level, which results in primary characteristic in flexures- smooth and precision motion for example in camera lens cap, laser scanning machine.

In this paper a flexural mechanism is designed to provide a linear motion in a compliant manner. Flexure mechanisms offer a number of advantages, such as increased precision, reduced friction and wear, simple (sometimes monolithic) construction, and reduced assembly. In many ways compliant mechanisms have developed similar functionality to rigid mechanisms. Flexure mechanisms could potentially offer an attractive choice to conventional linear motion mechanisms both in terms of improved functionality and decreased cost. Because flexure mechanisms gain some or all of their motion from deflection of the linkages, they have the potential to completely eliminate relative motion between linkages, and thus eliminate friction. As an added benefit, since mechanism members couple its energy storage with linkage motion as they deflect, stable positions can be integrated into the design. [3] [4]Several linear motion flexure mechanisms, including bi-stable mechanisms, have been developed, although they provide much less travel for their size compared to prismatic joints. Unfortunately, mechanisms that do have a longer travel often have significantly reduced off-axis stiffness due to the use of long flexural members.

2. Modeling and Analysis of XY Flexure mechanism

Based on the designs studied we found out that the all the mechanisms were based on flexural motion. An elastic strip is made to bend or twist causing distortion in its original dimensions and producing the desired motion. After studying various existing mechanisms, we tried designing our own mechanism based on Flexural Force Transmission

Trial Models

2.1 Single Beam(Rectangular) Hinge Type Flexure Mechanism

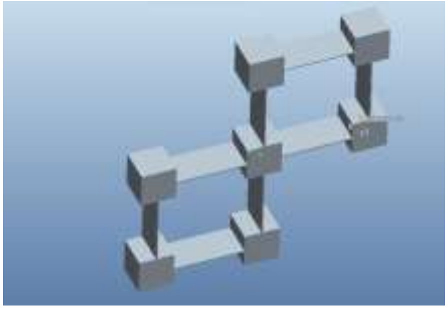

Figure 1: Single Beam Hinge type flexure mechanism

Single beam hinge type structure, where beams were fixed to the supports and their output was linear. In this model type version the angular motion was cancelled out by connecting two beams Parallel to each other. Unfortunately, the design produced comparatively less amplification than expected and because of the weight of the mechanism it is difficult to stabilize hence this design had to be rejected. A new design was sought for that was based on the flexural bending of the links and motion transmission caused displacement amplification. The design already existed and little modifications were made hence it lacks innovation. Also the design had complex linkages causing difficulty in manufacturing. Because of these reasons the design had to be rejected thus another design need to be sought for.

2.2 Single Piece Based Flexure Mechanism

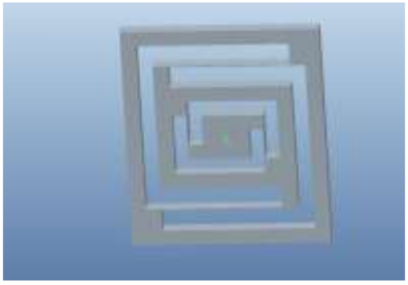

Figure 2: Single Piece Based Flexure Mechanism

To eliminate the mounting difficulties we decided to take Single piece mechanism in which whole mechanism is cut from the single block using Wire cut machining processes. It provides ease in mounting and avoids unnecessary displacement of the beams (strips), which provides good stability and accuracy. The stress developing in this design is maximum. It contains the angular motion of the beams because of its design and this mechanism provides linear motion.

2.3 Design Modification

As we have seen that in the previous designs the main drawback was due to the complexity of design and less displacement. This made the manufacturing infeasible and also hampered the results drastically. Objective is to produce linear motion. Thus in order to achieve this we have implemented this principle in two different directions in such a manner so that motion in X-direction and motion in Y-direction is nullified. These two links that are fixed are orthogonal to each other so that the motion in perpendicular direction is cancelled. The material selected is determined by the elastic modulus of the flexure strip that is required to produce the desired bending of Strips.

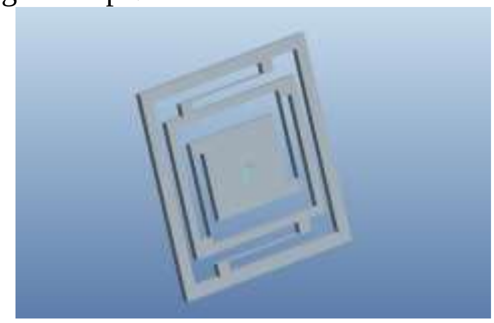

Figure 3: Present Design of Flexure Mechanism

3. Theoretical Deflection Calculation

Figure 4: Equivalent Cantilever Beam deformed n S-Shape

Buckling load of the beam is given by: [1]

$$P = \frac{\pi^2 EI}{L_e^{\,2}} \Rightarrow \frac{PL_e^{\,2}}{EI} = -9.87$$

Maximum allowable tip displacement is given by,

$$\left(\frac{\Delta y}{L}\right)_{max} \leq \frac{1}{3\eta}\frac{S_y}{E}\frac{L}{T}$$

Now for any flexure mechanism, these values range as given below: Δy = 0.01L to 0.1L We will aim to obtain results for deformation value as high up to 0.1L then flexure mechanism get the linear motion.

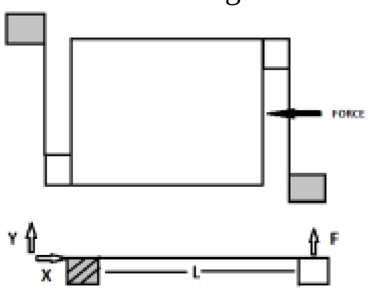

Figure 5: Force Acting on Mechanism

For S-shaped Deformation of beam we have from ref [1]

$$\begin{bmatrix}\delta_y\\ \theta\end{bmatrix} = \begin{bmatrix}1/6 & 1/2\\ 1/2 & 1\end{bmatrix}\begin{bmatrix}f\\ M\end{bmatrix}$$

Where,

$$\delta_y = \frac{\Delta y}{L}\ ,m = \frac{ML}{EI}\ ,f = \frac{FL^2}{EI}$$

Deflection of beam is given by above matrix we have,

$$\frac{\Delta y}{L} = \frac{1}{6}\frac{FL^2}{EI} + \frac{1}{2}\frac{ML}{EI}$$

Deflection of beam is given by equation,

$$\Delta y = \frac{FL^3}{6EI}$$

Where,

Δy - Deflection in beam in mm

F - Applied Force in N

L - Length of Beam in mm

E - Young's Modulus N/mm2

I - Movement Of Inertia.

To Find optimize design Considering the length variation in range 85mm to 100 mm. Thickness in Range 0.75-1.5 mm based on Design Consideration

4. FEA Analysis

The Geometrical Modeling of the Flexure mechanism is essential for the numerical analysis and graphical representation of the model. This is done on CAD software which in this case is Pro-E Wildfire 5.0

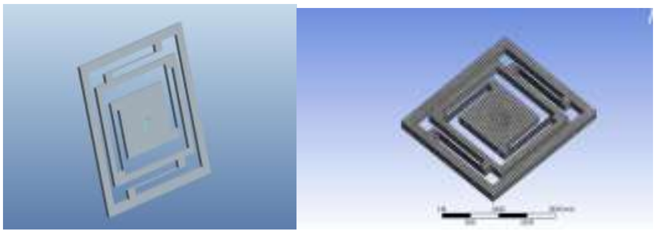

Figure 6: CAD Model Of Mechanism and Fine Meshing

4.1 Analysis of Mechanism

Import the model from external source (Pro-E) in parasolid (.x_t) format. Select the material for all the parts, Boundary Conditions, and Meshing.

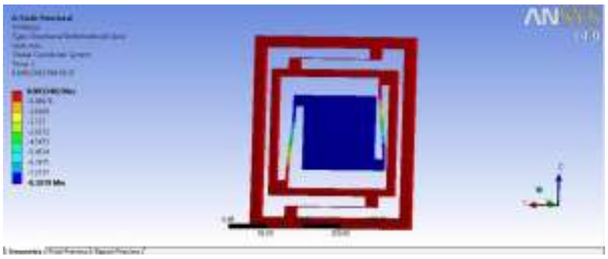

Figure 7: Graphical results of Displacement Analysis in X-Dir

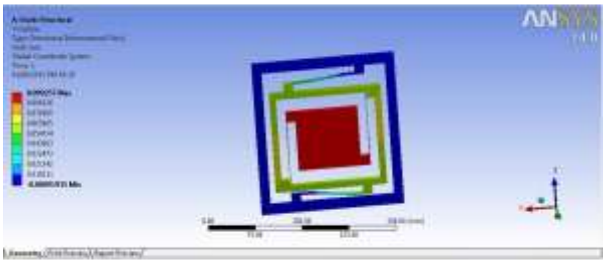

Figure 8: Graphical results of Displacement Analysis in Y-Dir

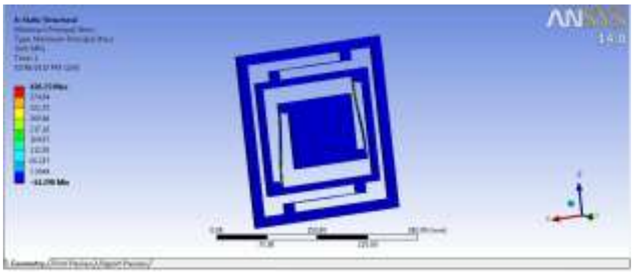

Figure 9: Stress Analysis in X- Direction

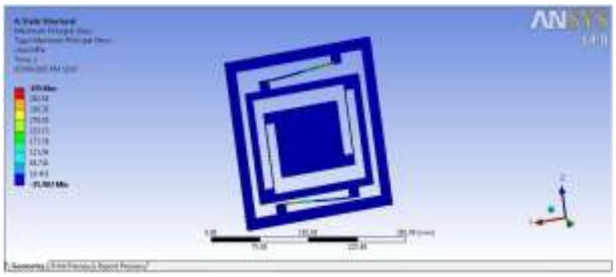

Figure 10: Stress Analysis in Y- Direction

As it is seen from the figure, the maximum stress created at the Flexure strip is 426MPa in X direction and 435MPa in Y direction. The material used for Flexure strip is Mild Steel. The Tensile Yield Strength of Mild Steel is 490MPa. Thus the maximum stress generated at the Flexure strip is within the permissible limit. Therefore the design is safe.

Table 1: Various Design Instances for 5N Force

Sr. No.	L (Length mm)	b (Width mm)	T(thickness mm)	Theoretical Deflection (mm)	FEA Stress (N/mm2)
1	90	15	1	4.2638	221.99
2	95	20	1.5	1.1164	75.85

DOI: 10.18535/Ijecs/v4i12.47

3	100	25	2	0.44664	37.031
4	90	10	1	6.4807	403.2
5	95	10	1.5	2.7905	160.83
6	100	10	2	1.1435	84.297
7	85	15	0.5	28.316	819.28
8	95	15	1.5	1.5094	104.43
9	100	15	2	0.75501	61.996
10	85	20	0.5	21.053	604.88
11	90	20	1	3.174	164.77
12	100	10	0.85	1.9384	42.485
13	85	25	0.5	16.719	484.11
14	90	25	1	2.5205	130.57
15	95	25	1.5	0.75978	58.6222

Based on above table the most optimized Design for Flexure mechanism is L = 100 mm, b = 10mm and t = 0.85 mm less Stress developed.

Designed Flexure Mechanism Specification:-

Length = 100 mm.

Width = 10 mm.

Thickness = 0.85 mm.

Moment Of inertia = 0.511770833

Yield Strength := 490 MPa

Young's Modulus = 2.1 x 10^5 N/mm2

Based on this Design We calculate deflection of beam for Force Acting from 0 to 20 N force

Table 2: Total Deformation and Max Stress by FEA

Sr No.	X Compo-nent	Y Comp-onent	Total Deformation	Maximum Equivalent Stress
Units	N	N	mm	Mpa
1	1	1	0.585393	21.66
2	5	5	2.926966	103.3
3	10	10	5.853933	216.6
4	15	15	8.780899	324.89
5	20	20	11.70787	433.19

Table 3: Directional Deformation in X- Direction

Sr. No	Y Direction Force	Y Deformation FEA	Deformation by Analytical
Units	N	mm	
1	1	0.40	0.38
2	5	2.04	1.93
3	10	4.09	3.87
4	15	6.14	5.81
5	20	8.18	7.75

Table 4: Directional Deformation in Y- Direction

Sr. No	Y Direction Force	Y Deformation FEA	Deformation Analytical
Units	N	mm	
1	1	0.42	0.38
2	5	2.10	1.93
3	10	4.20	3.87
4	15	6.31	5.81
5	20	8.41	7.75

5. Conclusion

After analyzing the results we can observe that the Input Displacement vs. Output Displacement characteristics for the FEA and numerical calculations remain the same. As well as stress are in permissible limit.

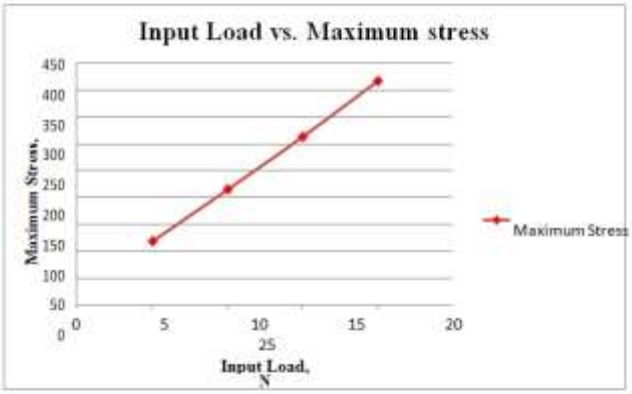

Figure 11: Input Load Vs Maximum stress X-Direction

DOI: 10.18535/Ijecs/v4i12.47

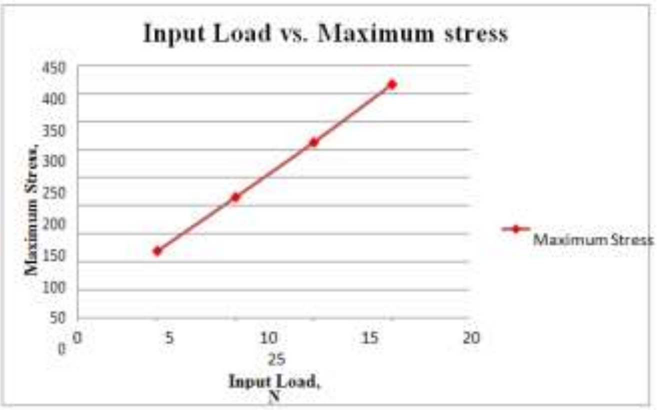

Figure 12: Input Load Vs Maximum stress Y-Direction

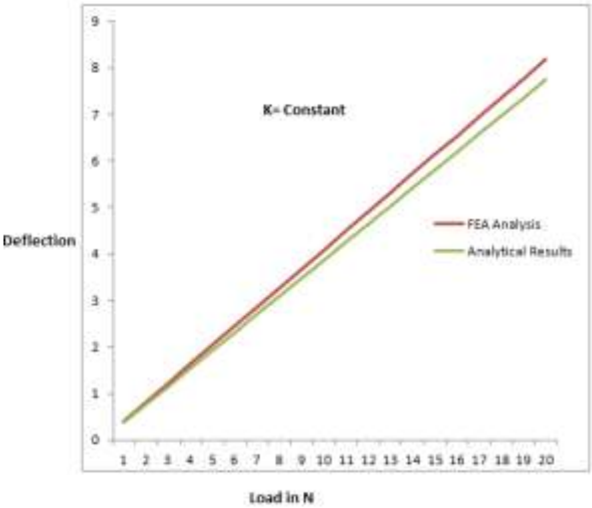

Figure 13: Input Load Vs Maximum Load

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Author Profile

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