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Comparison of Flexural Joints Used in Precision Scanning Mechanism Using FEA Tool

#1D M Bhoge, #2S P Deshmukh

1dhananjay.bhoge@rediffmail.com
2suhas.deshmukh@gmail.com

1,2Mechanical Engineering Department, Savitribai Phule Pune University
1Genba Sopanrao Moze College of Engineering, Balewadi, Pune
2 Sinhgad Academy of Engineering, Kondhwa, Pune

ABSTRACT

Precision scanning mechanisms is current demand of modern technologies such as micro-nano manufacturing, characterization equipments such as microscopes etc. Different flexural based mechanisms are developed for precise control/manipulation of position of object. These mechanisms operate based on flexibility of material and gives frictionless, backlash free motion. Current article attempts to compare the different flexural joints used for development of flexural mechanisms using FEA (Finite Element Analysis) tool ANSYS® Workbench™ 14.5. Different joints such as elliptical, circular, leaf used in different mechanisms are compared based on deformation, operation range, maximum stresses induced, and stiffness of the joint. Elliptical joint provides better performance compared to other joints. Further, using elliptical joint flexural mechanism with 5 mm scanning range is designed and experimental setup is developed for validation. It is observed that experimental results are having close match with FEA simulations and error involved is due to variation in material properties, actual dimensions of setup etc. Such mechanisms can be further used for precision applications such as micro welding, wafer alignment in lithographic micro-manufacturing.

Keywords— ANSYS®, FEA, Flexural Mechanisms, Scanning Mechanisms.

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I. INTRODUCTION

Precise control/manipulation of position of object demands the use of flexure based precision scanning mechanisms in micro-nano manufacturing, characterization equipments such as microscopes, wafer alignment in lithographic micro-manufacturing etc. Flexure joints based on elastic deformation are used in precision scanning applications as they offer the advantage of providing frictionless, backlash free, smooth and continuous motion with inherently infinite resolution .When flexure joints are used in a mechanism, the static and dynamic characteristics of the mechanism depend mainly on the performance of the flexure joints [1].

Performance evaluation of the flexural joints is done in first part of this work by investigating the influences of the geometric parameters on the characteristics of the flexure hinges. Performance characteristics like deformation, stress, stiffness, motion range and accuracy [2] are evaluated and compared for elliptical arc, elliptical, circular arc, right circular, leaf and multiaxis flexural joints through extensive parametric analysis using 'FEA' tool 'ANSYS® Workbench™ 14.5'. In order to achieve efficient compliant mechanisms for applications in micro- nano manipulation, it is important to correctly choose the geometric parameters and to predict and optimize the performance of flexure hinges [3].A catalog of design charts based on the parametric models qualitatively categorizing the joints

presented in this work will act as a useful tool for predicting and simulating the characteristics of these joints in the design stage.

Ultra-precision positioning over a large travel range is one of the trends for 'XY-stages' used in precision industries. For

example, ultra-precision 'XY-stages' with nano-metric positioning resolutions and tens of millimeters travel ranges are required in semiconductor manufacturing equipments and nano-measuring instruments. Displacement Amplifier mechanisms constructed with flexure hinges have been employed in the state-of-the-art 'XY micro-stages' for getting a stage travel range longer than the actuator stroke [4]. Second part of this paper presents a new XY planar scanning mechanism with a long travel range up to 5 mm in both X- and Y-directions developed using 1dof displacement amplifier. Parametric FEA analysis is performed to determine the maximum travel range of the mechanism. Experimental validation of the results showed 11% error with FEA simulations.

The remainder of this paper is organized as follows. Literature review is presented in section II, In Section III, parametric modeling of flexural joints with details of FEA simulations is presented, Results of FEA analysis are discussed in section IV. Section V presents development and finite element analysis of 'XY planar scanning mechanism' in two steps. Section VI compares experimental investigations with FEA outputs and conclusions are presented in Section VII.

II. Literature Review

Dongwoo Kang and Daegab Gweon [1] derived and verified the stiffness equations for cartwheel flexure hinge. The cartwheel flexure hinge was evaluated in terms of the motion range, stiffness and stiffness ratio and compared with the conventional right circular hinge to confirm the appropriateness of the cartwheel flexure hinge as a large displacement flexure joint. Brian P. Trease [2] presented new designs for revolute and a translational compliant joint. Parametric computer models were used to verify their superior stiffness properties and joint range of motion by using FEA. Both of these joints have advantages over existing flexures in the qualities of a large range of motion, minimal "axis drift," increased off-axis stiffness, and a reduced stress-concentrations. Y. Tiana and B. Shirinzadeh [3] investigated that new filleted V-shaped flexure hinges can provide both higher and lower stiffnesses than circular flexure hinges based on investigation of the influences of the geometric parameters on the characteristics of the flexure hinges. The accuracy of motion was also derived for optimized geometric design.

Yuki Shimizu and Yuxin Peng [4] developed a stage motion mechanism for an XY micro-stage employing leaf spring flexure, which has a compact size of 24 mm × 24 mm × 5 mm and a millimeter-order travel range. Q. Meng and Y. Li [5] proposed three empirical stiffness equations for a wide range of t/l ratios and large deformation for corner-filleted flexure hinges with fillet radius 0.1l based on FEA results. Comparisons made with the existing stiffness equations indicated that the proposed empirical stiffness equations enlarge the range of rate of thickness and ensure the accuracy under large deformation. H. Esteki and A. Shahidi [6] studied five different profiles with effective design parameters such as accuracy, flexibility and the maximum stress in the hinge to introduce the best profile of the hinge based on the application. Different profiles used in flexure hinges are compared and the behavior of accuracy is studied with deflection variation using FEM.

Lins S. and Erbe T. [7] in their work compared asymmetric notch contours regarding a reduced shift of rotational to conventional contours and the potential of simple asymmetric flexure hinges to minimize the shift of rotational axis was established. A ratio of fillet radius to notch length of r/l = 0.1 is generalized for symmetric rectangular contours with optimal corner fillets. Nicolae Lobontiu [8] derived the closed form compliance equations for displacement amplification, input stiffness and output stiffness calculations of single axis amplifying compliant mechanisms that contain symmetric corner-filleted or circular hinges. Parametric study of the mechanism performance based on the mathematical model developed was performed. Eskandari and P. R. Ouyang [9] presented design analysis and optimization for, a new XY planar motion compliant mechanical displacement amplifier (CMDA) based on the design of a symmetric five-bar compliant amplifier. Guimin Chen [10] presents a general closed-form compliance solution for flexure hinges in terms of elliptical arc flexure hinge which brings circular, right-circular, and elliptical profiles together under one set of equations.

III. Parametric Modelling and FEA Analysis of Flexural Joints

Single axis planar flexures derive the motion in desired direction through bending about the axis perpendicular to the plane of motion which usually passes through the center of flexure and lies in the cross section of minimum thickness. These flexures need to be more compliant about the bending direction and ideally rigid about all other axes. With a few exceptions, the generic loading for a single-axis, constant-width flexure hinge is visualized in Fig.1. This loading

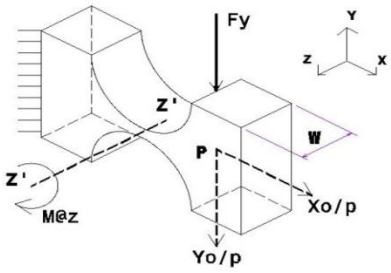
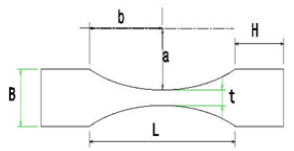
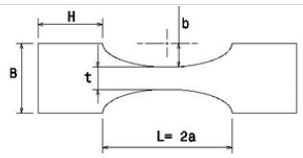
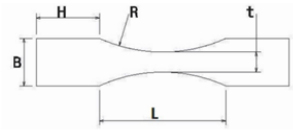
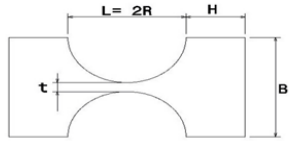
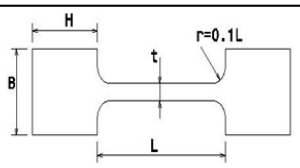
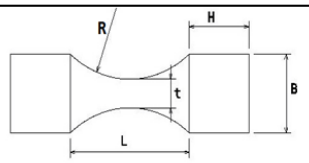


Fig.1 Generic loading conditions of a typical flexural joint

conditions are based on approximation of hinges as a cantilever beam and the same are used in FEA based parametric analysis to evaluate the influence of geometric parameters on performance of flexural joints.

The definitions of the geometric parameters of various flexures under consideration are given in Table I.

TABLE I
RANGE OF GEOMETRIC PARAMETERS DEFINING FLEXURES FOR PARAMETRIC ANALYSIS

 Elliptical arc	 Elliptical
a=minor axis radius, b=major axis radius Range:- a= 2,4 & 6mm, b= 8, 10 & 12mm	a=minor axis radius, b=major axis radius Range:- b= 8, 10 & 12mm
 Circular arc	 Right Circcular
R=arc radius Range:- R= 8, 10 & 12mm	R=arc radius
 Leaf	 Multi axis
r = fillet radius Range:- r=0.1mm constant	R=arc radius Range:- R= 8, 10 & 12mm
L= Hinge length, t= minimun hinge thickness, W= width of flexure, H= length of rigid end, B= total height of hinge. **Range:-** L= 5,10,& 15mm, t= 0.5, 0.75, 1.0 & 1.25mm, H & W= 5mm constant, B= varying with L & t.	

Out of all the listed parameters minimum hinge thickness 't' and hinge length 'L' have significant influence on deformation of flexures approximated as cantilever beams [3], [5], [6].

Accordingly the performance characteristics like lateral deflection, lateral stiffness, stress and range of motion of the hinges are derived for various combinations of 't' and 'L' parameters. Most of the planar flexural joints operate in the range of 5mm to 15mm for hinge length L and 0.5 to 1.25 mm of minimum hinge thickness which ultimately defines the range of L and t for analysis. L is varied in the steps of 5mm and t in the steps of 0.25 mm. Range of L and t along with the geometric constraints of individual joints defines the range of other associated parameters which are listed in Table I. It should be noted that width W and length H were kept constant at 5mm throughout the analysis for all the parametric models while length B varies with minimum hinge thickness t and hinge length L.

A. FEA Data

The 'ANSYS® Workbench™ 14.5' finite element software was used to run all finite element static-load simulations on several parametric models of all the flexures. For each flexural joint 10 node tetrahedron element with element size of 0.35mm was utilized. Mesh refinement technique yielded element size of 0.15mm at hinge faces generating 61224 numbers of total elements and 95963 numbers of nodes for the first set of parametric test model. In order to improve the computational accuracy, the mapping mesh method is utilized. The finite element model of the flexure hinge is shown in Fig. 2. It was considered that the flexures are made of structural

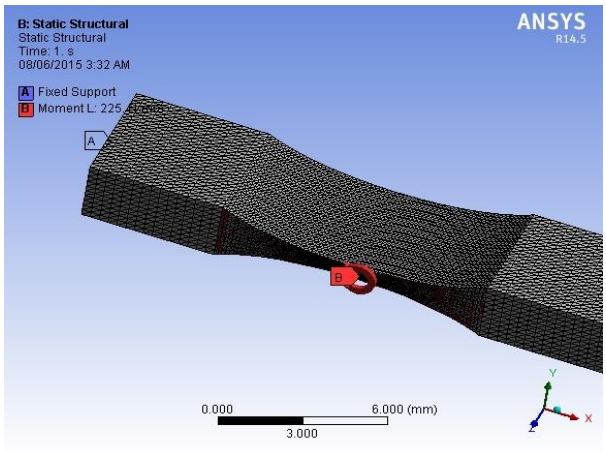


Fig. 2 Finite element model of flexural hinge

steel with a Young's modulus of 200 Gpa and a Poisson's ratio of 0.3 and the yield strength of 250 MPa.

In each static analysis, the boundary conditions are chosen as: the left block is fixed for all degrees of freedom, and the moment 'M@z' is applied about the bending axis of hinge i.e. about the 'z' axis. The values of moments are calculated considering the vertical forces 'Fy' sequentially applied on the upper right edge of hinge as shown in Fig.1. This is done to compensate stress singularities resulting from application of edge loads Fy. Vertical force Fy was varied from 5N to 20 N
in the steps of 5N and the corresponding vertical displacements Yo/p and parasitic axial displacements Xo/p were read at critical point p on central plane of hinges as depicted in Fig.1.This allowed the calculation of lateral deflection, lateral stiffness, and maximum stresses induced in hinges.

Over the wide range of number of geometric parameters considered along with the range in applied force permitted the evaluation of total of 432 parametric models of elliptical arc hinge, 96 of elliptical hinge, 144 each of circular arc and multi-axis hinge, 48 each of right circular and leaf hinge.

The complete set of the results for finite element analysis is demonstrated in terms of catalog of parametric graphs. Of all the studies, few significant ones pertaining to geometries of hinge that produces maximum deflection for 5N of lateral force are included here. Readers interested in referring complete set of charts may contact the authors.

IV. FEA RESULTS AND DISCUSSIONS

It should be noted that in the considered range of all the parameters maximum deflection for elliptical arc hinge occurs at b = 2mm and a = 12mm, While that of elliptical hinge maximum deflection occurs at b = 2mm and maximum deflection of circular arc and multi-axis occurs at R=12mm. Results corresponding to the hinges defined by these parameters are utilized for comparison in all the following graphs. For all the flexures lateral deflection (Yo/p) increases with hinge length (L) (not shown here) and decreases with increase in minimum hinge thickness. Maximum lateral deflection occurs at L=15 mm & t=0.5 mm for all the hinges in the given range. So comparison of

all the joints presented in this paper has been limited at maximum deflection i.e. at L=15mm.

A. Deflection

Fig.3 interprets that maximum deflection occurs in multiaxis followed by leaf, elliptical arc, elliptical, circular arc and right circular. As minimum hinge thickness increases deflection values for all the hinges decreases and approaches near to each other. Multiaxis, leaf and elliptical arc joints are more compliant sensitive to decrease in hinge thickness t.

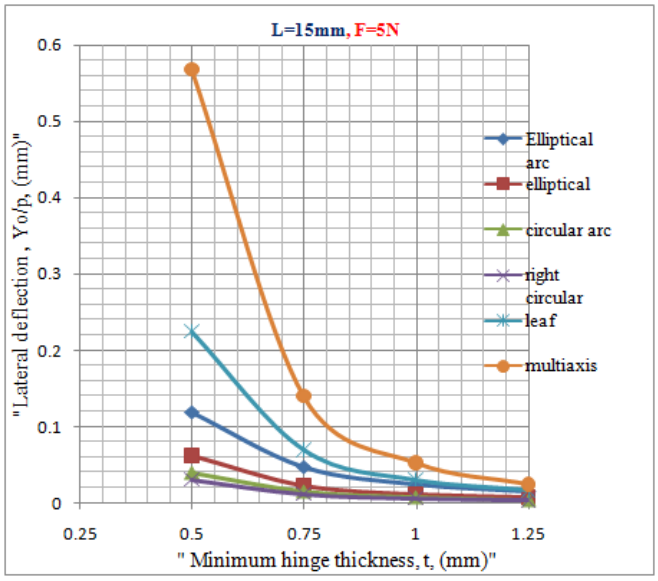


Fig.3 Lateral deflection over range of minimum hinge thickness

B. Lateral Stiffness

Fig. 4 shows that when the small thickness is used, the motion stiffness of all flexures has values in the similar range. As the larger thickness is used, the right circular and circular arc become more sensitive to increase in thickness. Least stiffness in lateral direction is observed for Multi-axis, leaf and elliptical arc as well they are less sensitive in stiffness over the entire range of t.

C. Maximum Stress

It is observed in Fig. 5 that the multiaxis and leaf have highest stress concentrations and are increasingly sensitive to decrease in thickness. Right circular and circular joints have least stresses induced and are almost in the same range. It should be emphasized from Fig. 3 to Fig. 5 that elliptical joint provides in between solution for stress, deflection, and

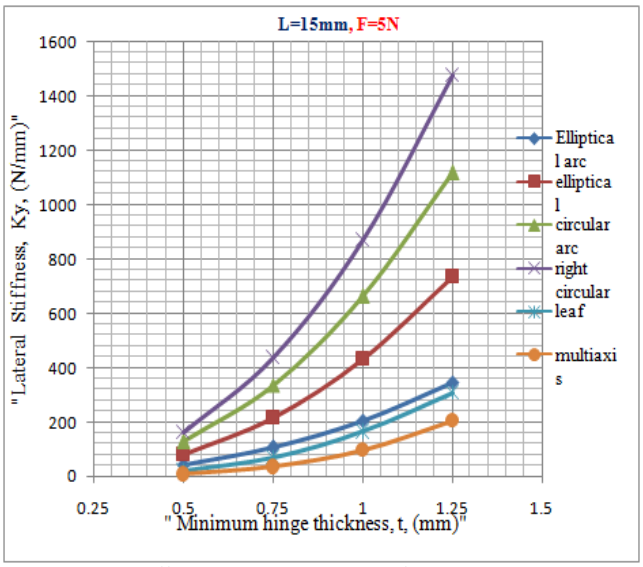


Fig.4 Lateral stiffness variation over range of minimum hinge thickness

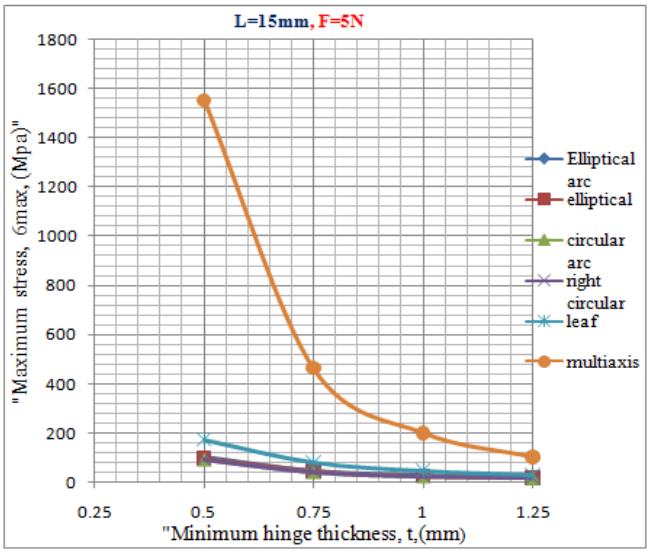


Fig.5 Maximum von-misses stress variation with minimum hinge thickness

stiffness in lateral direction. Location of maximum stress for all the joints is in the area of minimum hinge thickness as that of one shown for elliptical joint in Fig. 6.

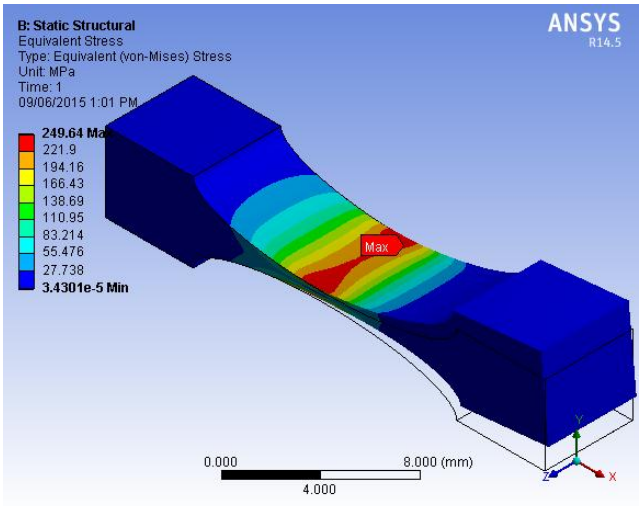


Fig.6 Location of maximum stresses induced in elliptical joint

These graphs also serve as a quantitative design tool for hinges working in the considered range of L and t. For e.g. when attempting to meet a given lateral stiffness, the given value corresponds to a horizontal line cut through the graph. Any point located below this line indicates the feasible design space left to meet any other design specifications.

D. Range of Motion

The range of motion is determined as the amount of deflection which causes yielding in the material [2].The operation range of all the joints is evaluated and compared within the specified range in L and t parameters. These results are given in Fig. 7. Elliptical arc flexure provides much larger

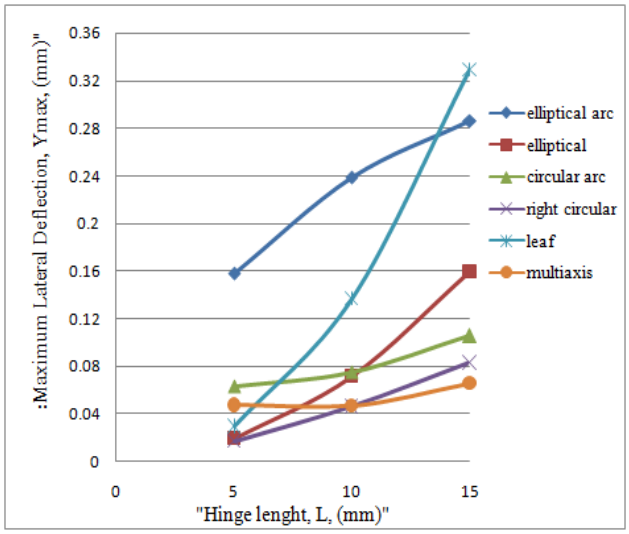


Fig.7 Operation range of different flexures

operation range over the entire span of hinge length L. Operation range of leaf increases with L and is more sensitive to increase in L compared to multiaxis, right circular and circular arc hinges. Operating range of elliptical hinge is less than elliptical arc and leaf but is more sensitive to increase in L compared to multi axis, right circular and circular arc. Hence it is evident that elliptical hinge provides the midway response.

As the maximum deflection for all the Hinges occur at minimum value of hinge thickness i.e. at t=0.5mm hence, the
maximum deflection values at t=0.5mm defines the operation range over the entire range of parameter t. Graph in Fig. 7 indicates the maximum lateral deflections for entire range in t at hinge lengths of L=5,10 and15 mm respectively. For e.g. Leaf will give maximum deflection of 0.3291 mm in the overall working range of t at hinge length of L=15mm while it will give maximum deflection of 0.13708 mm in the overall working range of t at hinge length of L=10mm. Also direct lateral load bearing capacities of flexural hinges in the specified range of parameters for their operating range are shown in Fig. 8.

E. Guiding Accuracy

Guiding accuracy of flexures is the measure of deviation from the axis of straight-line motion. While generating translation in motion direction flexures bend in rotation about the pivot axis which in fact also shifts due to deformation along the entire length of the flexible component. This is undesirable for the purpose of generating precise motion. Guiding accuracy of all the flexures was derived by plotting

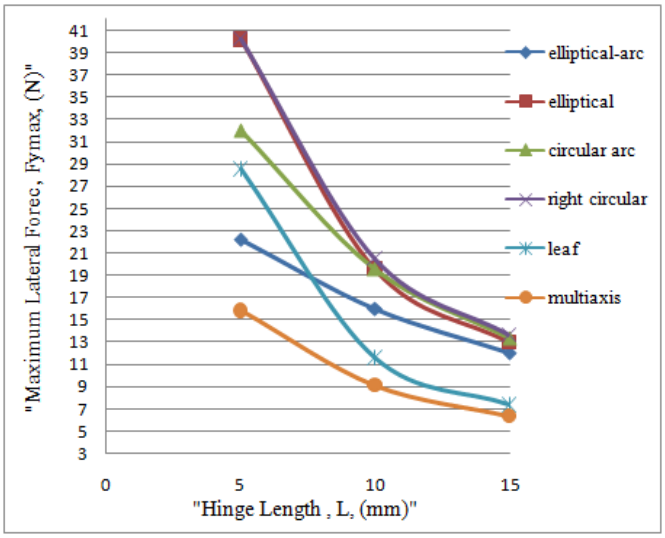


Fig.8 Lateral load bearing capacity of different flexures

parasitic displacement in axial direction (Xo/p) with respect to displacement in lateral direction (Yo/p) [7].

Fig. 9 Reflects that guiding accuracy of all the flexures decreases with increase in the lateral deflection. This is due to increase in parasitic motion along axial direction with increase in lateral deflection. It is noted that elliptical arc has the best

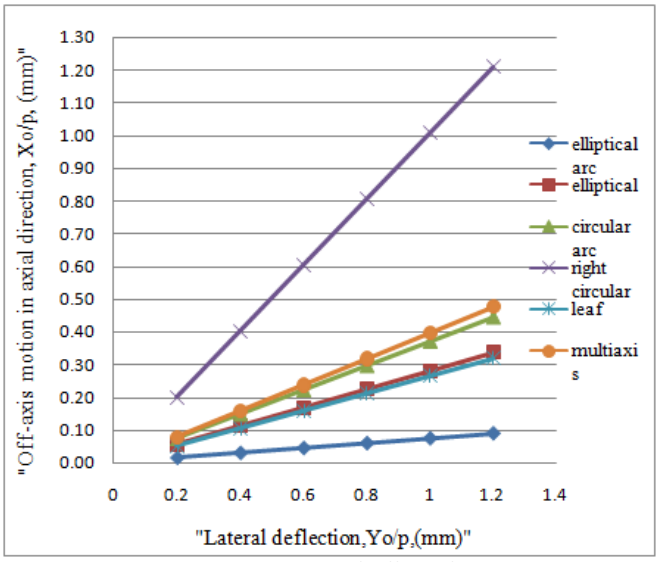


Fig.9 Guiding accuracy of different flexures

guiding accuracy which is also least sensitive to increase in lateral deflection while right circular has the least guiding accuracy which is highly sensitive to increase in lateral deflection. Elliptical hinge provides accuracy in between all the hinges.

Based on the above presented guidelines of FEA analysis choice preference graph is proposed here in Fig.10 which may guide the optimum selection of flexural joints based on the application and the requirement the designer is confronted with. It should be noted that the following graph in Fig.10 represents choice preference over entire range in t at hinge length of L=15mm only. Preference graphs over the

entire range of L and t are also derived but not presented in this paper. Numbers on Y axis of graph represents choice preferences for various design points. For e.g. If design demands the good guiding accuracy your first preference would be the choice of elliptical arc joint, second choice

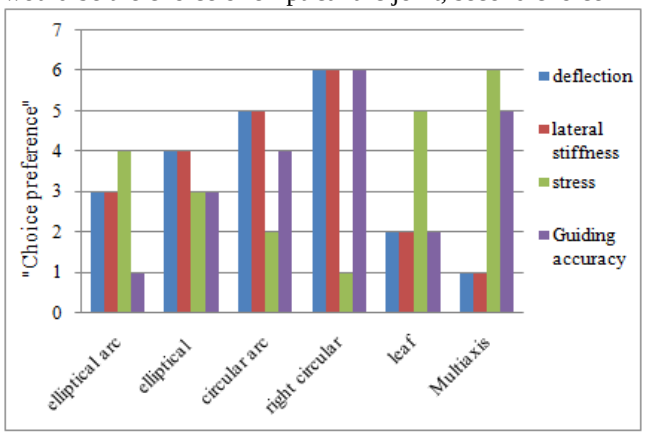


Fig.10 Choice preference chart for various performance parameters

would be leaf and so on. Fig. 10 reveals that elliptical flexural joint exhibits conservative results in terms of all four parameters considered together.

V. Development and FEA Analysis of Planar XY Mechanism

F. Step I

Amplifier mechanism with circular flexure hinge shown in Fig. 11 addressed by Nicolae Lobontiu and Ephrahim Garcia [8] in their work was used as a building block of XY planar mechanism here.

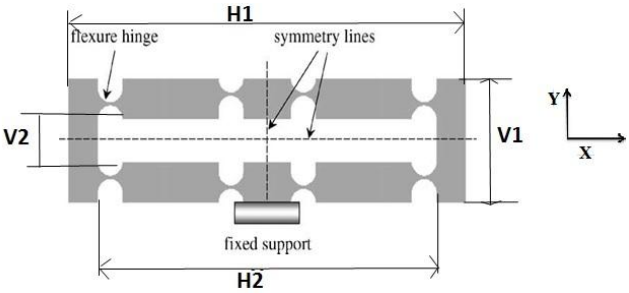


Fig.11 Single axis displacement amplifier

All the Circular flexure hinges were replaced by elliptical flexure hinges with hinge length L=10 mm, minimum hinge thickness t=1mm and semi-minor axis radius b=0.2mm. Mechanism is designed for 5mm range by parametric FEA analysis. The key structural parameters H1, H2, V1 and V2 considered for parametric simulations are varied in selected ranges simultaneously to reach the desired travel range in Y direction.

For FEA analysis the mechanism was modeled with 10 node tetrahedron elements with element size of 2mm for rigid arms and element size of 0.5 mm for hinge faces resulting in total of 17024 nodes and 8731 number of elements for the first test model. The material selected for mechanism is Aluminum Alloy having Yield strength of 270 Mpa, Poisson's ratio of 0.33,Young's modulus of 71000 Mpa and density of 2700 Kg/m^3.

Middle base of lower rigid arm is fixed in all degrees of freedom and axial force of 25 N was applied on each of the vertical arms and corresponding displacements of middle face of upper rigid arm are read in X and Y direction as shown in Fig 12.

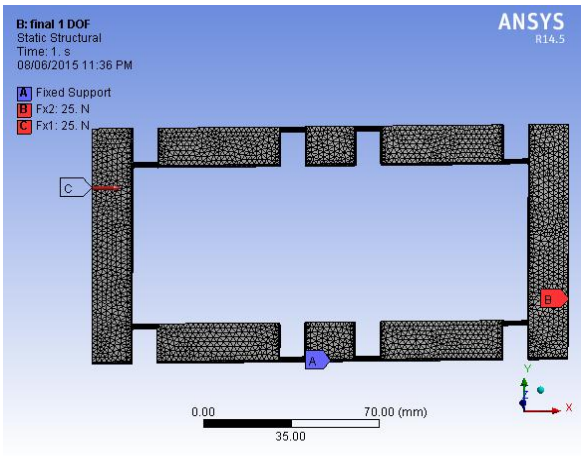


Fig.12 Analysis settings of 1DOF mechanism

The results giving 5mm Y direction displacement with an amplification ratio of 4.02 are depicted in Fig.13 and Fig.14.

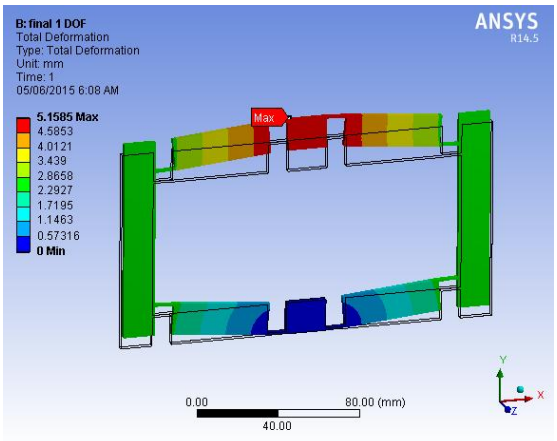


Fig.13 Maximum displacement in Motion direction.

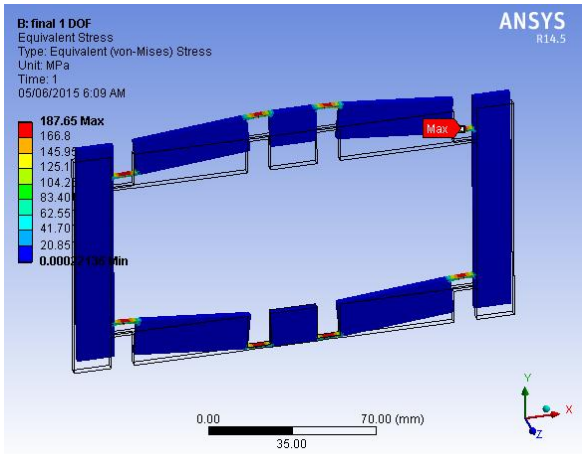


Fig.14 Maximum equivalent Von-Misses stress and its location

G. Step II

Goal in second step was to exploit the Geometry of 1 DOF displacement amplifier producing 5mm deflection in Y direction to propose 2 DOF 'XY planar mechanism' capable of giving the same range of motion in X and Y direction each. The symmetric arrangement of the proposed

mechanism is shown in Fig. 15 in which two pairs of the amplifier legs are
assembled at right angles by means of two arms connecting the motion stage .when the stage plate is driven by the driving amplifier 'A' in X-direction by means of PZT actuator, the vertical arm can guide the stage in Y direction while the Leg

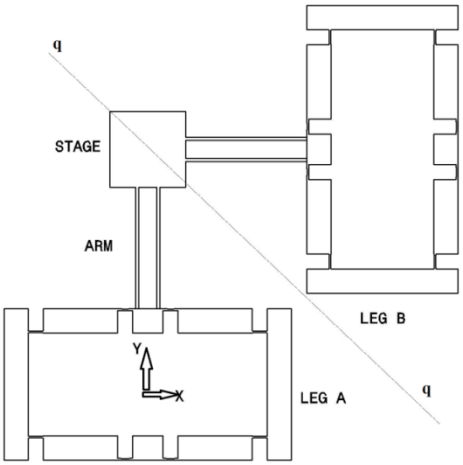


Fig.15 Geometry of XY planar mechanism

B does that in the X direction motion. Please note that both legs A and B are each constrained in all degree of freedoms from the base.

FEA simulations were carried out to estimate the range and force for mechanism. The mechanism was modelled with the same meshing data used in step I except the number of nodes and number or elements increased to **216432** and **120360** respectively. The calculated displacement and the stress when the stage was moved in Y-direction as shown in Fig. 16 are presented in Fig. 17 and Fig. 18.

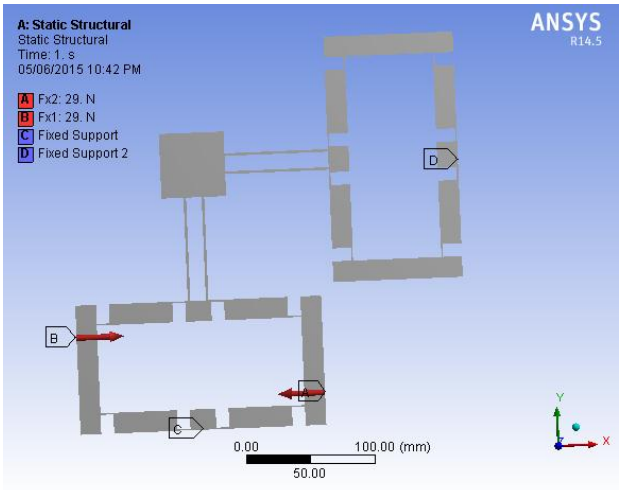


Fig.16 Analysis settings of XY planar mechanism

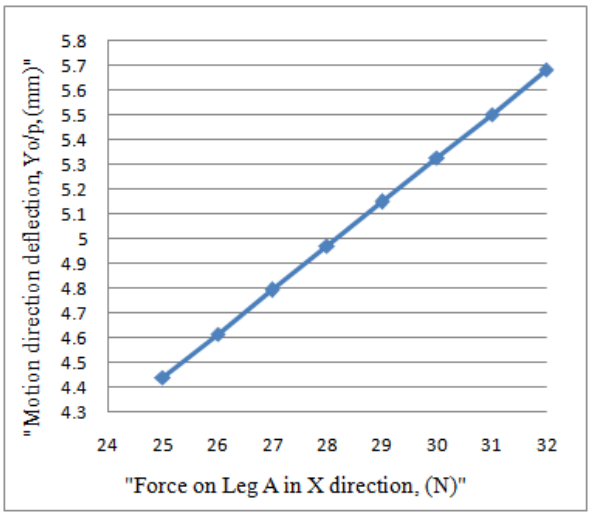


Fig.17 Motion direction Vs Force

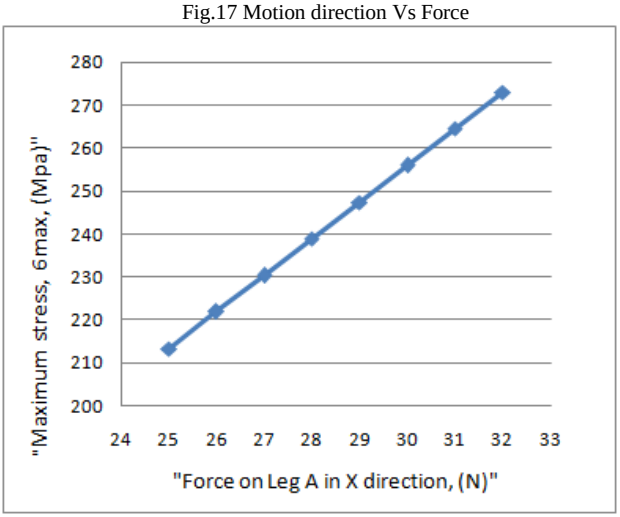


Fig.18 Von-Misses stress Vs Force

It is observed from parametric results that about 5.2 mm displacement of motion stage in Y direction is achieved for the force of 29 N applied at the amplifier ends and the mechanism has a operation range of 5.61mm and capacity of 31 N of force at which stress induced is 264.5 Mpa. Parasitic displacement of 0.98 mm in X direction is also associated with 5mm motion direction displacement. Corresponding FEA results for 5mm deflection are shown in Fig.19 and Fig. 20. It should be noted that almost the same results were obtained by the simulation on the X-directional stage motion.

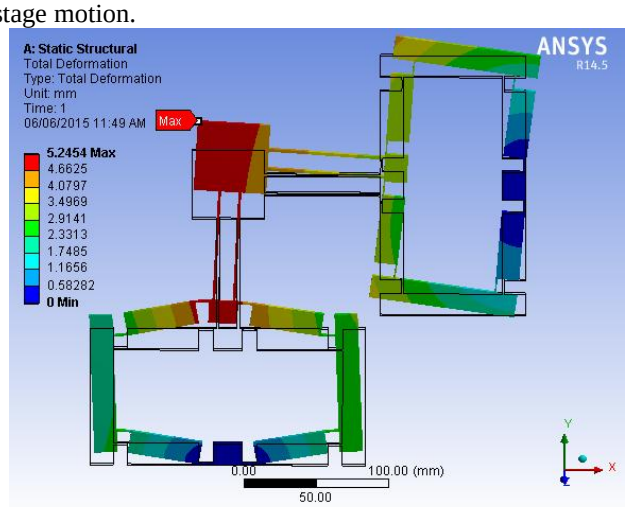


Fig.19. 5mm displacement of XY mechanism in Y direction

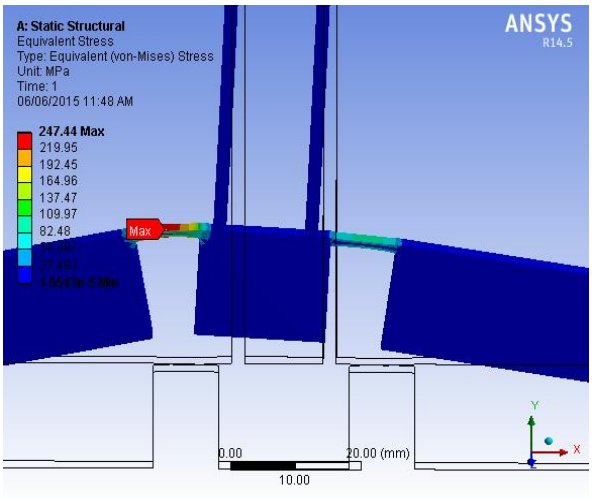


Fig.20 Maximum stress location at 5mm displacement

VI. Experimental Investigations

Monolithic XY mechanism made of Aluminium alloy Al-6061is fabricated using the wire-cut EDM (Electro Discharge Machining) in one set-up. The Experimental setup for testing the mechanism is shown in Fig.21.

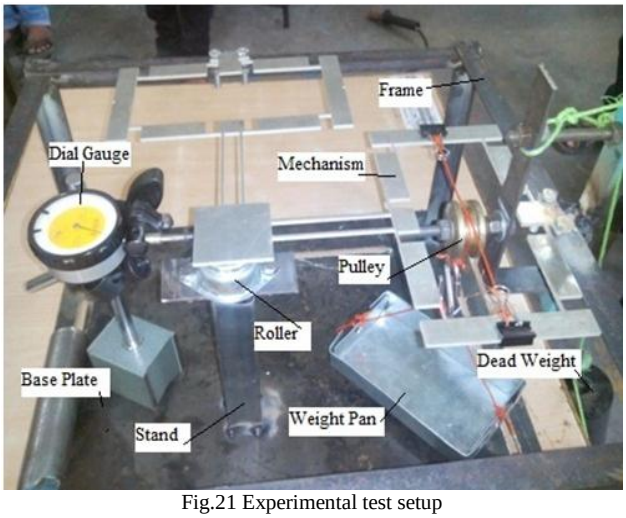


Fig.21 Experimental test setup

Perfect alignment of mechanism in single flat horizontal plane was ensured using spirit-level instrument while mounting the Mechanism on symmetric frame. Motion stage was supported by rolling bearing surface which also prevented out of plane deformation of motion stage. One leg was loaded axially using String, pan and pulley arrangement as shown and displacements of the stage in motion direction and off-axis direction were read on dial gauge and vernier respectively. The mechanism was loaded with six different loads in each of the three sets of testing.

The experimental results confirm the large range motion of
XY stage with 11 % error in motion direction deflection and 15 % error in off-axis motion, see Fig. 22 and Fig. 23. The deviations in the experiment from the FEA results may arise from manufacturing tolerances of the stage and misalignment errors of mechanism and errors induced due to manual readings. Use of actuators for precise loading and sensitive electronic equipments in reading measurements may further improve the accuracy of readings.

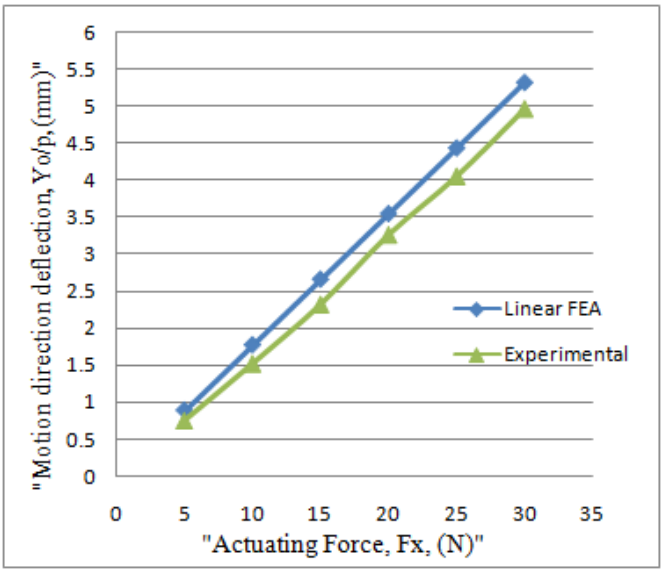


Fig. 22 Motion direction deflection comparison

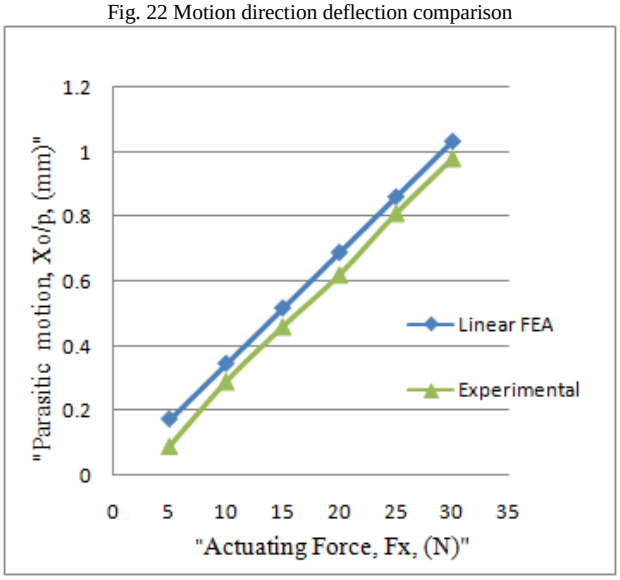


Fig.23 parasitic motion comparison

VII. Conclusions

This work evaluates six types of flexural joints in terms of stiffness, stress, deflection for the influence of geometric parameters on the performance of hinges. Further the operating range of each joint is stated within the considered parametric range in hinge length and minimum hinge thickness. Guiding accuracies defining the accuracy of motion are also derived. A catalog of design charts based on the parametric modeling using FEA tool ANSYS® Workbench™ 14.5, characterizing the joints are presented, allowing for rapid sizing of the joints for custom performance.

XY planar scanning mechanism employing elliptical flexure is designed to have a long travel range up to 5 mm in both X- and Y-directions, while having a size of 300mm × 300mm × 3 mm. In the proposed stage system, the stage would be driven by PZT (Piezo-Electric Amplifier) at amplifier legs considering the driving force in the range of 20 to 35 N. The experimental measurements validate the large travel range of the mechanism. Errors in motion

direction displacement and off axis displacement are near to 11 % and 15 % respectively.

Analysis of flexural joints providing actual numbers would require some normalization of parameters and is an interest to be considered in future research. Ongoing work includes dynamic analysis to determine natural frequency and mode shapes of the XY planar scanning mechanism presented in this paper.

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