

RESEARCH ARTICLE

OPEN ACCESS

IMPACT OF GEOMETRICAL PARAMETERS ON FATIGUE LIFE CYCLE OF COMPLIANT MECHANISM

Sandesh Solepatil^{*1}, Narendra Deore²

¹Department of Mechanical Engineering, Technology and Engineering, D Y Patil International University, Akurdi, Pune, India.

²Department of Mechanical Engineering, Pimpri Chinchwad College of Engineering, Akurdi, Pune, India.

¹<http://orcid.org/0000-0002-8005-4994>, ²<http://orcid.org/0000-0003-4710-3798>

E-mail: sandeshpatil310@gmail.com, narendra.deore@pccoepune.org

ARTICLE INFO

Article History

Received on: March 05, 2026

Reviewed on: April 05, 2026

Accepted on: July 01, 2026

Published on: July 31, 2026

Keywords:

Compliant Mechanism, Fatigue Life Cycle, Damage, Safety Factor, Square Motion Stage With Curve Beam, Circular Motion Stage With Curve Beam.

ABSTRACT

Compliant mechanisms, which derive motion through the elastic deformation of flexible members, offer distinct advantages over traditional mechanisms, including reduced part count, enhanced reliability, and smooth, frictionless operation. However, their fatigue life is a critical design concern due to the continuous cyclic stresses endured during operation. The fatigue performance of these mechanisms is strongly influenced by a combination of material properties, geometrical design, and loading conditions. Among these, geometrical parameters—particularly the radius of curvature—play a crucial role in determining stress concentration and overall durability. This study focuses on analyzing the impact of the radius of curvature on the fatigue life of compliant mechanisms using Finite Element Analysis (FEA). Four different configurations of square and circular motion stages with varying curvature radii were evaluated under constant loading conditions. The analysis considered key parameters such as damage factor, safety factor, and fatigue life cycles. Results indicate that increasing the radius of curvature significantly improves fatigue resistance by distributing stress more evenly and reducing critical stress concentrations. The findings suggest that appropriate geometric optimization, especially in curved beam regions, can substantially enhance the life span and safety of compliant mechanisms without compromising their function. These insights are particularly beneficial for the design of mechanisms in applications requiring high precision and longevity, such as biomedical devices, micro-positioning systems, and additive manufacturing equipment. This research contributes valuable design guidelines for enhancing the fatigue performance and operational reliability of compliant mechanisms.



Copyright ©2026 by authors and Galileo Institute of Technology and Education of the Amazon (ITEGAM). This work is licensed under the Creative Commons Attribution International License (CC BY 4.0).

I. INTRODUCTION

Fatigue refers to the progressive and localized structural damage that occurs when a material is subjected to cyclic loading. Over time, repeated deformation induces microscopic defects and cracks that grow, ultimately leading to failure. In compliant mechanisms, the primary source of fatigue arises from the cyclic bending, stretching, or twisting of flexural elements, often concentrated in areas of high stress or strain. The prediction and mitigation of fatigue failure are therefore critical for extending the service life of these mechanisms, particularly in applications requiring millions or billions of load cycles. The fatigue life of a compliant mechanism depends on several factors, including material properties, geometric design, stress distribution, and environmental conditions. Materials such as metals, polymers, and composites exhibit varying fatigue characteristics, influenced by their elastic limits, crack propagation resistance, and thermal stability. The geometry of the mechanism, particularly the radii of curved segments, fillets, and transitions between flexible and rigid regions, plays a crucial role in stress concentration and fatigue initiation. Compliant mechanisms have revolutionized modern engineering by leveraging the elastic deformation of materials to achieve motion and force transmission without the need for traditional joints or hinges. This innovative design approach leads to mechanisms that are often lighter, more compact, and capable of smoother and more precise movements compared to their rigid-body counterparts [1].

Applications of compliant mechanisms span a wide range of fields, including aerospace, robotics, biomedical devices, and microelectromechanical systems (MEMS) [2]. Despite their numerous advantages, compliant mechanisms are subject to continuous and repetitive deformation, which exposes them to fatigue. Fatigue, defined as the progressive and localized structural damage that occurs when a material is subjected to cyclic loading, is a critical concern for the longevity and reliability of these mechanisms [3-5]. Understanding and predicting the fatigue life of compliant mechanisms is essential to prevent unexpected failures and to ensure their safe and effective operation over time. Fatigue analysis involves studying how materials respond to cyclic loading and identifying the conditions under which they are likely to fail. For compliant mechanisms, this analysis is particularly important due to their reliance on material flexibility and continuous deformation [6]. Unlike traditional mechanisms, where fatigue may primarily occur at discrete joints or moving parts, compliant mechanisms experience distributed stress and strain throughout their structure. This makes fatigue analysis more complex and necessitates a detailed understanding of material behavior, stress distribution, and crack propagation [7-11]. Compliant mechanisms can be broadly classified into two categories based on their primary mode of deformation:

- **Linear Compliant Mechanisms:** These mechanisms achieve straight-line motion through bending or stretching. Examples include flexural beams and bistable springs.
- **Rotational Compliant Mechanisms:** These mechanisms enable rotational motion through flexural pivots or torsion. Examples include flexural hinges and rotational springs [12-14].

Both types of mechanisms offer unique advantages and challenges in terms of fatigue analysis. Linear mechanisms often encounter issues related to bending fatigue, while rotational mechanisms may experience torsional fatigue. Additionally, the geometry and material properties of compliant mechanisms play significant roles in determining their fatigue behavior. Material properties, geometric properties, and the combination of the components greatly influence the compliant mechanism and its performance. Stiffness, bending stress, beam deflection, and degree of freedom govern the Compliant Mechanism's various series and parallel beam arrangements. In contrast, payload capacity and resonance frequency decide the compliant mechanism's actuation. Many researchers have worked on the range of motion with different arrangements of beams, degrees of freedom, and actuation positions to get micro and nano ranges of motion with compliant mechanisms. Translational and rotational positioning systems are mainly used to achieve linear and rotational motion with various combinations of beams and hinges. A translational positioning mechanism provides a range of motion in X and Y directions. A serial-parallel kinematic chain is used in X-Y-compliant mechanisms to provide high load capacity, low inertia, and high resonant frequency [15] by maintaining accuracy, precision, and frictionless motion.

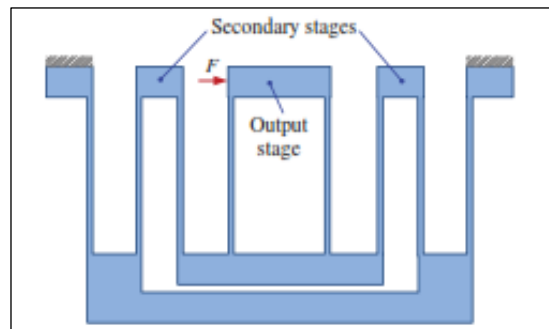


Figure 1: Translational positioning System.
Source: (15).

A Schematic of the parallel beam position is shown in Figure. 1 in which the translational motion stage is used to achieve single-directional displacement by applying actuating force on the vertical parallel beam. Figure. 2 demonstrates the combination of parallel beams stacked in two modules, wherein the second module rests on the first module, in laser sintering application for desired compliant values [15]. In this mechanism, the output motion is guided by a double parallel flexural manipulator used in combination located at the corners of the module. The second module is mounted orthogonally on the top of the first module to get a parallel kinematic and stacked structure. This modular design allows the cost reduction in both manufacturing and maintenance. When driven by the Voice coil actuator, four double parallel flexural manipulators on the top and two double parallel flexural manipulators at the bottom guide the motion along X and similarly for the motion along y direction. Hence, a decoupled two-dimensional translation is achieved by the XY stage to obtain the travel range of 10 mm [16].

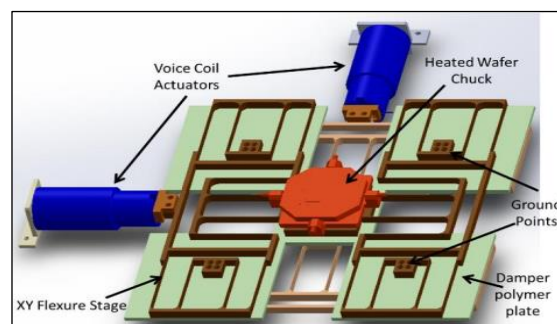


Figure 2: Selective laser sintering system.
Source: (16).

Rotational Movement in a compliant mechanism is achieved with a combination of beams, hinges, and radial flexures, as shown in Figure. 3. The relative motion between hinges and flexures leads to the desired compliant values due to actuating forces applied in compliant mechanisms [17] Radial flexure needs to be developed with a larger beam length of up to 100 mm, a larger outer radius of up to 10 mm, and a smaller thickness of up to 0.3 mm to achieve the desired rotational range [18].

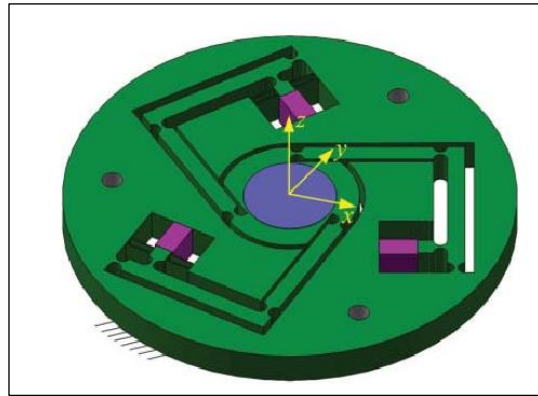


Figure 3: Rotational positioning System
Source: (16).

However, in practice, the compactness, restriction, manufacturing tolerance, and minimum stiffness criterion limit these geometrical parameters [19].

Challenges in Fatigue Analysis

Several challenges arise when analyzing the fatigue of compliant mechanisms. The primary challenges include:

1. **Material Selection:** The choice of materials significantly influences the fatigue life of compliant mechanisms. Materials commonly used in these mechanisms include metals, polymers, and composites, each with distinct fatigue characteristics [20].
2. **Stress Concentration:** The design of compliant mechanisms must account for stress concentration points, such as sharp corners or notches, which can significantly reduce fatigue life.
3. **Crack Initiation and Propagation:** Understanding where and how cracks initiate and propagate under cyclic loading is crucial for predicting the lifespan of compliant mechanisms.

Combined linear and rotational compliant mechanism is designed for biomedical applications requiring the linear range of motion from 2 mm to 4 mm in X and Y directions and rotational motion from 2° to 4°. These compliance values are achieved with various beams, hinges, and notches arrangements [21-25].

II. METHODOLOGY

Compliant mechanism is analyzed for actuating force of 80N which is the maximum load carrying capacity of PLA material. 4 Different configurations of mechanisms are developed for analyzing the effect of geometrical parameters on fatigue life cycle of compliant mechanism as follows:

Mechanism 1- Square motion stage with curve beam of 32mm

Mechanism 2- Square motion stage with curve beam of 40mm

Mechanism 3- Circular motion stage with curve beam of 32mm

Mechanism 4- Circular motion stage with curve beam of 40mm

Mechanism 1 and 2 - Square motion stage with curve beam of 32 mm and 40 mm

Compliant mechanism 1 is developed with the dimensions of (Length *width *height) 300mm*300mm*15mm . Motion stage is of 50 mm. beam thickness of mechanism is 2mm. Overall dimensions of mechanism is shown in Figure. 4. In this mechanism force is apply on Y direction and displacement of mechanism will occur in X direction. Elements of mechanism are as shown in Figure. 5. Radius of curvature of beam is 32 mm as shown in Figure. 6 and Radius of curvature of beam is 40 mm as shown in Figure. 7 is used to analyze the impact of actuating force on fatigue life cycle with variation in radius of curvature of beam.

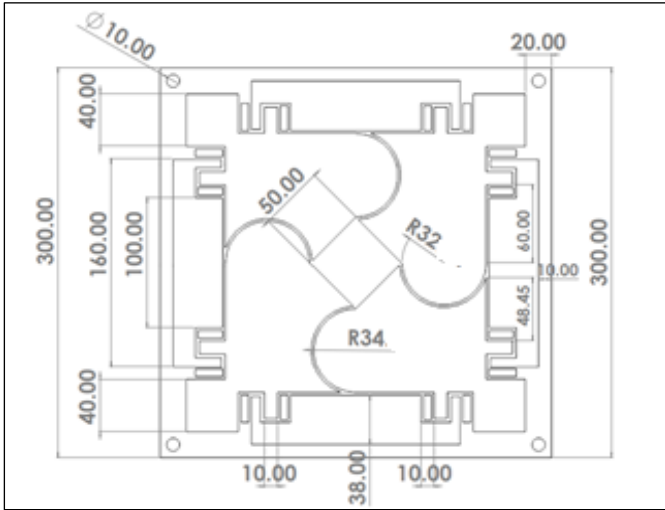


Figure 4: (A) Compliant mechanism with Square motion stage with beam radius 32mm (Mechanism 1).
Source: Authors, (2026).

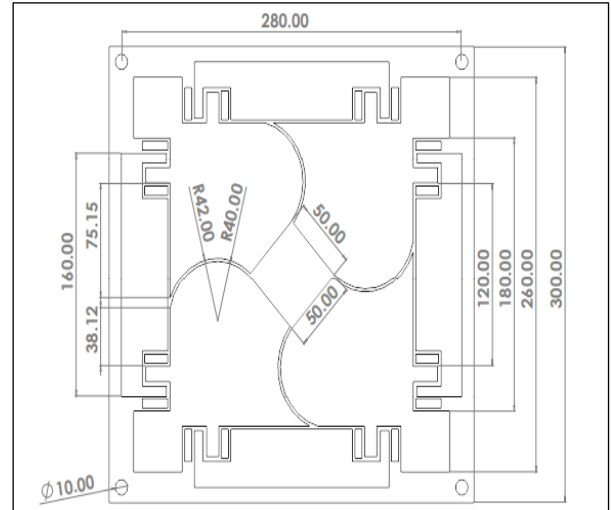


Figure 4 (B): Compliant mechanism with Square motion stage with beam radius 40mm (Mechanism 2).
Source: Authors, (2026).

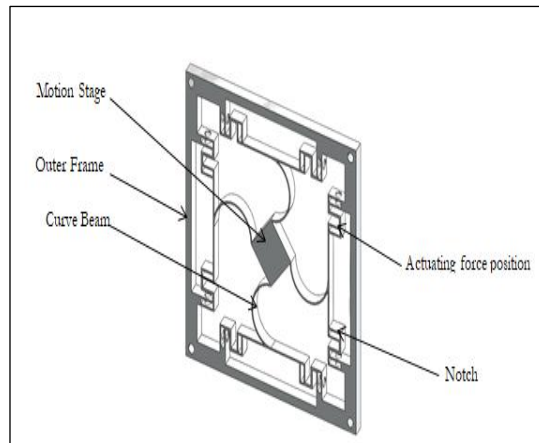


Figure 5: Elements of Compliant mechanism with Square motion stage.
Source: Authors, (2026).

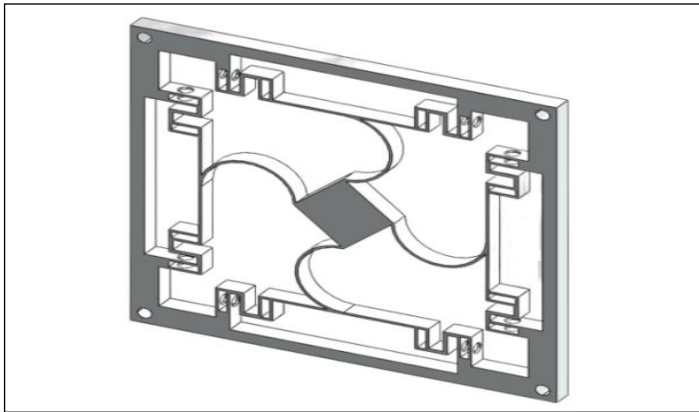


Figure 6: Square motion stage with curve beam radius 32 mm (Mechanism 1).
Source: Authors, (2026).

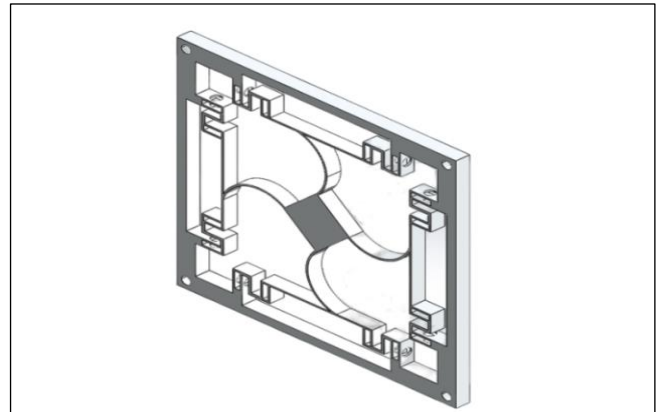


Figure 7: Square motion stage with curve beam radius 40 mm (Mechanism 2).
Source: Authors, (2026).

Mechanism 3 and 4- Circular stage with curve beam of 32 mm and 40 mm

Compliant mechanism is developed with the dimensions of (Length *width *height) 300mm*300mm*15mm . Motion stage is of 50mm, beam thickness of mechanism is 2mm, radius of curvature of beam is 32 mm. In this mechanism force is apply on Y direction and displacement of mechanism will occur in X direction. To evaluate fatigue life cycle actuating force of 80 N is used for mechanism. Radius of curvature of beam is 32 mm as shown in Figure. 8 and Radius of curvature of beam is 40 mm as shown in Figure. 9 is used to analyze the impact of actuating force on fatigue life cycle variation in radius of curvature of beam.

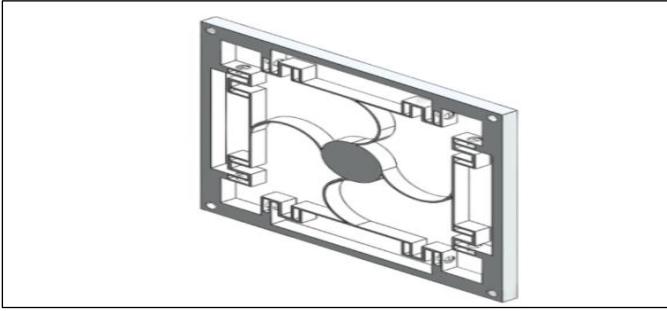


Figure 8: Circular motion stage with curve beam radius 32 mm (Mechanism 3).
Source: Authors, (2026).

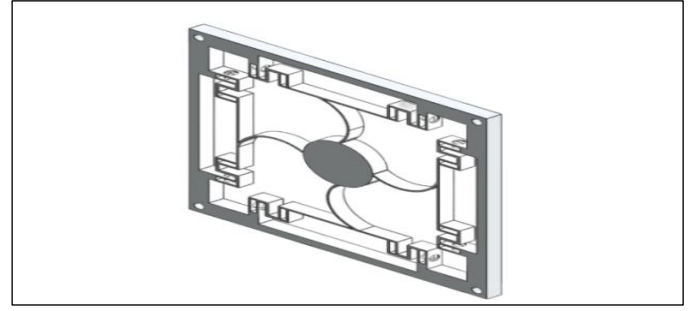


Figure 9: Circular motion stage with curve beam radius 40 mm (Mechanism 4).
Source: Authors, (2026).

Material

Polylactic Acid (PLA) is a biodegradable thermoplastic derived from renewable resources such as corn starch or sugarcane. It is widely used in 3D printing and various applications where its properties make it suitable for compliant mechanisms. When analyzing the fatigue behavior of compliant mechanisms made from PLA, several key material properties and considerations are relevant. Material properties of PLA are shown in Table 1. These values are taken from Tensile test of PLA Specimen with ASTM E8 standard.

Table 1: Material Properties of PLA.

Sr.No	Content	Values
1.	Young's Modulus	3986 Mpa
2.	Ultimate Tensile strength	59 Mpa
3.	Poisson's ratio	0.33
4.	Bulk Modulus	3907 Mpa
5.	Shear Modulus	1498 Mpa

Source: Authors, (2026).

Effect of geometrical parameters on fatigue life cycle of compliant mechanism

Fatigue analysis of a compliant mechanism involves evaluating the predicting the life of a component subjected to cyclic loading. The compliant mechanism is analyzed based on the stress-life (S-N) curve and the stress range at various points of the component. The material properties, loading conditions, and geometry of the mechanism are considered in the analysis. The fatigue life is estimated based on the number of cycles required to cause failure, taking into account the size and location of any cracks that may form. The results of the fatigue analysis are used to design components that can withstand the expected load cycles, ensuring that they will perform their intended function over their intended lifespan. This table 2 provides a comprehensive overview of the settings and parameters used in a fatigue analysis. The analysis is based on time, considers zero-based loading, and uses the Goodman mean stress theory. The fatigue strength factor is set to 1, indicating nominal strength is used. Results are displayed at the end of the analysis period, and the fatigue life is measured in cycles, with one cycle being equal to one actual cycle. The analysis type is stress-life, using the signed von-Mises stress component to evaluate fatigue life. Ansys 21 used for Numerical analysis of compliant mechanism.

Table 2: Input Parameters for Fatigue Analysis.

Object Name	Fatigue Tool
State	Solved
Domain	
Domain Type	Time
Materials	
Fatigue Strength Factor (Kf)	1.
Loading	
Type	Zero-Based
Scale Factor	1.
Definitin	
Display Time	End Time
Options	
Analysis Type	Stress Life
Mean Stress Theory	Goodman
Stress Component	Signed von-Mises
LifeUnits	
Units Name	cycles
1 cycle is equal to	1. cycles

Source: Authors, (2026).

III. RESULT AND DISCUSSION

Fatigue analysis of Square motion stage with curve beam of 32 mm (Mechanism 1)

Square motion stage with curve beam 1 is having dimension of 300*300*15 mm with curve beam radius of 32mm. Fatigue analysis of square motion stage is analyzed for 80N force. Force is applied at notch as shown in Figure. 10. Mechanism 1 is having meshing with 605938 nodes and 391526 triangular elements as shown in Figure. 11.

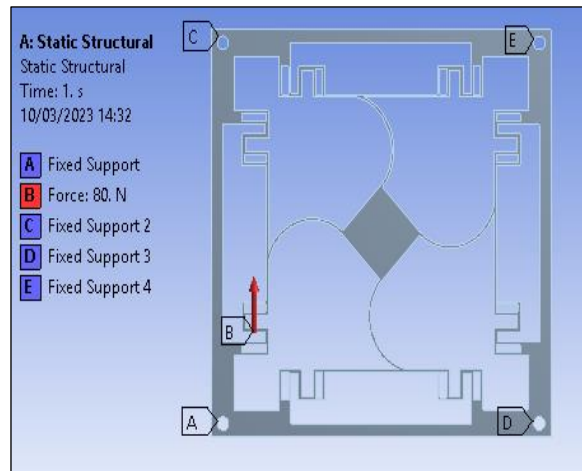


Figure 10: Boundary Conditions of Mechanism 1 for fatigue analysis.
Source: Authors, (2026).

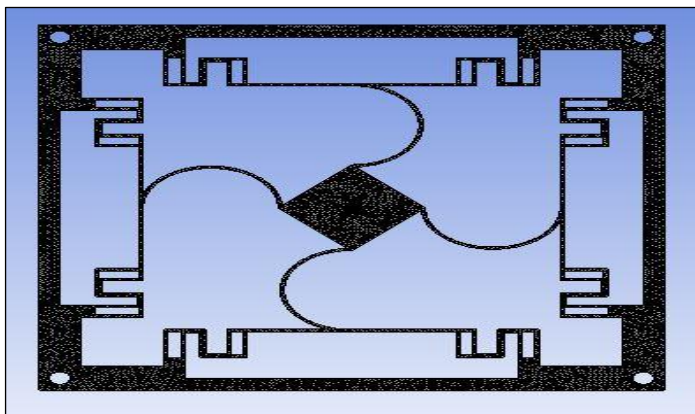


Figure 11: Meshing of Mechanism 1 for fatigue analysis.
Source: Authors, (2026).

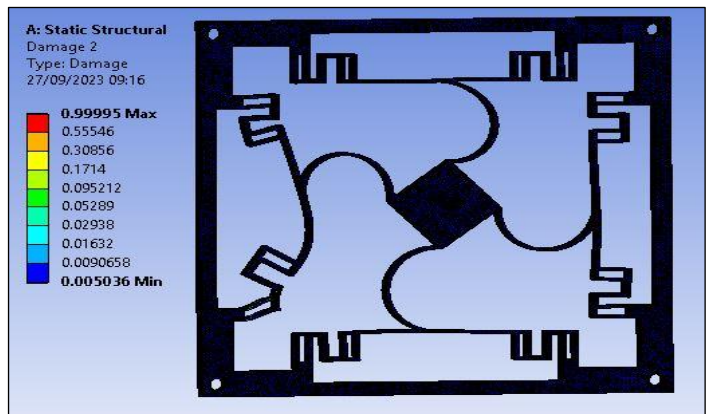


Figure 12: Damage of Mechanism 1.
Source: Authors, (2026).

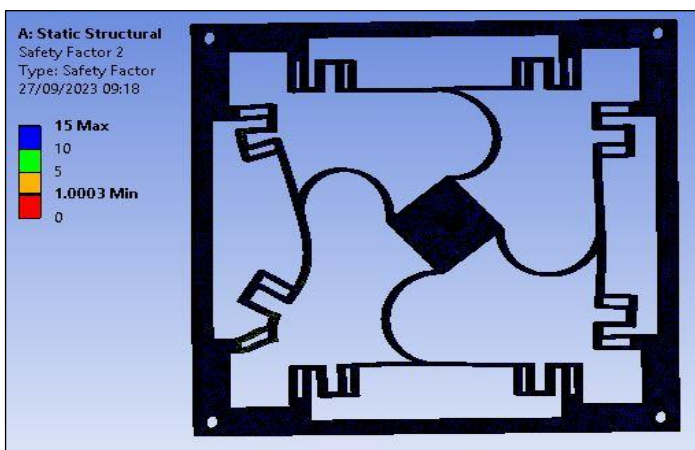


Figure 13: Safety of Mechanism 1.
Source: Authors, (2026).

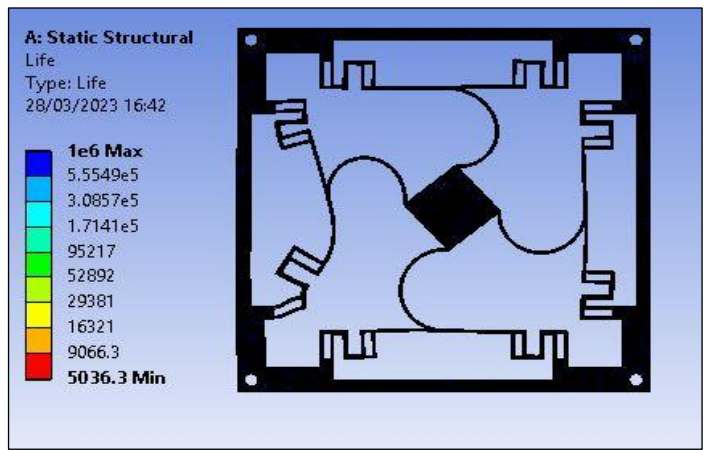


Figure 14: Life cycle for Square motion stage with curve beam 1.
Source: Authors, (2026).

To avoid fatigue failure, the damage value must be less than 1. Figure. 12 shows the damage assessment of the mechanism. According to Figure. 13, the safety factor of the mechanism is 1, which indicates that the mechanism is at the threshold of its design limits.

Figure. 14 presents the life cycle analysis of the mechanism. The square motion stage with Curve Beam 1 is deemed suitable for 5,036 life cycles, given that the damage factor is 0.99 and the safety factor is 1. This means the mechanism can withstand 5,036 cycles before potentially reaching its fatigue limit, as the damage factor is just under the critical value of 1, ensuring it does not fail prematurely.

Analytical Method to calculate Fatigue life Cycle, Damage and Safety factor for Mechanism 1

We'll use the same parameters for both numerical and analytical methods to ensure consistency.

- ❖ Material: PLA
- ❖ Tensile Strength (σ_t): 59 MPa
- ❖ Fatigue Strength Coefficient (σ_f): 2.05 MPa
- ❖ Fatigue Strength Exponent (b): -0.153
- ❖ Stress amplitude (σ_a): 0.52 MPa
- ❖ Number of cycles (N_f): Various levels for damage calculation
- ❖ Fatigue Life Cycle Calculation (N_f) using Basquin's equation:

$$\left(\frac{\sigma_a}{\sigma_f} \right)^{(1/b)} \quad (1)$$

$$N_f : \left(\frac{0.52}{2.05} \right)^{(1/-0.153)} \quad (2)$$

$$N_f : 7465 \text{ Cycle}$$

$$\text{Damage Calculation using Minors Rule (D)} : \sum_{i=1}^k \frac{n_i}{N_i} \quad (3)$$

Compliant mechanism is subjected to 9000 Cycle at 2 Mpa

$$\text{Damage (D)} : \left(\frac{7465}{9000} \right) = 0.82$$

$$\text{Safety Factor} : \left(\frac{\text{Endurance limit}}{\text{Applied stress}} \right) \quad (4)$$

$$\text{Safety Factor} : \left(\frac{2.02}{2} \right) = 1.01$$

Comparison of result

1. Fatigue life cycle by Analytical Method: 7465 Cycle
2. Fatigue life cycle by Numerical Method: 7375 Cycle
3. Damage by Analytical Method: 0.82
4. Damage by Numerical Method: 0.81
5. Safety factor by Analytical Method: 1.01
6. Safety factor by Numerical Method: 1.06

The results are reasonably close, indicating that the numerical model is validated by the analytical calculations. Small discrepancies can be attributed to differences in the precision of numerical simulations versus analytical approximations.

III.1 FATIGUE ANALYSIS OF SQUARE MOTION STAGE WITH CURVE BEAM OF 40 MM (MECHANISM 2)

Square motion stage with curve beam 2 is having dimension of 300*300*15 mm with curve beam radius of 40 mm. Fatigue analysis of square motion stage is analyzed for 80 N force. Force is applied at notch as shown in Figure. 15. Mechanism 2 is having meshing with 81886 nodes and 42871 triangular elements as shown in Figure. 16. To avoid fatigue failure, the damage value must be less than 1. Figure. 17 illustrates the damage assessment of Mechanism 2. According to Figure. 18, the safety factor of Mechanism 2 is 1, which indicates that it is at the threshold of its design limits. Figure. 19 presents the life cycle analysis of the mechanism.

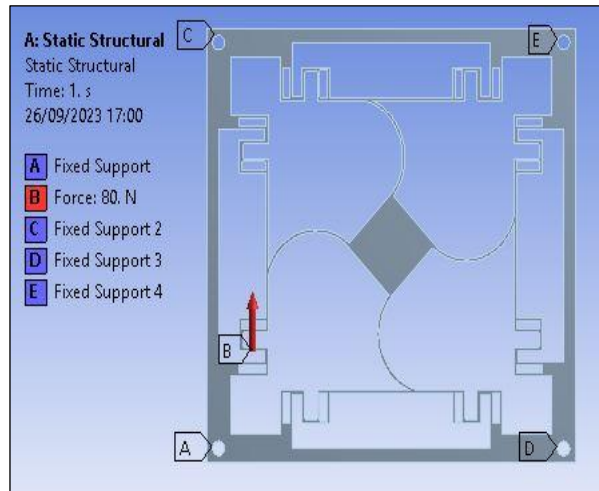


Figure 15: Boundary Conditions of Mechanism 2 for fatigue analysis.
Source: Authors, (2026).

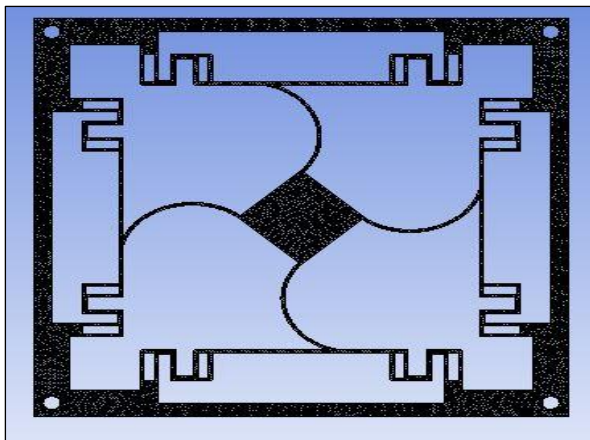


Figure 16: Meshing of Mechanism 2 for fatigue analysis.
Source: Authors, (2026).

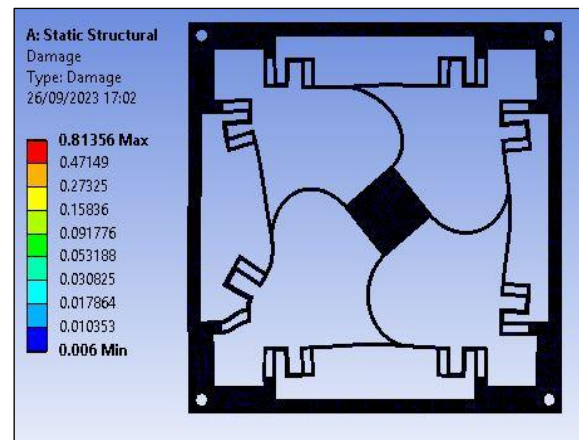


Figure 17: Damage of Mechanism 2.
Source: Authors, (2026).

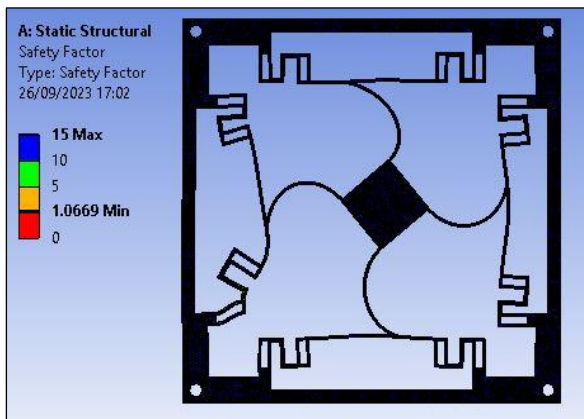


Figure 18: Safety of Mechanism 2.
Source: Authors, (2026).

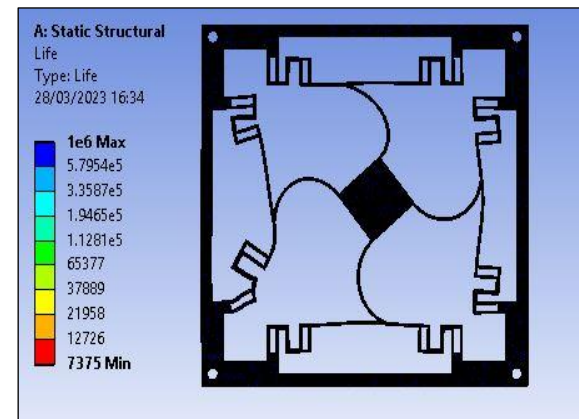


Figure 19: Life cycle for Square motion stage with curve beam -2.
Source: Authors, (2026).

The square motion stage with Curve Beam 2 is suitable for 7,375 life cycles, given that the damage factor is 0.81 and the safety factor is 1.06. This means the mechanism can endure 7,375 cycles before reaching its fatigue limit. The damage factor of 0.81, which is well below the critical value of 1, indicates that the mechanism has a lower risk of fatigue failure. The safety factor of 1.06 suggests a slight margin above the design limits, providing additional assurance of the mechanism's durability and reliability over its life cycle. A larger radius of curvature typically results in lower stress concentration at the curved sections. This means the stress is distributed more evenly, reducing peak stresses that can lead to fatigue failure. With a larger radius, the load is spread over a greater area, which reduces the overall stress experienced by any single point in the mechanism. A larger radius can also improve the flexibility of the mechanism, allowing it to better absorb and distribute applied forces, which can reduce the likelihood of fatigue failure.

Fatigue analysis of Circular motion stage with curve beam of 32 mm (Mechanism 3)

Circular motion stage with curve beam 1 is having dimension of 300*300*15 mm with curve beam radius of 32mm. Fatigue analysis of Circular motion stage is analyzed for 80N force. Force is applied at notch as shown in Figure. 20. Mechanism 3 is having meshing with 84333 nodes and 44134 triangular elements as shown in Figure. 21.

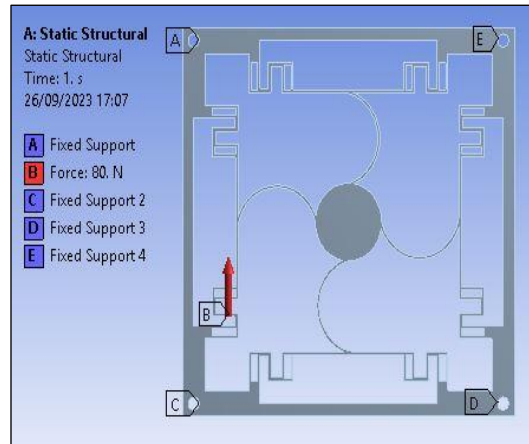


Figure 20: Boundary Conditions of Mechanism 3 for fatigue analysis.
Source: Authors, (2026).

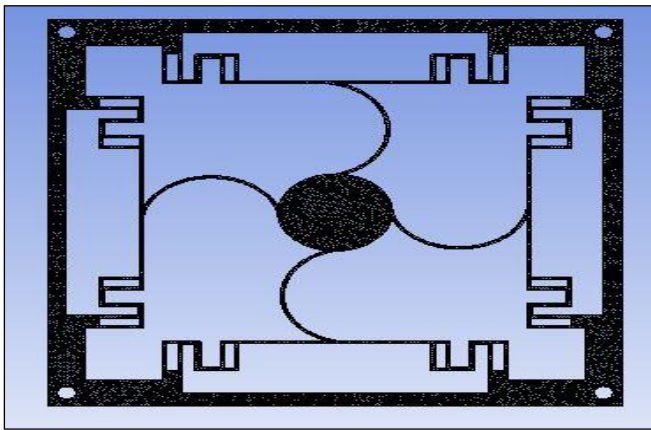


Figure 21: Meshing of Mechanism 3 for fatigue analysis.
Source: Authors, (2026).

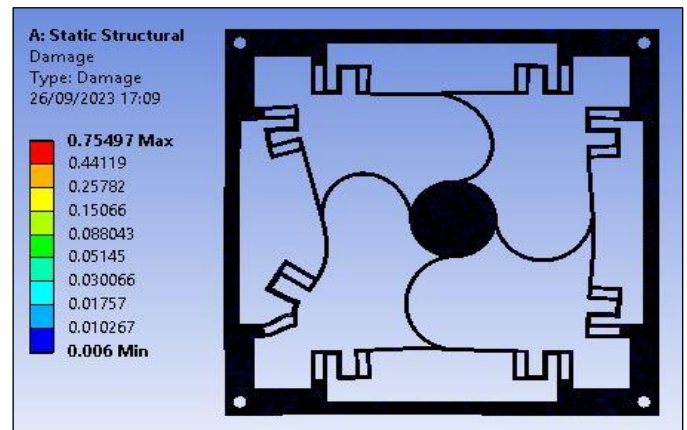


Figure 22: Damage of Mechanism 3.
Source: Authors, (2026).

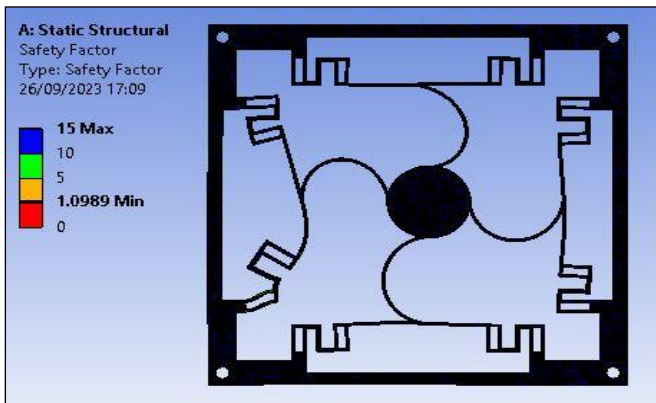


Figure 23: Safety factor of Mechanism 3
Source: Authors, (2026).

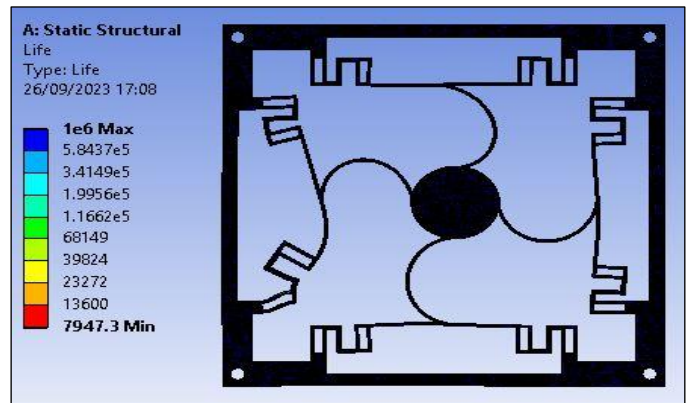


Figure 24: Life cycle for Circular motion stage with curve beam -1.
Source: Authors, (2026).

To avoid fatigue failure, the damage value must be less than 1. Figure. 22 shows the damage assessment of Mechanism 3. The safety factor of the mechanism, as indicated in Figure. 23, is 1.09, suggesting a slight margin above the design limits. Figure. 24 presents the life cycle analysis of the mechanism. The circular motion stage with Curve Beam 1 is suitable for 7,947 life cycles, with a damage factor of 0.75 and a safety factor of 1.09. This means the mechanism can endure 7,947 cycles before reaching its fatigue limit. The damage factor of 0.75, being well below the critical value of 1, indicates a low risk of fatigue failure. The safety factor of 1.09 provides an additional margin of safety, ensuring the mechanism's reliability and durability throughout its life cycle.

Fatigue analysis of Circular motion stage with curve beam of 40 mm (Mechanism 4)

The circular motion stage with Curve Beam 2 has dimensions of 300 x 300 x 15 mm and a curve beam radius of 40 mm. Fatigue analysis was conducted on this stage with an applied force of 80 N at the notch, as depicted in Figure. 25. Mechanism 4 features meshing with 85,536 nodes and 44,762 triangular elements, as shown in Figure. 26. To avoid fatigue failure, the damage value must be less than 1. Figure. 27 shows the damage assessment of Mechanism 4. The safety factor, depicted in Figure. 28, indicates the mechanism's capacity to handle stress beyond its designed load without failing. Figure. 29 presents the life cycle analysis of the mechanism.

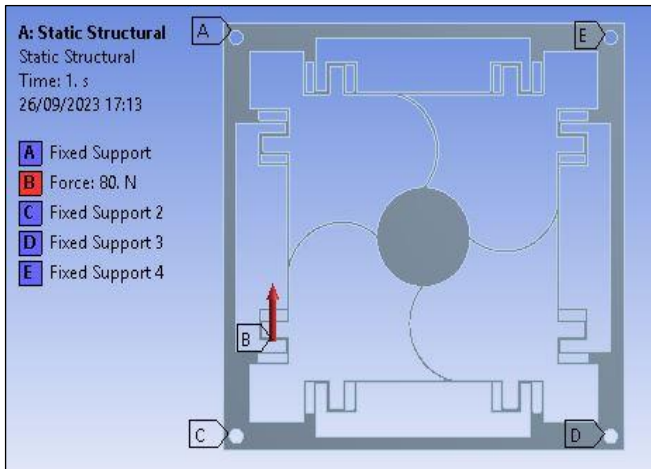


Figure 25: Boundary Conditions of Mechanism 4 for fatigue analysis.
Source: Authors, (2026).

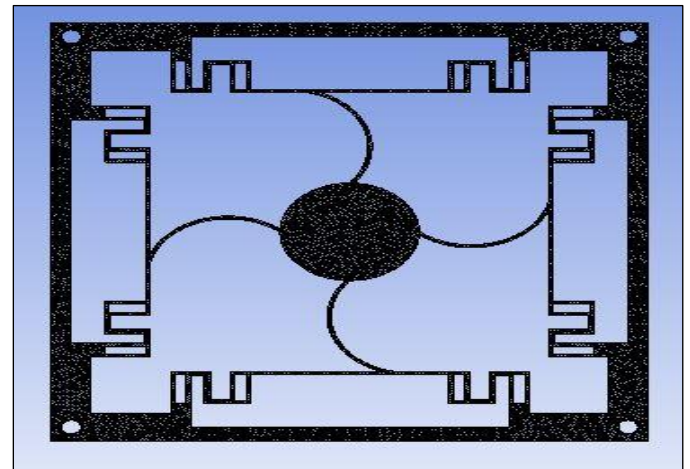


Figure 26: Meshing of Mechanism 4 for fatigue analysis.
Source: Authors, (2026).

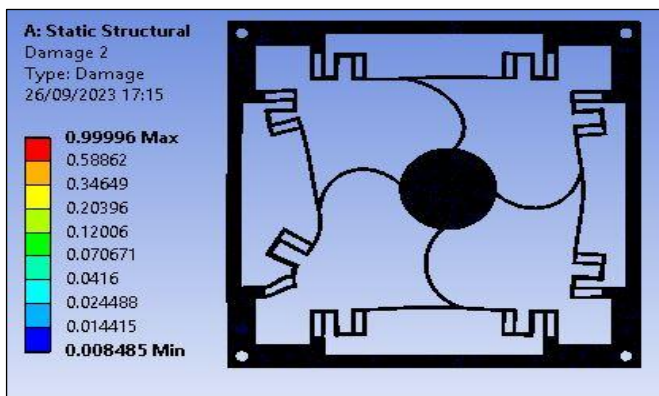


Figure 27: Damage of Mechanism 4.
Source: Authors, (2026).

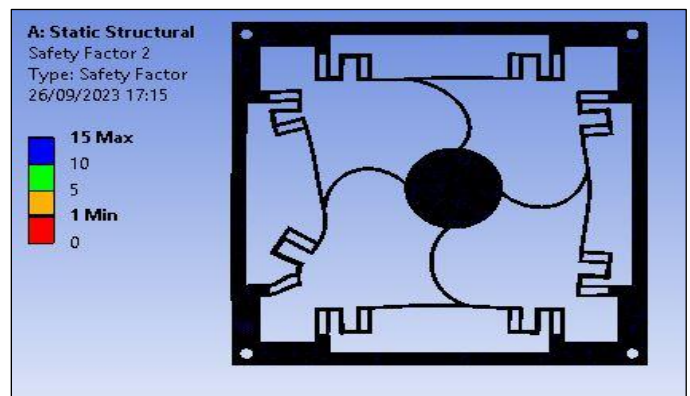


Figure 28: Safety factor of Mechanism 4.
Source: Authors, (2026).

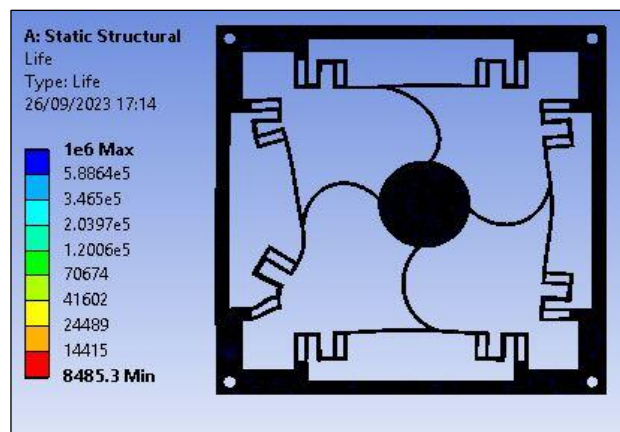


Figure 29: Life cycle for Circular motion stage with curve beam -2.
Source: Authors, (2026).

The circular motion stage with Curve Beam 2 is suitable for 8,485 life cycles, with a damage factor of 0.99 and a safety factor of 1. This means the mechanism can withstand 8,485 cycles before reaching its fatigue limit. The damage factor of 0.99, just under the critical value of 1, suggests that the mechanism is very close to its fatigue threshold. The safety factor of 1 indicates that the mechanism is designed exactly to its limit, with no additional margin for unexpected stresses.

Increasing the radius of curvature helps in mitigating stress concentrations and improving the distribution of loads, thereby enhancing the fatigue life of the compliant mechanism. Fatigue failures occur in compliant mechanisms, as they operate in a cycle of loading. Fatigue life is defined closely related to repeating loadings. Fatigue life is divided into three categories: low cycle fatigue (fatigue failure occurs between 1 to 10^3 cycles), high cycle fatigue (fatigue failure occurs more than 10^3 cycles), and infinite life (no fatigue failure at a given load).

Table 3: Comparison of fatigue life cycle of different mechanisms.

Sr.No	Type of Motion stage	Fatigue life cycle	Safety factor	Damage
1	Square motion stage with curve beam of 32mm (Mechanism 1)	5036	1	0.99
2	Square motion stage with curve beam of 40mm (Mechanism 2)	7375	1.06	0.81
3	Circular motion stage with curve beam of 32mm (Mechanism 3)	7947	1.09	0.75
4	Circular motion stage with curve beam of 40mm (Mechanism 4)	8485	1	0.99

Source: Authors, (2026).

The fatigue life cycle for the square motion stage with curve beam 1 is 5036 Cycles, and curve beam 2 is 7375 as shown in Table 3. The fatigue life cycle for the circular motion stage with curve beam 1 is 7947 cycles, and for curve beam 2 is 8485 cycles. A larger radius of curvature can reduce stress concentrations in a compliant mechanism. Lower stress concentrations are often associated with improved fatigue life because they help distribute the load more evenly.

IV. CONCLUSION

The fatigue life cycle analysis of compliant mechanisms demonstrates the significant impact of geometrical parameters, specifically the radius of curvature of the beams, on the durability and performance of these mechanisms under cyclic loading conditions. Both square and circular motion stages show increased fatigue life with larger radii of curvature. By increasing radius of curvature in Square motion stage fatigue life cycle is increased by 46.44% whereas by increasing radius of curvature in circular motion stage fatigue life cycle of mechanism is increased by 6.76%. This suggests that optimizing the geometric design of compliant mechanisms can significantly extend their operational lifespan. The safety factors for the different mechanisms indicate that larger radii not only improve fatigue life but also ensure that the mechanisms operate safely within their intended load conditions. A larger radius of curvature helps in distributing the load more evenly, reducing the localized stress and hence the damage values, contributing to better durability and reliability. The analysis underscores the importance of geometric optimization in the design of compliant mechanisms. By carefully adjusting the radius of curvature, designers can achieve better fatigue performance, enhance safety, and reduce damage, thereby ensuring that these mechanisms can sustain their intended functions over longer periods and under repeated loading conditions.

Abbreviation

- ❖ σ_t : Tensile Strength
- ❖ σ_f : Fatigue Strength Coefficient
- ❖ b : Fatigue Strength Exponent
- ❖ σ_a : Stress amplitude
- ❖ N_f : Number of cycles

V. AUTHOR'S CONTRIBUTION

Conceptualization: Sandesh Solepatil, Narendra Deore.

Methodology: Sandesh Solepatil, Narendra Deore.

Investigation: Sandesh Solepatil, Narendra Deore.

Discussion of results: Sandesh Solepatil, Narendra Deore.

Writing – Original Draft: Sandesh Solepatil, Narendra Deore.

Writing – Review and Editing: Sandesh Solepatil, Narendra Deore.

Resources: Sandesh Solepatil, Narendra Deore.

Supervision: Sandesh Solepatil, Narendra Deore.

Approval of the final text: Sandesh Solepatil, Narendra Deore.

VI. ACKNOWLEDGMENTS

I would like to extend my heartfelt gratitude to Dr. Mrs. A.V. Patil, Director, School of Engineering, Management and research, Prof. Manish Bhalla, Vice Chancellor, D.Y.Patil International University, Pune for providing the resources and environment that supported this research.

VII. REFERENCES

- [1] Chen G, Liu X, Du Y. Elliptical-Arc-Fillet Flexure Hinges: Toward a Generalized Model for Used Flexure Hinges. *J Mech Des.* 2011;133(9):1002–10. <https://doi.org/10.1115/1.4004441>.
- [2] Cullin M, Petersen T, Lobontiu N, Kirk T. Experimental determination of planar compliance values of flexures. *Measurement.* 2014;58:301–9. <https://doi.org/10.1016/j.measurement.2014.09.001>.
- [3] D’Souza RD, Mohamed AU, Tharakan OP, Mini A. Effect of Flexural Hinges in the design of a 2 DOF Compliant Microgripper. *Proc 3rd Int Conf Mechatron Robot Eng (ICMRE).* 2017:111–8. <https://doi.org/10.1145/3068796.3068819>.
- [4] Dirksen F, Lammering R. On mechanical properties of planar flexure hinges of compliant mechanisms. *Mech Sci.* 2011;2(1):109–17. <https://doi.org/10.5194/ms-2-109-2011>.
- [5] Fung RF, Hsu YL, Huang MS. System identification of a dual-stage XY precision positioning table. *Precis Eng.* 2009;33(1):71–80. <https://doi.org/10.1016/j.precisioneng.2008.04.002>.
- [6] Hunt J, Lee H. A new parallel actuated architecture for exoskeleton applications involving multiple degree-of-freedom biological joints. *J Mech Robot.* 2018;10(2):82–91. <https://doi.org/10.1115/1.4040701>.
- [7] Li Y, Xu Q. Modeling and performance evaluation of a flexure-based XY parallel micromanipulator. *Mech Mach Theory.* 2009;44(12):2127–52. <https://doi.org/10.1016/j.mechmachtheory.2009.06.002>.
- [8] Ling M, Cao J, Jiang Z, Li Q. Development of a multistage compliant mechanism with new boundary constraint. *Rev Sci Instrum.* 2018;89(1):015009-23. <https://doi.org/10.1063/1.5007890>.
- [9] Linß S, Schorr P, Zentner L. General design equations for the rotational stiffness, maximal angular deflection and rotational precision of various notch flexure hinges. *Mech Sci.* 2017;8(1):29–49. <https://doi.org/10.5194/ms-8-29-2017>.
- [10] Shang J, Tian Y, Li Z, Wang F, Cai K. A novel voice coil motor-driven compliant micropositioning stage based on flexure mechanism. *Rev Sci Instrum.* 2015;86(9):095001-7. <https://doi.org/10.1063/1.4929867>.
- [11] Shaw LA, Sun F, Portela CM, Barranco RI, Greer JR, Hopkins JB. Computationally efficient design of directionally compliant metamaterials. *Nat Commun.* 2019;10(1):43–52. <https://doi.org/10.1038/s41467-018-08049-1>.
- [12] Shinno H, Yoshioka H, Gokan T, Sawano H. A newly developed three-dimensional profile scanner with nanometer spatial resolution. *CIRP Ann Manuf Technol.* 2010;59(1):525–8. <https://doi.org/10.1016/j.cirp.2010.03.022>.
- [13] Sun X, Chen W, Zhou R, Zhang J, Chen W. Development of a flexure-based XY positioning stage for micro/nano manipulation. *Int Conf Manip Manuf Meas Nanoscale.* 2013:82–91. <https://doi.org/10.1109/3m-nano.2013.6737441>.
- [14] C Wang Hinges for compliant mechanism design. *Precis Eng.* 2010;34(3):408–18. <https://doi.org/10.1016/j.precisioneng.2009.10.002>.
- [15] Xu Q. Design of large positioning system. Hoboken: Wiley; 2016.
- [16] Roy NK. Design of laser sintering system. *Proc 26th Annu Int Solid Freeform Fabrication Symp.* 2016:1495–510.
- [17] Wan S, Zhang Y, Xu Q. Design and development of a new large-stroke XY compliant micropositioning stage. *Proc Inst Mech Eng C J Mech Eng Sci.* 2016;231(17):3263–76. <https://doi.org/10.1177/0954406216643341>.
- [18] Wu Z, Xu Q. Survey on recent designs of compliant micro-/nano-positioning stages. *Actuators.* 2018;7(1):5–11. <https://doi.org/10.3390/act7010005>.
- [19] Xu Q. New flexure parallel-kinematic micropositioning system with large workspace. *IEEE Trans Robot.* 2012;28(2):478–91. <https://doi.org/10.1109/tro.2011.2173853>.
- [20] Zhu Z, To S, Zhu WL, Li Y, Huang P. Optimum design of a piezo-actuated triaxial compliant mechanism for nanocutting. *IEEE Trans Ind Electron.* 2018;65(8):6362–71. <https://doi.org/10.1109/tie.2017.2787592>.
- [21] Solepatil S, Deore N. Displacement analysis of compliant mechanism. *Int J Eng Adv Technol.* 2019;9(1):4028–32. <https://doi.org/10.35940/ijeat.A1175.109119>.
- [22] Solepatil S, Deore N. Behavior of linear compliant mechanism using numerical and experimental method. *Mater Today Proc.* 2021;47:3190–4. <https://doi.org/10.1016/j.matpr.2021.06.299>.
- [23] Solepatil S, Deore N. Performance analysis of compliant mechanism for micropositioning stage. *Proc Eng Sci.* 2023;5(2):303–10. <https://doi.org/10.24874/PES05.02.013>.
- [24] Solepatil S, Deore N. Performance analysis of parallel beam XY compliant mechanism. *AIP Conf Proc.* 2024;3013(1):020009-1–9. <https://doi.org/10.1063/5.0202189>.
- [25] Solepatil, S. B., Deore, N. R., Dive, V. P., Mali, A. S., Kolekar, A. B. & Jadhav, G. Design, numerical and experimental investigation of linear XY compliant mechanisms for precision motion applications. *Advances in Science and Technology Research Journal*, (2026), 20(5), 400–415. <https://doi.org/10.12913/22998624/217145>.