Compliant Mechanisms and MEMS

ME 381: Introduction to MEMS
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ABSTRACT

Distributed compliant mechanisms, invented by Kota and Ananthasuresh in 1995[1], have been the focus of considerable research in recent years. Compliant mechanisms have applications in a wide array of applications including adaptive wing design, precision manipulation, and MEMS. Incorporation of compliant mechanisms into MEMS actuators is particularly promising because it provides the flexibility to tailor micro-actuators to meet desired output characteristics. This paper discusses the working principle of compliant mechanisms and their application to MEMS devices. It reviews the current state of the art methods used to synthesize and design compliant mechanisms. We also list laboratories where MEMS-based compliant mechanism research is conducted.

CONTENTS

1. INTRODUCTION

Distributed compliant mechanisms are flexible structures that use strain energy to transform intput energy components into a desired output force or displacement. In MEMS, due to an increase in surface to volume ratio, the frictional forces dominate. Therefore a MEMS mechanism must avoid joints. Due to the limitations of microfabrication methods, MEMS structures have to be monolithic and assembly must be avoided. Compliant mechanisms provide a jointless alternative to conventional rigid body mechanisms eliminating issues of friction, wear, lubrication, and backlash. Compliant mechanisms are monolithic structures and require no assembly, therefore they are very well suited for microfabrication (figure 1.1).

The remaining sections of this paper will discuss the working principles of compliant mechanisms focusing on their ability of transform energy in a controllable way. Some examples of macroscale compliant mechanisms are included, and the advantages of using compliance in MEMS actuators are addressed. This paper also reviews some of the techniques used to design, optimize, and analyze compliant mechanisms. Finally, several MEMS based examples are highlighted, and future applications are discussed.

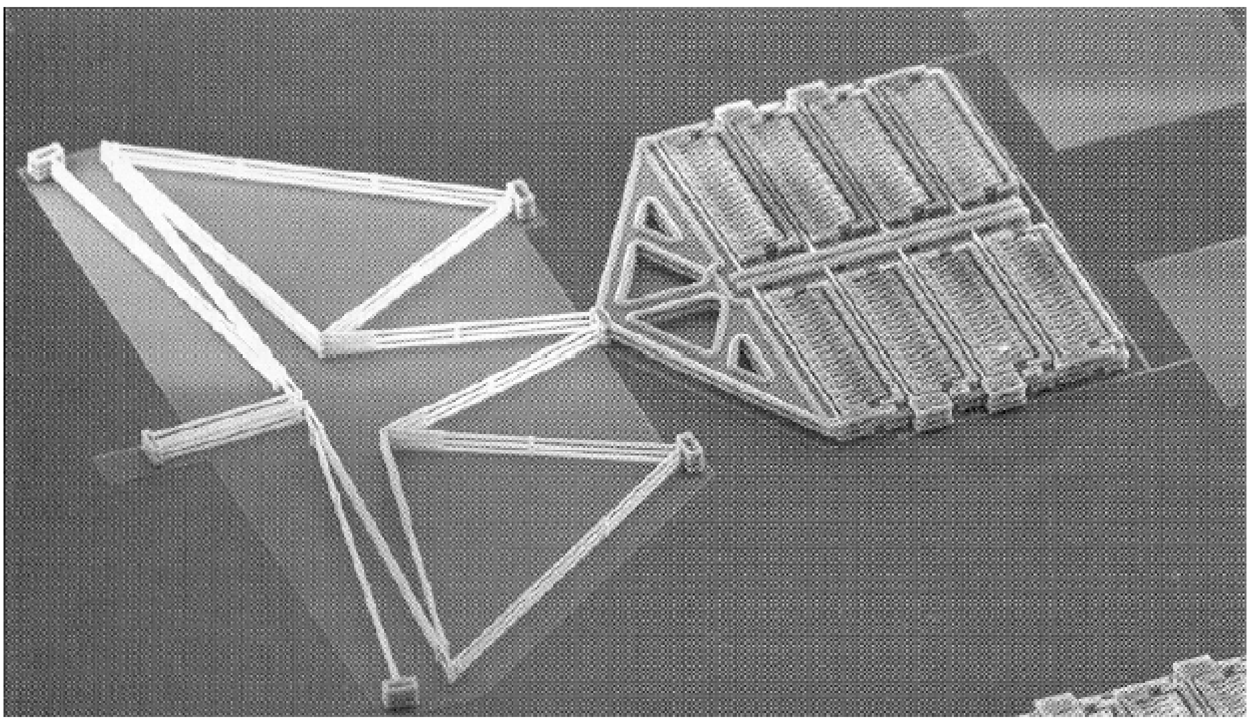


Figure 1.1 SEM image of a compliant mechanism attached to a comb-drive electrostatic actuator. The compliant mechanism has a geometric advantage of 20. The entire device occupies an area of 350 µm by 350 µm.[2,3]

2. WORKING PRINCIPLE

Traditional rigid body mechanisms consist of rigid links connected at movable joints and are capable of transforming linear motion into rotation or force into torque. A rigid mechanism does not have mobility, so it does not perform any work, it simply transfers energy. Since energy is conserved between the input and output, the output force may be much larger than the input force (mechanical advantage), but the output displacement is much smaller than the input displacement (geometric advantage) or vice versa

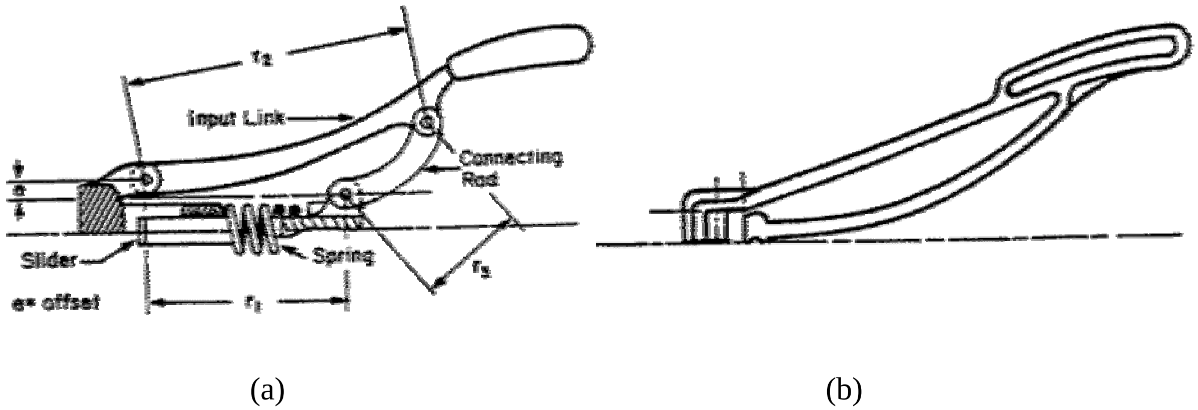


Figure 2.1 (a) A non-compliant crimp with rigid body links connected at movable joints. (b) A compliant crimp fabricated as one part.[4]

Compliant mechanisms rely on the deflection of flexible members to store energy in the form of strain energy. This stored energy is similar to the potential energy in a deflected spring. Thus compliant mechanisms can be used to easily store and/or transform energy that can be released at a later time or in a different manner. The simplest example of compliant mechanism is the bow, shown in figure 2.2, used to shoot arrows.

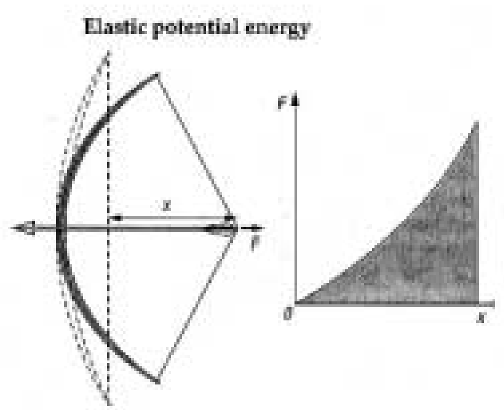


Figure 2.2 A bow (shown above) is a compliant mechanism since it uses deflection to store elastic potential energy like a spring and the energy is transferred as kinetic energy to an arrow.[5]

At the macro scale several innovative compliant mechanisms have been created such as the no assembly stapler and windshield wiper shown in figure 2.3. At the micro-scale compliant mechanisms have been implemented as displacement amplification transmissions for comb-drive actuators (as depicted in figure 1)[2,3] and in novel precision 6 DOF manipulation stages with nano-scale displacement resolution and mm range developed by the Culpepper group at MIT (figure 6.3).[7,8] Microscale compliant mechanisms typically take the form of folded beam structures.

(a) (b)

Figure 2.3 Macro-scale examples of a monolithic (a) stapler and (b) windshield wiper developed by the Kota Group at the University of Michigan using compliant mechanisms.[9]

In order to fully appreciate the benefits and limitations of compliant mechanisms one must first understand the concepts of geometric and mechanical advantage and the trade offs there in. Geometric and mechanical advantage are the ratios of the output displacement or force to the input displacement and force, respectively. Geometric and mechanical advantage are often described by the following equations.

$$GA = \frac{u_{out}}{u_{in}} \quad \text{and} \quad MA = \frac{F_{out}}{F_{in}} \,. \qquad (2.1\ \text{a,b})$$

Force and displacement are energy variables (i.e. the product of force and displacement has units of energy), and an ideal compliant mechanism is designed to conserve energy. Thus a compliant mechanism serves as a transmission that converts the input energy in a controlled way. Since energy is conserved the product of force and displacement must be a constant. So, if the output displacement increases then the output force must decrease and vice versa. Figure 2 depicts the tradeoff between force and displacement.

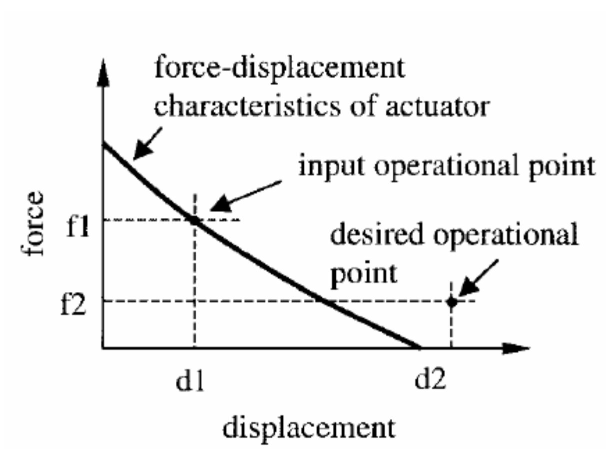


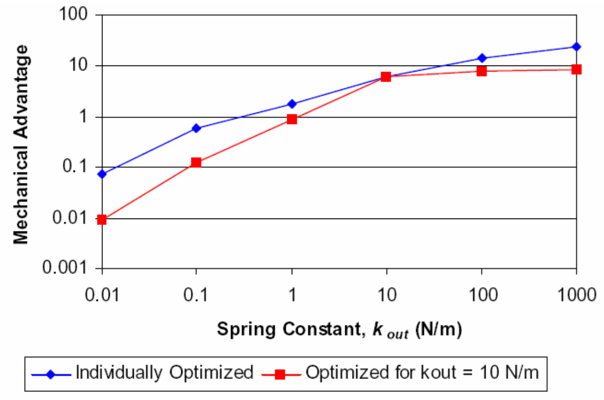
Figure 2.4 The above graph depicts the trade off between output force and displacement.[10]

3. COMPLIANT MECHANISMS AND MEMS

Compliant mechanisms can play an important role in the design of micro-actuators for MEMS applications. Compliant transmissions can be designed to perform complex mechanical functions and can be fabricated within the constraints of present day micro-machining processes. They require no assembly, which satisfies the demands of batch fabrication. They are also "joint less" which allows for precision motions and eliminates problems due of wear, lubrication, and backlash.

One of the primary benefits of compliant mechanisms is that as part of the design process one can specify the geometric or mechanical advantage allowing one to tailor the compliant mechanism output to meet specific design needs. This ability is particularly powerful for the modification of micro-actuators. The limitations of micro-actuators are well documented, and in many instances limit the development of MEMS based devices. Thus integrating compliant mechanisms into MEMS devices can extend the range of micro-actuators and allow them to be used in new and exciting applications such as MEMS-based micro-robotics, and bio-MEMS. Due to MEMS structure's small scale and high aspect ratio, they are naturally flexible making compliant mechanisms the only option in designing transmissions for micro-actuators. A compliant transmission can significantly improve the efficiency and precision of micro-actuators by amplifying the output forces or displacements in a controlled manner. As a result, compliant mechanisms in MEMS have helped realize the generation of forces and displacements on the order of μN and μm, respectively. Compliant mechanisms have outstanding robustness, endurance, and resistance to surface adhesion effects. A compliant mechanism similar to the one depicted in figure 1.1 has been successfully driven at its resonance frequency with an output displacement of 20 μm (input displacement of 1 μm) for more then 10^{10} cycles with no signs of fatigue.[3,6,10]

One of the primary drawbacks of compliant mechanisms is their limited displacement range due to a need to keep all of the mechanism's elements well within the linear elastic regime. Another draw back of compliant mechanisms is that the geometric and mechanical advantages as well as the efficiency are functions of the output stiffness. Thus when the output stiffness deviates from the designed value, the geometric advantage, mechanical advantage, and efficiency of the compliant transmission can drop quickly. This behavior is depicted in figure 2.5, which shows (a) mechanical and (b) geometric advantages as a function of the output stiffness. The blue line in figure 3.1 represents the optimized mechanical and geometric advantage for each value of spring stiffness.[11] The results in figure 3.1 imply that the behavior of a compliant transmission is largely dependant on the actuator and the object that is being worked on. An ideal configuration will adjust the geometry of the transmission to match the output stiffness. Professor Martin Caulpepper's group at MIT has developed a six degree of freedom nanomanipulator that utilizes dynamic compliant mechanisms that can achieve a wide range of geometric and mechanical advantages.[7,8]



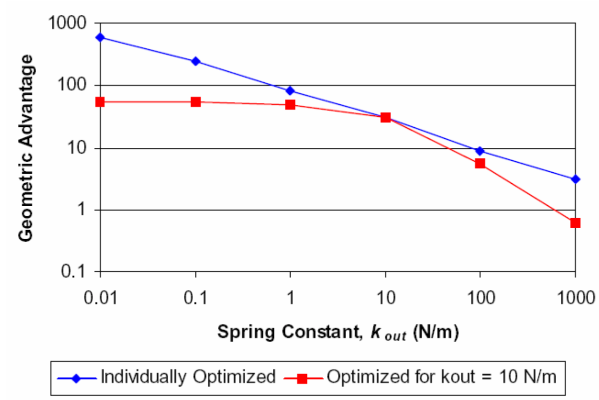


Figure 3.1 These graphs show the variation of (a) mechanical and (b) geometric advantage as a function of the output stiffness.[11]

4. DESIGN AND OPTIMIZATION

Traditional mechanism design utilizes infinitely rigid members connected by pins and joints to form a complex mechanical assembly. The first attempt at integrating compliance into mechanisms occurred in the 1960's when engineers began to replace pined joints with flexible members. These flexible members were limited to small a region where the joint was originally located and comprised a small part of the mechanism design. Burns and Crossley were some of the first researchers to present synthesis techniques for flexure-link joints.[12] Their synthesis techniques adapted established kinematic analysis techniques to include small regions of compliance. Integration of flexure-link joints into complex mechanisms significantly reduced total part numbers and manufacturing time. However, flexure link joints have small displacement ranges (in order to ensure that the flexure stays within the elastic regime), and have short lifetimes due to stress concentrations in the link. Distributed compliant mechanisms avoid issues of load concentrations and poor lifetimes by distributing the strain energy throughout the mechanism. In this way every member in the assembly undergoes elastic deformation transferring energy from the input to the output.

The design of efficient distributed compliant mechanisms that meet specific design criteria is a challenging task. The synthesis of distributed compliant mechanisms must deviate from the traditional kinematics and kinetostatics approaches, because the members are no longer rigid. The synthesis process is typically completed in two steps. The first step is to construct a kinematic geometry that generates the desired displacement and force output for a given input. This process is called topology synthesis which takes the form of a multi-criteria optimization problem.

The second step in the design of compliant mechanisms is size and shape optimization. In this step the shape of the individual elements is optimized to ensure that the mechanism achieves prescribed performance specifications. Performance specifications include prescribed geometric or mechanical advantage, minimization of energy losses in the mechanism, and output phase or resonant frequency for dynamic loading. In this step one can also impose local constraints for the minimization of buckling instabilities, enforcement of stress limitations to ensure the operation envelope of the device is within the elastic regime, minimization of stress concentrations, as well as enforcement of fabrication and weight limitations. The two step process is necessary because stress analysis in incompatible with the resizing methods utilized during the topology synthesis procedure due to an inability to continually re-mesh the geometry.[10]

4.1. TOPOLOGY SYNTHESIS

The purpose of topology synthesis is to generate a kinematic layout that is capable of converting a given input force and displacement into the desired output force and displacement. The key to effective topology optimization is the formulation of an objective function that successfully captures the need for compliance to achieve the desired output force and displacement as well as sufficient stiffness to resist external loads. Several objective function formulations have been proposed for the inclusion of compliance into mechanism designs.

One of the first successful objective function formulations consisted of a multi-criteria weighted sum that attempted to maximize the flexibility of the structure in addition to maximizing the device's stiffness.[10] The device stiffness is often represented by the inverse of the strain energy stored in the system. Thus the optimization routine attempts to maximize the ratio of the output displacement to the stored strain energy.[12] These formulations are often referred to as flexibility-stiffness formulation. Yin and Ananthasuresh have shown that that these methods, result in structures that are similar to the rigid-body designs with localized compliance.[13] This is to be expected since rigid members connected by revolute joints minimize the stored strain energy in the system. Figure 4.1 (a) depicts a compliant mechanism topology synthesized using the flexibility-stiffness objective function, and shows the concept of localized compliance. Compliant transmissions designed using the flexibility-stiffness formulations suffer from the same limitations discussed above for the flexible-link joints.

One of the most common objective functions and simplest to grasp is the energy efficiency formulation developed by the *Compliant Mechanisms Design Laboratory* at the University of Michigan.[1] In the energy efficiency formulation one attempts to maximize the ratio of the output energy to the input energy. The energy efficiency formulation is preferred over the flexibility-stiffness formulation, because it results in well behaved optimization routines. It is also useful, because the resultant topologies have compliance distributed throughout. Figure 4.1 (b) depicts a compliant mechanism topology synthesized using the energy-efficiency objective function, and shows the concept of distributed compliance. The energy efficiency formulation is also preferred because the same objective function can be used for size and shape optimization.

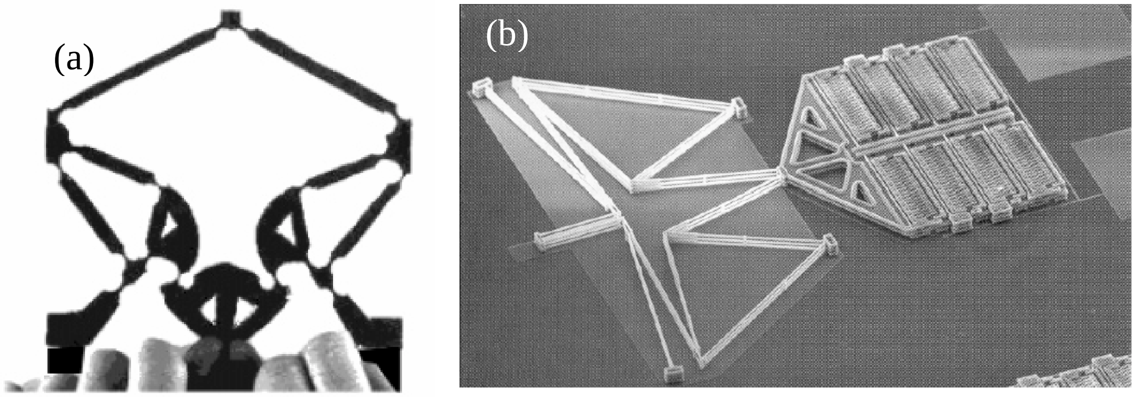


Figure 4.1 (a) A compliant mechanism designed using the flexibility-stiffness objective function. This design depicts the concept of localized compliance.[13] (b) A compliant mechanism designed using the energy efficency objective function. This design depicts the concept of distributed compliance.[2,3]

4.1.1. Energy Efficiency Formulation

The energy efficiency formulation uses the following objective function which is a ratio of the output energy to the input energy.

$$\eta = \frac{(work)_{out}}{(work)_{in}} = \frac{\int F_{out}(t) u_{out}(t) dt}{\int F_{in}(t) u_{in}(t) dt} \qquad (4.1)^{10}$$

In the above equation F_{out} and F_{in} are the output and input force components respectively, and u_{out} and u_{in} are the output and input displacement components respectively. The benefit of using the above objective function is that by maximizing the efficiency one will generate a topology that satisfies the basic kinematic restraints as well as minimizing the elastic energy stored in the system. Kota *et al.* have also shown that equation (4.1) generates topologies with distributed compliance where the strain energy is evenly distributed throughout the mechanism maximizing the compliance of the design.[10] Another advantage of the above formulation is that the work integrals in equation (4.1) need not be in the mechanical domain allowing one to use other energy domains such as electrical (voltage and charge) or thermal (temperature and entropy). This allows designers to include the behavior of a wide variety of actuators, such as piezoelectric or thermo-elastic actuators, into the design of the compliant mechanism ensuring that the compliant mechanism is optimized for the proper actuator.

The key to properly implementing the energy efficiency formulation is the selection of a spring that models the resistance of the object being worked on. By varying the stiffness of the of the output spring the one can optimize the design for mechanical advantage (high stiffness) or geometric advantage (low stiffness). As discussed in section 3.0, the selection of a stiffness that best represents the output is critical to obtaining the maximum efficiency and the desired geometric or mechanical advantage.[11]

One can simplify equation (4.1) for analysis by applying the following boundary conditions, depicted in figure 4.2, which are applied in two stages:

1. Apply the external load, F_{ex}, while keeping the input fixed (i.e. un-actuated case). The force needed to resist the external force is $F_{in,1}$ and the output displacement is $u_{out,1}$.
2. With the external load applied, the input is displaced a prescribed amount, u_{in}, with an applied force of $F_{in,2}$ (i.e. actuated case). The displacement of the output is $u_{out,2}$.[3,10]

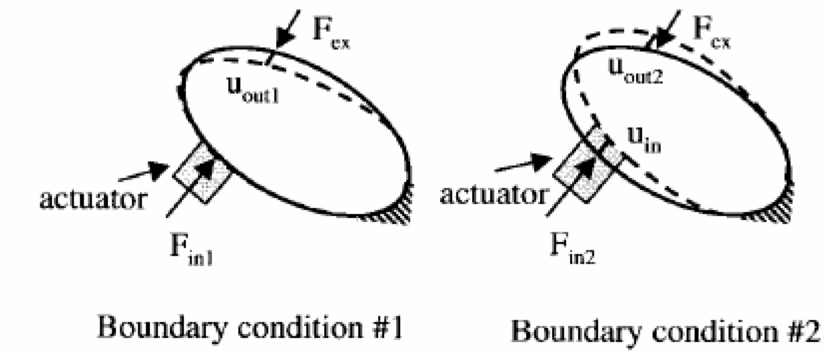


Figure 4.2 The above figures display the application of the boundary conditions to an arbitrary geometry, and the corresponding force displacement history. Note that $u_{out,0}$ is the output when the eternal force is removed.[3,10]

Now neglecting any dynamic or damping loses in the material one can show that equation (4.1) becomes,

$$\eta = \frac{F_{ex}\left(u_{out,2} + \frac{1}{2}u_{out,1}\right)}{\frac{u_{in}}{2}\left(F_{in,1} + F_{in,2}\right)}. \qquad (4.2)^{3,10}$$

This equation derived by Kota *et al.* can be implemented as the objective function in both topology synthesis as well as size and shape optimization.[3,10]

The overall size of the compliant mechanism is regulated by restricting the design domain of the optimization problem. The optimization problem is evaluated by discretizing the device area into nodes that are connected by beam elements which serve and an initial guess. Figure 4.3 shows two examples of discritizing a design domain into an array of nodes and beam elements**.** Certain nodes are fixed to simulate locations where the device is anchored, and the remaining nodes are allowed to wonder within a controlled space defined by the user.[3,10] The optimization process is carried out by varying the cross sectional area of each of the beam elements and using finite element analysis to evaluate the objective function. Kota *et al.* ensure that the optimization process is well behaved by defining constraints for the cross-sectional area of the beam elements.[10,13] The optimal design is determined by eliminating all elements that reach the minimum value. In this way one eliminates all beam elements that do not contribute to the compliance of the mechanism.

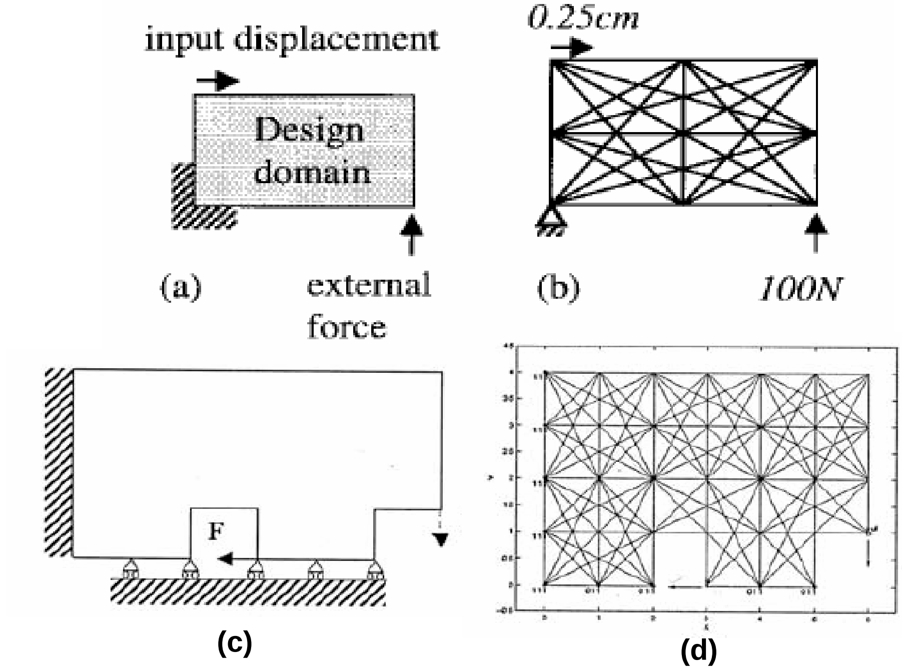


Figure 4.3 The above figure show two examples of discritizing a design domain into an array of nodes and beam elements. (a) and (c) The design domain used in the topology synthesis of a displacement amplification transmission and a micro-gripper respectively. (b) and (d) The ground structure of beam elements which provides an initial guess for the optimal topology. [3,10]

The topology optimization function which utilizes equation (4.2) has the following form,

$$\max(\eta)$$

Subject to:

$$a_{i,\min} \le a_i \le a_{i,\max} \tag{4.3}$$ [10]

$$V \le {}^{Volume}\!\!\diagup_{Max\,\mathrm{Re}\,source} - 1$$

In the above equation $a_{i,min}$ and $a_{i,max}$ are the user defined maximum and minimum areas of the beam elements, and V is a total volume constraint.[10] The above optimization problem can be evaluated using linear finite elements to solve the equilibrium equations and determine a value for equation (4.2). The inputs to the above optimization problem are the modulus of elasticity for the material, the thickness of the planer device, the minimum and maximum cross sectional area of the beam elements, the maximum volume of the mechanism, and the applied input force or displacement.

4.2. Size and Shape Optimization

Once one has generated an optimal topology that meets the kinematic constraints, the next step is to optimize the size and shape of each of the beam elements such that they meet the desired performance characteristics. Size and shape optimization is a relatively new concept, which was developed by Ananthasuresh and Kota in 1995.[1] Since size and shape optimization of compliant topologies is so new, there are very few objective functions that have been successfully implemented. As indicated above, one of the benefits of using the energy efficiency formulation for the objective function is that the same function can be used for both the topology optimization and the size and shape optimization.

4.2.1. Energy Efficiency Formulation

The energy efficiency formulation can also be used as the objective function for the size and shape optimization; however, in this case the problem is subject to different constraints. As indicated above, the added constraints are used to enforce desired performance characteristics of the compliant transmission. These performance characteristics can include weight, geometric and mechanical advantage, minimization of stress concentrations, and avoidance of buckling instabilities. In the size a shape optimization problem the beam cross sectional area and resource constraint are applied again to ensure that the topology and volume are maintained from the topology synthesis routine. Thus the optimization routine listed in equation (4.4) searches for a way of distributing the resource volume such that the objective function is maximized. The following equation is the optimization function for the size and shape optimization that utilizes equation (4.2) as the objective function.

$$\max(\eta)$$

Subject to:

$$a_{i,\min} \le a_i \le a_{i,\max}$$

$$V \le {}^{Volume}\!/\!_{Max\,\mathrm{Re}\,source} - 1 \qquad (4.4)^{3,10}$$

$$h_1 = \left(\frac{F_{out}}{F_{in}}\right)\left(\frac{1}{MA}\right) - 1 \text{ or } h_1 = \left(\frac{u_{out}}{u_{in}}\right)\left(\frac{1}{GA}\right) - 1$$

$$FS = {}^{\sigma_i}\!/\!_{\sigma_{\max}} - 1$$

Equation (4.3) also includes constraints for a specified geometric or mechanical advantage, h_1, as well as a constraint on the maximum stress found in the mechanism, *FS* (i.e. minimization of stress concentrations).[10] It is important to note that equation (4.3) is just an example and the kinematic constraint, h_1, and stress concentration constraint, *FS*, do not need to be applied in order to solve equation (4.3).

In the size and shape optimization problem the topology meshed using beam elements. One can once again define a region around each of the nodes in which the node is free to travel. This allows for small topology changes that can be critical to the maximization of equation (4.2).[10] Figure 4.4 shows the process of size and shape optimization where (a) is the input topology and (b) is the optimized geometry. Note that the boxes around the nodes are the regions where the nodes can wonder. One should also note that this topology was optimized using tapered beam elements.[3] The optimization problem is again evaluated using the finite element method. The inputs to the above optimization problem are the modulus of elasticity for the material, the thickness of the planer device, the minimum and maximum cross sectional area of the beam elements, the maximum volume of the mechanism, and the applied force or displacement.

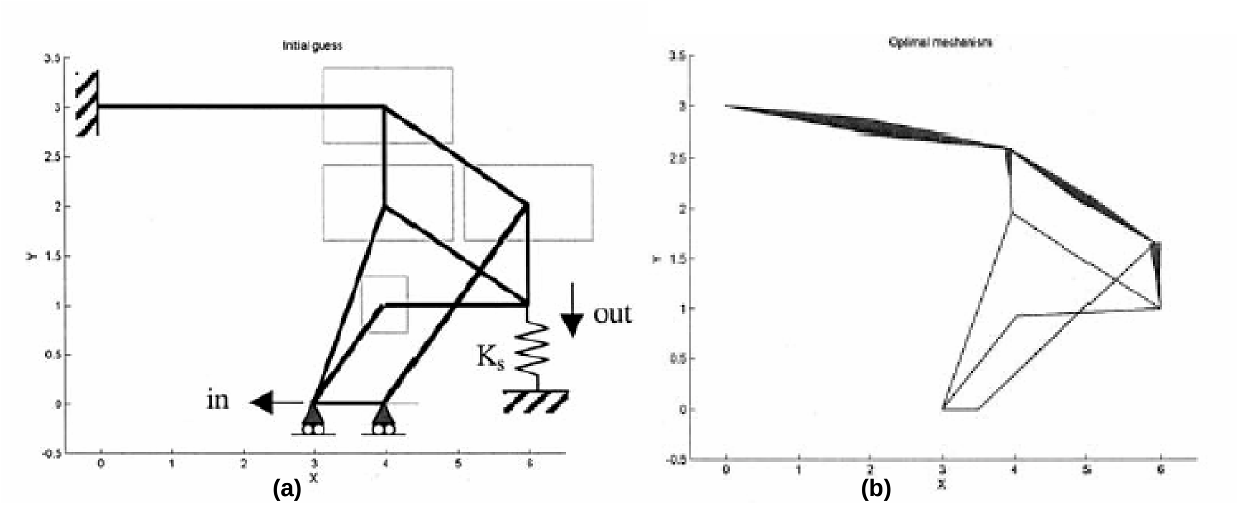


Figure 4.4 (a) The optimized topology of a MEMS gripper input into the size and shape optimization. The rectangles around the nodes mark the region where the nodes are allowed to wonder. (b) The optimized geometry. Note that tapered beam elements where used in the optimization of this geometry.[3]

5. ANALYSIS

A computational procedure that can search through all possible configurations to select the best one based on qualitative criteria of functional behavior as well as quantitative criteria of performance is still an eluding goal.[15] Once the topology synthesis method is completed the optimal geometry undergoes further refinements to the size and shape as indicated above. These refinements are subject to various factors including sensitivity, efficiency, stress range, and dynamic characteristics of the device.

5.1. STRESS ANALYSIS

Stress analysis of compliant mechanisms can be completed using various methods. Two of the most common methods are the Beam Element Model and the Plane Element Model. A Beam Element Model is formulated according to the Pseudo-Rigid-Body Model developed by Professor Larry Howell at BYU, in which kinematic pairs of rigid bodies are used at the place of localized bending. The Plane Element Model is formulated as a plane stress problem, and is more efficient at handling distributed compliance problems.

Consider the topology problem statement as shown in figure 5.1 (a) and (b) and the corresponding optimized topology with the required deformation shown in figures 5.1 b and c. It is important to note that in the following example only half of the mechanism is designed due to symmetry.

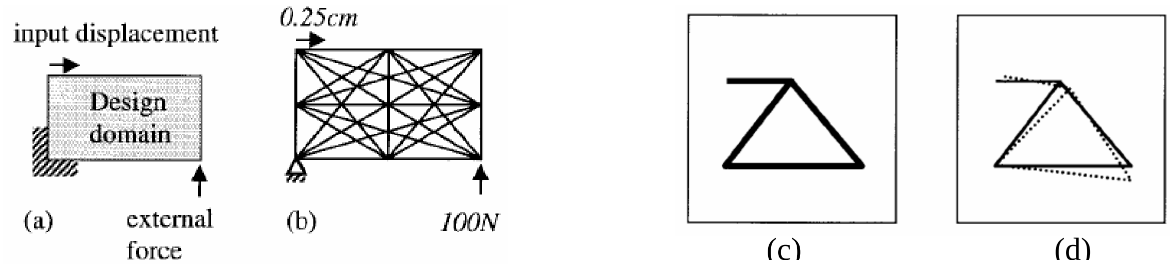


Figure 5.1 (a) Topology Problem Statement. (b) Structure of the initial guess. (c) and (d) Optimized topology with the expected deformation.[10]

As mentioned above, prior to size and shape optimization certain nodes are labeled active and allowed to wonder in a region defined by the user. This wondering region is important to ensure that the optimal topology is maintained during the optimization procedure. The active regions also ensure that the optimization process is well behaved and convergence can be established. Figure 5.2 shows the active nodes and their wandering range.

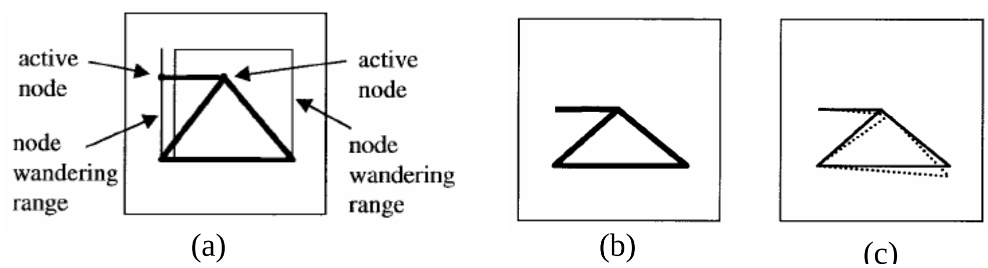


Figure 5.2 (a) The optimized geometry generating using the topology synthesis. This topology serves as an input to the size and shape optimization where stress constraints are taken into account. (b) and (c) The results of size and shape optimization with a kinematic constraint of MA = 1:2.[10]

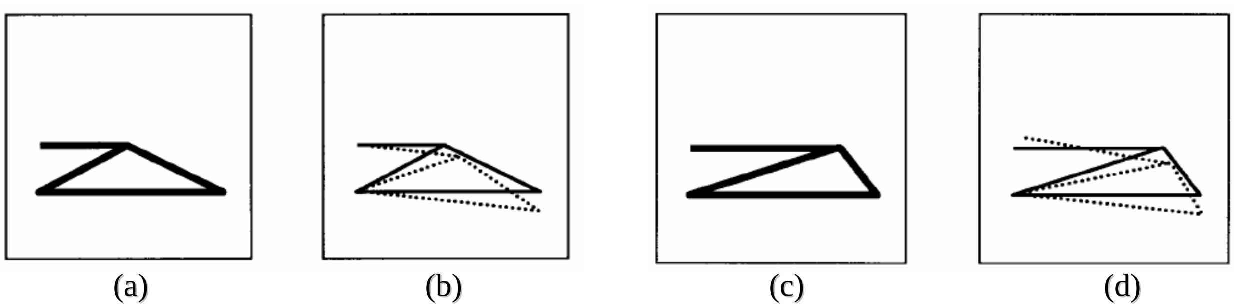


Figure 5.3 (a) and (b) The optimal compliant design with out any stress constraint. (c) and (d) The optimal compliant design with the stress constrain enforced.[10]

Figure 5.3 shows the optimal compliant design following size and shape optimization. Figures (a) and (b) show the optimal design if the stress constraint indicated in equation (4.4) is not enforced, and (c) and (d) show the optimal design if the stress constrain is enforced. It is important to note that that the mechanical advantage is the same in both cases indicated above; however there are significant differences in the efficiency.

5.2. DYNAMIC ANALYSIS

The optimization methods discussed in section 4.0 are purely based on static analysis and often do not consider the dynamic response of the device. Dynamic analysis of compliant mechanisms has largely been ignored by the research community. The authors were only able to find two papers which discussed the dynamic behavior of compliant mechanisms. Characterization of the dynamic behavior of compliant mechanisms is typically carried out on established designs.

Dynamic differential equations have be developed by Kota *et al.* to calculate the natural frequencies of compliant mechanisms. The first four natural frequencies of the stroke amplification compliant mechanism shown in figure 5.4 (a) are 3.8 kHz, 124.0 kHz, 155.5 kHz and 182.1 kHz.[6,10] The first natural mode is shown in figure 5.4 (a). Also observe the two characteristic plots which can help predicting the behavior of the mechanism at a given running speed.

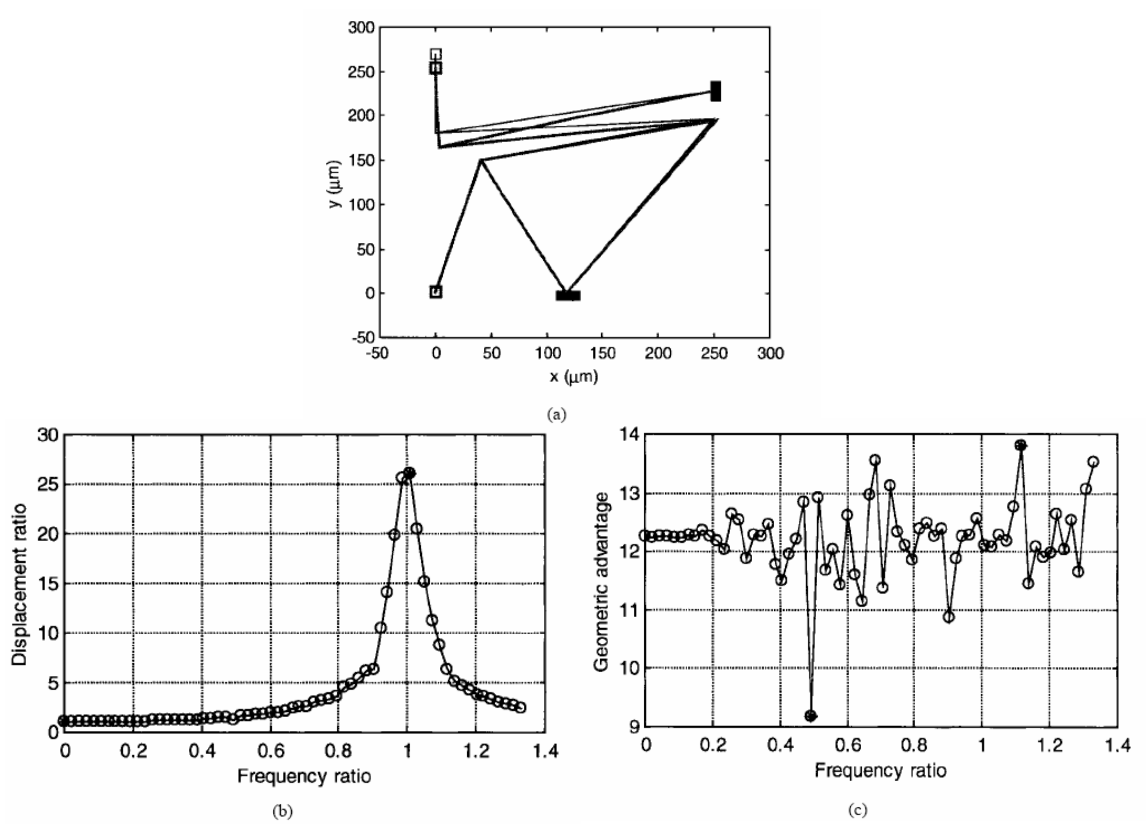


Figure. 5.4 (a) First natural mode of the compliant transmission. (b) Amplitude –frequency characteristic of the output displacement. (c) GA-frequency characteristic of the compliant transmission.[10]

Since the fundamental frequency is much smaller than the higher order frequencies, we can say that the motion shape is dominated by the first natural mode. Displacement ratio in figure 5.4 (b) is the ratio of actual displacement to the static displacement of the output is plotted against the frequency ratio (ω/ω_n). These results indicate that the displacement is amplified by a factor of ~25 if the device is operated near the resonant frequency which is to be expected. Figure. 5.4 (c) depicts the geometric advantage as a function of the frequency ratio. It can be seen that GA fluctuates around the static design result of 12:1 and is very close to that at lower frequencies. Thus if one needs to maintain the designed mechanical advantage then it is advisable that the device be operated at frequencies much less then the natural frequency.

6. EXAMPLES OF COMPLIANT MEMS

The following examples represent a small cross section of how compliant mechanisms are being implemented into MEMS devices. As indicated below considerable improvements can be realized by incorporating compliant mechanisms with micro-scale actuators.

6.1. Double V-beam Suspension for Linear Micro Actuators

The double-V-beam design shown in figure 6.1 provides an order of magnitude higher ratio of transverse to longitudinal stiffness (Ky/Kx) and a 30% longer stroke with linear force-displacement compared to the conventional folded-beam elastic suspension used in MEMS.[10] Improvements in the ratio of the transverse to longitudinal stiffness ensure that the electrostatic actuator is stable for a wider range of applied voltages and therefore a much larger range of displacements.

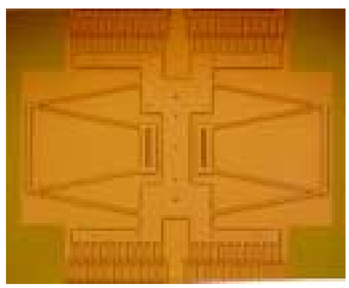


Figure 6.1 Double V-beam suspension used in electrostatic resonator.[10]

6.2. MicroEngine- Electrostatic Actuator

In the electrostatic actuator shown in figure 6.2 the addition of compliant stroke amplifier produced several improvements. 80% of the actuator size was reduced going from non-compliant to compliant.[2] Reduction in size results in reduction of production cost. The power density of the system increased by a factor of 100 and there were reduced stiction problems due to smaller size.

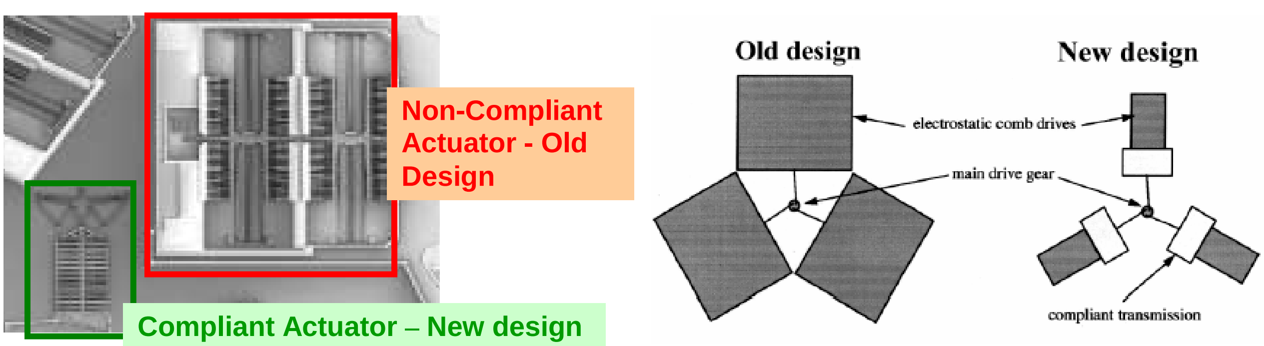


Figure 6.2 Micro-drive Electrostatic actuator and Comparison of compliant verses non-compliant design.[10]

6.3. HEXFLEX NANO-MANIPULATOR: ADAPTIVE COMPLIANCE

The HexFlex Nano-manipulator shown in figure 6.3 is a planer compliant mechanisms that is capable of generating 6-degree of freedom motion with nanoscale resolution. The HexFlex employs adaptable compliant mechanisms that allow for a wide rage of geometric advantages. Thus a single mechanism can complete both coarse motion (on the order of millimeters) while maintaining nanoscale resolution.[7,8] This is a truly powerful tool for nanotechnology. The Culpepper group at MIT is developing the individual elements of the technology such as new mechanism designs, software/analysis tools, actuators and manufacturing processes, for the design/manufacture of the next generations of these mechanisms.

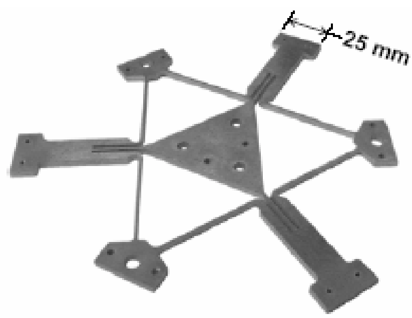


Figure 6.3 HexFlex 6-degree of freedom nanomanipulator.[7,8]

7. AVAILABLE SOFTWARE

A few Matlab and web based simulation software programs are available to assist in designing aspects of the complaint mechanisms. The research group of Prof. G. K. Ananthasuresh has developed *PennSyn 1.0* which generates compliant topologies, animates the resulting motion, creates an IGES file of the solution, and stores the solution for later use. It is implemented using Matlab 5.3. The *PennSyn 1.0* software package is available on the web at http://www.seas.upenn.edu/~gksuresh/mylinks.html.

Professor Ole Sigmund at the Technical University of Denmark (DTU) and his group has developed a web-based topology optimization program *TOPOPT*. This program solves the general topology optimization problem of distributing a given amount of material in a design domain subject to load and support conditions, such that the stiffness of the structure is maximized. The type and size of problems that can be solved using the web-based TOPOPT program are very limited. To access the program, link to http://www.topopt.dtu.dk/. Prof. Sigmund's group has also developed TopOpt3D program for 3D Stiffness optimization using Topology Optimization and *top.m, a* 99 line topology optimization code written in MATLAB. This is developed for engineering education in structural optimization. It can be used to do extensions such as multiple load-cases, alternative mesh-independency schemes, passive areas, etc.[16]

8. THE COMPLIANT MEMS COMMUNITY

Institution	University of Michigan
Lab	Compliant Systems Design Laboratory
Faculty	Prof. Sridhar Kota
Research	a. Disk Drive Microactuation b. Compliant stroke multiplier in MEMS

Institution	Brigham Young University
Lab	Compliant Mechanism Research
Faculty	Prof. Larry L. Howell
Research	a. Thermal Actuators b. Micro-Compliant Clutches c. Bistable Mechanisms d. Compliant Pentographs

Institution	Univ. of Illinois at Chicago
Lab	Micro Systems Mechanisms and Actuators Laboratory (µSMAL)
Faculty	Prof. Laxman Saggere
Research	Compliant Micromanipulator for Microassembly

Institution	Univ. of Penn.
Lab	Computational Design
Faculty	Prof. (Suresh) G. K. Ananthasuresh
Research	a. Electro-Thermal-Compliant (ETC)micro devices b. Micromanipulation

Institution	M.I.T
Lab	Precision Compliant Systems Lab
Faculty	Prof. Martin L. Culpepper
Research	3D Six-axis compliant mechanisms (HexFlex Nanomanipulator)

Institution	Technical Univerisity of Denmark (DTU)
Lab	Topology optimization
Faculty	Prof. Ole Sigmund
Research	Algorithms and software for Topology optimization

9. CONCLUSION

The development of MEMS devices is currently limited by the displacement and force output characteristics of microscale actuators. Compliant mechanisms enable the tailoring of actuators to meet the design and performance constraints and are thus important in MEMS devices. These mechanisms allow for amplification of output forces or displacements in a user defined manner providing greater flexibility and precision. Integration of compliant mechanisms with MEMS devices has potential for nanoscale manipulation with high spatial resolution and minimization of hysterisis. Improvement of performance characteristics of the MEMS actuators using compliant mechanisms will help realize MEMS-based microscale robotics and bio-MEMS. The design and implementation of compliant mechanisms is in its infancy and more research is necessary.

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