SYNTHESIZING HIGH-PERFORMANCE COMPLIANT STROKE AMPLIFICATION SYSTEMS FOR MEMS

Category: Electronic Design, Modeling, and Synthesis Aids

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ABSTRACT

We have recently designed, fabricated, and demonstrated a new class of compliant stroke amplification mechanisms that are exceptionally well suited for microelectromechanical system (MEMS) applications. Manufactured in Sandia's advanced 5-level surface micromachining technology known as SUMMiT-V [1,2], these computer generated structures provide high work and area efficiency in designs that are highly compatible with the fabrication process (see Figure 8). The actual devices display outstanding yield, robustness, endurance, and resistance to surface adhesion effects during the final release process. One device has been driven to a 20-μm output displacement at resonance for more than 10^{10} cycles with no apparent fatigue. This paper focuses on the unique methodology employed to design and analyze these compliant stroke amplification systems. The same approach, however, can be used to design many other compliant structures for fabrication in a MEMS technology. Compliance in design leads to creation of jointless, no-assembly, monolithic mechanical device [3].

INTRODUCTION

Even though the SUMMiT-V process provides for batch fabrication of jointed micro-mechanisms [1], compliant structures can offer superior alternatives in cases where joint play becomes an issue. Although simple deformable (compliant) structures such as beams and diaphragms have performed adequately in many micro devices, more sophisticated micromechanical functions can be realized by fully exploiting the preferred uses of elastic deformation via compliant mechanisms. Besides, the small scale and high aspect ratio of micromechanical structures makes them inherently flexible. Therefore, in MEMS, a compliant design that needs no assembly is not merely a prudent choice it is a necessity.

Nature has realized the pivotal role that compliance plays at the realm of microorganisms, the level at which MEMS fit. Nearly 90 percent of living creatures are invertebrates and the percentage of invertebrates increases as we go down the dimension scale where compliant structures reign [3].

Traditionally, engineered artifacts are designed to be strong and stiff. Designs in nature, on the other hand, are strong but not necessarily stiff – they are *compliant*. Compliant mechanisms are single-piece flexible structures that deliver the desired motion by undergoing elastic deformation as opposed to jointed rigid body motions of conventional mechanisms. Compliance in design leads to jointless, no-assembly (Figure 1), monolithic mechanical devices and is particularly suited for applications with small range of motions. The Compliant stapler shown in Figure 1 illustrates this paradigm of no-assembly. Conventional flexural mechanisms employ flexural joints that connect relatively rigid links as depicted in Figure 2. Reduced fatigue life, high stress concentration and difficulty in fabrication are some of the drawbacks of flexural joints. Our focus is on designing complaint mechanisms with *distributed compliance* which employs flexural links (see Figure 3) and have no joints (neither pin nor flexural joints) for improved reliability, performance, and ease of manufacture. Distributed compliant mechanisms derive their flexibility due to topology and shape of the material continuum rather than concentrated flexion at few regions.

In this paper, we discuss a systematic method of design of compliant micro mechanisms and present a stroke-amplification mechanism for MEMS application as a design example. In particular, we present a brief overview of mathematical procedures employed for design of compliant mechanisms for: (i) Topology synthesis - which involves generation of a functional design in the form of a feasible topology starting from input/output force/motion specifications, and (ii) Size and shape optimization - to meet performance requirements such as max stress, motion amplification or force amplification etc.

DESIGN OF COMPLIANT MECHANISMS

The first step in the design of a compliant mechanism is to establish a kinematically functional design that generates the desired output motion when subjected to prescribed input forces. This is called topological synthesis. Although the size and shape of individual elements can be optimized to a certain extent in this stage, local constraints such as stress and buckling constraints cannot be imposed while the topology is being determined. Once a feasible topology is established, performance constraints can be imposed during the following stage in which size and shape optimization are performed. Performance constraints may include minimizing the energy loss in the mechanism, obtaining desired motion amplification (geometric advantage) or force amplification (mechanical advantage), or ensuring that none of the elements buckle under the action of applied forces and external loads.

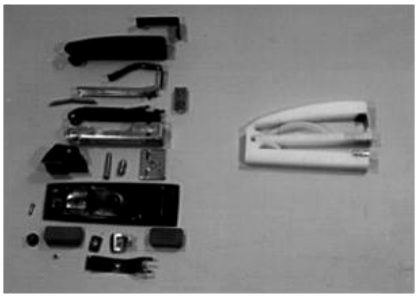


Figure 1: A Single-piece Compliant Stapler [4]

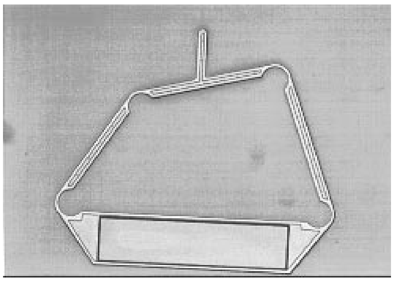


Figure 2: A micro compliant four-bar mechanism with *lumped* compliance.

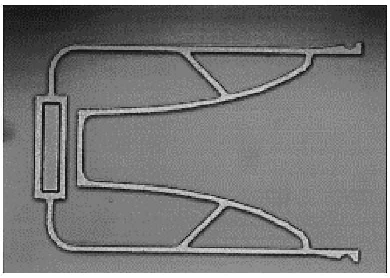


Figure 3: Micro compliant crimping mechanism with *distributed* compliance

In this section, we briefly explain systematic methods of design of compliant mechanisms starting from functional specifications. First, we describe a method of deriving the topology (configuration) of a compliant mechanism given the desired input forces and output displacements. Next, we describe the method of optimizing the size and shape of various elements of a compliant mechanism in order to satisfy prescribed mechanical or geometric advantage, stress constraints, size constraints etc. The work reported in this paper assumes linear elastic models. Readers who are interested in geometric non-linearities due to large deformation, or dynamic performance of compliant mechanisms should contact the authors for on-going research on these topics.

Topology Synthesis

The goal of this first stage in compliant mechanism design is to establish a feasible topology (configuration) to meet prescribed input-output force-displacement relationship. Although multiple input forces and multiple output displacements can be prescribed, we will describe only a single input-output case for the sake of simplicity.

Given, a single-force-input and a single-displacement-output design specifications, first, we formulate an objective function that captures the need for (a) compliance to undergo desired deformation (kinematic requirement), and (b) stiffness to resist external loads (structural requirement) once the mechanism assumes the desired configuration. We then employ a formal structural optimization technique to *synthesize* a form, which is an optimal topology, shape and size of a compliant mechanism that performs the intended function [5,6,7]

Problem Formulation

As shown in Figure 4, to satisfy both the kinematic and structural requirements in compliant mechanism synthesis, a two-part problem is posed in terms of potential energies. The first part, the "mechanism design", is where the kinematic requirements are met by maximizing the deflection at a specified point along a specified direction. This is achieved by applying a fictitious force at the point of interest, B, along the direction of the desired output deflection, Δ. This "dummy load" is denoted by f_B and as shown below for a general design domain subject to an applied force, f_A, at the point A and some specified boundary conditions. Maximizing the deflection at the point B in the direction of f_B is equivalent to maximizing the mutual potential energy, $v_B^T\ K\ u_A$, where u_A is the deflection field due to f_A, v_B is the deflection field due to f_B, and K is the global stiffness matrix. The constraints are two equilibrium equations, one due to the applied load, and one due to the dummy load.

The second part of the two-part problem is the structure design, where the structural requirements are met by maximizing the stiffness. Here the point A is considered fixed, and the resistance of the workplace is accounted for by applying the force f_B at point B in the opposite direction. Maximizing the stiffness is equivalent to minimizing the strain energy, $u_B^T\ K\ u_B$, where u_B is the deflection field due to this set of loading, and K is the global stiffness matrix. The constraint is a single equilibrium equation.

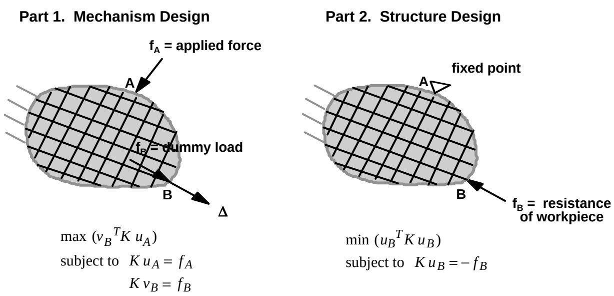


Figure 4 Illustration of problem formulation [5]

The combined problem is shown as equation (1).

Combined Problem

$$\max \left[\frac{\text{mutual energy}}{\text{strain energy}}\right] = \max \left[\frac{v_B^T K\, u_A}{u_B^T K\, u_B}\right]$$

$$\begin{aligned} \text{subject to}\quad & K\,u_A = f_A \\ & K\,v_B = f_B \\ & K\,u_B = -f_B \\ & \text{total resource constraint} \\ & \text{lower and upper bounds} \end{aligned} \qquad (1)$$

The mechanism design and structure design are then combined into a single problem via multi-criteria optimization [5]. The design example, given below, illustrates the ground structure approach using an array of beam elements. Reference [7] discusses an improved problem formulation and its implementation using beam elements.

<u>An Array of Beam Elements</u>

In this method, the prescribed design domain (this is the area within which the mechanism should fit) is first divided into a number of nodes. Each node is connected to several other nodes via modular array of <u>beam elements</u>. This serves as an initial guess. Certain nodes are "fixed" to imply the points where the mechanism is anchored to the substrate. The cross sectional area of each beam element serves as the design variable with specified upper and lower bounds. The resource constraint provides less material than the available space. The objective then is to distribute the material in a way that maximizes the objective function. During the optimization process, those beam elements whose cross sectional area reaches the lower bound are removed (deemed unnecessary) leaving only a network of beam elements whose area reached the upper bound. This defines the topology of the compliant mechanism.

The results of the automated synthesis method are illustrated by the following example of a compliant gripper mechanism (Figure 5). The design specifications are that the applied force, F, causes the motion, D, at the indicated location, which will allow the device to grip some object at that point as shown in Figure 5(a). The design domain shown in Figure 5(a) represents the upper-half view since this is assumed to be a symmetric problem without any loss of generality in the solution procedure. The dashed line represents the desired space within which the mechanism should fit.

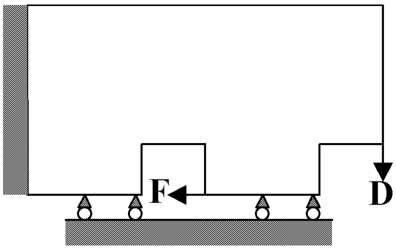
The initial guess is a modular beam structure (Figure 5(b)) with a uniform distribution of cross-sectional areas. When the algorithm converges, the solution consists of beam members whose design variable reached (or is close to) the upper bound. The beam members whose design variables reached the lower bound constraint are eliminated. The optimized solution and corresponding finite element model are shown in Figure 5(c)), where the un-deformed shape is denoted by the dashed lines and the deformed shape is denoted by the solid lines. Compliant grippers based on this design were fabricated in nylon using a rapid prototyping machine (Stratasys 3D Modeler). The methodology described here applies for three-dimensional problems as well as design problems with multiple sets of input/output force/displacements [6] requirements particularly for shape-change applications [2].

Size and Shape Optimization

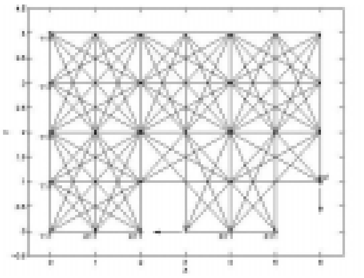
Topology optimization provides qualitative results in that it provides a kinematically functional mechanism. It cannot provide a mechanism with prescribed performance characteristics. Therefore, once the topology is established, the next logical step is to perform a size and shape optimization.

In order to produce practical compliant mechanism designs, the following design criteria must be addressed: (i) required kinematic motion (both magnitude and direction), (ii) required stiffness to an external load, (iii) design space, and (iv) materials properties. (v) Stress limitations, (vi) buckling instabilities, (vi) dynamic considerations, and (v) weight limitations.

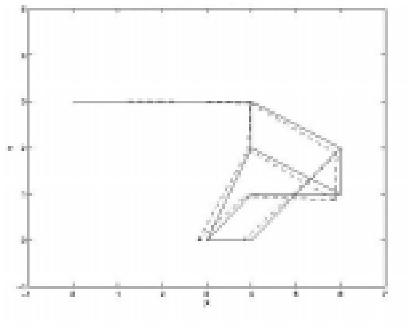
For structural optimization of compliant mechanisms, the stiffness of the mechanism must be quantified order to achieve maximum performance. A compliant mechanism absorbs energy as the mechanism deforms. Therefore, one way to quantify (and thereby optimize) the performance of a compliant mechanism is to maximize the energy efficiency. This forms the basis for determining the cross-sectional size, and shape of individual beam elements and the location of end-points of each beam element (geometry).



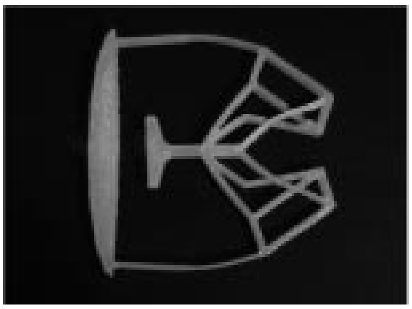
(a)



(b)



(c)



(d)

Figure 5: Compliant design methodology [7]. From (a) or (d) are the steps for synthesizing a 2-D compliant gripper. (a) Define the design problem and desired forces. (b) Create an initial guess of what the structure may look like. (c) Obtain the solution through structure optimization (deformed shape shown with dashed lines). (d) Fabricate design. Note that only half of the structure is modeled, and the design is mirrored to create the physical device.

<u>Energy Efficiency Formulation</u>

Considering a linear elastic body, work can be measured at both the input and output ports by assuming the following boundary conditions:

- The input is "actuated" by controlling the displacement of the input port on the body. The input displacement in effect controls the maximum range of travel of the mechanism.
- Work performed at the output is measured by applying an external resistive load which opposes the desired direction of the output port on the body. The external load can be considered to be a worst caseload (on an average load) arising from the environment of the mechanism.

As shown in Figure 6 below, these boundary conditions are applied in two separate stages. First, the external force is applied to the body while the input is held fixed. Second, the input is actuated a finite distance with the external load applied. Triangular regions in Figure below at the input and output illustrate the energy absorbed due to loading and flexure. The shaded areas at both ports represent the reciprocal work or a fixed kinematic relationship between the input and the output. The mechanical efficiency can be formulated as equation (2).

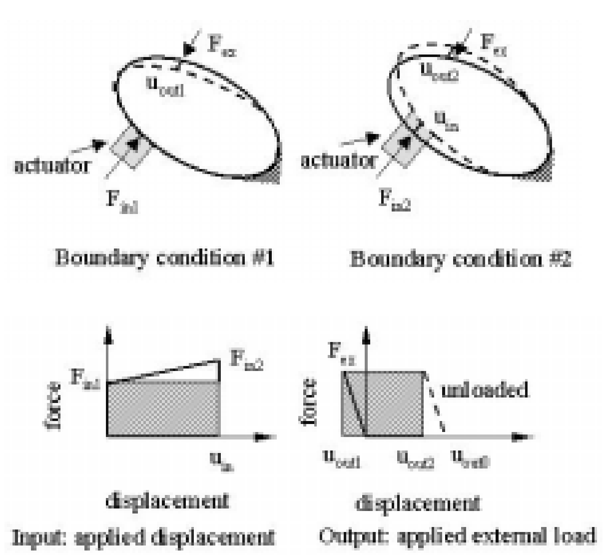


Figure 6 Energy efficiency formulation used in size and shape optimization [6]

$$\eta_{efficiency} = \frac{F_{ex}\left(u_{out2} + \frac{1}{2}u_{out1}\right)}{\frac{u_{in}}{2}\left(F_{in1} + F_{in2}\right)} \qquad (2)$$

MEMS Multiplier

One example of the size and shape optimization based on the energy formulation is a MEMS Multiplier presented in Figure 7. Prescribed performance requirements included (i) desired geometric advantage of 12, (2) maximum permissible stress, (3) minimum in-plane beam-width of 1μm, and (4) The energy efficiency of the final design is 72.9%. The mechanism amplifies the input motion by a factor of twelve; i.e. the mechanism is said to have a Geometric Advantage (G.A) of 12. Figure 7 shows an electrostatic comb-drive actuator integrated with a compliant mechanism.

The primary driver for this work was the need to create core actuation components that are much smaller than existing devices, thus saving valuable die area and considerably reducing manufacturing costs. This size reduction will also allow more complex microelectromechanical systems to be realized. One of the authors presented a paper at Transducers '99 [9], which highlighted the level of complexity that has already been achieved with 5-level surface micromachining. Not highlighted was the fact that the most complex of the systems presented already requires 2/3 of the available die width, therefore greater complexity cannot be achieved without making the micro-components even smaller (Figure 8).

<u>Dynamic Analysis</u>

The energy efficiency method described above is based on kinetostatic design specifications and does not consider the dynamic effects. For high-speed applications, the dynamic analysis is necessary in order to predict the dynamic characteristics of the mechanism. The dynamic differential equations of the mechanism are derived using the finite element method and has the form of

$$[M]\{\ddot{D}\}+[K]\{D\}=\{R\} \qquad (3)$$

where $[M]$ and $[K]$ are system mass and stiffness matrix respectively, $\{D\}$ is the generalized coordinates representing the translation and rotation deformations at each element node, $\{R\}$ is the generalized force corresponding to $\{D\}$. Through mass normalized principal coordinate transformation, the differential equations (3) can be decoupled as

$$\ddot{z}_i + 2\xi_i\omega_i\dot{z}_i + \omega_i^{\,2} z_i = p_i \qquad (i=1,2,\ \dots\ ,n) \qquad (4)$$

where z_i and p_i are the ith normalized coordinate and force, ξ_i is the viscous damping coefficient added to the decoupled equations according to the convention (ξ_i =0.02 is used in calculations), ω_i is the ith circular natural frequency of the system, n is the total number of elastic degrees of freedom of the system. The differential equations can be used for frequency and mode analysis, dynamic responses and spectrum analysis.

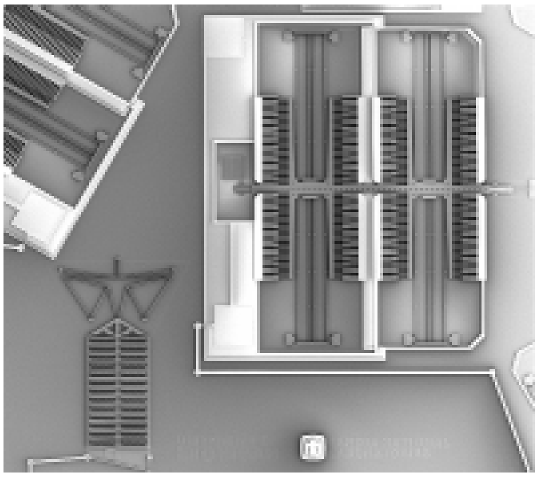


Figure 7 Compliant based actuation system (short stroke comb drive with stroke amplifier) is considerably smaller than the comb drives currently used (patents pending).

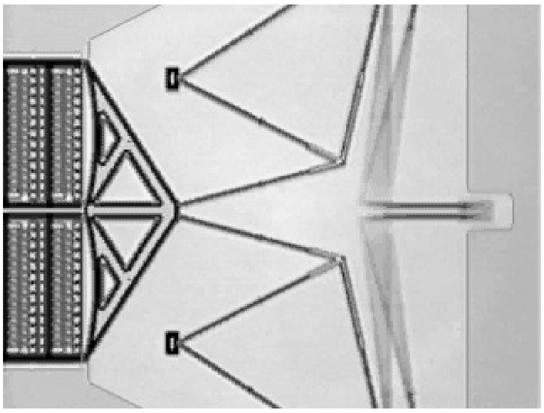


Figure 8: Optical micrograph showing blur envelope created by the output beams of an actual device that is being driven at resonance. This combination has an overall system resonance of 26.9 kHz. Total output displacement is approximately 20 μm. Higher magnification is required to accurately determine input displacement (patents pending).

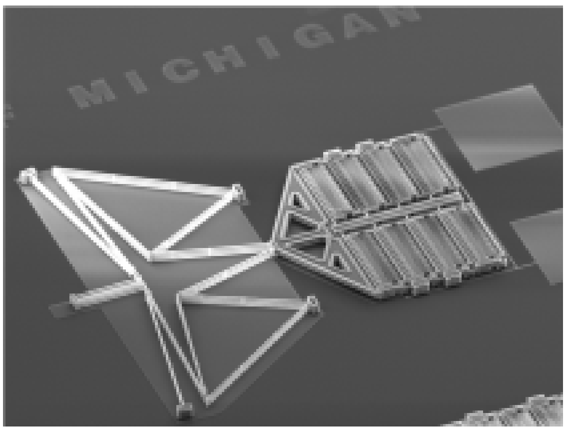


Figure 9: The electrostatic actuator combined with the stroke multiplier increased the force per unit area by 220 times. Note the size of the actuator relative to the size of the bond pad (patents pending).

The first four natural frequencies of the stroke amplification compliant shown in Fig. 9 mechanism are: f_1 = 3.88 KHz, f_2 = 124 KHZ, f_3 = 155 KHz, and f_4 = 182 KHz.

Figure 10 shows the characteristic of geometric advantage (GA) of the compliant multiplier based on our dynamic simulation model. The geometric advantage is the ratio of the output to the input displacement. The static displacement of the output in this case is approximately 12 times the input displacement, i.e.; the GA is 12 at low frequency. The operating frequency of this mechanism is about 0.5 times the natural frequency, at which point the GA is nearly 20. More advanced designs are now in fabrication, and the latest results will be included in the conference presentation.

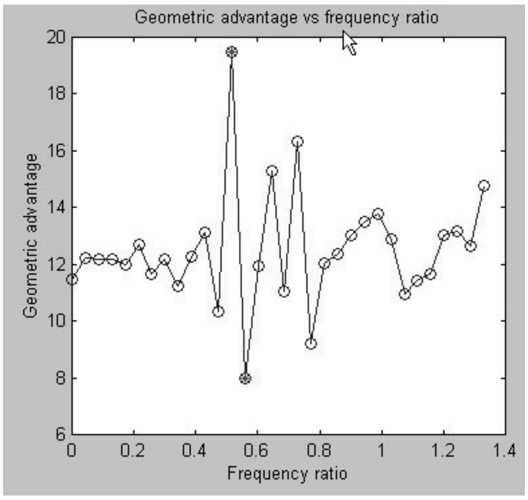


Figure 10. Dynamic analysis of the stroke amplifier shows the geometric advantage as a function of the frequency ratio.

CONCLUSIONS

Complaint mechanisms play an important role in the design of micro mechanical structures for MEMS applications. These monolithic mechanical structures can be designed to perform complex mechanical functions and fabricated within the constraints of present day micromachining processes [9]. Based on linear elastic models, we have developed methods of synthesis of compliant mechanisms to meet kinematic and static stiffness requirements. Our future work includes non-linearities due to large deformation, and dynamic aspects of micromechanical structures. The design methods developed to-date can generate micro mechanism designs for a variety of applications including, motion/force amplification, static shape change, and multiple input/output force-displacements.

Interested readers should refer to the following references for more details on various steps in the design synthesis methodology and different classes of problem formulations. Ref [5] provides a multi-criteria formulation for topology synthesis and its implementation using truss ground structures. Ref [6] presents the size and shape optimization method using energy efficiency formulation including design of stroke amplification mechanisms. Ref [7] presents an improved problem formulation for topology synthesis and its implementation using beam elements. Ref [10] provides a overview of some of the early work on design of compliant mechanisms using homogenization method [9] and its applications to MEMS.

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