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SYNTHESIS OF ADAPTIVE AND CONTROLLABLE COMPLIANT SYSTEMS WITH EMBEDDED ACTUATORS AND SENSORS

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ABSTRACT

This paper presents a framework for the design of a compliant system; that is, the concurrent design of a compliant mechanism with embedded actuators and embedded sensors. We focus on methods that *simultaneously* synthesize optimal structural topology and placement of actuators and sensors for maximum energy efficiency and adaptive performance, while satisfying various weight and performance constraints. The goal of this research is to lay a scientific foundation and a mathematical framework for distributed actuation and sensing within a compliant active structure.

Key features of the methodology include (1) the simultaneous optimization of the location, orientation, and size of actuators *concurrent* with the compliant transmission topology and (2) the concepts of controllability and observability that arise from the consideration of control, and their implementation in compliant systems design. The methods used include genetic algorithms, graph searches for connectivity, and multiple load cases implemented with linear finite element analysis. Actuators, modeled as both force generators and structural compliant elements, are included as topology variables in the optimization. Results are provided for several studies, including: (1) concurrent actuator placement and topology design for a compliant amplifier and (2) a shape-morphing aircraft wing demonstration with three controlled output nodes. Central to this method is the concept of structural orthogonality, which refers to the unique system response for each actuator it contains. Finally, the results from the *controllability* problem are used to motivate and describe the analogous extension to *observability* for sensing.

INTRODUCTION

The basic premise of a *compliant system* is the integration of <u>motion/force transmission via elastic deformation</u> with <u>embedded actuation and sensing</u>. Current electromechanical systems are generally molded in the rigid-and-discrete paradigm where one first designs a rigid structure with mechanical joints and then adds actuators and sensors, with the design of controls only following as an afterthought. In spite of the "mechatronic revolution" of the early 1990s, this paradigm is still prevalent today. The goal of this research is to lay a scientific foundation and a mathematical framework for distributed actuation and sensing within a compliant active structure. In past studies of compliant mechanisms (CMs) and their synthesis, single-actuator mechanisms have been considered, with the determination of the actuator's type, orientation, size, and location occurring *outside* of the automated design synthesis, at the designer's option. The objective of this research can be stated as a *systems* approach to synthesis of mechanism, actuator, and sensors, thereby advancing the path from traditional mechanical design to *systematic* compliant-mechatronic design, as graphically depicted in Fig. 1. The steps made include systematic design tools and algorithms. We are developing methods that *simultaneously* synthesize optimal structural topology and placement of actuators and sensors for maximum energy efficiency and adaptive performance. From these we seek specific insights that will lead to general engineering design guidelines for embedded systems.

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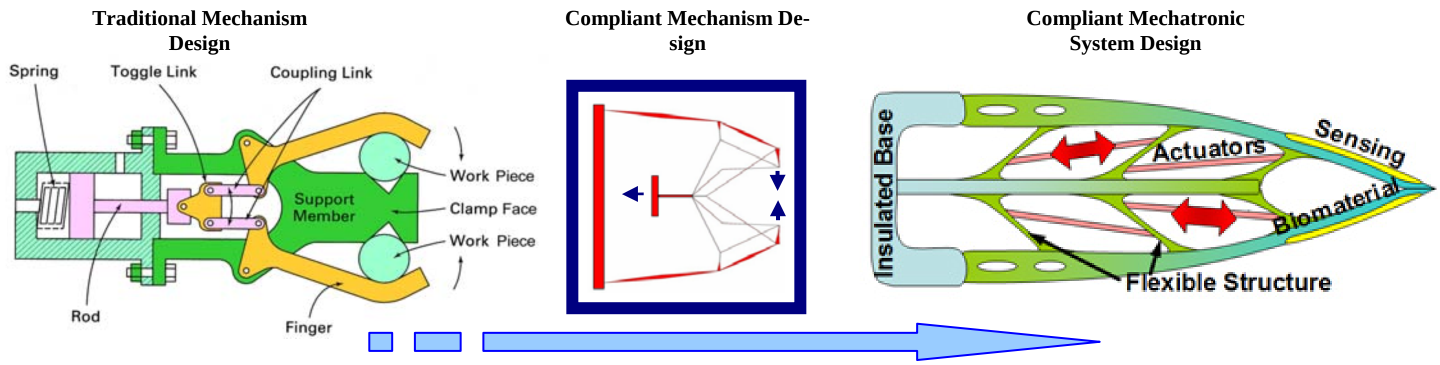

Figure 1: Transition from "Traditional Mechanical Design" to "Mechatronic Design with Distributed and Embedded Compliant Systems". Current State-of-the-Art is "Compliant Mechanism Design".

The physical layout of these components is optimized for energy efficiency, controllability, and responsiveness, while satisfying various weight and performance constraints (e.g. for autonomous robots). Figure 3 shows an example of the problem definition and a conceptual solution generated algorithmically based on the proposed synthesis framework. Allowing for multiple inputs in compliant mechanism design also leads to the question of control. A method for incorporating consideration of control *during* the design synthesis can enhance the controllability and responsiveness of an adaptive system. Central to this method is the concept of structural orthogonality, which refers to the unique system response for each actuator it contains.

The focus of this research lies at the intersection of four system components: structure, actuation, sensing, and control (see Fig. 2). While the ultimate goal is achieved by simultaneously integrating all of these in the design synthesis process (**A**+**B**+**C**+**D**), we have broken these down and focused on the interaction of the individual components.

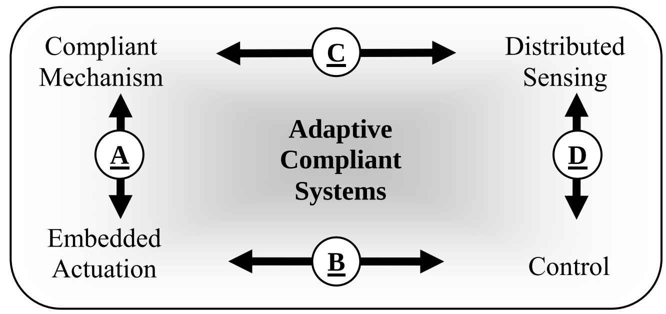

Figure 2: Grand Vision for Synthesis of Adaptive Compliant Systems (A, B, C, and D refer to *interactions*)

We are currently investigating these basic research issues: *(1)* optimal layout of the compliant structure and actuators (**A**), *(2)* a framework for embedded controls in adaptive compliant systems (**A**+**B**), and *(3)* the logical extension of the embedded *actuator* framework to create a parallel method for a distributed *sensor* network (**C**+**D**). In all of these, we seek a generalized synthesis scheme in which the design requirements are captured in a mathematical form to transform an initial grid of elements into an optimal layout of elastic beams, sensors, and actuators. This paper presents a specific methodology for the first two issues, **A** and **A**+**B**, along with numerical results and discussion. The extension to **C**+**D** is then explained in theory within the framework of synthesis and its implementation is beyond

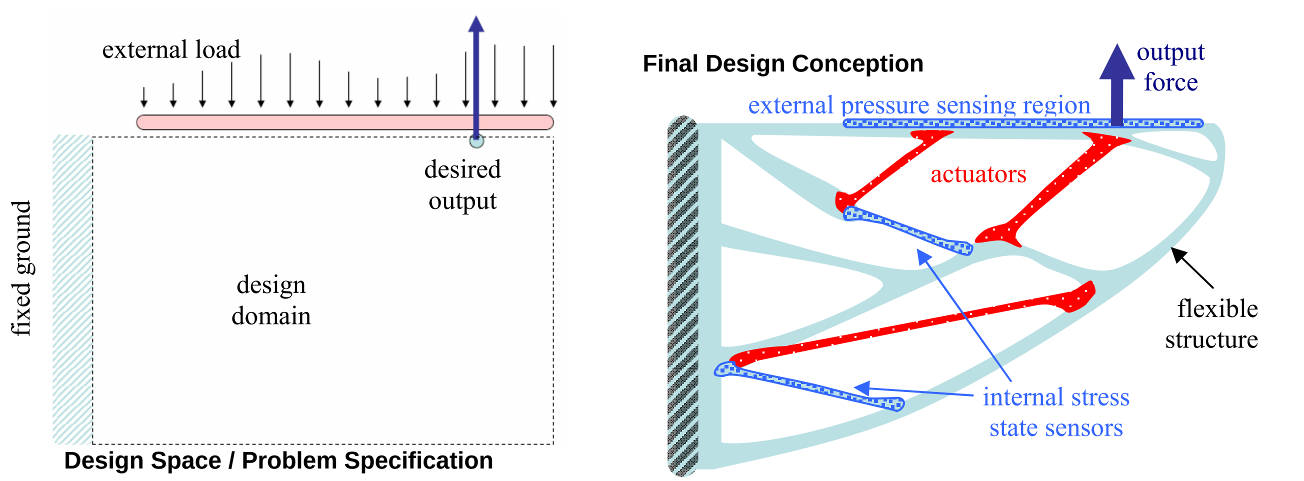

Figure 3: Vision of an Internalized and Adaptive Compliant System. Beginning with a specified design space, external loading conditions, and desired mechanical task, we seek to synthesize a fully-compliant system with embedded and distributed actuators and sensors.

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the scope of this paper.

Motivation

The motion-and-force-transmission part of the compliant system problem has been researched for more than a decade in the field of jointless monolithic devices called compliant mechanisms (CMs). These devices use flexure and deformation to transmit motion and force, rather than rigid bodies with conventional mechanical joints, providing several benefits such as elimination of mechanical joints, joint-wear, and joint-clearance. This design paradigm is inspired by nature, where strength and *compliance* are observed in its designs, as opposed to the goal of strength and *stiffness* employed in traditional engineering. In fact, we refer to such systems in terms of both mechanism *and* structure; in this paper, both conceptions are used. The interchangeability of these terms highlights the significance of the multiple roles played by this technology and blurs the distinction between a (flexible) structure and a (jointless) mechanism.

This research extends the biological analogy to the creation of truly monolithic *systems* – <u>autonomous, adaptive, efficient, self-contained devices</u>. The design methods and the applications of CMs already apply to many domains in micro-, meso-, and macro-scales. The past success with CMs motivates a new level of sophistication in synthesis of <u>compliant systems</u> including controls and adaptive structures. Frecker [1] illustrates the gaps in knowledge amongst these synergies. Her survey of the fields shows that work has been done in (1) actuator placement on non-compliant predetermined structures, (2) the actuator material distribution problem, and (3) the coupling structure for a prescribed actuator, which are incremental steps in the progression of adaptive structure technology. However, the topic of systems-level *concurrent* synthesis of actuator component placement and topology design, a fundamentally different approach to the design of smart mechatronic systems, has yet to be addressed. By combining these fields, a new paradigm emerges for comprehensive designs that offer a basis for autonomy via the improved energy efficiency and adaptability that the inherent compliance provides.

Benefits

The benefits of this reported work are two-fold: (1) those that arise from the actuator placement paradigm and (2) those that arise from the concepts of controllability and observability. The former improves prior compliant mechanism design with better efficiency and reduced actuator size and power requirements. Such design also takes place in a greater, unconstrained design space, and allows for more complex deflections and shape-changes. The concepts of controllability and observability (degree to which the external environment can be sensed) arise from the consideration of control – a natural question after the leap from single to multiple actuator structures. These concepts represent the structural adaptability of the system. The use of a compliant mechanism and structure then enables increased sensitivity and output function with fewer actuators/sensors. In other words, a reduced number of *internalized* sensors and actuators can effect and sense a large range of *externalities* over a large physical region.

The use of multifunctional elements as both structural members and actuators, the integration of multiple materials, and the freedom from dependency on external drives are specifically attractive to many fields. Many biomedical applications require these qualities, including surgical tools and grippers, active-assist joints, and active valves. At a larger scale, we can create sense-and-control response within compliant shape-morphing aircraft wings. Other applications, proposed in the Future Research section, include variable stiffness design and adaptive-fit orthotics and prosthetics.

Background

Compliant mechanisms can be designed for any desired input-output force-displacement characteristics including specified volume/weight, stiffness, and natural frequency constraints. As flexure is permitted in these mechanisms, they can be readily integrated with unconventional actuation schemes including artificial muscles, thermal, electrostatic, piezoelectric, and shape-memory-alloy actuators. The synthesis of these mechanisms has been well studied [2-5], and their advantages have been well documented, including: energy storage, ease of manufacture, and absence of wear, backlash, and friction. Synthesis typically involves two stages: (I) generation of the mechanism topology and (II) determination of optimum size, geometry, and shape of various constituent elements of the mechanism. Starting with functional requirements of desired forces and displacement, a conceptual design is automatically created in Stage I topology synthesis. Based on material constraints (i.e. permissible stress, strain), fabrication constraints (minimum feature size, etc.), external loads, and desired mechanical advantage, the exact size, shape, and geometry of each of the beam elements are optimized in Stage II.

Distributed Compliance. A significant difference exists between the CMs discussed in this paper and conventional flexures. Conventional flexures have relatively rigid sections connected by very thin flexural joints. These flexures localize the deformation and are prone to high stresses and reduced fatigue life. Such flexures have been known for a long time ([6], 1965) and have been successfully employed in less-demanding applications (ex. shampoo bottle lid). The mechanisms discussed here have distributed compliance, i.e. they deform as a whole and do not have any joints, flexural or conventional. The property of distributed compliance has enabled the study of compliant mechanisms for use in shape-morphing applications [3,7-8] that are predecessors of this current work.

Controllability and Observability. There is a collection of related literature regarding the optimal placement of actuators and sensors for acoustics and vibration control, which is summarized in [9]. While many of these studies use genetic algorithms (sometimes simulated annealing), they focus on structure rather than mechanism (motion transmission) problems. Several researchers have investigated the placement of actuators on vibrating plates [10,11] and at least one [12] considers actuator and sensor placement on simple, fixed structures (1 to 3 beams), also for vibration suppression. Most authors examined only fixed topologies, except for the case of Begg and Liu, who did investigate simultaneous topology optimiza-

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tion and actuation placement for cantilevers [13] and trusses [14]. Again, these are for vibration suppression in structures, not mechanical control of compliant systems. It remains to be seen whether the same metrics used for controllability and observability of smart structures for vibration suppression can be applied to compliant systems, which do not suppress vibration but perform prescribed mechanical tasks.

Actuator Architecture. There are a few literature references to the optimization of *actuator architecture*, which are similar in appearance to some aspects of the reported research. However, these methods are of different scope and employ different methods. Actuator architecture problems aim to lay out the material makeup of an actuator, but do not address how that actuator relates to the rest of a physical system. Most of these studies focus on the distribution of only the actuator material, without any supporting structural elements. (Silva: piezoelectric actuator material distribution, [15], and Anusonti-Inthra & Frecker: actuator material layout in a flexible airfoil, [16].) Their results show networks of only actuators. Such a structure may be suitable when considering the entire device as a single actuator component, but is highly impractical in describing implementation of discrete actuator components within a mechanical system. Bharti and Frecker [17] consider the passive and active material distribution problem. However, the choice of actuator or structure is not a variable but a post-processing decision, and again, the results are not applicable to practical component layout. Their results are better interpreted as the layout of passive and active material within a single actuator component. However, the need for discrete actuator placement is stated, but no method or rationality is yet provided for how a systematic and accurate interpretation is to be made. Such limitations in the related research further motivate our pursuit of this current research.

Some readers may also be familiar with the concurrent role of actuators and structure within tensegrity systems [18]. However, such systems are fundamentally different from our research in material, methodology, and suitable applications. By their nature, tensegrity structures consist of axially-loaded cables and struts and are often used in self-deployment applications, while compliant systems consist of compact structures performing mechanical tasks based on the controlled bending of beams.

Concurrent Actuator Placement. Bharti et al. [19] and Johnson and Frecker [20] are also investigating optimal multiple actuator placement. They employ different elements types than presented in this paper for their structure (i.e. trusses and tensegrity structures instead of beams) and determine active (piezoelectric) and passive material distribution within a structure by using genetic algorithms.

OVERVIEW OF METHODOLOGY

The basic outline of the research is shown in both a general problem flowchart (Fig. 4) and a pictorial example (Fig. 5). First, the design specifications are identified, from which the problem type is determined (Fig. 5(a)). The available solutions to the problem potentially contain compliant mechanism, actuator, and sensor components. The design space must be properly parameterized in terms of these components (Fig. 5(b)), which is critical for the success of the optimization. The high-level goal of optimization in the embedded actuator problem (**A**) is to minimize actuator number and size, to maximize energy effi-

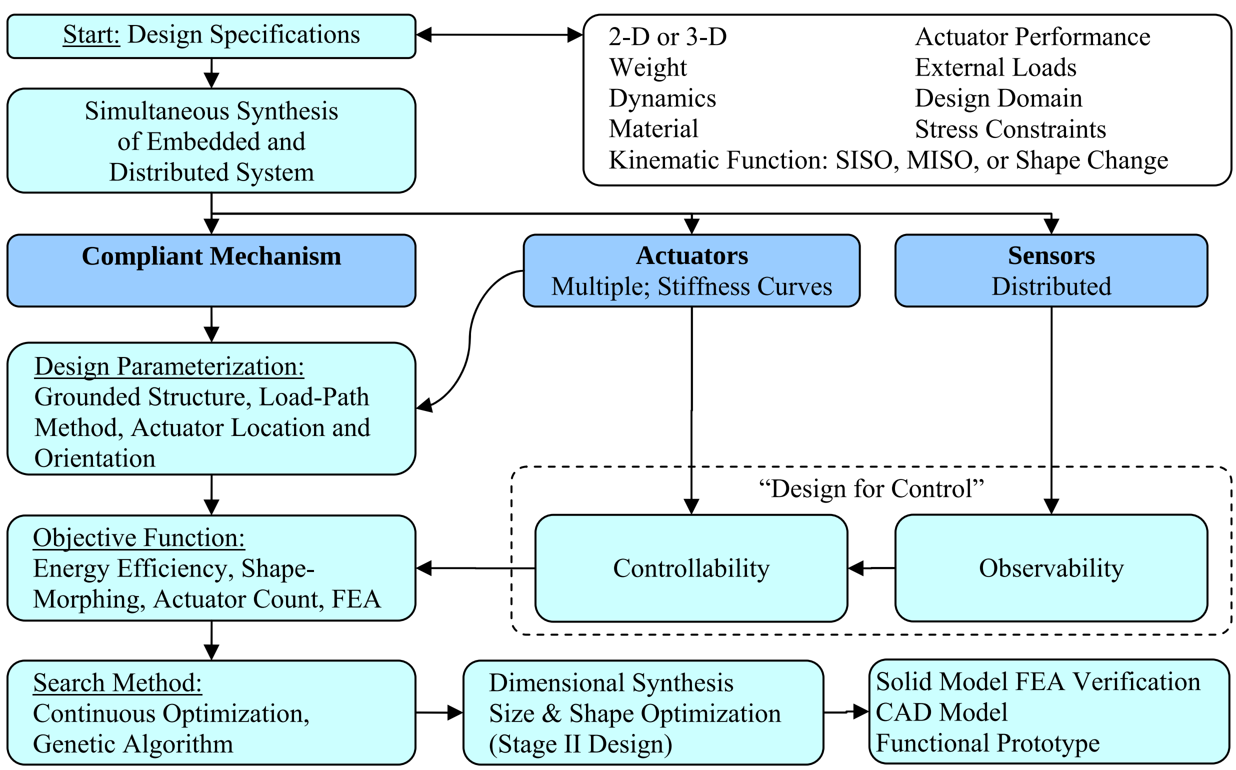

Figure 4: Overview of the Synthesis Method

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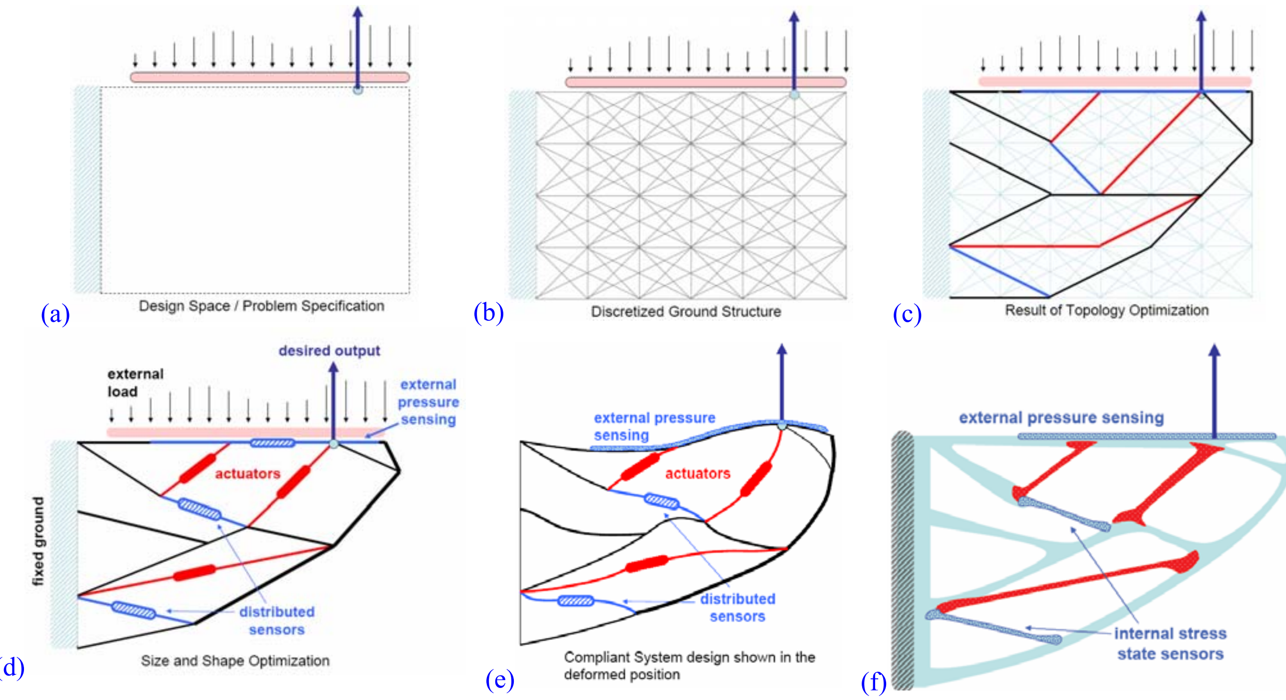

Figure 5: Various steps in the synthesis scheme. (a) design specifications (b) initial array of beam elements as a ground structure (c) optimized topology of beams, actuators, and sensors (d) size optimization (e) deformed position (f) physical interpretation.

ciency, and to meet other constraints such as size, weight, displacement, and stress constraints. For that, we intend to include other components (actuators and sensors (**A**+**B**+**C**+**D**)) to achieve true *system* optimality. Application of a search method finds the optimal system (Fig. 5(c)). Results are further refined in size and shape optimization (Fig. 5(d)). Finally, there will be verification of the design and methodology via solid model finite element analysis and testing of functional prototypes (Fig. 5(e,f)). This methodology is now illustrated via example.

Example 1 – Multiple Actuator Placement

We report two examples of the methodology in use. The first is the traditional task of displacement amplification, accomplished with multiple actuators. The second task is a controllable shape-adaptive wing, possible only within the new multiple actuator formulation. Both problems use the same genetic algorithm (GA) for optimization and Grounded Structure Approach (GSA) [4, 21] for parameterization. The next section describes all the steps required for the first problem. The following section then discusses the second problem and the associated changes from the first method.

Parameterization by the Discrete Grounded Structure Approach. In the code, the workspace is broken down into a grid of nodes, which are interconnected by beam elements (Fig. 6). A design variable exists for every element and determines whether or not the element exists. A value of 0 deactivates the element, removing it from the structure; other values represent three discrete thickness choices. In addition, there is a variable for every actuator to be used in the structure. This variable has a value between 1 and the total number of elements. Its value marks the element selected to be the actuator in a given structure. This approach guarantees a specified number of actuators for each problem and will prevent results like those seen in related literature. However, when using the GSA with a GA, a number of connectivity problems can occur; a variety of graph-checking constraints are added to resolve the issue.

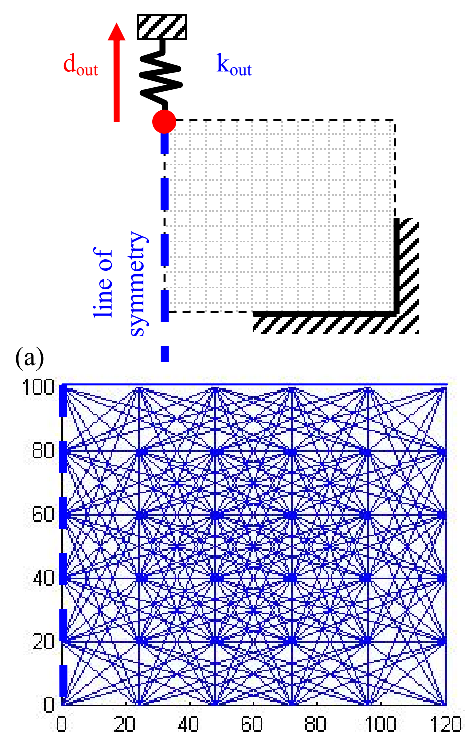

Figure 6: Design Domains and Discretization for Symmetric Amplifier Problem, 6x6 grid, 250 elements

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Objective Functions. A key aspect of the research is implementing the proper objective function to achieve meaningful results. The choice of objective function depends on the type of problem being solved, which is described as single-input-single-output (SISO), multiple-input-single-output (MISO), or multiple-input-multiple-output (MIMO, shape-change, adaptive, etc.) Most objectives include energy efficiency with regards to input work and output work. The maximization of energy efficiency leads to distributed compliance in our structures, and the new methodology allows for more complex constraints, motions, and shape changes. Other objective functions may be applied to minimize actuator power consumption, minimize the total number of actuators used, or to maximize the combined effect of multiple actuators. Shape-change objectives can be implemented via least squares formulations, by calculating the deviations of points on a deflected structure from corresponding points on a target curve. Additional objective function terms are available to maximize responsiveness and adaptability and to minimize total length of all elements present (much like a volume constraint). At this time, all objective function terms are calculated from the results of linear finite element analysis (FEA), with only one beam element placed between each of the nodes in the topology being analyzed. Finally, actuator segments serve as both force generators and structural elements.

In the first example, the objective function is to maximize energy efficiency, expressed in term of a spring-based efficiency, η. That is, while the actuators act against the structure at the input ports, springs (k_{out}) are placed at the output ports to simulate the reaction loads (See Fig. 7).

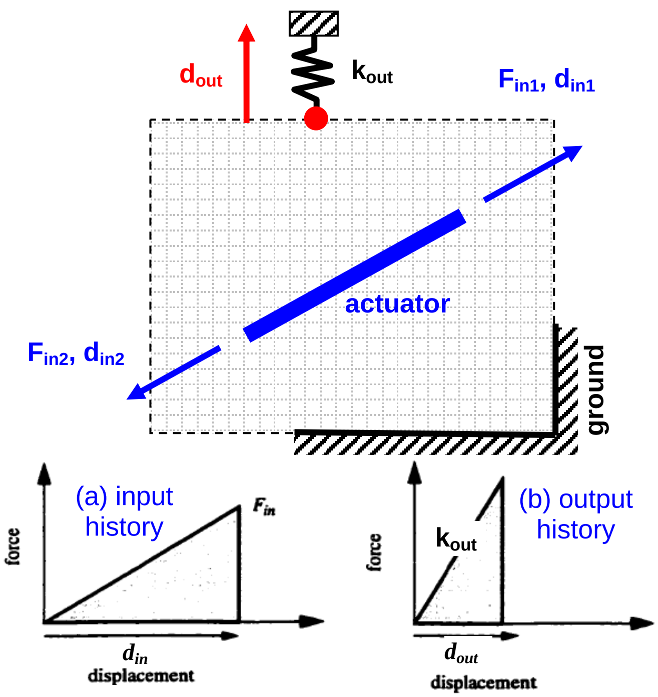

Figure 7: (Top) Basis for calculation of Energy Efficiency. (Bottom) The Input and Output Work History (Spring Formulation) (Hetrick, 1999)

This enables the easy calculation of work output (W_{out}) divided by work input (W_{in}) in Eq. (1), where all forces and displacements are measured parallel to the intended direction of motion. Thus, this formulation does not include output work that does not contribute to the mechanism's function. Hetrick [21] provides a derivation for this formulation, shown in Fig. 7 (bottom).

Constraints. Of particular importance are constraints that guarantee the ground, input, and output are all connected to the same structure, and that there are no other floating structures. We have used connectivity constraints as a filter rather than an objective function term, by simply discarding and replacing the designs with connectivity violations. Other possible constraints include weight, peak-power consumption, and displacement requirements, of which we have implemented the last.

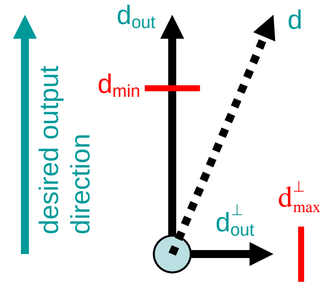

Figure 8: Values used to constrain minimum output displacement and maximum tangential displacement

Two constraint penalties are added to the objective function to monitor deflection. The first is a constraint that requires the output deflection in the desired direction to be greater than a minimum value, d_{min} (Fig. 8). The second requires the output deflection perpendicular to the desired direction be less than a maximum value, $d^{\perp}_{max}$. The final form of the objective function is shown in Eq. (2), where η is from Eq. (1) and w_1 and w_2 are relative weighting constants. From the formulation, the penalties are applied when $d_{out} < d_{min}$ or $d^{\perp}_{out} > d^{\perp}_{max}$. (The tangential constraint is not needed in our symmetric example problem.)

$$\eta = \frac{W_{out}}{W_{in}} = \frac{W_{out}}{\sum W_{in}} = \frac{k_{out}\, d_{out}^{2}}{\sum F_{in}\, d_{in}} \le 100\% \tag{1}$$

$$\text{maximize}\left[\eta + w_1 \times (d_{out} - d_{min}) + w_2 \times (d^{\perp}_{max} - d^{\perp}_{out})\right] \tag{2}$$

Discrete Nonlinear Topology Optimization and Synthesis. Given a model of the design space, the variables can be optimized via a genetic algorithm (GA) to yield a mechanism topology. GAs are a form of nonlinear optimization that seeks global optima as opposed to local optima and are especially suited for discrete, nonlinear problems. (See Eiben [22] and Goldberg [23] for more information.) Genetic algorithms convert the design variables into a long sequence of genes that can be propagated from generation to generation with the effects of cross-over and mutation. The best-fit indi-

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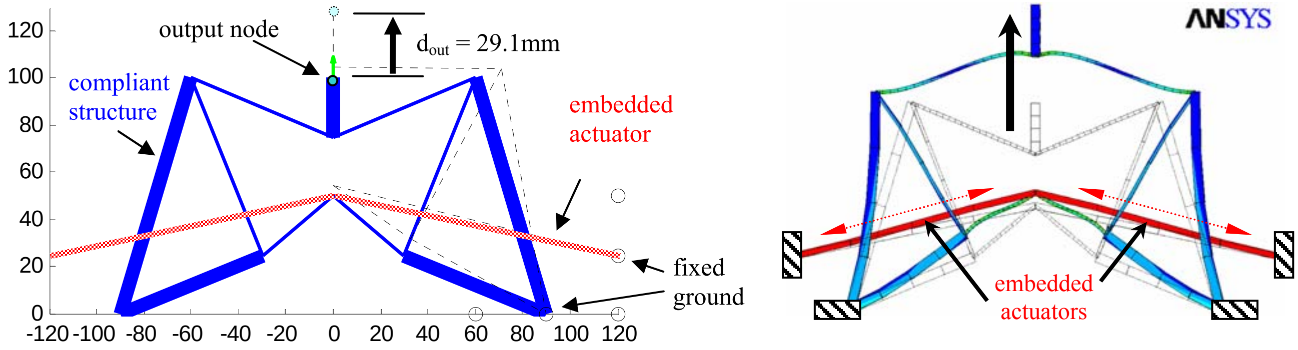

Figure 9: Dual-Actuator Symmetric Motion Amplifier. d_{out} = 29.1mm. d_{in} = ~0.5mm / actuator. Dashed lines represent deformed structure. Left: Matlab optimization results. Right: ANSYS nonlinear finite element analysis

viduals survive by scoring high against an objective (fitness) function.

The GA starts with a population of randomly generated designs. The selection scheme in GA is based on the 'survival of the fittest.' In our work we also implement "genetic engineering", i.e. the trimming of elements before fitness function and "Lamarckian trimming", i.e. the trimming of elements after the fitness evaluation. The highest scoring members of the combined parent and offspring population become the parent population for the next generation, guaranteeing that good designs are not lost.

Results. Again, the task for this example is to amplify an input displacement into an output displacement along the device's line of symmetry. Such devices have played an important role in the design of MEMS actuators [24]. The design specifications are actuator block force: 90N, element modulus: 2,480MPa, actuator modulus: 2,000MPa, output stiffness: 0.05N/mm, and minimum output deflection: 20mm. Figure 9 shows results for the best of a population of 150 over 1,000 generations, with only one actuator variable.

We observed that the GA tends to increase the number of elements per design over time, to maintain connectivity requirements. To limit the number of elements, the above example used a "random element elimination mutation" in addition to normal mutation. Later runs instead implemented a total structural volume constraint and also yielded acceptable results, not shown due to space constraints.

Discussion. Encouraging results (Fig. 9) show that the algorithm attains the goal of distributed actuation and display the desired attribute of fully-distributed compliance. The Symmetric Amplifier problem is a benchmark in the field of compliant mechanisms ([4-5,21]). The design depicted above is similar in topology to the more successful of previously reported high-gain results, such as MEMS amplifiers [24], tailored actuator transmissions [25], and results using a load-path method [3].

Note on Size / Shape Optimization. We have focused only on topology optimization, considered the more difficult of the two design stages. Further dimensional (size and shape) synthesis can be implemented with standard continuous optimization methods to address many other practical constraints and performance requirements. These include materials (permissible stresses and strains), desired fatigue life, prevention of localized buckling, natural frequency, manufacturability constraints on geometry, etc. During this process, the nodes or interconnection points are allowed to wander within a certain window thereby modifying the geometry without altering the topology; in size and shape optimization the topology is fixed and only the node locations and beam width and thickness change.

Example 2 – Design for Controllability

The next line of research (**A**+**B**) is the merger of compliant mechanism synthesis and "design for control" within the synthesis methodology for compliant systems. Our initial approach to integrate controls with CMs is not to optimize the controller, but to optimize the controllability *characteristics* of the system, defined as measures of how thoroughly a given set of internal actuators/sensors can achieve/sense a full range of output states on the external environment. Controllability and observability of the structure will be calculated in terms of the linear independence of the actuators and sensors, which will be incorporated into the optimization. The presented work on *controllability* provides the method for optimizing *observability*. We hypothesize that if the actuators independently affect the output displacement, then they can compensate for a variety of unknown loading conditions.

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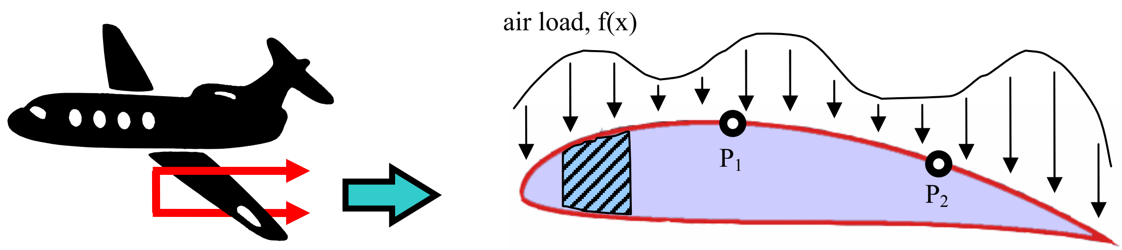

Figure 10: Aircraft wing cross-section subject to unknown and arbitrary air load, f(x). Points P_1 and P_2 are to be controlled by internal actuators and monitored by internal sensors.

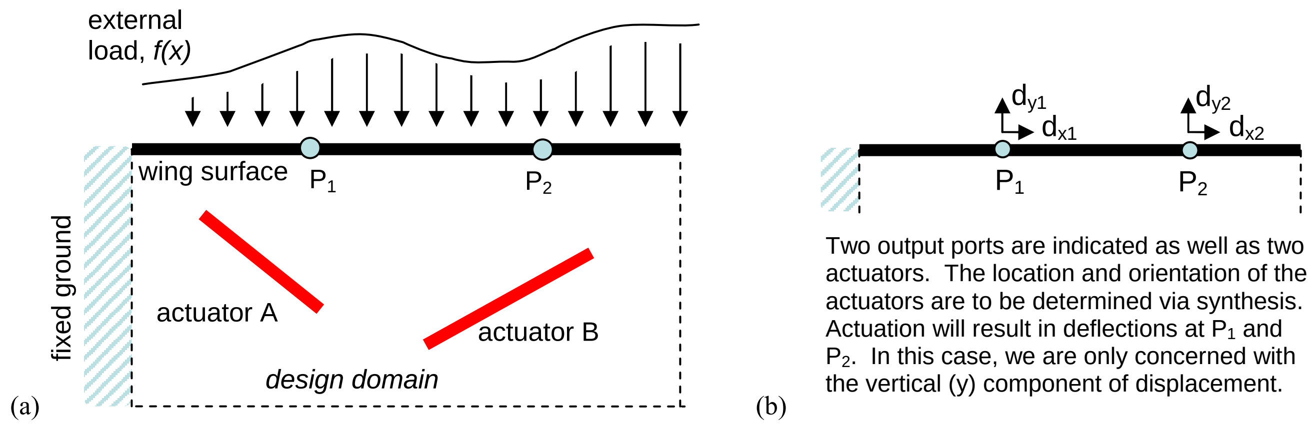

Figure 11: Design Space for the "Design for Control" synthesis problem.

Problem Formulation. As stated, the task of a compliant system is to serve as a mechanism with a specified functionality while also maintaining a structural stiffness to support external loads. We now look at adding adaptability to that definition, so that compliant systems can detect and appropriately respond to unknown and arbitrary loading. Consider an example problem with an output region defined by two output points, P_1 and P_2. Imagine a simple airfoil that is to remain in a particular configuration despite changing pressure loads, as depicted in Fig. 10. These points will be subject to unknown loading in the vertical direction, f(x). Each point represents a degree of freedom of the system, and thus two actuators (A, B) are required (at a minimum) to fully control the output points.

Design Space. The design space (Fig. 11) is nearly the same as that used in the basic embedded-actuation problem (see Fig. 6). A network of enumerated beams represents the underlying structure, with each beam being a design variable. The left-side of the design space is fixed to the ground. To guarantee a continuous airfoil, the top edge elements are not included in the optimization, but are pre-specified as known-thickness elements. For actuation, we use axial force actuators with specified stiffness (including bending) and output blocked force. The actual output load and deflection depend on the stiffness of the entire compliant system, and are determined by the FEA solution. The actuators are not binary (on or off) but continuously controlled, allowing for flexibility in mixing the independent effect of each actuator on the structure.

Loading. When calculating the fitness function within the synthesis procedure, the structures are first loaded by only the internal actuators, one at a time, with no external load. While the goal is to support external loads, those loads are not yet known. Instead, we attempt to optimize the actuator layout within the structure to result in the greatest freedom to control the displacement output points. Such a structure is believed to best handle variable load. The second example given in this section does consider external load, applied in an additional load case with no active actuators. The structural response in this case is added to the objective function in a minimization term.

Ultimately, we will consider the shape-change problem where the airfoil has an initial curve and a *target deflected curve* to achieve a prescribed function. The system will need to be adaptable and robust to unknown external loading in *both* configurations. However, in this initial research, no shape change is applied; the desired target position is the neutral starting position.

Objective Function and Constraints. To develop a measure of controllability, the effect of each actuator within the structure is first tested via FEA. The output points P_1 and P_2 will have displacements (d_{1x}, d_{1y}) and (d_{2x}, d_{2y}). We are interested in the coupled output of both actuators in the vertical direction, represented by the vector: $\bar{d}_i = [d_{1y}, d_{2y}]_i$, where "$i$" represents the active actuator. Thus, for actuator A, $\bar{d}_A = [d_{1y}, d_{2y}]_A$. Likewise, a second FEA is run with actuator B activated, resulting with: $\bar{d}_B = [d_{1y}, d_{2y}]_B$.

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Linear Independence. To increase the controllability of the structure, it is desired that the $\bar{d}_i$ vectors be linearly independent. This condition means that the effect of each actuator will not be redundant, allowing for the controller to be effective. Thus, the combination of the two actuators ($\bar{d}_{combined} = w_A\bar{d}_A + w_B\bar{d}_B$) can be used to cover as broad a range of output combinations as possible. An example of potentially orthogonal vectors is shown in Fig. 12. By the nature of compliant structures, all output points are affected by all actuators, making it impossible to have each output point simply controlled by only one input. That is, a scenario such as $\bar{d}_A = [1,0]$ and $\bar{d}_B = [0,1]$ is not possible, but other linearly independent combinations should be achievable, such as $\bar{d}_A = [0.5,0.5]$ and $\bar{d}_B = [0.5,-0.5]$.

Controllability. We define controllability as the degree of independence between the two vectors, evaluated as a number ranging between zero and one, obtained by creating a matrix of the $\bar{d}_i$ vectors and then dividing its determinant by the product of the magnitudes of the vectors, as shown in the formula below:

$$\eta_c = \left| \frac{\det[\bar{d}_A \; \bar{d}_B \cdots \bar{d}_m]}{|\bar{d}_A||\bar{d}_B|\cdots|\bar{d}_m|} \right| \tag{3}$$

(where m is the number of outpoint points, equal to the number of actuators)

Here, a value of zero implies linear *dependence*, while values closer to one imply orthogonality of the *m* vectors. For the 3-point case, we also recognize $|\bar{d}_A||\bar{d}_B||\bar{d}_C|\eta_c = \det[\bar{d}_A \; \bar{d}_B \; \bar{d}_C]$ from Cartesian linear algebra as the volume of the parallelepiped formed by the three vectors $\bar{d}_A$, $\bar{d}_B$, and $\bar{d}_C$. (a.k.a. *triple product*.) When the vectors are all orthogonal, the maximum volume is achieved, indicating that larger values of η_c are desirable for our objective. Hence, the controllability, η_c, serves as the objective function to be maximized during the genetic algorithm. (Note: orthogonality is a rather basic and limited estimation of linear independence. In our continuing work, we are implementing more common measures based on the Jacobian of the transformation, such as condition number and the ratio of singular values.)

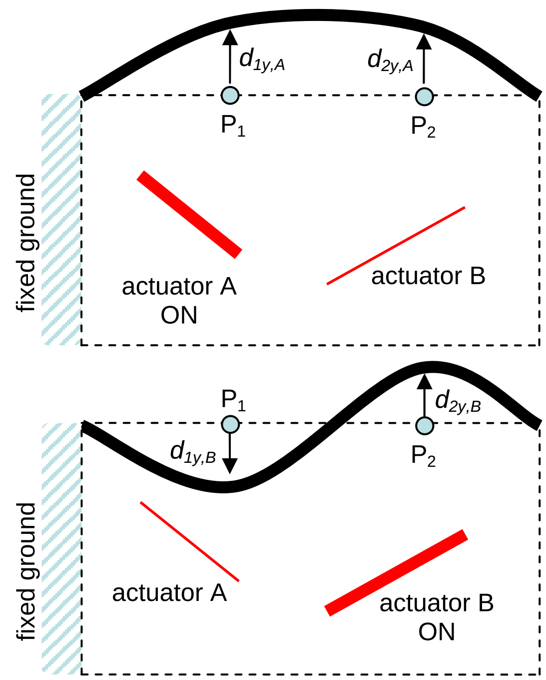

Figure 12: An example of two possibly orthogonal actuation modes. It is seen how the deformations due to each actuator are not only different, but also not redundant.

Other Constraints and Terms to be Optimized. While the primary objective is the orthogonality of the output vectors, it is also required that these vectors surpass a minimum displacement magnitude so that the structure can sufficiently deform. This can be checked by making sure that for every output point at least one vector meets the minimum displacement required of that point. The volume is also minimized in each of the following designs, while the second example also minimizes deflection under an external load applied at the output nodes opposite the desired direction of motion.

Results. Again, the task for these examples is to control a simple airfoil exposed to unknown and variable pressure loads along its external surface. The design specifications are actuator block force: 90N, element modulus: 2,480MPa, and minimum output deflection: 5mm. Each problem used the same design domain, similar to that in Fig. 11, but with three output nodes. Figure 13 shows results for the best of a population of 200 over 680 generations. Note the orthogonality of the output curves; the shapes are more complex than the trivial solution (where each actuator primarily affects only one output node): all of the nodes are moving, yet high orthogonality (99.98%) is still achieved. However, the first and last skin nodes are cantilevered and have questionable integrity if under external loads, leading us to the second example with the additional external load case present. Figure 14 shows the results for the best of a population of 200 over 1,500 generations, with an additional load case to establish structural integrity. High orthogonality is achieved (99.99%) and the maximum vertical deflection for a downward load of 10mN simultaneously applied at all three output nodes is 0.022mm.

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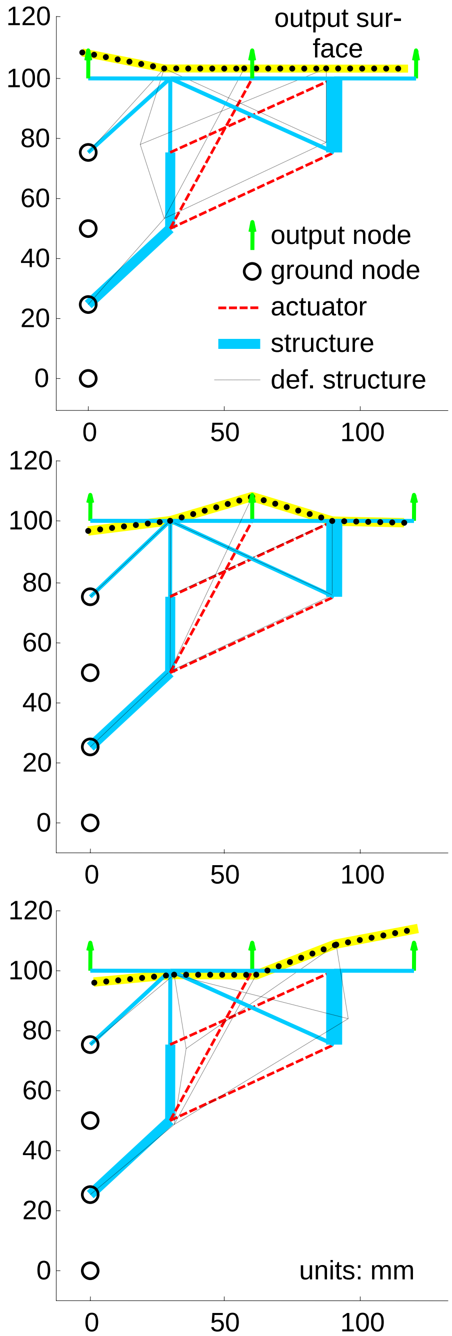

Figure 13: Results of Optimization for Controllability. No external load-case considered. Each figure shows the response of a different actuator (dashed red line) firing. Dotted black lines indicate deformed structure. Parameters: Population=200, Generations=680, Actuator modulus=500MPa. Orthogonality=99.98%

Discussion. We are very encouraged that the results are not trivial designs, i.e. actuators merely connected directly from the output nodes to the ground (or to floating nodes effectively grounded by stiff connections to the ground). Over the course of many (20+) runs, it was also observed that values for orthogonality tend to be below 10% or greater than 90%, for *all* structures, both random and optimized. This discontinuity, not yet fully understood, makes it difficult for the optimizer to

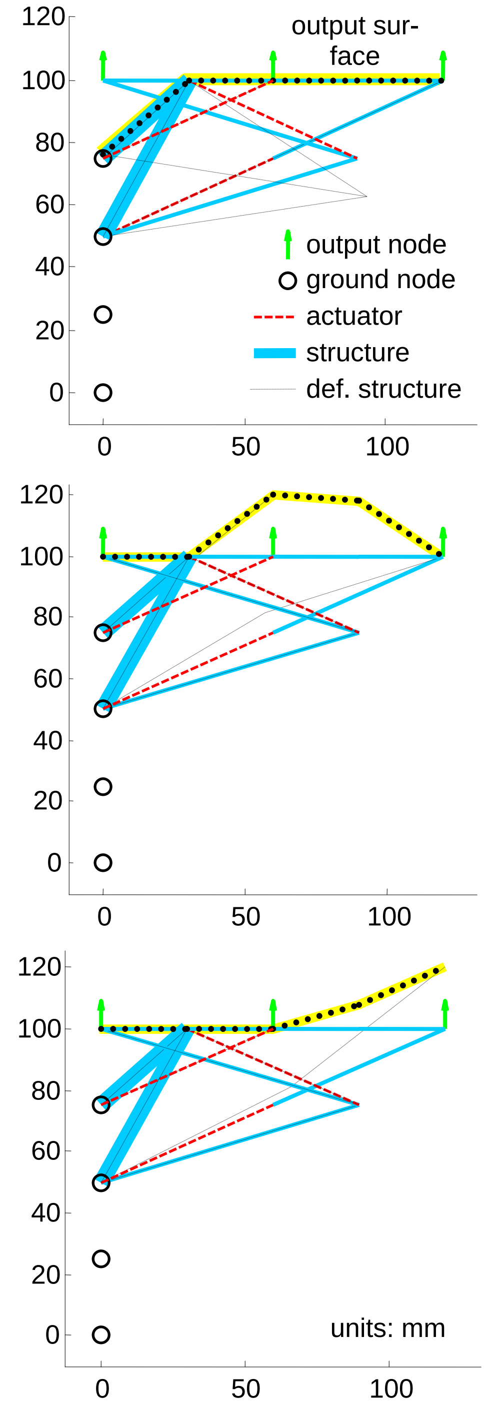

Figure 14: Results of Optimization for Controllability, with an external load-case considered. Each figure shows the response of a different actuator (dashed red line) firing. Dotted black lines indicate deformed structure. Parameters: Population=200, Generations=1,500, External load= -10mN at each output node, Actuator modulus=500MPa, Skin element thickness=0.5mm. Orthogonality=99.99%, Max. deflection under load=0.022mm.

balance the requirements for orthogonality and large displacements. Typically, one or the other dominates the objective function and the other falls far behind. Many parameters have been identified which might make this balance more even; some have been implemented in the second example. The bal-

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ance is very sensitive to the values for skin modulus, actuation modulus, skin grounding, and the presence of an output load; these all not only affect the design, but specifically affect the displacement/orthogonality conflict. Connection of the skin elements directly to the ground constrains both the deflection magnitude and the flexibility required for controllability. Skin elements that are too compliant or actuator elements that are too stiff make orthogonality overly easy to achieve, resulting in its dominance. In effect, these conditions decsrease the coupling between the three output nodes. Manipulating all of these initial parameters, in the presence of the volume constraint, has made for the design seen in Fig. 15. Only one actuator response is shown, and while it may appear an inelegant mess, analysis has shown that every element is critical to the four functions this device is achieving: orthogonality of three actuator responses at 99.88% and deflection under load of only 0.86mm. In fact, the device maintained the large number of elements even after a run of 1,500 generations under pressure for minimal volume.

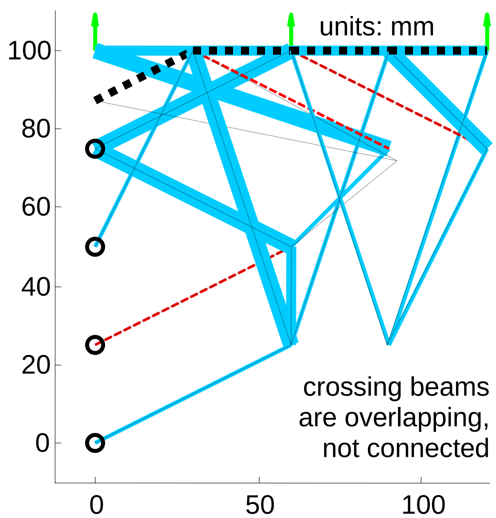

Figure 15: Finding the balance between orthogonality and deflection. Parameters: Population=200, Generations=1,500, External load = -1N at each output node, Actuator modulus=1,000MPa, Skin element thickness=1mm. Orthogonality=99.88%, max. deflection under load=0.86mm.

EXTENSION OF METHODOLOGY TO EMBEDDED SENSING

Sensors are also required in any controlled system. In this section we will develop a means for measuring *observability* that is very analogous to the method already given for measuring controllability. Through an inversion of inputs and outputs, the framework for embedded actuation (**A**+**B**) can next be extended to distributed sensing (**C**+**D**). While these are analogous, some new issues are sure to arise and will be the subject of near future research. Of key significance is that this is not the traditional ad hoc design; sensor design will be simultaneous and synergistic, with sensor placement affecting structural topology and vice versa. This new sensing paradigm corresponds to an "*internal* state of stress" that relates to the *external* conditions of the system environment.

Before proceeding it is necessary to point out the advantages of and requirements for non-collocated actuators and sensors. Using sensors in tandem with the actuators (or using the actuators *as* sensors), requires that they take advantage of the same deformation mode. However, most actuators provide force in the *axial* direction and most sensors (strain or MIS) respond best to *bending* deflection, as bending provides more deflection and strain than does axial loading. It is not likely or desirable for both of these conditions to occur at the same time in the same element. Further, the actuators must lie directly in the load path and require additional stiffness, while the sensors will perform better on more compliant members not directly in the load path of the structure. Finally, designs with *internally* located sensors are safe from the dangers of the *external* environment.

Formulation

The goal is now to describe the conditions at the output points described in the previous section, P_1 and P_2. It may be desired to know the pressure load acting on those points or the shape of the deformed curve. We consider these equivalents as they are related by the structural elasticity. In this example, we will look at the ability of the sensors to detect pressure loadings of unknown shape.

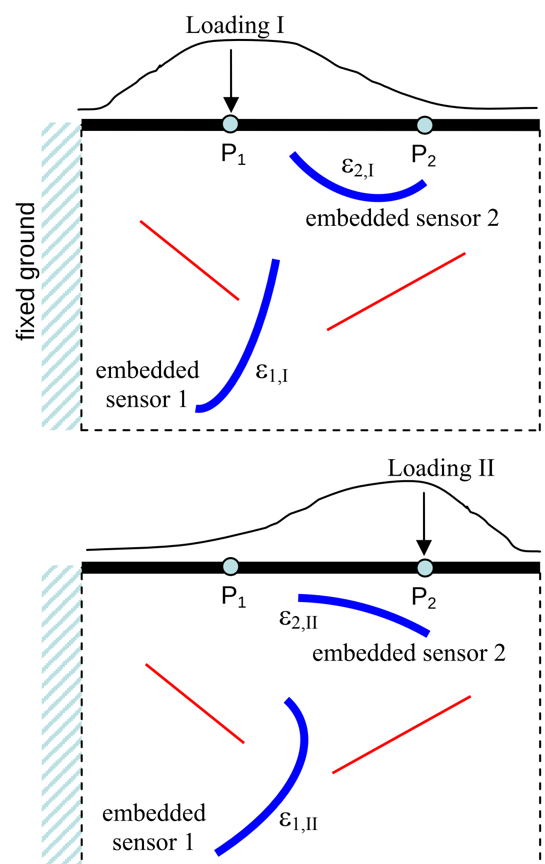

Figure 16: Response of "embedded and distributed sensors" to different applied loads. The loadings are design to be orthogonal, with the intent that the actuators respond differently and uniquely to each loading case.

Sensor elements are now to be integrated with the structure, which are an addition to the embedded actuator problem origi-

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nally stated. The integration is slightly different though, as there will be no "beam replacement" as there was for the actuators. Rather, the nature of the embedded sensors allows them to be directly added to the beams, without changing the beam properties. Two sensing strategies are identified as applicable: strain gauges placed directly on the beams or embedded *mutual inductance* sensing (MIS, see Peshkin U.S. Patent [26]).

There are two general ways to approach design for optimal sensing. The *first* occurs within the optimization after the FEA: rather than let sensor location be a variable, we could analyze every acceptable structure generated by the GA and determine which beams are best for embedded sensing. We have chosen the *second* option, which is to actually treat sensor location as a variable, similar to the way actuator location is determined. The embedded sensors for each structure are then chosen by the GA, using the genetic engineering steps to guarantee connectivity. This second option gives the sensors a much greater role in *determining* topology, rather than being determined *by* topology.

The task of optimal distributed sensing is to find beam deflections that correspond to orthogonal external loading. Analogous to actuation, we seek to place the sensors so that they can detect as wide a range of loading as possible. Rather than subjecting the structure to a series of random loading, we will apply specific loads designed to be independent of each other. For example, we consider two orthogonal loads by loading only one output point at a time (Fig. 16). Just as two FEA runs were required before for finding the output vectors for each actuator ($\bar{d}_A$ and $\bar{d}_B$), two FEA runs are required to find the sensing vector for each loading condition ($\bar{s}_I$ and $\bar{s}_{II}$). The sensing vector comprises the total signal (ε) found in each of the embedded sensor beams: $\bar{s}_i = [\varepsilon_1, \varepsilon_2]_i$

Observability
Observability is now defined as the ability of the system to measure its state, specifically, the condition being imposed at its external boundary. Again, we use the determinant formula (Eq. (4)) for linear independence to calculate a single value for observability, η_o. Again, the values range from zero (poor observability) to one (high observability). This value will be added as another weighted term in the objective function to be maximized.

$$\eta_o = \left| \frac{\det[\bar{s}_I \ \bar{s}_{II} \cdots \bar{s}_m]}{|\bar{s}_I||\bar{s}_{II}|\cdots|\bar{s}_m|} \right| \tag{4}$$

Finally, with both controllability and observability addressed in the optimization, an effective, linear feedback is established for operation of the system. A simple PID controller can then monitor the load measurement vectors and accordingly adjust the actuators to maintain the specified position. The same controller can also be used to effectively carry out the shape-change functions that are required of the adaptive structures in later stages of this research.

DISCUSSION
Considering the low fidelity of the modeling, the attainment of results that display all of our design requirements is very promising. The addition of nonlinear analysis and additional analysis elements for each beam will allow greater flexibility in any given structure and increase the likelihood of more and better multi-functional solutions. These studies also pointed out other basic research issues, particularly in parameterization and forming objective functions. Convergence required complex, non-standard "genetic engineering" of our design population members during the GA: trimming of superfluous substructures, computationally expensive graph searches, and identification of ineffective rigid-body segments. The unbalanced tradeoff between controllability and large deflection has also been identified, along with a family of parameters to balance the struggle. Finally, the GSA works as a benchmark, but its complexity motivates exploration of other approaches, such as the Load-Path Method described by Lu [3,8].

FUTURE RESEARCH
Further work may enable new applications in the compliant paradigm, such as medical implants (low back pain/disk replacement), exoskeletons, exotendons, prosthetics, orthotics, artificial organs, micro-air-vehicles based on flapping wings, and reconfigurable surfaces for a wide range of applications. *Variable stiffness* of CMs is an interesting open research question that can be tackled by dividing our actuators into "function" groups and "adjustment" groups. With a socket system capable of sensing the external pressure load and changing the shape to minimize any stress concentrations, discomfort can be alleviated and proper fit maintained. Such devices could also be used for functional changes in different scenarios, such as an orthosis that could adjust to increase stiffness during running and before jumping. Another possibility is the continued study of a "sense-and-adapt" shape-changing aircraft wing or aquacraft fin that could sense a variety of external pressure load profiles and respond with a variety of shape changes. Such sense-and-control variable geometry wings could adjust to the most appropriate and efficient configuration for both predicted and unpredicted flight conditions.

Specific paths to explore include alternate parameterization, such as Load Path (connectivity-based discretization), and nonlinear FEA analysis within synthesis. The imminent extension from controllability to observability will be followed by the combination of both terms within the same objective function. The next step is to then combine both controllability and observability with a compliant mechanism shape change problem. One example is a wing that can change between different trailing edge configurations *and* be adaptable to various loading conditions while in *either* configuration.

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