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3D compliant mechanisms synthesis by a finite element addition procedure

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ABSTRACT

This work presents an element addition strategy for 3D compliant mechanisms design. The proposed procedure is based on an extension of the evolutionary structural optimization (ESO) method, which has been successfully applied to several optimum material distribution problems, but not for 3D compliant mechanisms optimization.

Even if most investigations for compliant mechanism design have been oriented for planar systems design, this technology may be useful also for 3D mechanisms design, for instance in making devices for micro- and nanomanipulation, like the popular hexapods mechanisms used for six axis positioning. These 3D structures and mechanisms (rigid or compliant) must be carefully manufactured and assembled from many precision components, and there are still many aspects that must be examined to accomplish the topology optimization and ensure the performance of these precision manipulators. The present paper aims to progress on this line, and will apply an alternative approach derived in this investigation, which improves the solutions obtained by this specific method. The proposed method has been tested in several numerical applications and benchmark examples to illustrate and validate the approach, and satisfactorily applied to the solution of 3D examples.

1. Introduction

Compliant mechanisms obtain their mobility from flexibility of their parts, in contrast with traditional mechanisms made of rigid counterparts, where mobility is obtained from hinges, bearings or sliders. As a result, compliant mechanisms can be built using fewer parts, require fewer assembly processes and need no lubrication. An important application of compliant mechanisms lies in Micro Electro Mechanical Systems (MEMS) design, where due to the small size, hinges and bearings cannot be used due to friction problems that would dominate at the small scale. Therefore these types of mechanisms must be built and designed as compliant mechanisms etched out of a single piece of material.

Topology optimization has been successfully applied to optimize compliant mechanisms in many practical engineering designs with the use of finite element analysis, since this technique enables systematic design directly from the behavioral specifications. This method is able to allow for a change in the number and position of elements, because holes may be added or deleted to modify the connectivity of the structure during the course of the optimization problem without relying on an intuitive initial design. The structural topology design problem is formulated as a material distribution problem within a given design domain, where material should be placed and connected to some portions of the boundary with some number of holes inside to optimize an objective function. The design goals for structures and compliant mechanisms are quite similar, and the same topology optimization methods may therefore be adapted to design both types of elements. To realize such a design methodology, formal techniques of structural optimization are adopted but the design goals were modified specifically to suit the functional requirement of compliant mechanisms. Although the intended functions of compliant mechanisms and stiff structures are inherently different, structural optimization algorithms can be used for the synthesis of compliant topologies. A fundamental difference in the two design problems is that in compliant mechanisms, adequate flexibility is deemed essential for their structural reconfiguration to afford the required displacement at the point of interest. Additionally, a compliant mechanism also needs to be stiff enough to be able to sustain external loads. Thus, there are two design objectives to be met simultaneously when designing a compliant mechanism. The most used objectives in compliant mechanisms synthesis are the geometrical and mechanical advantages, which describe the ratios between output and input port displacements and forces, respectively. Furthermore, even if a simplified lineal analysis may be used as a first step into compliant mechanisms design, one must be aware of the limitations that such modelling imposes and the use of geometrically nonlinear finite element analysis becomes essential in this type of optimization.

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The pioneering numerical implementation of the material distribution idea based on homogenized materials for finding the optimal topology of a structure was first described by Kikuchi and Bendsoe [1], treating the reference domain as if it was made of composite material consisting of a solid and void periodic microstructure. The idea of using a penalized variable density approach (SIMP) for numerically approximating a material-void design problem was first tested in Bendsoe [2] and Rozvany et al. [3]. This possibility has proven very popular and extremely efficient, where a density variable is associated with each finite element and exponential law is applied to compute effective properties ($E=E_0\rho^p$), where E_0 denotes de isotropic material elastic modulus, ρ is the density and p represents de penalization factor. It was not originally intended to correspond to a physical microstructure but recently it was concluded that there exist real microstructures for SIMP functions with a suitably chosen exponent [4]. Since the beginning of the 1990s, several new methods have emerged, which can be used alongside traditional methods to complement them. Among these methods, a number of heuristic or intuition based methods have been proposed to minimize compliance or other objective functions, like genetic algorithms [5], which show to become prohibitively expensive for large systems, or the evolutionary method, also known as evolutionary structural optimization (ESO) [6], although a more appropriate term for this method would be sequential element rejections and admissions technique, suggested by Rozvany et al. [7]. The recently developed level-set method, originally introduced by Osher and Sethian for numerically tracking fronts and free boundaries[8], has been successfully used in the field of optimization[9] and seems to be tremendously promising, even if it still is in its early stages.

In the field of compliant mechanisms topology optimization for continuum synthesis approach, the first applications appeared in Ananthaturesh et al. [10]. A later approach by Sigmund [11] modelled the output load by a spring that captures the nature of the workpiece held at the output port of the compliant mechanism and allows control of the input–output behavior using the mechanical advantage as objective function. An equivalent but different approach is based on the maximization of the ratio of two mutual energies, where two different finite element problems are considered [12]. The main differences between the above mentioned topology optimization approaches consist mainly in the formulations of the optimization problem and the "best" formulation probably still remains to be defined. Also Frecker et al. presented the synthesis of compliant topologies with multiple input and output ports, using as objective function a combination of the mechanical and geometrical advantage of the mechanism [13]. Path generating mechanisms have been also treated in the work by Saxena and Ananthaturesh [14], as well as compliant thermal microactuators topology optimization [15].

Concerning the parameterization methods employed for the solution of the topology optimization problem for compliant mechanisms, we can cite the microstructure based homogenization method [16], the SIMP interpolation [17], the level-set method [18] or a simple version of the ESO method, successfully applied by this research group for planar compliant mechanisms design [19]. Progressing in this line of work, this paper presents an enhanced and more general additive version of this method for 3D compliant mechanisms design, which is quite difficult to find in topology optimization literature [20] but may become very useful, for instance in conceiving mechanisms like compliant hexapods for micro- and nanomanipulation or complex positioning processes [21]. Even if the validity of the ESO method has been examined critically and several arguments have been made against it, since it may lead to highly nonoptimal solutions in the same circumstances and presents some drawbacks compared with other methods, this paper shows that actually it can be used for 3D compliant mechanisms topology design by means of an additive version of the method. Here an alternative mechanical advantage based objective function is applied, which allows to find the optimum design very efficiently when this method is applied. The unwanted formation of checkerboard patterns is prevented by the classical smoothing technique frequently adopted when evolutionary topology optimization is applied. This paper presents also an enhanced variable smoothing technique to circumvent these numerical problems, especially critical when dealing with 3D problems. The procedure has been implemented as part of a general optimization computer program called Odessy [22] and tested in several numerical applications and benchmark examples to validate the approach.

2. Topology optimization problem for compliant mechanisms design

Consider a linear elastic body occupying three dimensional domain Ω where the mechanism is assumed to lie, with given loading and boundary conditions shown in Fig. 1a, where P_1 is the input force, and u_{out} is the expected output displacement. Displacements at the input and output ports of the compliant mechanism can be found by discretizing it using finite elements and solving equilibrium equations for two load cases. The first load case consists of the actual input load P_1 and the second case load is a unit dummy load applied at the output port in the direction of the desired displacement (see Fig. 1b). First, the equilibrium problem must be solved for each case:

$$\mathbf{K}\mathbf{u}_1=\mathbf{f}_1, \quad \mathbf{K}\mathbf{u}_2=\mathbf{f}_2 \tag{1}$$

where $\mathbf{K}$ is the global stiffness matrix of the structure, $\mathbf{u}_1$ and $\mathbf{u}_2$ the nodal displacement vector due to the input and unit dummy load and $\mathbf{f}_1$ and $\mathbf{f}_2$ the nodal force vectors containing the input and dummy force. Once displacements are computed, u_{ij} displacements can be obtained as

$$\begin{aligned} u_{11}&=\mathbf{u}_1^T\mathbf{K}\mathbf{u}_1/p_1, \quad u_{12}=\mathbf{u}_2^T\mathbf{K}\mathbf{u}_1/p_1 \\ u_{22}&=\mathbf{u}_2^T\mathbf{K}\mathbf{u}_2/p_2, \quad u_{21}=\mathbf{u}_1^T\mathbf{K}\mathbf{u}_2/p_2 \end{aligned} \tag{2}$$

where indices ij indicate displacement at port i due to a load at port j. In this work mechanical advantage is posed as objective function, that is, the ratio of induced output force to input force, a functional specification frequently used for many mechanisms design like, for instance, crunching or gripping mechanisms. This ratio depends on the stiffness of the elastic workpiece that fills the gap under the output force, shown in Fig. 2a, and modelled by a spring with stiffness k_s. Fig. 2b shows the load in the input port and the reaction force at the output port. By means of the reciprocity theorem we have the following relationship:

$$u_{12}p_1+u_{22}R=u_{out}p_2 \tag{3}$$

where R denotes the reaction force in the spring and is related with the displacement at the output port, u_{out}, which can be written as

$$u_{12}p_1-k_S u_{out}u_{22}=u_{out}p_2 \rightarrow u_{out}=\frac{u_{12}p_1}{p_2+k_S u_{22}} \tag{4}$$

Using this expression to formulate the mechanical advantage we get the same equation obtained by Sigmund [11]:

$$M=\frac{F_{out}}{F_{in}}=\frac{k_S u_{out}}{p_1}=\frac{k_S}{p_1}\frac{p_1u_{12}}{p_2+k_S u_{22}}=\frac{u_{12}}{\frac{p_2}{k_S}+u_{22}}=\frac{1}{p_1}\frac{u_{21}}{\frac{u_{22}}{p_2}+\frac{1}{k_S}} \tag{5}$$

where the displacements shown in this equation can be found using equations in (2). Additionally, practical limits on the input displacement are usually introduced, i.e. $u_{in}\leq u_{in}^{max}$. This