A two-stage topological optimum design for monolithic compliant microgripper integrated with flexure hinges

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Abstract. The formation of hinge-like portions typically is encountered in the structural topological synthesis. This paper proposes an optimum synthesis integrated with the analytic single-axis flexure hinge as a second-stage design process to obtain optimum flexure configuration and location for promoting the overall performance of a compliant microgripper after the topology optimization. The locations of hinge-like in topology synthesis are verified to be the exact locations of flexures. A simplifying optimization applying ANSYS is consequently done in the second-stage optimum design. The proposed process can be extended to other types of compliant structural design.

1. Introduction

Compliant mechanisms and structures are single-piece flexible constructions and well suited for MEMS because of the small length scale and problems with assembly, friction and wear that prohibit use in conventional rigid-body mechanisms [1]. Topology optimization [1-4] is a known technique that can be effectively applied to the design of compliant mechanisms. There are two difficult problems: the first one is checkerboard formation [5]. Another numerical problem encountered is the formation of single-node that should not the part of the final design. Such the singular portion actually represents the hinge to possess a localized compliant behavior that is unrealizable in a real mechanical system. A procedure by Yoon et al [6] in which a multiscale wavelet-based topology optimization formulation to obtain hinge-free result was presented. This post-processing can change the portion of single-node shown in Figure 1(a) to a design of Figure 1(b). Although the design in Figure 1(b) takes away the singular hinge portion, it still does not match the configuration of the real mechanical flexure hinge. Certainly, it is also not an optimum design.

![Figure 1.](image)

Figure 1. (a) A single-node hinge (b) Compliant hinge by post-processing

Since a mechanical compliant flexure joint can be included in a flexure-based gripper, under a small deflection, the flexure joint can be modeled as a rigid hinge joint attached by a linear torsional
spring for the analysis and design [7]. This idea inspires us to examine the formulation of a mechanical flexure hinge applying to the post-topology optimization. Consequently, we propose a post-processing technique as called the second stage for topology optimization design and it is further simplified in recent development. The proposed two-stages topology based design process in dealing with mechanical flexure hinges indeed promotes the overall performance of a compliant structure.

2. Mechanics of flexure hinge

The geometrical shape surrounding flexure hinges is critical for the output performance of a single-piece compliant mechanism, as recognized by researchers. Figure 2 shows dimensions that define the hinge, various forces and moments for which the angular and linear compliances are calculated. A single-axis flexure hinge must be flexible about the input axis or sensitive axis. Usually, the hinge must be as stiff as possible about the cross axis and along the longitudinal axis. For an input moment $M_z$ causes the flexure to deflect through angle $\alpha_z$. The simplify compliance equation and an equivalent torsional spring rate can be derived. Linear deflection $\Delta y$ may be caused by moment $M_z$. Thus, the simple form of compliance for the case of applied moment and the equivalent linear spring rate along $y$-direction can be developed. All fundamental mechanics concerning this single-axis flexure hinge appear in the publication of Paros and Weisbord [8].

![Figure 2. Hinge dimensions and various forces and moments](image)

3. First stage optimization: topology synthesis

![Figure 3. Domain for a compliant mechanism design](image)
Figure 3 shows the sketch of general domain of a monolithic compliant structural topology design problem, as recognized the first stage optimization in this paper. A spring at the output port simulates the resistance from the work-piece. The goal of the optimization problem is to maximize the work performed on the spring. A topology optimization solves the problem of distributing a limited amount of material in the design domain such that the output displacement is maximized. The relative density in each element is a design variable.

In this paper, the design domain in Figure 4 of a compliant gripper/magnifying mechanism is the example for the demonstration. The size of the design domain is 8.0 mm×4.6 mm and the thickness is 0.5 mm. The gripper is built in PU material that has Young’s modulus 4.95×10^7 Pa and poison ratio 0.47. An input force $F_{in} = 0.34g$ is applied at the center of 2.70 mm from the left edge and the output spring (0.02 N/mm) is mounted at the output of the right edge. All the material property, the overall size and related information is the same as the thesis of Cheng [9]. The input spring is 29.7 N/mm and the volume is restricted to be 30 per cent of the design domain. Due to the symmetry, only the half of the design domain is discretized using 1840 4-node finite elements. The optimized topology synthesis can be obtained as shown in Figure 5.

![Figure 4. Design domains for the gripper amplifying mechanism](image)

Two elliptic circles are marked on the Figure 5(a) that shows hinge-like portions required to be modified in a practical sense. How can one modify the hinge-like portion to be a mechanical compliant flexure-hinge for manufacturing availability and simultaneously be an optimum compliant mechanism design? In other words, it requires modify the situation in Figure 5(a) to be the state in Figure 5(b) so that it can fit the model of single-axis mechanical flexure [8].

![Figure 5. (a) Hinge-like portion of the compliant gripper. (b) Compliant flexure hinge](image)
4. Second stage optimization: Post-design modeling
A proposed second stage optimization modeling [10] is developed in this work concentrating on the middle portion of gripper, as shown in Figure 6 expressed in a half configuration. There are two single-axis flexure hinges required to design. Nine parameters written in $l_1, l_2, l_3, r_1, r_2, C_x, d_1$ and $d_2$ are considered as design variables. Once a horizontal input force toward left applies to the gripper, an obviously vertical output displacement can be resulted.

![Figure 6. Half configuration of design model after topology optimization](image)

From the structure of Figure 6, the effects of both equivalent torsional springs stiffness should be small in order to create a maximum output motion. Therefore, the two equivalent torsional spring rates require to be minimized. The linear stiffness of outside hinge requires a minimum stiffness. Similarly, the linear stiffness of inside hinge requires a maximum stiffness. To maintain the stable grasping effect, the output vertical displacement should be large and the output horizontal displacement should be small, as compare to input horizontal displacement. This means that magnification factor (MF) requires to be maximized. Therefore, from the above description, this optimization problem is formulated as: determine nine design variables, five objective performances, six inequality constraints and single equality constraint. A multi-objective optimization technique has been applied for solving the current problem. The similar and clear formulation can be obtained from the publication of Lin and Shih [10]. The optimum MF is summarized in Table 1. It is noted that the locations of single-axis flexure hinges obtained by this stage optimization are (1.191, 3.530) mm and (1.805, 3.293) mm. A continuously description regarding this part is given in the next paragraph.

5. Simplified optimum design of flexure hinges
We measure the locations of hinge-like rotational axis obtained by the first-stage topology optimization are (1.25, 3.55) mm and (1.75, 3.35) mm, as indicated in Figure 7. When we compared this phenomenon to last paragraph, we found that the location of hinge-like rotational axis obtained by the first-stage topology optimization, as verified in our size optimization, is exactly same and correct. Therefore, the design variables in second optimization for flexure hinges can be reduced for determining its locations. The configuration of Figure 7 is re-sketch to be Figure 8, the point “a” and “b” indicates the locations of hinges, for further simplified optimization to flexure hinges.
At the beginning of simplifying design, the design variables in local coordinate as shown in Figure 9(a) is setup, where \( C_p \) indicates the center and \( R \) indicates the radius of adopted mechanical flexure hinges. The area of Figure 9(b) represents the complete part of a flexure and then using Boolean operation to deduct the circular area determined by \( C_p \) and \( R \). As a result, there are four design variables, \( \{C_p, R\} \) for locations “a” and “b” in Figure 8, in simplified optimization mode.

For avoiding the geometrical confliction, the value of \( R_C p \) must be reasonably restricted. The induced stress must be less than material’s yielding strength (41.37MPa). The displacement perpendicular to working direction shall be as smaller as possible; we constrained it in an upper bound. Consequently, the mathematical formulation of the optimization problem can be written as:

\[
\text{Find } X = \begin{bmatrix} C_{p,y} \end{bmatrix}_t, \begin{bmatrix} C_{p,y} \end{bmatrix}_b \\
\text{s.t. } 50 \mu m \leq C_{p,y} - R \leq 150 \mu m \\
\sigma_{\max}(X) \leq 41.37 \text{MPa} \\
U_{out,nongrip}(X)/U_{in} \leq 1.15
\]
where $\sigma_{\text{max}}$ represents the maximum induced stress in the gripper. Notations of $U_{\text{in}}$ and $U_{\text{out}}$ represent the input and output displacement, respectively. The upper and lower bound corresponding to each design variables are $(200,150)\mu m \leq (C_{\text{p},y},R) \leq (300,250)\mu m$ and $(150,200)\mu m \leq (C_{\text{p},y},R) \leq (100,150)\mu m$. This problem applied APDL programming with batch mode interface in ANSYS sub-problem optimization solver using PLANE82 element. The final design are $(273,2,203.2)\mu m$ and $(198.5,146.3)\mu m$. The resulted configuration of the optimum microgripper is shown in Figure 10. The performance of MF is analyzed by ANSYS and summarized in Table 1. The MF is about equally effective in both normal and simplifying design model.

![Figure 10. Final microgripper design using simplifying flexure design mode](image)

### Table 1. Final performances for microgripper optimization design

|                  | Cheng [9] | First stage | Second stage | Simplifying mode |
|------------------|-----------|-------------|--------------|------------------|
| Output displacement | 55.95 $\mu m$ | 68.15 $\mu m$ | 91.42 $\mu m$ | 90.57 $\mu m$ |
| MF               | 3.486     | 4.166       | 6.687        | 6.545            |

### 6. Conclusions

The presenting paper introduces a two-stages optimization process of handling flexure hinges design for monolithic compliant mechanism after the topology synthesis. The geometry and mathematical model of the second stage post-design are on the basis of analytical single-axis mechanical flexure hinge to obtain optimum flexure configuration and its location. The locations of hinge-like in topology synthesis are consequently verified to be the exact locations of flexures. A simplifying optimization using ANSYS can be applied for the second-stage optimum design. The performance of the second-stage design shows that the magnification factor of output displacement to input displacement is successfully improved. The presenting concept and the appropriate approach of post-design after topology optimization is helpful for general compliant mechanism design.

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