



Design and optimization of a XY compliant mechanical displacement amplifier

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Abstract. Piezoelectric actuators are increasingly becoming popular for the use in various industrial, pharmaceutical, and engineering applications. However, their short motion range limits their wide applications. This shortcoming can be overcome by coupling the piezoelectric actuators with a mechanical displacement amplifier. In this paper, a new design for a XY planar motion compliant mechanical displacement amplifier (CMDA) based on the design of a symmetric five-bar compliant mechanical amplifier is introduced. Detailed analysis with Finite Element Method (FEM) of static and dynamic characteristics of the proposed XY CMDA design is also provided. Finally, the optimization process and results to increase the Amplification Ratio (AR) of the proposed XY compliant mechanism with minimal compromise in Natural Frequency (NF) is discussed.

1 Introduction

High precision manipulation systems have a variety of uses in many industrial applications, especially where the positioning of components with high accuracy (i.e. in micrometer or nanometer scales) is required (Yong and Lu, 2009). Examples of these applications may include the alignment of fibre-optics and lasers, the positioning of specimens in a scanning-electron-microscope, the positioning of masks in lithography, cells manipulation in micro-biology, and assembly and manipulation of micro-scale components in micro-assembly applications (Yong and Lu, 2009).

Piezoelectric (PZT) actuators are micro motion generators capable of producing a high displacement resolution and low strain with high force outputs (Ouyang et al., 2008a). However, due to their relatively short motion ranges, the functions of PZT actuators become limited or infeasible for many of the above mentioned applications. One technique to overcome the mentioned shortcoming is to integrate a PZT actuator with a mechanical displacement amplifier (Ouyang et al., 2008b). Such an amplification mechanism can be based on a compliant mechanical displacement amplifier (CMDA) (Timoshenko and Gere, 1961; Howell, 2001; Hull and Canfield, 2006; Lu and Kota, 2006; Su and McCathy, 2007). A CMDA has many advantages such as no friction losses,

no need for lubrication, no tolerance, and et al. over conventional rotating pin-joint mechanisms (Lobontiu, 2002). Hence, the primary goal of a CMDA is to achieve a large output displacement in desired direction(s) for a given input displacement generated by a PZT actuator, and to keep a high positioning resolution at the same time. This, however, may cause a reduction in the structure's generated output force and the natural frequency (NF) of the mechanism (Ouyang et al., 2008b). Nevertheless, the consequent reduction of the output force due to a CMDA can be tolerated since PZT actuators are capable of generating large amount of force, and a relatively high NF can be achieved through properly structural design.

The topic of design, characteristics, and application of CMDAs is not a new one for one direction motion amplification. Numerous literatures regarding this topic can be found in Ananthasuresh and Saxena (2000); Bharti and Frecker (2004); Furukawa et al. (1995); Kota et al. (1999); Pokines and Garcia (1998); Tian et al. (2009); Yang et al. (1996). In general, the performance of a CMDA is a function of some important parameters such as the material properties and the flexure hinge profile of the mechanical displacement amplifier. Xu and King (1996) introduced and compared the performance (in terms of flexibility and accuracy) of three topologies of flexure hinge structures that

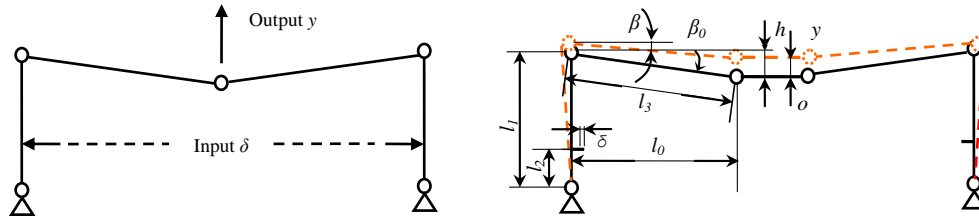


Figure 1. Symmetric five-bar topology.

comprise the elemental components of the majority of compound amplifiers. Shuib et al. (2007) also conducted a review of existing compliant technologies, their applications, and their design limitations. Ma et al. (2006) suggested that increasing the thickness of the flexure hinges will increase the NF of the CMDA. However, this will be accompanied by a significant decrease in the total output displacement. The design of compliant mechanisms with multiple optimally placed and sized PZT actuators for obtaining maximized output deflection and force was also discussed by Bharti and Frecker (2004).

Ouyang et al. (2008b) proposed a new design of CMDA based on a symmetric five-bar topology and compared its performance characteristics to other existing topologies for displacement amplification purpose. But the compliant mechanism developed in Ouyang et al. (2008b) can only produce one direction motion and has its limitations for real applications. It is nature to expand the previous design from one direction motion to two direction motions that suits most micro precision applications. The performed optimization analysis in this paper is the extension of the previous research to XY planar motions and also leads to the introduction of the parameters which have the most effect on the performance of the device.

There are some other advantages of using piezoelectric actuators with mechanical displacement amplifiers with the redundancy in design. That is to say, such a system has a potential of redundancy in providing actuations in robotics. Such redundancy may be utilized for improving the resilience of the system and for improving dynamic performance of the system (Ouyang, 2011). It should be mentioned that the designed CDMA can be used as the micro motion part for a Hybrid macro-micro mechanism (Sun et al., 2011). Therefore, the designed CMDA has its potential in different applications.

2 Topology of a symmetric five-bar structure

The design and optimization of the planar motion generator CMDA, discussed in this paper, is based on the symmetric five-bar structure proposed by Ouyang et al. (2008b). This particular CMDA was designed to be symmetrical in its configuration. As shown in Fig. 1a a PZT actuator is used to produce the input displacement required for simultaneous ro-

tations of two driving links in opposite directions. This then generates an output that can be constrained in one direction only. This topology is actually a combination of a symmetric four-bar topology and a lever arm topology (Yong and Lu, 2009). In another word the vertical bars are of lever arm structures, and between the two lever arms is a symmetric four-bar structure, see Fig. 1a. The advantage of this configuration over the symmetric four-bar topology is its high NF and Amplification Ratio (AR) in a compact size. Figure 1b also shows a pseudo-rigid body model (PRBM) of this model. The PRBM model can be used to predict the output displacement and overall stiffness of the CMDA. A detailed PRBM analysis on the symmetric five-bar structure CMDA is provided by Ouyang et al. (2008b).

It should be mentioned that, from the PRBM in Fig. 1b, the AR is directly related to the ratio of l_1/l_2 , h and l_0 . Also it is certain that the profile of the flexible hinge has significant contribution to the AR and the NF. Please note that all the design parameters and their importance on the performance of the 5 bar CMDA has been discovered and explained in details. The purpose of this paper was mainly design a planar motion generator based on the design of the 5 bar CMDA and optimize the performance of the device base on the proposed design parameters mentioned (Ouyang et al., 2008b).

3 Design requirements and constraints

As it is mentioned earlier, the symmetric five-bar structure CMDA is only capable of generating an output displacement constrained in only one direction. However, many industrial applications of micro motion devices require manipulators be capable of generating an XY planar motion with high NF. The goal of this research is to exploit the advantages of the symmetric five-bar structure CMDA and propose a novel XY CMDA capable of generating large displacements in both X and Y directions.

The proposed planar motion CMDA comprises of two identical legs located in X and Y directions as shown in Fig. 2. Each leg then is connected with a pair of arms to the output of the device. Leg A of the device translates the horizontal driving force and motion of the PZT to the Y direction motion, while the Leg B does that in the X direction motion. Please note that both legs A and B are each constrained in all degree of freedoms from the base.

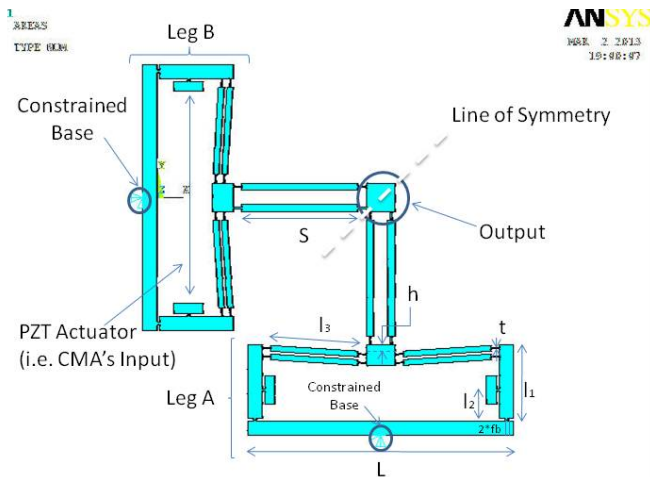


Figure 2. Planar motion generator CMDA based on symmetric five-bar topology.

According to Fig. 2, the proposed planar motion generator CMDA has the following characteristics:

1. The mechanism is a symmetric compliant mechanism with two degree of freedoms (DOF) due to two identical legs in X and Y directions.
2. A CMDA is applied to amplify the stroke of a PZT actuator for each leg.
3. The designed mechanism can be used to generate both planar and linear micro motions.
4. The moving output or end-effector of the device is design to be a cube.
5. Each leg of the planar motion generator CMDA consists of two pairs of the inclined bars of the length l_3 to increase the NF of the designed mechanism.

Some of the key parameters that have significant effect on the performance of the CMDA are also shown in Fig. 2. Please note that the parameters shown on the Leg A can be used to construct the rest of the device since the mechanism is symmetric about the illustrated line of symmetry.

Table 1 below also lists the initial values of the most important parameters used to design the planar motion generator CMDA. Please note that the minimum dimensions that parameters l_2 and L can adopt are related to the dimensions of *NPA50SG* actuators that will be integrated within each leg. S is determined to make two directional motions possible in a compact size. Parameter h also stipulates the inclined angle of the double parallel bars, referring to Fig. 1b. Finally parameters t and fb represent the thickness of the rectangular and circular flexure hinges, respectively. The thickness of the CMDA is 10 mm.

Table 1. Design parameters of the planar motion generator CMDA (unit: mm).

Parameters	L	l_1	S	t	l_2	h	fb	l_3
Dimension	93	26.696	40.5	0.35	11	2.5	0.35	30.57

Table 2. Stainless steel material properties.

Modulus of Elasticity [GPa]	200
Density [kg m^{-3}]	8000
Poisson's Ratio	0.285
Yield Strength [MPa]	703

4 Initial FEM static and dynamic analysis

A 3-D model of the planar motion generator CMDA was constructed using ANSYS Mechanical APDL (ANSYS, 2009) with the primary parameters listed in Table 1. Then by assigning the stainless steel material properties (Table 2) to the model, the CMDA was meshed using SOLID 186 element type. SOLID186 is a higher order 3-D 20-node solid element that exhibits quadratic displacement behavior. The element is characterized by 20 nodes having three translational degrees of freedom per node in the nodal X, Y, and Z directions. This element supports plasticity, hyperelasticity, creep, stress stiffening, large deflection, and large strain capabilities (Ouyang, 2011).

The initial static and dynamic FEM analyses for the following four cases are performed:

- Case 1: input displacements of $10\ \mu\text{m}$ were applied to each of the device's four inputs to obtain the mechanism's overall AR and NF.
- Case 2: input displacements of $10\ \mu\text{m}$ were applied to each of the two inputs of the Leg A of the device only to get the AR of the mechanism in vertical direction.
- Case 3: input displacements of $10\ \mu\text{m}$ were applied to each of the two inputs of the Leg B of the device only to get the AR of the mechanism in horizontal direction.
- Case 4: two forces of 5 [N] each were applied to the devices output (as shown in Fig. 3) while an input displacement of $10\ \mu\text{m}$ were applied to each of the device's four inputs. Then the mechanism's overall AR and NF were obtained by performing FEM analysis.

Please note that the AR is the ratio of the maximum output displacement (y) of the device to the input displacement (δ) created by PZT actuator:

$$\text{AR} = y/\delta \quad (1)$$

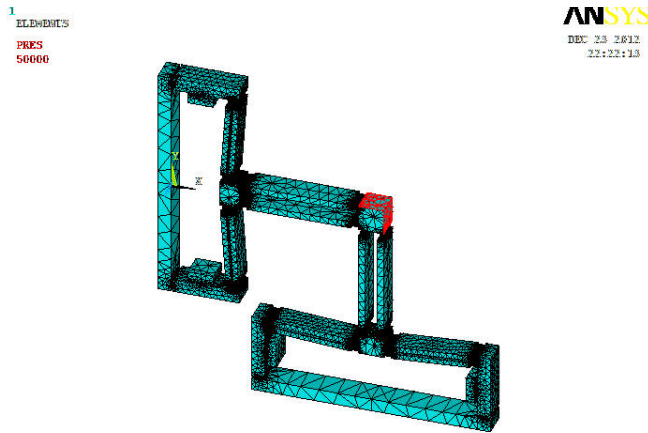


Figure 3. FEM model of the planar motion generator CMDA with the locations of the applied forces shown in red.

Table 3. Initial dynamic and static analysis results.

Case Number	NF [Hz]	AR	Cross-Talk
1	293.09	33.3	–
2	289.27	24.3	19.5
3	289.27	24.3	19.5
4	293.09	27.6	–

The cross-talk of the CDMA was calculated using the following two formulas (Loberto et al., 2004):

$$A_{yx} = \frac{\Delta y}{\Delta x_s} \quad (2)$$

$$A_{xy} = \frac{\Delta x}{\Delta y_s} \quad (3)$$

where Δy is the desired displacement in the Y direction and Δx_s is the spurious displacement found in X direction when attempt to displace in Y, while in the case of A_{xy} , Δx is the desired displacement in the X direction and Δy_s is the spurious displacement found in Y direction when attempt to displace in X.

The results obtained from the above analysis are listed in Table 3.

As it can be seen from Table 3 the mechanism has an overall AR of 33.3 and a NF of 293.09 [Hz]. The NF of the system remains the same for Case 4. However; the AR reduces from 33.3 for Case 1 to 27.6 for Case 4 due to device being under loading for this scenario, as the external force will act to resist the output motion of the CMDA. Table 3 also shows that the mechanism possess a NF of 289.286 [Hz] and an AR of 24.3 in each vertical and horizontal directions when only one direction motion is produced using only one PZT actuator. The cross-talk level between Y and X displacement as well as the X and Y displacement for both cases 2 and 3 were also found to be 19.51.

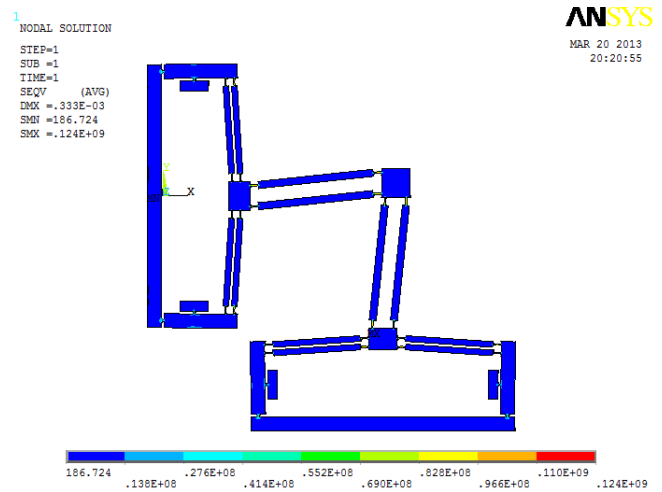


Figure 4. Contour plot of the Von-Mises stress of the initial CMDA.

Figure 4 illustrates the contour plot of the Von-Mises stress of the CMDA when displacements of 10 [μm] were applied to all of its inputs. From Fig. 4, it can be observed that the CMDA experiences a maximum stress of 124 [MPa] at the rectangular flexure hinge attached to the left vertical arm that connects the Leg A of the device to cubic output. This high stress area is marked on the figure by MX. Certainly, the maximum stress is much less than the Yield Strength of the material.

It should be mentioned that this initial design is a sub-optimal design based on previous one dimensional CMDA presented in Ouyang et al. (2008b), and it will be used as the first iteration in the following optimal design process.

5 Optimal design

5.1 Design parameters

The goal of an optimization process for the planar motion generator CMDA is to obtain the values of some significant design parameters that maximize the AR of the device with taking into account the maximum stress occurred in the flexural hinges and the maximum force available by PZT actuators to create a desired input displacement. These significant parameters were discovered in Ouyang et al. (2008b) and were regarded as having the most effects on the AR of the symmetric five-bar CMDA. Table 4 lists the design parameters used for the optimization of the planar motion generator CMDA with their corresponding allowable ranges. It should be noted that selected ranges of the design parameters are based on some sub-optimal design results.

5.2 Optimization process

As mentioned earlier, one of the optimization constraints is the maximum stress that flexural hinges can undergo. This

Table 4. Design parameters and their range used for optimization (unit: mm).

Design Parameter	Range
l_1	26.696–28.700
t	0.300–0.500
l_2	10.500–12.500
h	2.000–3.000
fb	0.250–0.400

Table 5. Design parameters obtained after optimization (unit: mm).

Design Parameter	Value
l_1	26.696
t	0.303
l_2	10.526
h	2.090
fb	0.347

was defined by stainless steel yield strength listed in Table 2. In another word during optimization process, the maximum stress (σ) experienced within flexural hinges was not allowed to exceed stainless steel yield strength (σ_y) with a safety factor (SF). Another constraint considered for the optimization was the force required to create a given amount of input displacement. This force also was set to not exceed *NPA50SG* actuators push load capacity of 1000 [N] during the optimization process.

Therefore, the optimization problem of the XY motion CMDA can be described as:

$$\text{Maximum : } AR = y/\delta \quad (4)$$

Subject to:

$$\sigma < \sigma_y/SF \quad (5)$$

$$F + (20E6) \cdot (\delta) < 1000 \quad (6)$$

where F is the force required to create a displacement at each of the device's input, and the constant $20E6$ in Eq. (5) is the actuator inherent stiffness.

As there is no explicit formula to build the connections between the design parameters and the objective function and constraints, a numerical optimization analysis using ANSYS is approached. ANSYS first order optimization method in conjunction with random design method were used to find the design parameters (within the given range) that yields the best AR for the proposed design for a constant 10 [μm] input displacement, while two constraints are satisfied at the same time. The optimized parameters under this specific condition are obtained through ANSYS and listed in Table 5.

Figure 5 is the plot of the design parameters versus the design objective (DMAX). As it can be seen from the graph and

Table 6. Optimized dynamic and static analysis results.

Case Number	NF [Hz]	AR	Cross-Talk
1	243.91	39.2	–
2	239.86	28.5	15.52
3	239.86	28.5	15.52
4	243.91	32.8	–

Table 5 design parameters l_1 , l_2 , and h have the most effect on the AR of the device. On the other hand, parameters fb and t maintained relatively constant values throughout the optimization process. The thickness t of the rectangular flexure hinge is very close to the minimum value of the allowable range, while the thickness fb of the circular flexure hinge is determined mainly by the allowable Please note that smaller values of these two parameters yield a higher AR. However, thinner flexure hinges can sustain less stress for a given input displacement and will result in the CMDA failure.

5.3 Static and dynamic FEM analysis on the optimized design

After obtaining the optimized design parameters, static and dynamic FEM analysis were performed on the proposed optimized CMDA design for four different cases. These analyses were done for the same four cases as of the initial design. Table 6 lists the dynamic and static results of the proposed optimized design for each case.

Figure 6 illustrates the contour plot of the Von-Mises stress of the optimized CMDA when displacements of 10 [μm] were applied to all of its inputs. From Fig. 6, it can be observed that the location of the maximum stress experienced by the CMDA has changed to the location of rectangular flexure hinge attached to the top horizontal arm S that connects the Leg B of the device to cubic output. This high stress area is shown on the figure by MX and the maximum stress experienced by the hinge is 132 [MPa]. Such an increase in the amount of the stress experienced by the rectangular flexure hinge is due to the fact that the thickness of the hinge has reduced from 0.35 [mm] for the un-optimized CMDA to 0.303 [mm] for the optimized one. The same reason has also led to the decrease of the cross-talk level between Y and X displacement as well as the X and Y displacement from 19.51 for the un-optimized case to 15.52 for the optimized one.

From the optimized results, it can be observed that the overall AR of the device increased from 32.8 to 39.2 while the NF decreased from 293.09 [Hz] for the initial design to 243.91 [Hz] for the optimized design. Such a result is expected as the goal of the optimization is to achieve high AR value with a small sacrifice of NR. Figure 7 shows the corresponding mode shape to the found NF for the first case. Similar results have also been obtained for cases 2 to 4. Comparing Table 6 with Table 3, one can see that the ARs are

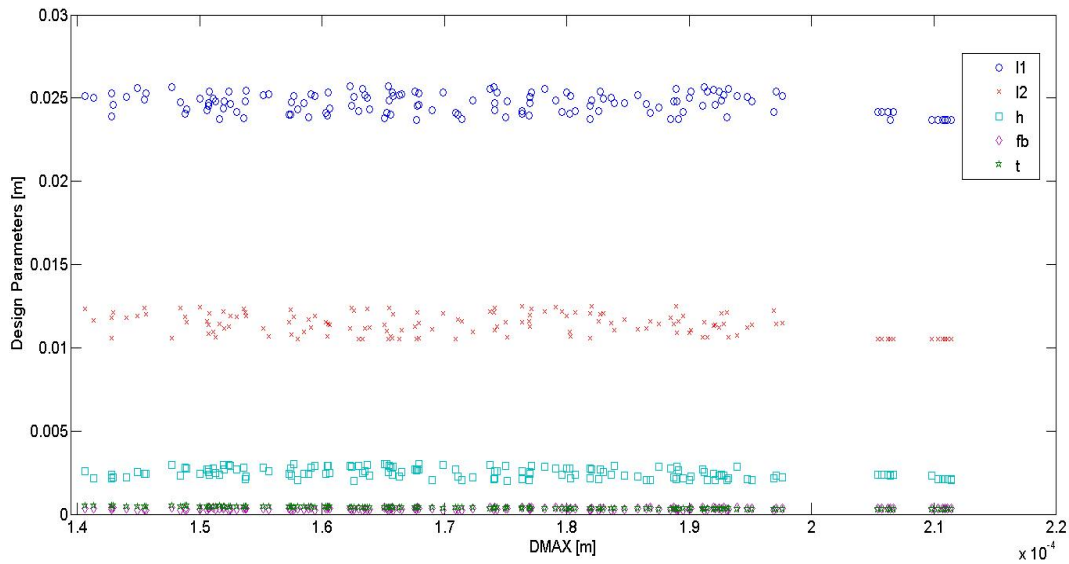


Figure 5. Plot of design parameters versus design objective (unit: m).



Figure 6. Contour plot of the Von-Mises stress of the optimized CMDA.

larger and the NRs are less for the optimized designs. All these results are expected as the objective function for the optimization is the maximum amplification ratio in the design. Figure 8. also shows the mode shape corresponding to the second case where displacements of 10 [μm] in the X direction were applied to the Leg A of the device.

6 Conclusions

In this paper, the design, modeling, and optimization of a planar motion generator CMDA based on the design of a symmetric five-bar CMDA are presented. The goal of optimization process is to achieve high amplification ratio for the device and have a relative high natural frequency. The planar motion generator CMDA is capable of converting and amplifying the linear motion of PZT actuator to a large rang planar (2-D) output motion. Some important parameters are identified and the optimized values are obtained for the purpose of the maximum output displacement of the designed CMDA for a selected specified PZT actuator. Detailed analysis with finite element method of static and dynamic characteristics of the proposed XY CMDA design is also provided. Through the optimization process, a XY compliant mechanical displacement amplifier with high AR compared to the initial sub-optimal design is provided. In the future, a full scale model of the proposed device will be manufactured and tested to compare the obtained FEM results to that obtained by real experiments.

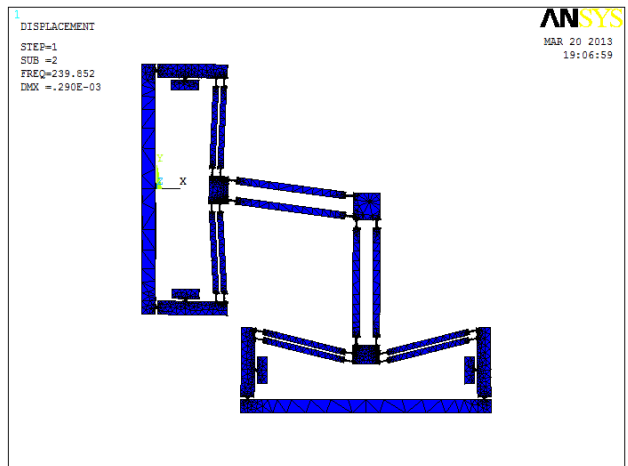
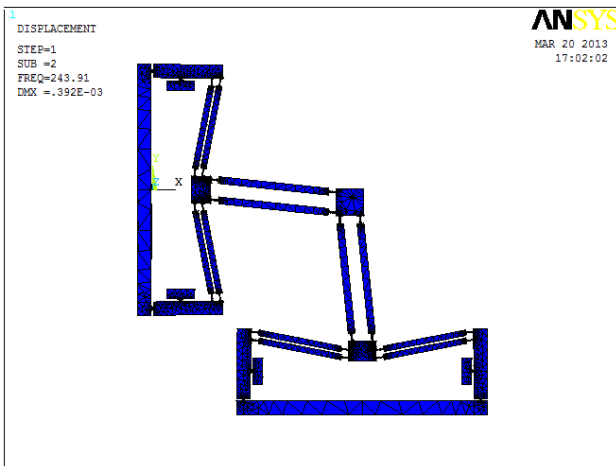
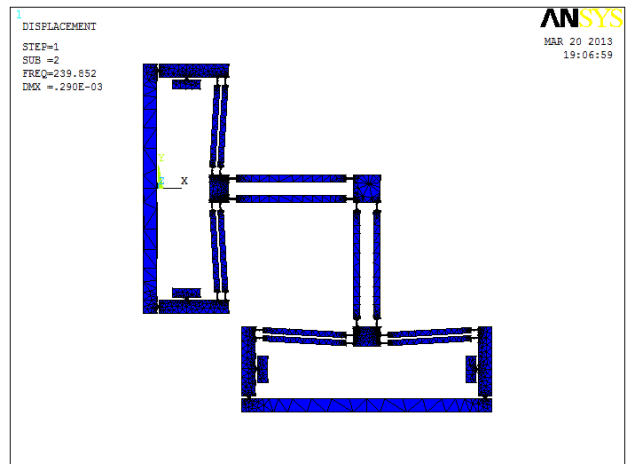
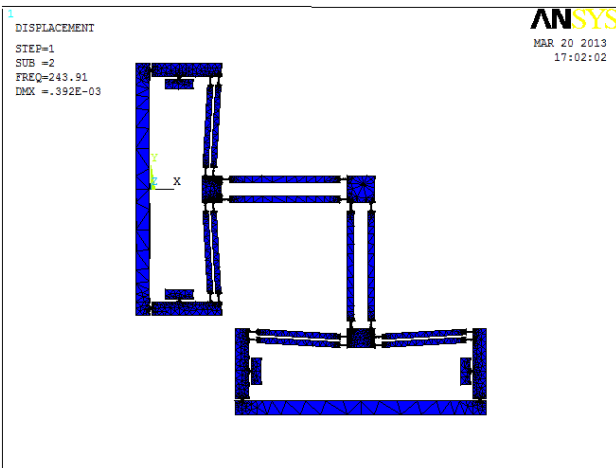
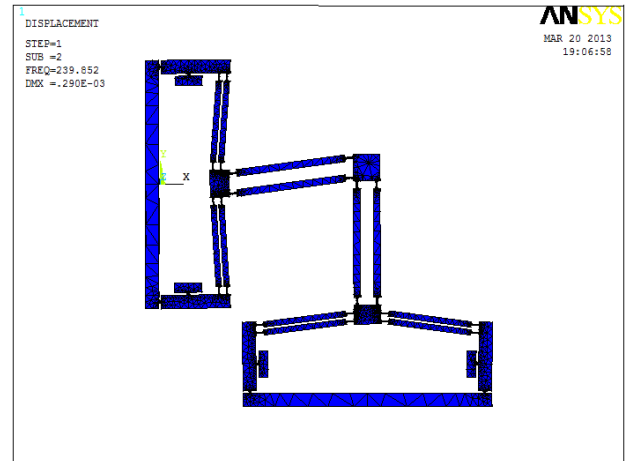
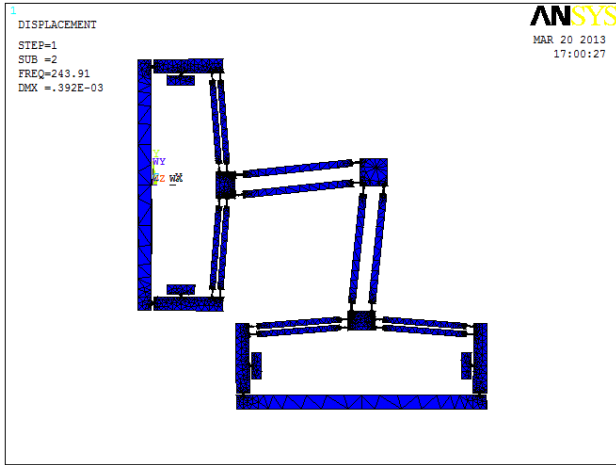


Figure 7. Mode shape of the device corresponding to the first case
 Fig. 8. Mode shape of the device corresponding to the second case.

Figure 8. Mode shape of the device corresponding to the second case.

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References

- Ananthasuresh, G. K. and Saxena, A.: On an optimal property of compliant topologies, *Structural Multidisciplinary Optimization*, 19, 36–49, 2000.
- ANSYS 12.0: Help Manual, 2009.
- Bharti, S. and Frecker, M. I.: Optimal design and experimental characterization of a compliant mechanism piezoelectric actuator for inertially stabilized rifle, *J. Intel. Mat. Syst. Str.*, 15, 93–106, 2004.
- Furukawa, E., Mizuno, M., and Doi, T.: Development of a flexure-hinged translation mechanism driven by two piezoelectric stacks, *Int. J., Ser. C*, 38, 743–748, 1995.
- Howell, L. L.: *Compliant Mechanisms*, New York, Wiley and Sons, 2001.
- Hull, P. V. and Canfield, S.: Optimal synthesis of compliant mechanisms using subdivision and commercial FEA, *Mech. Des.*, 128, 337–348, 2006.
- Kota, S., Hetrick, J., Li, Z., and Saggere, L.: Tailoring unconventional actuators using compliant transmissions: design methods and applications, *IEEE/ASME Trans. Mechatronics*, 4, 396–408, 1999.
- Loberto, G. A., Mechhl, A., Nader, G., and Silva, E. C. N.: Development of a XY piezoelectric nanopositioner, *ABCM Symposium Series In Mechatronics*, 1, 662–671, 2004.
- Lobontiu, N.: *Compliant Mechanisms, Design of Flexural Hinges*, in: N. Lobontiu, *Compliant Mechanisms, Design of Flexural Hinges*, Newyork, CRC Press LLC, 2002.
- Lu, K. J. and Kota, S.: Topology and dimensional synthesis of compliant mechanisms using discrete optimization, *Mech. Des.*, 128, 1080–1091, 2006.
- Ma, H. W., Yao, S. m., Wang, L. Q., and Zhang, Z. L.: Analysis of the displacement amplification ratio of bridge type flexure hinge, *Sensor. Actuator.*, 132, 730–736, 2006.
- Ouyang, P. R.: A spatial hybrid motion mechanism: design and optimization, *Mechatronics*, 21, 479–489, 2011.
- Ouyang, P. R., Tjiptoprodjo, R. C., Zhang, W. J., and Yang, G. S.: Micro-motion devices technology: The state of arts review, *Int. J. Adv. Manuf. Technol.*, 38, 463–478, 2008a.
- Ouyang, P. R., Zhang, W. J., and Gupta, M. M.: A New compliant mechanical amplifier based on a symmetric five-bar topology, *J. Mech. Design*, 130, 104501–104505, 2008b.
- Pokines, B. J. and Garcia, E.: A Smart material microamplification mechanism fabricated using LIGA, *Smart Mater. Struct.*, 7, 105–112, 1998.
- Shuib, S., Ridzwan, M. I. Z., and Kadarman, A. H.: Methodology of compliant mechanisms and its current developments in applications: A Review, *Am. J. Appl. Sci.*, 4, 160–167, 2007.
- Su, H. J. and McCathy, J. M.: Synthesis of bistable compliant four-bar mechanisms using polynomial homotopy, *Mech. Des.*, 129, 1094–1098, 2007.
- Sun, Z. H., Zhang, B., Cheng, L., and Zhang, W. J.: Application of the redundant servomotor approach to design of path generator with dynamic performance improvement, *Mech. Mach. Theory*, 46, 1784–1795, 2011.
- Tian, Y., Shirinzadeha, B., and Zhang, D.: A flexure-based five-bar mechanism for micro/nano manipulation, *Sensor. Actuator.*, 153, 96–104, 2009.
- Timoshenko, S. P. and Gere, J. M.: *Theory of Elastic Stability*, New York, McGraw-Hill, 1961.
- Xu, W. and King, T.: Piezoactuator displacement amplifiers: flexibility, accuracy, and stress considerations, *Precis. Eng.*, 19, 4–10, 1996.
- Yang, R. Y., Jouaneh, M., and Schweizer, R.: Design and characterization of a low-profile micropositioning stage, *Precis. Eng.*, 18, 20–29, 1996.
- Yong, Y. K. and Lu, T. F.: Kinetostatic modeling of 3-RRR compliant micro-motion stages with flexure hinges, *Mech. Mach. Theory*, 38, 1156–1175, 2009.