REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. REPORT DATE (DD-MM-YYYY)	2. REPORT TYPE	3. DATES COVERED (From - To)
10/13/2017	Final Technical Report	5/1/2014 to 4/30/2017

4. TITLE AND SUBTITLE
Self-Biased Dual Field Energy Harvesting Systems for Structural Health Monitoring Sensors.

5a. CONTRACT NUMBER

5b. GRANT NUMBER
N00014-14-1-0354

5c. PROGRAM ELEMENT NUMBER
16PR03726-00

5d. PROJECT NUMBER

5e. TASK NUMBER

5f. WORK UNIT NUMBER

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Office of Naval Research
875 North Randolph Street
Arlington, VA 22203-1995

10. SPONSOR/MONITOR'S ACRONYM(S)
ONR

11. SPONSOR/MONITOR'S REPORT NUMBER(S)

12. DISTRIBUTION/AVAILABILITY STATEMENT
Article 51 of the Core Terms and Conditions (June 2011) and Article 29 of the ONR Agency Specific Requirements (Feb 2011) encourage publication of results of this research. There are no access or dissemination controls on the research results so this project can be conducted as fundamental research.

13. SUPPLEMENTARY NOTES
N/A

14. ABSTRACT
We have shown how the current strategy of using highly compliant 2D beam shapes to harvest energy from low frequency vibrations creates performance reducing torsion. Through analytical and finite element modeling, and experimental measurements, we have shown that through deliberate design, the Elephant Harvester was able to yield a 2,675% increase in power production with only a 4% increase in resonance frequency, compared to the Zigzag harvester. Moreover, the Elephant harvester has a large effective beam tip whereby large tip mass may be placed while retaining a low profile, resulting in a low volume harvester, and subsequently large power density.

15. SUBJECT TERMS
Vibration energy harvesting; 2D beam shape, Mode shape combination; Power density

16. SECURITY CLASSIFICATION OF: a. REPORT	b. ABSTRACT	c. THIS PAGE	17. LIMITATION OF ABSTRACT	18. NUMBER OF PAGES	19a. NAME OF RESPONSIBLE PERSON
					YongKe Yan
U	U	U	SAR		19b. TELEPHONE NUMBER (Include area code) 540-231-7183

Technical Progress

2D Shape optimization for vibration energy harvesting: We initiated research by examining the current art of 2D beam shapes, by defining the Zigzag beam shape shown in the schematic drawing of Fig. 1(a). Since stress transfer from the beam to the piezoelectric material is of principal importance, a finite element stress analysis of the Zigzag beam shape was conducted, as presented in Fig. 1(b), for the first bending mode, found in Fig. 1(c). Fig. 1(b) shows the distributed nature of the stresses in the undeformed Zigzag shape, colored with stress magnitudes resulting from first bending mode vibrations, where warm colors represent tension, cold colors represent compression, with green representing zero stress. Simulations are done in the Stress Analysis environment of Autodesk Inventor Professional 2013.

Distributing stress makes for a more compliant (i.e. less stiff) beam, with low natural frequency. However, this creates a problem when determining where to place the piezoelectric layer(s). If the piezoelectric material is placed throughout the beam, it must be separated and poled in opposition to adjacent segments, as discussed by Berdy et al., due to the alternating sign of stress, as shown in Fig. 1(b). Furthermore, the closer one gets to the free-end of the beam, the lower the magnitude of stress. This results in the situation where each successive segment of piezoelectric becomes less effective and connecting like-poled segments in parallel leads to detrimental charge redistribution and loss of efficiency. For this reason, we chose to place the piezoelectric material only on the first segment of our Zigzag harvester, where stress is the highest.

In order to solve the problem of distributed stress in 2D beam shapes, we created a structure whereby stress is focused onto a single beam segment, upon which the piezoelectric material may be most effectively placed. Another problem to consider is the presence of torsional forces in the Zigzag design. Torsional stresses are not readily harvestable by the flat rectangular profile of these planar 2D beam shapes, due to the resulting orthogonality of electric field to the material polarization and electrodes, and are therefore undesirable. Harvester performance should increase if these forces are removed or ideally transformed into bending forces. Torsional forces are also present in spiral and arc-segment based designs. Berdy et al., have addressed this problem by connecting two Zigzag beam shapes at their free-ends, creating a fixed-fixed configuration. Exploiting symmetry to reduce the onset of torsional forces is valuable; however, a cantilever configuration is a better performing option to a fixed-fixed configuration. Therefore, we introduced the symmetric zigzag cantilever, termed 'Flex', and presented in Fig. 1(e). It can be noted that the stress in this design is more concentrated in the first segment near the fixed end, as shown in Fig. 1(f). This is due to the decrease in torsional forces by symmetry, allowing for a more pure bending motion to occur. In this case, rather than placing a unit force on the tip of the beam, as in Fig. 1(b), we place a half unit force on both terminating free ends of the Flex beam shape. The bending motion is also reflected in the mode shape of Fig. 1(g).

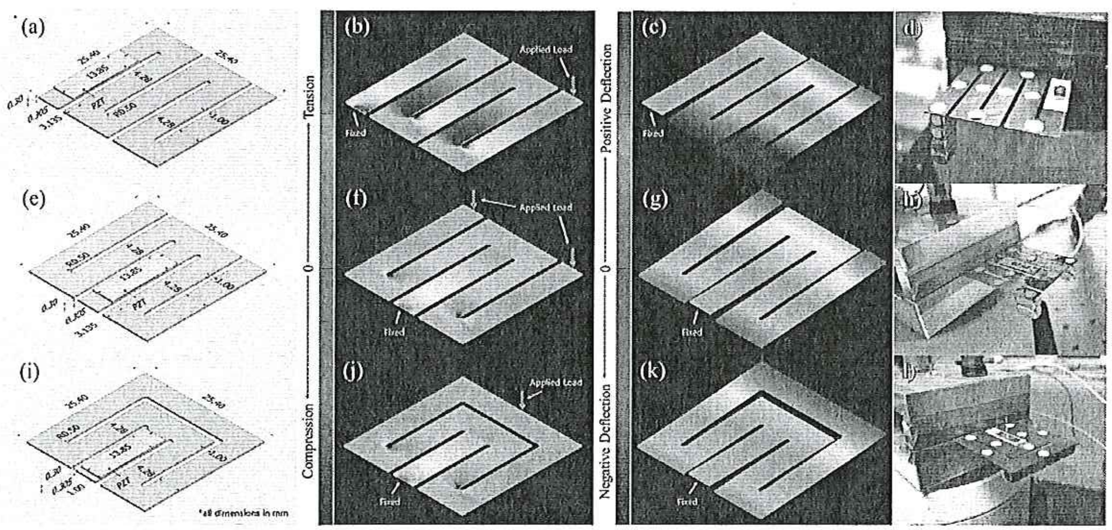

FIG. 1. Dimensioned drawing, finite element stress analysis for first bending mode, first mode shape, and picture of the fabricated device in test setup for Zigzag (a-d), Flex (e-h), and Elephant

(i-l) beam shapes, respectively. Coloring of (b,f,j) are all with respect to the same arbitrary stress scale, and coloring of (c,g,k) are all with respect to the same arbitrary modal displacement scale.

With the Flex design, torsional forces are still present due to the two ends being free (i.e. unsupported). It does seem counterintuitive; the merit of the cantilever is the fixed-free configuration, however, in the 2D case, the presence of the free-end not being collinear with the fixed-end, creates undesirable torsional effects. It is in the spirit of eliminating free-ends, but somehow maintaining a cantilever-like configuration, that we developed the closed-circuit symmetric meandering configuration, termed 'Elephant', displayed in Fig. 1(i). From Fig. 1(j), a high concentration of stress can be observed in the first beam segment, due to the optimally pure bending motion of the mode shape, rendered in Fig. 1(k). This is accomplished by joining the meanders on either side of the plane of symmetry at the top of the beam, forming a closed-circuit, whereby torsional effects are forced to cancel out.

The designed harvesters were subjected to varying frequency input (base excitation) and load resistances. Base excitation was held constant at 0.1g acceleration across all frequencies. Both frequency and resistance were varied manually until maximum values were located. Figs. 2(a-c) show the experimental results for each of the three 2D beam shapes, excited at their first bending frequency, with insets illustrating the experimentally measured mode shape and maximum power/load resistance. The predicted mode shapes have been experimentally validated, by mapping magnitude and phase information from transfer function taken at each point (reflective dots in Figs. 1(d,h,l)), to its corresponding physical location in space. It follows from Fig. 2(a) that the Zigzag harvester is capable of producing 2.93μW across 0.75MΩ at 65.6Hz. Inspecting Fig. 2(b), we see that the Flex harvester produced 32.2μW across 1MΩ at 62.0Hz base excitation frequency. In both designs, the electromechanical coupling is quite small, as no frequency shift was observable between the short-circuit ($R_L \approx 10^3 \Omega$) and open-circuit ($R_L \approx 10^7 \Omega$) frequencies. However, for the Elephant harvester, we note a substantial shift between short-circuit and open-circuit frequencies, as well as, a large power output of 81.3μW across 1MΩ at 68.125Hz, as shown in Fig. 2(c). From these findings, we can conclude the merits of the Elephant design, and how it is beneficial towards efficient low-frequency piezoelectric energy harvesting. It should be noted that while the dimensions of the piezoelectric elements for all three designs are identical, to facilitate fair comparison, since the Elephant has a shorter fixed-end segment, the piezoelectric element is mounted slightly closer to clamped boundary than the other two designs, giving it access to slightly higher stresses. This bias does not alter the superiority of the Elephant design. Rather, the relative performance of the Zigzag and Flex designs may be marginally closer to that of the Elephant design, had the piezoelectric elements been bonded closer to the clamped boundary. It should be mentioned that the electromechanical coupling coefficient exhibited by the Elephant harvester is larger than its counterparts. This results in a strong increase in the level of the harvested power. Also noteworthy is the large observed damping ratio of the Zigzag design. This is responsible, in part, for the lower observed power, and comes from the higher number of segments away from the fix end, causing more sign changes of the stress, observed in Fig. 1. Indeed, the Zigzag has four segments away from the fixed end while the Flex and Elephant have two, resulting in approximately twice as much damping.

In summary, we have shown how the current strategy of using highly compliant 2D beam shapes to harvest energy from low frequency vibrations creates performance reducing torsion. Through analytical and finite element modeling, and experimental measurements, we have shown that through deliberate design, the Elephant harvester was able to yield a 2675% increase in power production with only a 4% increase in resonance frequency, compared to the Zigzag harvester. Moreover, the Elephant harvester has a large effective beam tip whereby large tip mass may be placed while retaining a low-profile, resulting in a low volume harvester, and subsequently large power density.

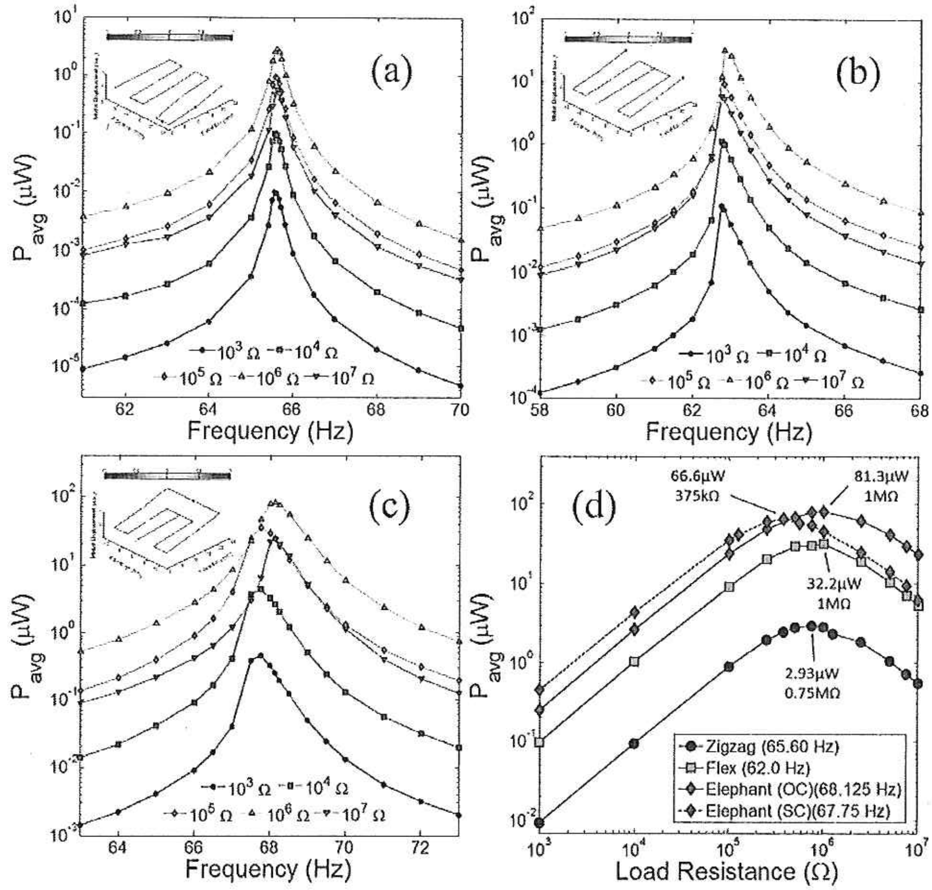

FIG. 2. Experimental results for average power production as a function of frequency and electrical load resistance with insets for measured mode shape for (a) Zigzag, (b) Flex, and (c) Elephant configurations, with (d) average power as a function of electrical load resistance at each harvester's respective resonance frequency. All input vibrations were at 0.1g base acceleration.

Mode Shape Combination in a Two-Dimensional Vibration Energy Harvester through Mass Loading Structural Modification: Vibration energy harvesters are most often designed to operate at resonance, where the resonant frequency of the energy harvester matches that of the dominant frequency of the vibration source. This matching of source and harvester mechanical impedances allows for the most efficient transfer of power from the source to the harvester. Availability of more power to the harvester results in higher transduction. The drawback of such designs is that outside the narrow resonant frequency band, little mechanical power is transduced. Addressing vibration energy harvester bandwidth and increasing power output have been the major focus of research.

Vast majority of harvester designs employ some variation/modification of a single-degree-of-freedom one-dimensional cantilever beam and the mode shapes of such beams are well known. This is especially true for the first bending mode which i has been selectively the chosen vibration mode for harvesting. The term 'one-dimensional' here implies emphasis on a specific dimension, typically the length, along which variations in geometry, materials, mass, etc. occur. One-dimension designs are not conducive towards mode-shape optimization. Thus, two-dimensional beam shapes (e.g. curved, meander, spiral) have gained research interest as they offer the ability to modify the vibration characteristics. Wu et al. have proposed a two-degree-of-freedom design by folding a two mass cantilever back on itself thereby creating a 'cutout cantilever'. This design improves on the continuous cantilever with two masses by effectively altering the stiffness matrix which was demonstrated through a lumped parameter model. This architecture allowed bringing two bending modes closer to each other resulting in a broadband response.

Building upon the Elephant beam shape discussed in the previous section, it was found that the second bending mode of the Elephant configuration possesses an intriguing mode shape relative to the first bending mode. For the harvester geometry shown in FIG 3(a), the first and second bending modes are rendered in FIG. 3(b) and (c), respectively. Inspection of these mode shapes indicates that they are nearly opposite of each other. In first bending mode, the section with highest deflection is furthest from the clamped section in the y-direction while small deflection is observed in the sections closest to the clamped section in the y-direction. In contrast, highest deflection in the second bending mode is observed in the sections closest to the clamped section in the y-direction, and very little deflection is observed in the section furthest from the clamped section in the y-direction. The fact that the first two mode shapes do not share areas of common deflection, at least to a significant degree, implies that they are nearly orthogonal to each other. Thus, the modal frequencies of each shape could be adjusted independently through mass loading structural modification. Modifying the beam structure through mass loading has the effect of lowering the natural frequency associated with each mode, as $\omega_i = \sqrt{k_i/m_i}$, where k_i and m_i are the equivalent stiffness and mass associated with each mode, resulting in modal frequency ω_i. In a single-degree-of-freedom system, this mass loading takes the form of 'tip mass' or 'proof mass', and is used to tune the harvester dynamics to that of the source dynamics for the most efficient power transfer. As a secondary effect, power output is also increased by this practice, as kinetic energy is increased due to the presence of higher moving mass in the system.

We hypothesized that there must exist a modified structure whereas mass is added at the points of highest deflection of each mode (to have the greatest effect) it results in the shifting of second bending modal frequency (by virtue of near orthogonality) near enough to that of the first bending modal frequency to create the combined mode shape as depicted in FIG. 3(d). Combining these opposite modes would result in the entire beam vibrating in phase. This mode shape is desirable for energy harvesting because the entire mass of the structure is in motion and moves in phase, thereby increasing the kinetic energy, and subsequently stress in the piezoelectric layer. Based upon this hypothesis, it was proposed that this combined mode shape would yield higher power output.

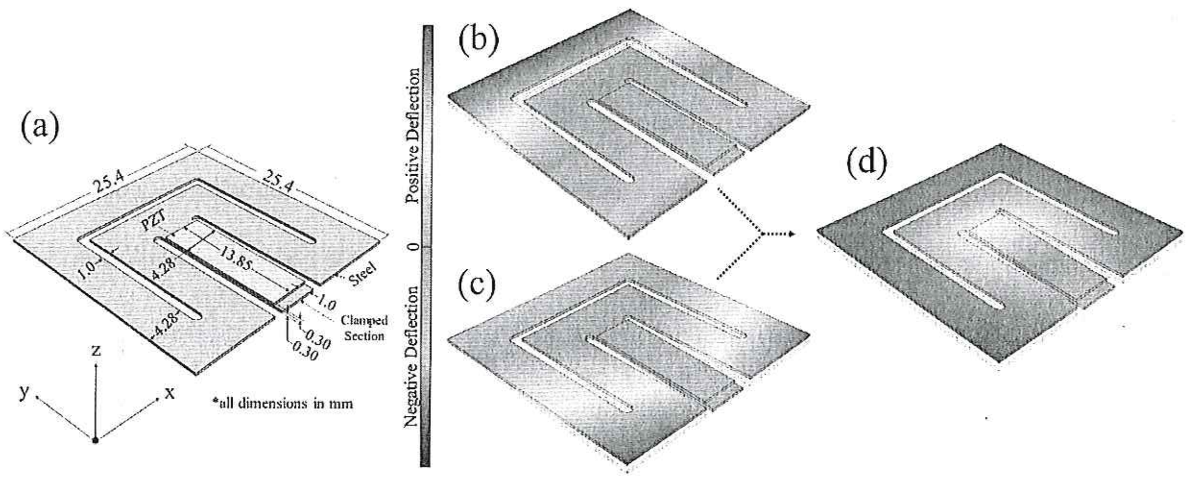

FIG. 3. (a) Dimensioned schematic of 'Elephant' two-dimensional vibration energy harvester, (b) 1st bending mode of Elephant, (c) 2nd bending mode of Elephant, and (d) proposed combined mode shape.

Starting from the zero added mass case, in FIG. 4(a), it is clear that the mode shapes predicted by the finite element modal analysis (FIG. 3(b-c)) are validated. By adding mass, the first bending mode is largely unchanged in shape, frequency, and amplitude all the way up to 2020 mg. For the second bending mode, however, while the mode shape remains the same, the amplitude of vibration increases. At 2360 mg, a change in the mode shape is observed as the curves approach one another. In both the first and second bending modes, the entire beam vibrates in phase. The first mode shape appears to pivot about the clamped end while the second mode transforms into the combined mode shape hypothesized in FIG. 3(d). Around 2700-3000 mg added mass, the mode shapes begin to vary significantly. Above 3000 mg added mass, sufficient mass is added to constitute a new system. Clearly, the modal frequency of the first bending mode is no longer roughly constant, but decreases with mass. The first mode shape of this new system appears to pivot about the middle (on the y-axis) of the beam, while second mode shape looks again like the combined mode shape and vibrates in phase.

It can be noticed that the second bending mode shapes in FIG. 4 are not symmetric about the center of the beam (in the x-axis). Originally, this was thought to be due to slight asymmetry in mass placement. However, these apparent twists in the second bending mode shape are actually caused by the coupling of the first torsion mode with the second bending mode. The first torsion mode does not contribute to energy harvesting, so it is not readily observable in the transfer function between the base excitation and the voltage output. It is however evident in the transfer function between base excitation and vibrometer reading. The torsion mode shapes as a function of added mass are plotted in FIG. 5 along with several additional measured mode shapes of the second bending mode where it most strongly interacts with the torsion mode. After confirming the hypothesis of the existence of a combined mode shape, the next step in the study was to examine if this mode shape really produces more power. Power production is typically measured in response to broadband excitation (white noise/random), or at a particular frequency or set of discrete frequencies (sine sweep). The maximum power produced was measured to be 43 μW at 2360 mg added mass. This mode shape corresponds with the mode shape hypothesized in FIG. 3(d). Harvester half-power bandwidth was not significantly affected by the mass loading structural modification (approximately 1% of the given modal frequency). This coupled mode shape was found to produce the highest level of power under single frequency sine wave excitation, compared to all other mode shapes and mass configurations, due to the extra kinetic energy in the system and extra stress in the piezoelectric layer provided by the entire beam moving in phase.

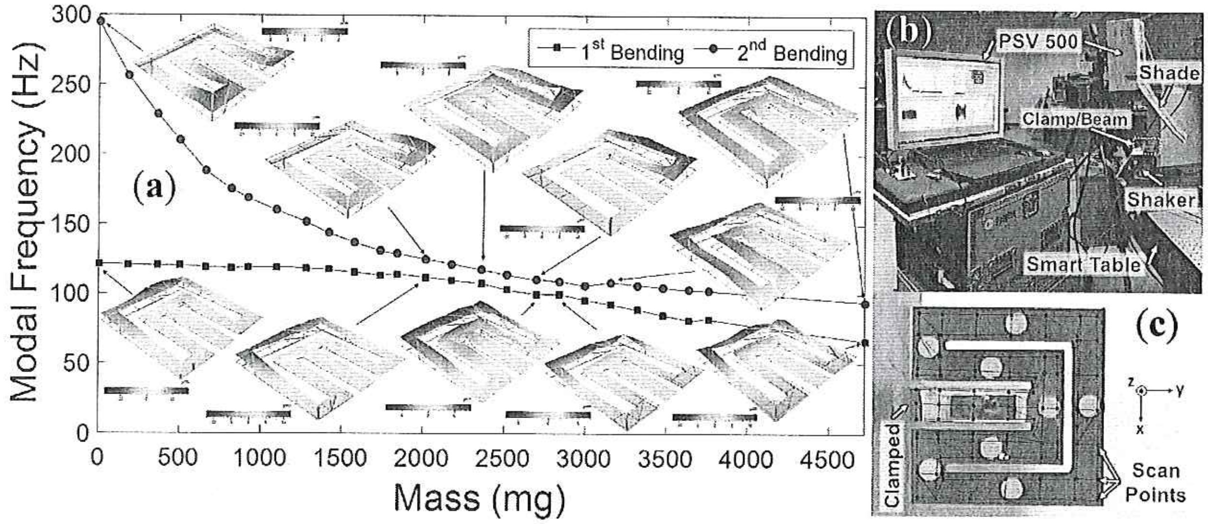

FIG. 4. (a) Selected scanning laser vibrometer measured mode shapes for the 1st and 2nd bending modes of the Elephant beam as a function of mass loading, (b) experimental setup, (c) scanning pattern over the beam, and detailed view of the clamped beam.

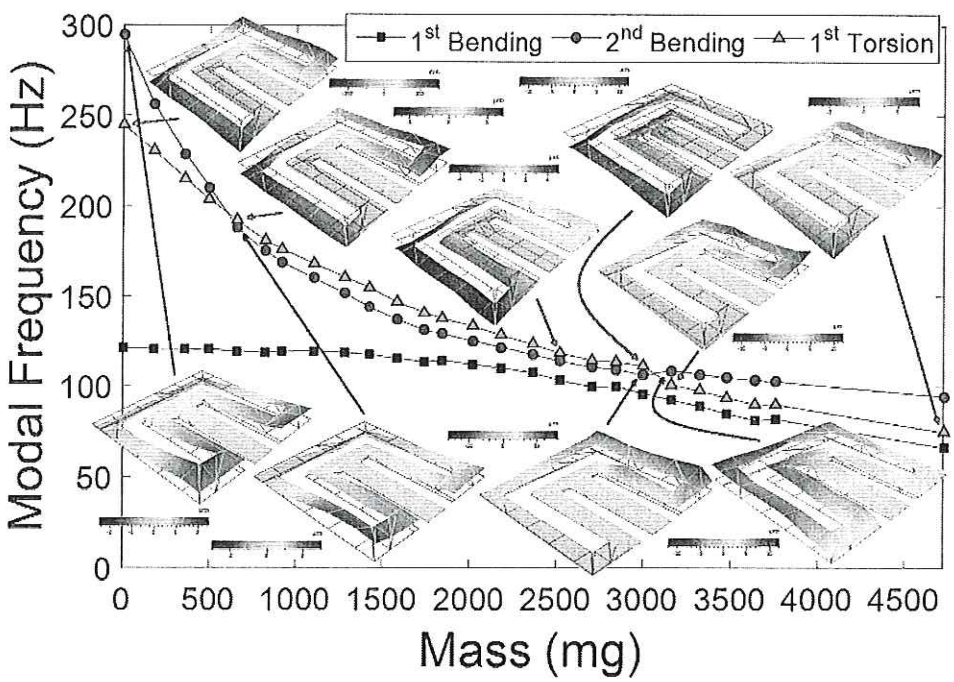

FIG. 5 Selected scanning laser vibrometer measured mode shapes for the 2nd bending and 1st torsion modes of the Elephant beam as a function of mass loading.