Design and Analysis of a Novel Compliant Tensile Testing Module Based on Buckled Fixed-Guided Beam

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ABSTRACT The need to accurately measure the mechanical properties of freestanding thin films such as carbon nanotube reinforced nanocomposite thin films and various buckypaper is increasingly urgent. This paper presents the design and control of a novel compliant tensile testing module based on buckled fixed-guided beams aimed at accurately measuring the mechanical properties of freestanding thin films. The piezoelectric stack actuator that can provide high resolution for the output displacement is chosen to actuate the tensile testing module. The advantages of the module are its compactness, no friction, no wear, low cost and good repeatability. The performance is further evaluated by the established analytical model using the elliptic integral approach and the nonlinear finite-element analysis (FEA). A metal prototype of the tensile testing design has been fabricated and the performance of the fabricated prototype is investigated through open-loop displacement response experiments. Experimental results demonstrate the effectiveness of the proposed ideas for the mechanism. The design holds great potential applications in measuring mechanical properties of freestanding thin-film materials such as carbon nanotube reinforced nanocomposite thin films and various buckypaper as well as other freestanding thin-film materials.

INDEX TERMS Compliant mechanism, negative stiffness, buckling beam, piezoelectric stack actuator.

I. INTRODUCTION

With the development of ultrahigh precision manipulation engineering, more attentions have been focused on ultrahigh precision small-scale measurement systems [1]–[3]. Small-scale mechanical testing systems mainly consist of the fixturing parts to hold the specimen, actuation parts to apply a force or displacement history, the parts of sensors to measure the force and the specimen deformation. To date, the small-scale mechanical tester has been studied by many researchers. Ruud et al. from Harvard University [4] developed a micro tensile test platform which uses compliant mechanism driven by piezoelectric actuators to measure the mechanical response of self-supporting thin film materials. The limitation of this system is that the stroke of small piezoelectric actuators is too small. Sharpe et al. from Johns Hopkins University [5] developed JHU microsample tensile experimental platform. The system has air bearings to overcome the friction in the stretching process at one end of the fixture, but this system is too complex. Other tensile test platforms developed by National Institute of Standards and Technology [6] and other scientific research institutions [7], [8] are driven by one or two piezoelectric stacks. In particular, Shigeaki Murata et al. from Toyota Motor Corporation [9] used piezoelectric stacks in series to increase the driving stroke, and carried out compression tests to measure the mechanical response of polymeric thin films. In our research, piezoelectric stacks are also used to drive the system.

Existing mechanical testing systems employ lead screws and gears to integrate these parts into a whole mechanism testing system [7], [8]. However, assembling these components can lead to measuring errors. The backlash brought by the gears may lead to deformation measuring errors, and the friction between the fixturing parts to hold the specimen and the actuation parts may lead to force measuring errors.

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errors. Some designs use linear air bearings to overcome the frictions, but the complexity of such measuring systems is undesirable in real applications [8]. Unlike traditional gears, compliant mechanisms exhibit the merits of no friction, no wear, low cost and high repeatability. In addition, the flexure elements of compliant mechanisms can be used for force measurements because of its linear force-displacement characteristics [10]. Hence, a compliant mechanism made from a monolithic piece of a material is adopted in our work to make a small-scale mechanical tensile testing system.

While there exist many kinds of flexures [10]–[12], this work focuses mainly on negative stiffness mechanism. In some literatures negative mechanism works as a displacement amplifier, which is used to increase the motion range of the piezoelectric actuator. The existed amplifiers can be classified into lever mechanism, bridge-type mechanism and Scott-Russell mechanism [10]. Dong et al. [11] developed a highly efficient bridge-type mechanism based on negative stiffness. The negative mechanism acts as an additional energy stream to maintain the total energy constant in the system. Zhang et al. [12] developed a bidirectional torsional negative stiffness mechanism for energy balancing systems. In their design the negative stiffness mechanism is coupled to the target positive stiffness system to produce an energy balancing system. Braun et al. [13] developed positive–negative stiffness actuators. In their design a passive negative stiffness mechanism is employed in conjunction with a tunable positive stiffness mechanism to reduce the complexity of actuators. Wang et al. [14] studied cushion performance of cylindrical negative stiffness structures. In their research normal and gradient cylindrical negative structures are designed and approaches to improve their cushion performance are investigated. Zhou et al. [15] developed bistable mechanism with linear negative stiffness and large in-plane lateral stiffness. In their research analytical models based on a comprehensive elliptic integral solution of bistable mechanism are established. Erwin et al. [16] developed a negative stiffness mechanism for measuring dissipation in a mechanical oscillator at low frequency. Holst et al. [17] investigated buckling modes and deflection of fixed-guided beams in compliant mechanisms. The method of the arithmetic-geometric mean and descending Landen transforming is adopted in their research [17]. Hoetmer et al. [18] developed negative stiffness building blocks for statically balanced compliant mechanisms. In their research a negative stiffness is added to cancel the positive stiffness of the compliant mechanism. Kashdan et al. [19] designed and evaluated a meso-scale negative stiffness system fabricated with selective laser sintering (SLS) technology. Niu et al. [20] developed and analyzed a quasi-zero stiffness isolator using a slotted conical disk spring as the negative stiffness structure. The configurative parameters are optimized to achieve a wide displacement range around the equilibrium position in their research. Dong et al. [11] developed a highly efficient bridge-type mechanism based on negative stiffness.

The proposed compliant module can deliver and transmit force, energy and repeatable motion between the actuator and the specimen. The conceived module also functions as the actuator guide. Due to the compactness of the monolithic module, it is more convenient to orient the directions of the testing axis compared with other kinds of testing modules. Integrated fabrication of the module provides perfect alignment of the measurement axis.

The main contribution of this work is the mechanism design of a novel compliant tensile testing module. The novelty of the proposed mechanism design is to meet the need to measure the mechanical properties of free-standing thin films such as carbon nanotube reinforced nanocomposite thin films and various buckypaper, as well as polymeric thin films. As compared with previous mechanical testing designs, the designed tensile testing module exhibits the attractive merits of more compact size, no friction, no wear, low cost and good repeatability. Analytical modelling, finite-element analysis (FEA) simulations and experimental investigations have been conducted to verify the effectiveness of the compliant tensile testing system.

**FIGURE 1. Module arrangement of the tensile testing system.**

**II. DESIGN, SIMULATION & MODELING**

Fig. 1 shows the schematic diagram of the micro scale tensile platform system to be built, which includes the micro scale tensile module, the sensor signal processing module and the computer. The specimen will be located at the gap between the fixed grip of the fixture and the moving grip of the fixture. The upper part of the tensile module has a home position and a tensile operation position with loading. When the actuator generates output force to drive the fixed grip of the fixture, the moving grip of the fixture generates displacement along the x-axis under the guidance of the compliant mechanism. At the same time, the non-contact displacement sensor and the stress sensor measure the real-time signals and transmit the signals to the signal processing module.

Before designing and fabricating the structures in detail, a performance prediction should be performed and analyzed. Scholars has developed several methods to evaluate the performance of inclined fixed-guided beams [21], such
as pseudo-rigid-body models [22]–[24], elliptic integral models [25]–[27] and finite element models [28]. In all the above mentioned methods, the elliptic integral solution is generally considered to be the most accurate method for analyzing large deflections of thin beams in compliant mechanisms [25]. In this work, the elliptic integral approach is employed to develop analytical models and the nonlinear FEA is conducted to verify the models.

A. ELLIPTIC INTEGRAL METHOD

As shown in Fig. 2, the fixed-guided beam is deformed when driven by the force \( f, m \) denotes the reaction torques and \( \alpha \) denotes the angle of the force \( f \) with respect to the \( X \) axis. The length of the fixed-guided beam is \( L \).

![FIGURE 2. Model parameters of the inclined fixed-guided beam.](image)

Firstly, the negative stiffness mechanism is designed and conducted. If a straight beam is located with an inclined angle and there is a force acting on one end of the beam, it is regarded as a fixed-guided beam undergoing bending deformations. The core of the elliptic integral approach is the Bernoulli Euler equation. As shown in Fig. 2, for a coordinate \( s \) along the beam’s length, the Bernoulli Euler equation for the moment \( m \) in the beam at point \( P(x, y) \) gives:

\[
El\frac{d\beta}{ds} = m + fy \cos \alpha - fx \sin \alpha
\]

where \( x \) and \( y \) represent the corresponding coordinates of point \( P \), \( \alpha \) represents the tilted angle of the beam, \( \beta \) represents the angle between the tangential line of the deflected beam and the horizontal line at point \( P \), \( E \) is the Young’s modulus of the beam’s material, and \( I \) represents the rotational inertia of the inclined beam.

To solve the Bernoulli Euler equation, variables \( x \) and \( y \) are eliminated:

\[
El\frac{d\beta}{ds} = m + fy \cos \alpha - fx \sin \alpha
\]

Integrating Eq. (2) with respect to \( (\beta-\alpha) \) yields

\[
0.5El(d\beta/ds)^2 = C - f \cos(\beta - \alpha)
\]

The constant \( C \) can be derived from the integral of the above differential equation:

\[
C = (2k^2 - 1)f
\]

The parameter \( \gamma \) is introduced as follows:

\[
\cos(\beta - \alpha) = 2k^2 \sin^2 \gamma - 1
\]

Then the expression can be simplified as follows:

\[
0.5El(d\beta/ds)^2 = 2k^2 \cos^2 \gamma
\]

\[
ds/d\beta \text{ and } d\beta/d\gamma \text{ can be represented by the function of } \gamma:\]

\[
ds/d\beta = \sqrt{El/(4k^2 \cos^2 \gamma)}
\]

\[
d\beta/d\gamma = 4k^2 \sin \gamma \cos \gamma / \sqrt{1 - \cos^2(\beta - \alpha)}
\]

Then the expression can be further simplified as follows:

\[
ds = (ds/d\beta)(d\beta/d\gamma)d\gamma = \sqrt{El/f} \cdot d\gamma / \sqrt{1 - k^2 \sin^2 \gamma}
\]

The non-dimensional reaction force can be obtained as follows:

\[
\sqrt{\frac{L^2f}{EI}} = \int_{\Gamma_1}^{\Gamma_2} \frac{d\gamma}{\sqrt{1 - k^2 \sin^2 \gamma}}
\]

The first-kind and second-kind incomplete integral are:

\[
F(\Gamma, k) = \int_{0}^{\Gamma} \frac{d\gamma}{\sqrt{1 - k^2 \sin^2 \gamma}}
\]

\[
E(\Gamma, k) = \int_{0}^{\Gamma} \sqrt{1 - k^2 \sin^2 \gamma} d\gamma
\]

The horizontal and vertical displacements of the buckled fixed-guided beam are derived as follows:

\[
a_B = -\sqrt{EI/f} \left\{ 2k \sin \alpha(\cos \Gamma_2 - \cos \Gamma_1) \right\} + \cos \alpha[2E(\Gamma_2, k) - 2E(\Gamma_1, k) - F(\Gamma_2, k) + F(\Gamma_1, k)]
\]

\[
b_B = -\sqrt{EI/f} \left\{ 2k \cos \alpha(\cos \Gamma_2 - \cos \Gamma_1) \right\} + \sin \alpha[2E(\Gamma_2, k) - 2E(\Gamma_1, k) - F(\Gamma_2, k) + F(\Gamma_1, k)]
\]

The kinetic energy of the mechanism should be regarded as 0. Using the quasi-static design principle, the energy equation is:

\[
\frac{dU_b}{ds} = \frac{El}{2} \left( \frac{d\beta}{ds} \right)^2
\]

The energy function represented by the parameter \( \gamma \) is derived as follows:

\[
U_b = \int_{0}^{L} dU_b = \int_{0}^{L} 2k^2f \cos^2 \gamma \sqrt{\frac{El}{f}} \frac{d\gamma}{\sqrt{1 - k^2 \sin^2 \gamma}}
\]
The mathematical model of the force and displacement can be used to determine the geometric specifications of the fixed-guided beam. These equations based on the elliptic integral approach are solved by MATLAB software.

**FIGURE 3.** The energy change process of the bending fixed-guided beam.

**B. CALCULATION RESULTS**

Fig. 3 shows the energy change process of the bending fixed-guided beam. In Fig. 3, the in-plane width of the beam is 0.5 mm. As shown in Fig. 3, as the deformation increases, the elastic energy starts to increase at first and then the magnitude of the elastic energy reaches the first peak. After that, the elastic energy decreases as the deformation increases. The elastic energy stops decreasing and starts to increase again at a minimum. Compared to the force-displacement relationships in Fig. 5, it is known that when the elastic energy stops increasing and begins to decrease, the displacement of the deformation still increases, but the force begins to decrease. At the point of the first peak in Fig. 3, it can be concluded that the stiffness of the fixed-guided beam changes the direction from the positive to the negative compared to the curve in Fig. 5(a). The coordinate of the first peak point in Fig. 3 is (0.0067, 0.0138). It is because the direction of the force changes the positive to the negative at the same point as shown in Fig. 5(a). According to Hooke’s Law, the stiffness changes from the positive to the negative.

The elastic energy equals the work done by the force. The magnitude of the elastic energy can be obtained by integrating the force with respect to the displacement. As shown in Fig. 5(a), when the force $F$ keeps positive, the elastic energy increases with the increasing displacement. When the force reaches 0, the magnitude of the elastic energy (the work done by the force) stops increasing and reaches a maximum as shown in Fig. 5(a) and Fig. 3.

**TABLE 1.** Units for the negative stiffness beam properties.

| Symbol | Parameter | Value  |
|--------|-----------|--------|
| $L$    | Length    | 75.0 mm|
| $h$    | In-plane width | 1.0 mm|
| $w$    | Thickness  | 15.0 mm|
| $\theta$ | Inclined angle | 5°    |

beam is 5°. The above size of the beam is chosen according to the piezoelectric transducer’s (PZT) specification, the PZT’s output performance and manufacturing restrictions.

**FIGURE 4.** Deflection shapes of the fixed-guided beam.

**C. SENSITIVITY ANALYSIS**

Fig. 5 shows the simulation results of nonlinear stiffness. In the $oxy$ coordinate system, the $x$ axis is the horizontal displacement of the beam tip under the applied force. The origin of the $x$ axis stands for the original position of the beam tip. When the applied force exceeds the critical buckling value, the inclined beam will buckle under the compression load, and the elastic force $F$ will decrease with the increasing displacement of the beam tip. The elastic force $F$ keeps positive until the beam tip approaches a critical unstable position where the elastic force changes to zero. Then, the elastic force becomes negative as the displacement of the beam tip increases, which indicates the beam will snap to the other stable position. By regulating the tilted angle and dimensions of the beams, different force-displacement curves can be obtained to fulfill practical applications. In real tensile testing applications, only the displacement range with constant negative stiffness is useful. The range between the point with the maximum positive force and the maximum negative force meets the requirements of our tensile testing applications. The influences of the length, the in-plane width, and inclined-angle on the force-displacement performance are investigated here.

These curves in Fig. 5 are obtained using the analytical model. One geometrical parameter of the fixed-guided beam is altered, when other geometrical parameters are kept constant. As shown in Fig. 5(a), Fig. 5(b) and Fig. 5(c), to generate a larger range of constant negative stiffness for the flexure mechanism, a smaller in-plane width $h$, a longer length $L$, and a larger inclined angle $\theta$ are required. It can be seen from Fig. 5(d) that the out-of-plane width $w$ has a small influence on the range of constant negative stiffness.
FIGURE 5. (a) The force displacement relationship as the in-plane width $h$ varies from 0.5 mm to 1.0 mm; (b) The force-displacement relationship as the length $L$ varies from 72.0 mm to 77.0 mm; (c) The force displacement relationship as the tilted beam’s angle varies from $5^\circ$ to $10^\circ$; (d) The force displacement relationship as the out-of-plane thickness varies from 10 mm to 15 mm.

It can be seen that the force increases with the increasing of the in-plane width $h$ in Fig. 5(a). According to Eq. (18), the theoretical stiffness $k$ increases with the increase of the in-plane width $h$. According to Hooke’s Law, if the deflection is constant, the force $F$ increases with the increase of the stiffness $k$. Thus, the force $F$ increases with the increase of the in-plane width $h$.

Fig. 5(b) shows that the beam length $L$ has a relatively small influence on the force-displacement relations of the beam when the length $L$ (72 mm $\sim$ 77 mm) is much larger than the in-plane thickness $h$ (0.5 mm $\sim$ 1 mm). It is also shown that the force increases with the increasing of the inclined angle $\theta$ in Fig. 5(c). It is observed that the force $F$ increases with the increasing of the out-of-plane width $w$ in Fig. 5(d). According to Eq. (18), the theoretical stiffness $k$ increases with the increase of the out-of-plane width $w$. According to Hooke’s Law, if the deflection is constant, the force $F$ increases with the increase of the stiffness $k$. Thus, the force $F$ increases with the increase of the out-of-plane width $w$. It can be concluded that the in-plane width $h$ of the inclined beam has the greatest influence on the force-displacement relations of the beam as compared with the other three parameters. However, the smallest width $h$ and the inclined angle $\theta$ are constrained by the manufacturing technique. The longest length $L$ is restricted by the size of the PZT actuators and other compactness requirement. And the largest angle $\theta$ is limited by the size of the PZT actuators and the maximum tensile strength of the Al material.

FIGURE 6. Equivalent spring diagram of the tensile testing system.

D. DESIGN OF THE ENTIRE COMPOUND FLEXURE MECHANISM

Fig. 6 shows the equivalent spring diagram of the tensile testing system. $K_l$ and $K_u$ are the stiffnesses of the lower
part of the flexure and the upper part of the flexure. \( \lambda_1 \) and \( \lambda_2 \) are the deflections of the lower the flexures and the upper flexures. \( K_s \) is the stiffness of the specimen when the specimen is stretched at the length of \( \lambda_s \). \( F_t \) is the controllable stimulus generated by the actuator. The relationship between the imposed force and the displacement response can be obtained by applying force balance. The expressions are listed as follows:

\[
\frac{\Delta \lambda_s}{\Delta F_t} = \frac{K_l}{K_l K_a + K_l K_s + K_a K_s} \quad (17)
\]

Since a simple leaf spring mechanism behaves like a cantilever and has a small rotation, which is parasitic motion in our applications, the design of the parallelogram flexure mechanism is adopted. The parallelogram flexure mechanism is realized by combining two simple springs in series. It is a more symmetrical design, in which the curvilinear of the simple spring is compensated. The stiffness of such a compound flexure mechanism will be half of that of a simple leaf spring mechanism since it is composed of two leaf springs in series. Because of a small magnitude of the tilted value, the stiffness of the parallelogram flexure with tilted angle is similar to the compound flexure without tilted angle. The theoretical output stiffness of the mechanism is as follows:

\[
k = \frac{2Ewh^3}{L^3} \quad (18)
\]

where \( L, h, \) and \( w \) represent the length, in-plane width and the thickness, respectively.

![FIGURE 7. FEA simulation results and the analytical model results of the force-deflection relationship [10].](image)

### E. FEA SIMULATION STUDY

To verify the force-displacement performance of the fixed guided beam, finite element analysis (FEA) was conducted with respect to the fixed guided beam. The offset force technique is widely used in a finite element analysis to verify the performance of the designed negative stiffness mechanism [10]. Fig. 7 shows FEA simulation results and the analytical model results based on the elliptic integral method of the force-deflection relationship for the negative stiffness mechanism in Xu’s work [10]. As shown in Fig. 7, it is observed that the FEA method predicts similar results as compared with the analytical elliptic integral model. The simulation result validates the accuracy of the developed analytical model. In this study it is shown that the motion range of the adopted fixed-guided beam is 7.7 mm. Scholars have demonstrated that the results of the analytical model can be further improved by considering the axial deflections in the solution [17].

| Symbol | Parameter Description | Value |
|--------|-----------------------|-------|
| \( E \) | Modulus of Elasticity | 68.90 GPa |
| \( \mu \) | Poisson’s Ratio | 0.33 |
| \( \rho \) | Density | 2730 kg/m\(^3\) |
| \( \sigma_0 \) | Tensile Strength | 310 MPa |
| \( \sigma_t \) | Tensile Strength, Yield | 276 MPa |

In detail, the beam is meshed using 3D elastic beam element in ABAQUS software. The material parameters of the simulated mechanism adopted are listed in Table. 2. The structural parameters are listed in Table. 1. All 6 degrees of freedom are constrained on the fixed surfaces. The input displacements in Fig. 8 applied on the input surface are 10 mm. All of the mesh models are created by the element C3D8R. In Fig. 8, the analysis step is static stress analysis and it belongs to nonlinear geometric analysis. During the nonlinear analysis, the command NLGEOM is turned on. In Fig. 9, the analysis type is linear perturbation analysis, and the selected eigen solver is set as Lanczos.

![FIGURE 8. FEA simulation results of (a) Mises and (b) Maximum principal stress distribution of the stage with 10 mm translational displacement applied.](image)

The output stiffness of the mechanism is analyzed by FEA. When a constant force is applied to one end of the moving grip along the X axis to examine the corresponding output displacement, the output stiffness can be calculated by comparing the force over the output displacement along the actuation direction. It is found that the output stiffness is 0.05 N/mm, indicating the FEA results match well with the
FIGURE 9. Modal analysis by FEA: (a) 1st modal, (b) 2nd modal, (c) 3rd modal, (d) 4th modal.

analytical calculations. The deviations of the analytical result compared with the FEA simulation results is 8.1%.

Fig. 8(a) and (b) show the FEA simulation results of Mises stress and Maximum principal stress distributions of the stage with 10 mm translational displacement applied respectively. The stress distribution of the stage is simulated with input deflections of 10 mm along $X$ axis, which is applied on the moving grip of the module. The maximum stress values are 4.48 MPa, which are all far below the yield stress (MPa) of the aluminum (6061-T6) materials. Therefore, the tester will operate within elastic deformation range over the whole history displacement within 10.0 mm, which is long enough for deformation measurement.

The modal analysis of the stage is implemented through FEA. As shown in Fig. 9, the first four natural frequencies of the stage are 8.76, 90.07, 93.34, and 106.13 Hz. It is obvious that the first mode is translational motion in the $X$ axis. The second frequency, as well as the third and the fourth frequencies, is over ten times of the first resonance frequency, which demonstrates a robust translational motion along the $X$ axis.

FIGURE 10. Experimental setup for open loop displacement response test.

F. EXPERIMENTAL RESULTS

A prototype of the tensile testing stage based on compliant mechanism shown in Fig. 10 has been fabricated to demonstrate the performance of the proposed design. The major parts of the prototype are made of aluminum materials (Al-6061). The properties of the adopted aluminum material are listed in Table. The displacement response performances of the mechanism are investigated. A series of sinusoidal voltage signals with peak-to-peak values ranging from 10 mV to 160 mV are applied to the piezoelectric actuator to drive the tensile testing stage.

The test setup is shown in Fig. 10. It consists of the piezoelectric actuator (PK2FSP2, America THORlabs), a signal generator (AFG1062, America Tektronix), a power amplifier (BP4610, Japan), a laser displacement sensor (LK-G300, Japan). An optical table is chosen as the foundation support which shields the environmental interference. The driving signal is generated by the signal generator and then is amplified by the amplifier before inputting to the piezoelectric actuator. The displacement response is detected by the laser displacement sensor and recorded. The laser displacement sensor from Keyence corporation can afford an accurate measurement with a 10 nm resolution. In real tensile testing experiment, the specimen is fixtured between the fixed grip of the fixture and the moving grip of the fixture.

FIGURE 11. Tracking result of 5 Hz sinusoidal trajectory.

FIGURE 12. Tracking result of 10 Hz sinusoidal trajectory.

The open-loop tests are conducted. The test results are given in Fig. 11 and Fig. 12. The peak-to-peak amplitude of the sinusoidal voltage signal input to drive the stage in Fig. 11 is 160 mV. The peak-to-peak amplitude of the sinusoidal voltage signal input to drive the stage in Fig. 12 is 100 mV. The desired sinusoidal trajectories are 11.0 $\mu$m and 5.0 $\mu$m respectively. The frequency in Fig. 11 is 5 Hz, while it is 10 Hz in Fig. 12. In the two cases, the actual positioning
trajectories track well with the desired input. It is shown that the positioning errors are approximately 14.0% and 16.0%, respectively in Fig. 11 and Fig. 12. The positioning errors are mainly caused by the machining errors of the tensile stage and the nonlinear output performance of the piezoelectric actuator. It is concluded that the desired performance of the designed stage has been achieved. Meanwhile, some other researches, such as a more precise machining of the stage and closed-loop control strategies, should be investigated in future work, so as to reduce the errors and realize a better test performance.

III. CONCLUSION
In summary, this paper presents the design, modeling, FEA simulation, fabrication and experimental study of a novel compliant tensile testing module based on buckled fixed-guided beams. The design of the tensile testing module is aimed at measuring the mechanical properties of free-standing thin films such as carbon nanotube reinforced nanocomposite thin films and various buckypaper, as well as polymeric thin films. The analytical model of negative-stiffness structure has been derived. Sensitivity analysis has been performed. Further the performance of the negative stiffness structure has been verified by performing FEA analysis. The motion range of the adopted fixed-guided beam is 7.7 mm. Static analysis and modal analysis through finite element method have been conducted. Results show that the maximum stress of the negative stiffness satisfies the material requirement and the design exhibits robustness. A metal prototype has been fabricated and tested. The experimental results show that the desired performance of the designed stage has been achieved. The ideas presented in this paper can further use in other one-dimensional micro positioning stages or constant force stages. In future work, the compliant module will be integrated with the actuation parts and the control parts as well as the sensors. In addition, the mechanical properties of free-standing thin films such as carbon nanotube reinforced nanocomposite thin films and various buckypaper will be studied using the designed measuring system.

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