Development of a Modular and Compliant Microassembly Platform with Integrated Force Measurement Capabilities

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INTRODUCTION

There are many challenges associated with microassembly that are not present in macro-assembly such as poorly understood interactions and difficulty in taking measurements at the microscale. As a result, full automation of microassembly has not yet been achieved [1]. This has caused microassembly to take longer than automated macroassembly and to become a bottleneck in many assembly processes. Additionally, during the microassembly process, force measurements are needed in order to protect the parts from being damaged [2]. It is therefore beneficial for a microassembly platform to be able to measure microscale forces and to allow compliance in order to avoid damage. Previous work has been done to produce such a system used for micromanipulation of rigid and non-rigid parts [3]. However, this system was limited to force and compliance measurements along one axis and allowed only one predefined amount of compliance.

This paper presents the design and analyses of a modular assembly platform capable of measuring forces, which due to its modular nature, can be combined to achieve different levels of compliance with one or two degrees of freedom. The experimental characterization that conforms to the theoretical and simulation results demonstrates that the performances of the developed platform are suitable for automated microassembly applications.

The paper is organized as follows. First, the design and fabrication of the platform is presented. Afterwards, theoretical and simulation results are discussed. The experimental results and their comparison with the previous analytical results end the paper.

DESIGN OF APPARATUS

The following section describes the design of a modular, compliant platform for micro-manufacturing. Each platform module of the device was required to have a force measurement range of 0-10mN and the ability to be combined with other modules in order to achieve two degrees of freedom. The platform, as shown in Fig. 1, is a single module consisting of a table attached to a base by four fixed-fixed beams, a *Femtotools* FT-S270 force sensor, and a S*marAct* SL-0610 linear positioner used to manipulate the force sensor.

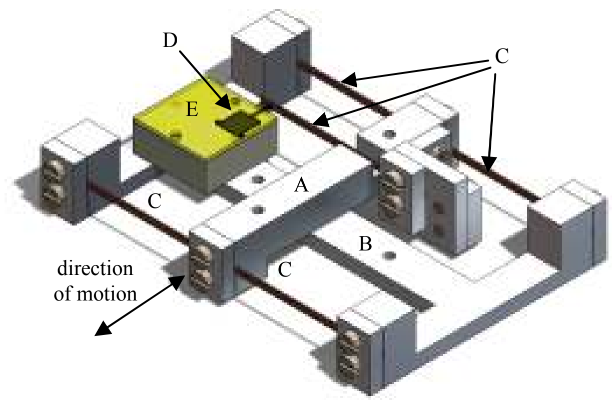

Fig. 1: Design of a modular, compliant platform consisting of: (A) table, (B) base, (C) beams, (D) force sensor, and (E) linear positioner

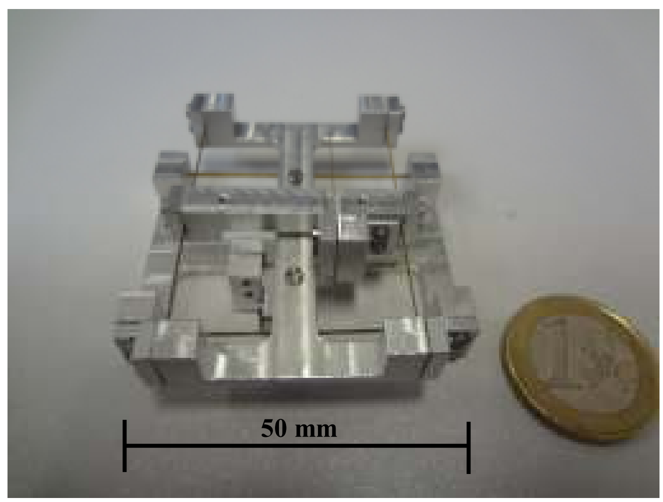

Fig. 2: Compliant platform prototype in 2-DOF configuration

The fixed-fixed beams are parallel, allowing positive and negative displacement of the table in the plane passing through all four beams and normal to the beam axes. Before the platform is used, the linear positioner is used to move the force sensor into contact with the cantilever beam until a desired preload is achieved. Upon which the coordinate system of the table is reset to zero. While the platform is in use, the force sensor is kept stationary, resulting in an equivalent system of five springs in parallel (i.e., the four fixed-fixed beams and one cantilever beam). Subsequently, the deflection of the cantilever beam at the point of contact with the force sensor is therefore equal to the displacement of the table with respect to the base during use. Using Euler-Bernoulli beam equations, the contact force between the sensor and the cantilever beam can be used to determine both the displacement of the table with respect to the base as well as the component of the force applied to the table in the direction of the displacement, refer to Fig. 3.

In order to measure different force ranges, the position of the force sensor can be modified. Different levels of compliance can also be achieved by replacing the fixed-fixed beams with beams of different material or thickness. These changes cannot be performed while the device is in use so it is proposed that multiple modules are created with varying force ranges and compliance. These modules can be placed one on top of another with the same orientation in order to obtain intermediate ranges of force measurement and compliance, stacked modules being analogous to springs in series. Two modules can also be stacked on top of one another with one module rotated 90 degrees about the axis normal to the tabletop in order to obtain 2-DOF force measurement and compliance.

STRUCTURAL ANALYSES

A. EULER-BERNOULLI BEAM ANALYSES

The behavior of the platform can be described using Euler-Bernoulli beam theory where the deflection of the cantilever is given by

$$\Delta_c = \frac{P_c l_c^3}{3 E_c I_c} \tag{1}$$

where P_c is the load applied to the beam by the force sensor, l_c is the length of the beam, E_c is the elastic modulus of the beam, and I_c is the area moment of inertia. The deflection of the ends of the fixed-fixed beams can be represented by the Euler-Bernoulli equation for the deflection of the center of a fixed-fixed beam with twice the length and a point load applied to the center. It is also important to note that since there are two sets of collinear beams, the load must be halved when solving for the deflection. As a result, a multiplication factor of four is applied to the standard equation for a point load applied to the center of a fixed-fixed beam. The equation for the deflection of these beams therefore is given by

$$\Delta_{ff} = \frac{4 P_{ff} l_{ff}^3}{192 E_{ff} I_{ff}} \tag{2}$$

where P_{ff} is the load applied to the beam by the force acting on the table, l_{ff} is the length of the beam, E_{ff} is the elastic modulus of the beam, and I_{ff} is the area moment of inertia.

Since the range of force applied to the table was chosen as 0-10 mN, it is interesting to know the corresponding load acting on the force sensor. The maximum force, calculated with the above Euler-Bernoulli equations, was found to be 0.54 mN. The resulting magnitude gain is therefore 0.054. Since the range of the force sensor is 0-2 mN, the device should be capable of measuring any forces applied to the table within the prescribed range. Next, the beam equations were used to determine the load on the force sensor when the table is displaced the prescribed maximum of 50 µm. This force was determined to be 0.54 mN, which is equal to the sensor load for the maximum prescribed load on the table.

B. COMSOL STRUCTURAL ANALYSES

In order to validate the Euler-Bernoulli models used to predict the mechanics of the structure, static structural analyses of the system were performed using *COMSOL Multiphysics* simulation software. First, a single module was simulated.

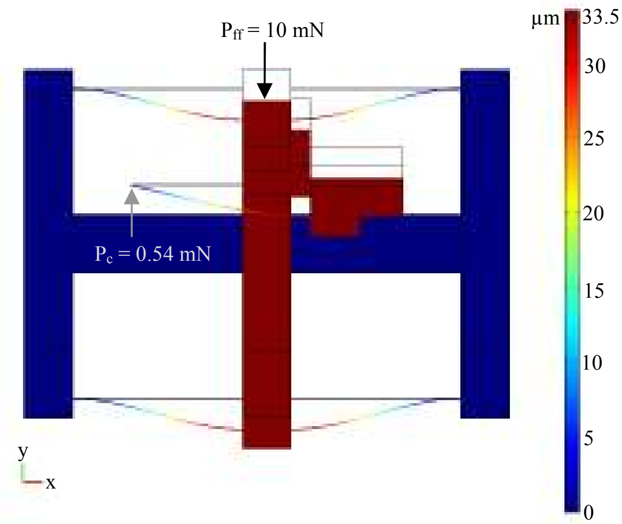

Fig. 3: COMSOL static force analyses of 1-DOF module configuration with 10 mN force applied in the negative y-direction, showing displacement in the negative y-direction

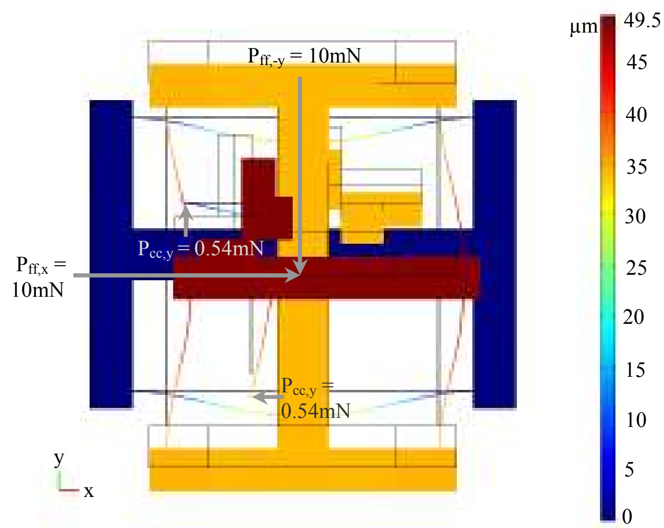

Fig. 4: COMSOL static force analyses of 2-DOF module configuration with 10 mN forces applied in both the negative y- and positive x-directions showing total displacement

The bottom face of the base of the platform was fixed and point forces were applied to the table and cantilever beam based on the calculations from the Euler-Bernoulli analyses. The force applied to the table was 10 mN in the negative y-direction and the force applied to the cantilever beam was 0.54 mN in the positive y-direction. The displacement of the fixed end of the cantilever beam was equal and opposite the displacement of the free end of the beam, thus resulting in zero total displacement of the free end of the beam. This validated the parallel spring-mass system behavior and above theoretical modeling.

Next, a simulation was performed for two modules stacked with one module rotated 90 degrees. A force was applied to the table with components in the negative y-direction and positive x-direction with values of 10 mN for each component. Additional forces were applied to the cantilevered beams in directions opposite the corresponding table force components with magnitudes of 0.54 mN. As in the case with the single module, the free ends of the cantilevered beams had zero net displacement, conforming to the parallel spring-mass system behavior for each degree of freedom. It should be noted that due to their staged orthogonal motion constraints, the two degrees of freedom are decoupled and the zero net displacement for each cantilever beam at its point of contact with the sensor is maintained,

MODAL ANALYSES

A. LUMPED MASS MODEL

In order to determine the modal characteristics of the mechanism, a lumped mass model was developed. The model used two equivalent springs, one that represented the fixed-fixed beams and another used to represent the cantilever beam. In this model, which is shown in Fig. 5, the equivalent stiffness and damping coefficients of the fixed-fixed beams are given by $4k_{ff}$ and $4c_{ff}$ respectively. The mass of the assembly of parts on the free ends of the cantilever beams is given by m. Using *SolidWorks*, the mass of this assembly was determined to be 4.42 g. The stiffness and damping coefficients of the cantilever beam are given by k_c and c_c respectively. In order to determine the damping coefficients, the Euler-Bernoulli beam equations were used. By solving for the deflection for a given load, the stiffness could be described as

$$k_{ff} = \frac{P_{ff}}{\Delta} \quad \text{and} \quad k_c = \frac{P_c}{\Delta} \tag{3}$$

The load used to calculate these stiffness values was in 10mN and 0.54 mN for the fixed-fixed beam and cantilever beam respectively. The effective stiffness of the lumped parameter model is the sum of the cantilever and fixed-fixed stiffness. The stiffness values were calculated for deflections in both the x and z-directions and are given in Table 1.

Table 1. Stiffness of fixed-fixed and cantilever beams in x and z directions in N/m.

	Fixed-fixed	Cantilever	Effective
x-direction	277	15	292
z-direction	12300	657	12957

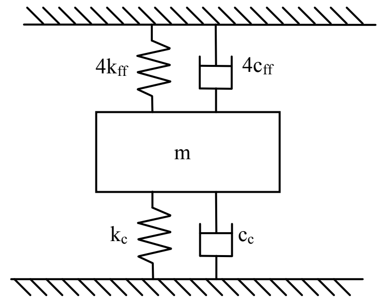

Fig. 5: Lumped-mass model of a single module of the compliant table; where k_{ff}, c_{ff}, k_c and c_c are the stiffness and damping coefficients of the beams, and m is the mass of the assembly

The effective stiffness values were used to determine the natural frequency of the first mode for both the x and z directions. The equation used to solve for the natural frequencies is given below.

$$\omega_{n,x;n,y} = \sqrt{k_{x;y}/m} \tag{4}$$

The resulting natural frequencies of the first modes of the lumped-mass system were 40.9 Hz in the x-direction and 273 Hz in the z-direction. It is important to note that the mass used for these calculations was for a single module and thus the weight is simply that of the center bar. When using two modules, the weight includes both the center bar and the weight of the top module.

B. COMSOL EIGENFREQUENCY ANALYSIS

In order to validate the findings of the lumped mass model, a *COMSOL* eigen-frequency analysis was performed. The first two modes of the system were found to have natural frequencies of 43.5 Hz and 286 Hz, the former being the first mode in the x-direction and the latter being the first mode in the z-direction. These natural frequencies correspond to the natural frequencies found using the lumped mass model with a percent difference of 6.3% and 4.8% of the magnitude of the former natural frequencies. The first three modes of the system are shown in Fig. 6. All other modes of vibration were found to have frequencies of greater than 1000 Hz.

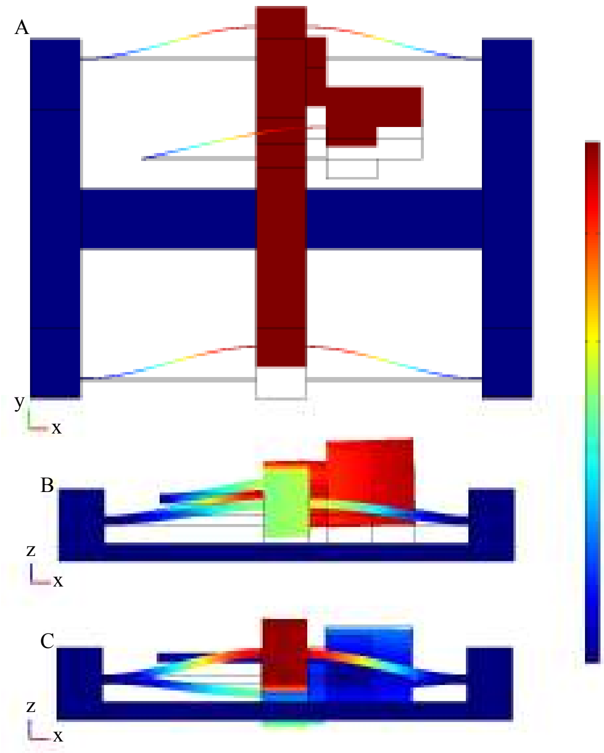

Fig. 6: COMSOL plot of the first three modes of the single module system, where **A** *is the first mode,* **B** *is the second mode and* **C** *is the third mode, corresponding to eigenfrequencies of 43.5Hz, 286Hz and 427Hz respectively*

EXPERIMENTAL VALIDATION

In order to validate the results of the COMSOL simulations and Euler-Bernoulli beam analyses, experiments were performed. The experimental apparatus included the compliant platform, two laser position sensors, a *Femtotools* FT-S270 force sensor and a *NanoCube* XY piezo stage. The sensors and stage were connected to a *dSPACE* processor board and the data was processed using *ControlDesk* and *Simulink*.

The compliant platform, shown below, did not include a linear positioner as in the original design. The linear positioner had a very long procurement time so an external *NanoCube* positioner was used in its place. While the *NanoCube* positioner was more capable than the *SmarAct* SL-0610 linear positioner would have been in the 1-DOF system—it had a higher resolution and closed loop rather than open loop feedback—its size made it impossible to implement in the 2-DOF system. As a result, the experimental validation of the system was limited to the 1-DOF system. The setup of the experimental apparatus can be seen in Fig. 8.

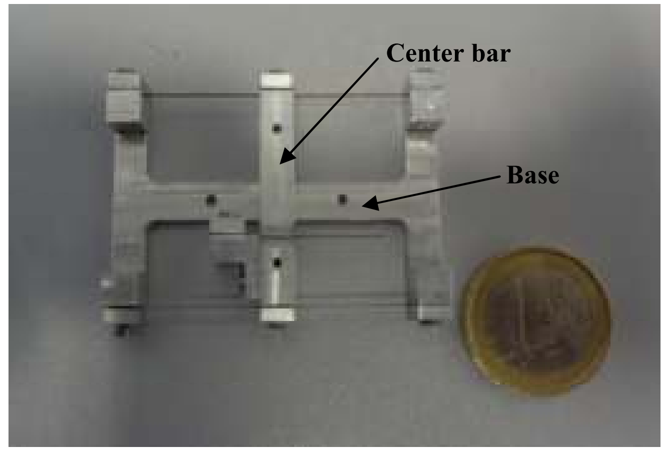

Fig. 7: 1-DOF compliant platform prototype without an integrated force sensor or linear positioner

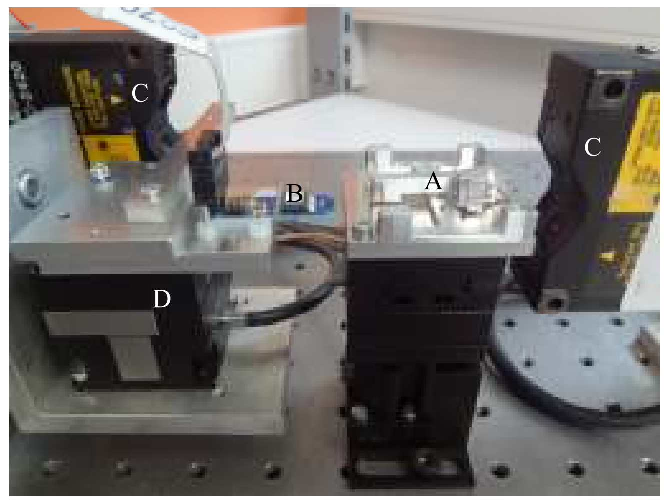

Fig. 8: Experimental apparatus including (A) compliant platform, (B) force sensor, (C) laser position sensors, and (D) **NanoCube** *XY stage*

In order to determine the stiffness of the fixed-fixed beams, the force sensor was moved into contact with the center bar using the *NanoCube* XY stage. Once contact was made, the positions of the force sensor and center bar were recorded. The force sensor was then displaced 9 µm toward the compliant platform along the axis of the center bar using the *NanoCube* XY stage. The resulting force was then measured and filtered using an open-loop, simple moving average with a sampling period of 100 data points or 0.1 s.

The force measurement data collected had very large disturbances with a frequency of approximately 110 Hz. The displacement showed no such disturbances. By plotting the force with respect to displacement over the range of [0.78, 8.86] µm, the stiffness of the system was determined to be 252 N/m and showed a strong linear correlation between the two variables. This stiffness is 11.1% less than the predicted stiffness of 277 N/m. Plots of the experimental data can be seen in Figs. 9-11.

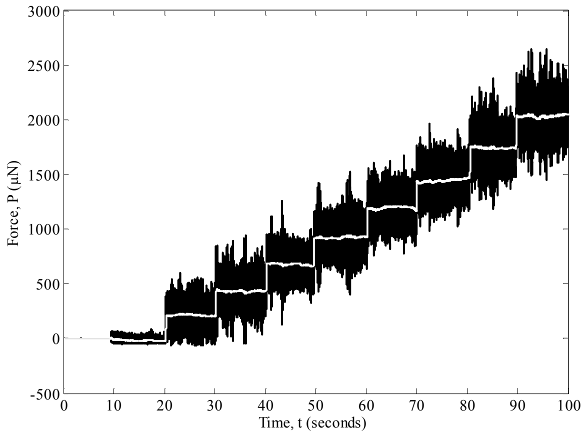

Fig. 9: Contact force with center bar with respect to time, where the white line is the filtered data obtained from the unfiltered measurements in black

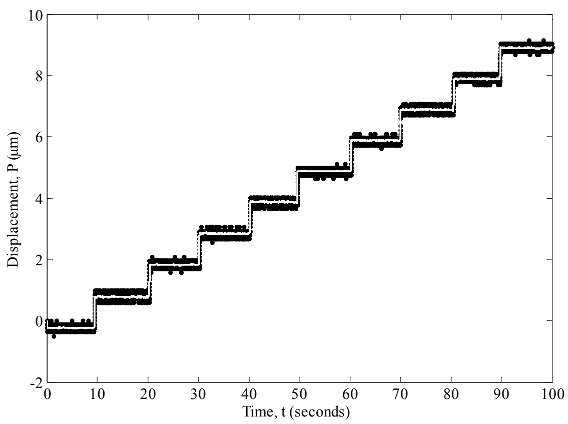

Fig. 10: Displacement of center bar with respect to the base of the compliant platform (y-axis) plotted with respect to time (x-axis), where the white line is the filtered data obtained from the unfiltered measurements in black

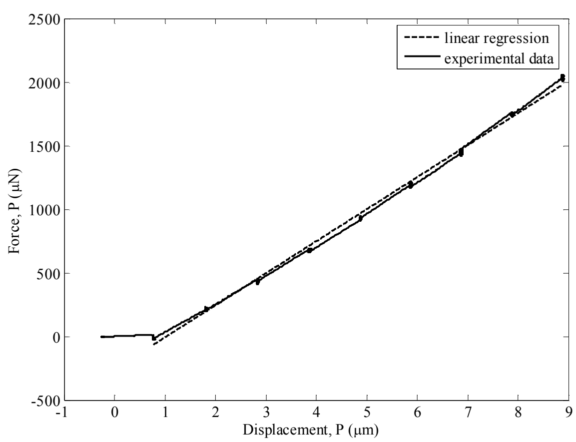

Fig. 11: Contact force with the center bar with respect to the displacement of the center bar, where the black line is the filtered experimental data and the dotted line is a linear regression of that data

RESULTS AND DISCUSSION

The static analyses using *COMSOL* showed that Euler-Bernoulli beam theory can be used to accurately describe the displacement of the cantilevered beams. Therefore, the contact force between the sensors and the cantilever beams can be used to determine both the force applied to the table as well as the compliance of the table in either a single or double module configuration. The analyses also showed that forces in the range of 0-10mN and compliance in the range of 0-50μm can be achieved.

The experimental analysis also confirmed that the stiffnesses obtained from Euler-Bernoulli beam theory and COMSOL were correct. However, large, unexpected disturbances in force made it difficult to determine the natural frequency of the system. The disturbances occur at a frequency of 110 Hz, which is approximately twice the frequency of the first mode in the x-direction determined from beam theory and COMSOL, which were 40.9 Hz and 43.5 Hz, respectively. It is possible that in this mode shape the center bar is locate at a node and therefore does not show any displacement as was the case in the experimental data. However, the force at a node should also be zero. Other possibilities are that vibrations of computer equipment on the work table caused internal excitation in the system or that electoral disturbances in the force sensor itself were responsible for these errors.

In future work, the compliant platform will be modified in order to increase the natural frequency of the system. The disturbances in force measurements will also be addressed by performing experiments on a damped breadboard with no contact to computer equipment or other moving parts unrelated to the experiments. Integration of force sensors into the platform will also be a priority in future developments as will testing of the platform with actual microassembly processes.

ACKNOWLEDGEMENTS

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