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Development of Silicon-on-Glass Electrostatic Microactuators with a High Aspect Ratio and vertical configuration

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Abstract

Actuators are the key components that produce the mechanical output and perform physical functions of particular systems. Electrostatic actuators are widely used in micro-electromechanical systems (MEMS) applications for their combining versatility and simple technology. In this thesis, an electrostatic microactuator fabricated on silicon-on-glass (SOG) is studied.

The electrostatic microactuator utilizes a vertical configuration of comb-drive working plane to the mounting plane of driving components, which are different from the planar configuration of comb drives. The vertical configuration provides more space area for more comb drives to be arrayed, making low driving voltage possible. The electrostatic microactuators have high-aspect-ratio structure, high structural strength, and low cost. The proposed microactuator can be used in the hard disk drives (HDD) to meet their high-capacity and high-performance trend. The microactuator can be used as a dual-stage actuator in the head-positioning system to move the magnetic heads, while providing high speed and accuracy of the positioning.

To fulfill the high-bandwidth and high-accuracy requirements of the electrostatic actuators, the flexures that support the microactuator should have a high stiffness and at the same time be flexible enough to have the desired displacement in the operational direction. The straight-beam, folded-beam, symmetric-quad and asymmetric-quad flexures were studied. The analytical and finite element analysis proved that the symmetric-quad flexure has the highest stiffness among the four flexures.

Fabrication of the electrostatic microactuator was realized by using silicon-on-glass wafer process. The fabrication started with a 200 μm -thick n-type silicon wafer, and then bonded to a Pyrex glass substrate, and finally the comb structures were formed by high-anisotropy deep reactive ion etching (DRIE) technique. The fabrication processes

of glass deep wet etching, wafer bonding and high-aspect-ratio DRIE have been developed. This SOG process is alternative to the SOI process for some applications but at a much lower cost.

The prototypes of the microactuators were characterized. With a driving voltage of 40 V, the microactuators with straight flexures, folded flexures and asymmetric-quad flexures have the displacement of 0.94 μm , 1.73 μm and 0.26 μm , respectively. They have the primary resonant frequency of 7.2 kHz, 5.89 kHz and 15.85 kHz, respectively. The characterization results were compared and discussed with the theoretical results. The fabrication process, especially the DRIE process, generated various geometrical tolerances of the comb drive actuators, basically including the profile tolerance and undercut tolerance. Studies showed that these fabrication influences have significant impact on the device performance, such as the electrostatic force, flexure stiffness, displacement and quality factor. The study also shows that the 3-D electrostatic microactuator with symmetric-quad flexures or asymmetric-quad flexures can be used as the dual-stage actuator in HDD, and in other applications such as inertial sensors.

Keywords: electrostatic, microactuator, high-aspect-ratio, silicon-on-glass, flexure, symmetric-quad flexure, DRIE, wafer bonding.

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Nomenclature

α	Ratio of moment inertia, $\alpha = I_a / I_b$
β	Reciprocal of characteristic distance, $\beta = 1/\delta$
δ	Characteristic distance of the lateral driven structures in comb drives (m)
ε_0	Vacuum permittivity, $\varepsilon_0 = 8.85 \times 10^{-12}$ F/m
ω_n	Natural frequency of the actuator (Hz)
ζ	Damping ratio of the electrostatic microactuator, $\zeta = C / (2m\omega_n)$
θ	Tapered angle of the electrodes etching profile
ϕ	Tapered angle of the flexure etching profile
μ	Absolute viscosity of air or dynamic viscosity of air (N·s/m ²)
ρ	Density of material (kg/m ³)
A	Area of electrode plate (m ²)
A_m	Effective area in the comb drive (m ²)
A_p	Top area of the comb plate in the comb drive (m ²)
A_b	Top area of the flexure in the comb drive (m ²)
A_t	Top area of the connecting truss in the comb drive (m ²)
C	Capacitance (F)
C_p	Parasitic capacitance (F)
d	Air gap between the two electrode plates (m)
E	Young's modulus of the material (Pa)
F_e	Electrostatic force generated in the microactuators (N)
F_x	Force applied to the actuator in the x - direction (N)
F_y	Force applied to the actuator in the y - direction (N)
f_n	Natural frequency of the actuator (Hz)
I	Moment inertia of the flexure (m ⁴)
I_a	Moment inertia of the vertical flexure segment (m ⁴)
I_b	Moment inertia of the vertical flexure segment (m ⁴)
K_x	Stiffness or spring constant of the flexure in the x - direction (N/m)
K_y	Stiffness or spring constant of the flexure in the y - direction (N/m)
L	length of the flexure (m)
l_a	Length of central vertical flexure segment (m)

l_0	Length of extended vertical flexure segment (m)
l_b	Length of transverse flexure segment (m)
M	Moment applied to the flexure (N·m)
m	Mass of the movable electrodes of the actuator (kg)
n	Numbers of the movable comb fingers
P_{cr}	Critical load of flexure buckling (N)
Q	Quality factor of the actuator, $Q = 1/(2\zeta)$
Q_d	Quality factor due to the air damping under the comb drive
Q_δ	Quality factor due to the air damping above the comb drive
Q_c	Quality factor due to the air damping between comb drive electrodes
r_1	Ratio of the flexure segment length, $r_1 = l_0 / l_a$
r_2	Ratio of the flexure segment length, $r_2 = l_b / l_a$
t	Thickness of the electrodes (m)
u_i	Flexure's displacement at the end point i in the x - direction (m)
v_i	Flexure's displacement at the end point i in the y - direction (m)
θ_i	Flexure's displacement at the end point i in the rotational direction
w	width of the flexure (m)
w_a	width of vertical flexure segment (m)
w_b	width of transverse flexure segment (m)
V	Voltage applied to the electrostatic actuators (V)
DRIE	Deep reactive ion etching
FEM	Finite element method
FEA	Finite element analysis
HDD	Hard disk drives
ICP	Inductively coupled plasma
LPCVD	Low pressure chemical vapor deposition
MEMS	Microelectromechanical systems
PECVD	Plasma enhanced chemical vapor deposition
SOG	Silicon-on-Glass
SOI	Silicon-on-Insulator
SEM	Scanning electron microscopy
VCM	Voice coil motor

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Chapter 1

Introduction

1.1 MEMS and Microactuators

Micro-Electro-Mechanical Systems (MEMS) technology is a rising technology that is developed to fulfill the miniaturizing tendency. It is the integration of mechanical elements, sensors, actuators, and electronics on a common silicon or non-silicon substrate through the utilization of microfabrication technology. MEMS technology promises to revolutionize nearly every product category by bringing together silicon-based microelectronics and complex micromachining technology, thereby making possible the realization of a complete system-on-a-chip. MEMS technology is truly an enabling technology allowing the development of smart products by combining the computational ability of microelectronics with the perception and control capabilities of microsensors and microactuators [1-2].

The primary distinctive features defining MEMS technology are miniaturization, multiplicity, and microelectronics. MEMS devices have very wide scope of applications, such as in optics, transportation and aerospace, robotics, fluidic and chemical analysis systems, biotechnologies, medical engineering, data storage, and microscopy using scanning microprobes. MEMS devices are inherently smaller, lighter, faster, lower in cost than their macroscopic counterparts; and in many cases, they are also more precise.

Microactuators are the key parts of micro systems to perform physical functions. Many types of microactuators have been successfully devised and operated [3]. Microactuators are classified by their actuation mechanisms [4-5]. The actuating force can be generated following two main principles:

1) The first principle is external forces which are generated in the space between stationary and movable parts using electrostatic [6-7] or electromagnetic fields [8-9] and electrochemical [10] or thermopneumatic [11] effects.

2) The second principle is the inner forces that use special materials (also called functional elements) having intrinsic actuation capabilities, for example, piezoelectric [12], thermomechanical [13], shape memory alloy [14], electrostrictive [15] and magnetostrictive effects [16].

Each actuation principle has its own advantages and disadvantages. The right choice and optimization of the actuation mechanism should be made according to the requirements of a particular application, such as the structural dimensions, technology, response time, force and torque as a function of displacement, and maximum power consumption [17]. Generally, the electrostatic actuator is more suitable for performing tasks that can be completed within a chip (e.g., positioning of devices/heads/probes, sensors with servo feedback for self-test or readout, light deflection and modulation) since it is easily integrated on a chip, easily controlled, and consumes little power. On the contrary, the other types of actuators are more robust, more capable of producing larger forces, and more suitable for performing external tasks (propulsion, or manipulation of objects).

1.2 Microactuators in HDD

Since the first hard disk drive (HDD) was introduced by IBM in 1956, HDD has developed at an astonishing speed in terms of their characteristics; and HDD has probably improved more than any other PC component [18]. The amazing progressing trend in HDD can be summarized as high areal density, miniaturized size, high access speed and positioning accuracy. This progress is supported by many breakthroughs in

various fields, such as the magnetic recording media, flying height, vibration control of spindle, slider and actuator. Among them, the actuators determine the positioning speed and accuracy on the tracks and greatly affect the performance of HDD.

In HDD, the actuator (also called head actuator assembly) is the positioning mechanism used to move the suspension arms and consequently the magnetic heads, back and forth over the disk surface, thus the heads can read/write data on disk tracks. The development of actuators used for HDD has come through two general varieties: stepper motors and voice coil motors (VCM) [19].

Originally, HDD used a stepper motor to position the heads. The stepper motor rotated by reacting to stepper pulses, and only stopped at predefined "steps" as it turned around. Each position defined a track on the disk surface. As a stepper motor had no way to compensate for the expansion or contraction of the disk, tracks would be written fairly wide so that the heads could find them. This greatly limits the track density, thus it is not suitable for high track density demand of HDD. The access speed of stepper motor is between 30 ms and 70 ms.

Nowadays, HDD uses a VCM to actuate the head arms. A coil, attached to the head assembly, moves toward or away from a permanent magnet, controlled by the amount of current flowing through the coil. This is a servo system, with the exact amount of movement controlled by the exact amount of current applied. The general access time of VCM is between 10 and 20 ms. In the HDD trend, the track density will be greatly increased and seeking time will be reduced; these require a high spindle speed and a high bandwidth of the actuator. However, the high spindle speed will cause a large windage disturbance and disk flutter, thus making the read/write data on disc unreliable. Furthermore, due to the limited natural frequency of the

actuator assembly, the bandwidth and access speed are limited. Therefore, the VCM actuator will not be able to keep up with the high acceleration of data storage.

To realize the high-bandwidth high-accuracy positioning system in the evolution trend of HDD, a dual-stage actuator system has been developed in the last decade. There are two actuators employed in the actuators system: one is the current VCM that served as a first stage to generate coarse and slow movement; the other one is the MEMS-based actuator, which is the secondary stage to provide fine and fast positioning. The microactuator can be very small with a high positioning bandwidth and accuracy. So far, many types of microactuators have been proposed, such as electrostatic microactuator [20], piezoelectric microactuator [21], and electromagnetic microactuator [22], and their application in HDD have been studied, respectively.

1.3 Objective and Scope

The objective of this project is to study and develop a high-aspect-ratio bulk-micromachined electrostatic microactuator, with high-bandwidth and high-precision actuation, which can be used in a dual-stage actuator positioning system of HDD.

Corresponding to the objectives, the scope of the thesis involves the following:

- 1) To investigate the development of the silicon-on-glass (SOG) electrostatic microactuator, which is suitable for the dual-stage positioning system of HDD;
- 2) To study the mechanical flexures for the high-bandwidth and high-precision control of the microactuators;
- 3) To explore the fabrication process of the high-aspect-ratio SOG electrostatic microactuators;
- 4) To characterize the fabricated devices and study the influence of fabrication tolerance on the performances of the devices.

1.4 Organization of Thesis

The thesis is organized as follows:

Chapter 2 reviews the development of comb drive actuators, and introduces the flexures for the precision design of the actuators. Review of typical MEMS technology employed to fabricate the microactuators is also carried out. Lastly, the development of dual-stage microactuators for HDD is studied.

Chapter 3 presents the design of the 3-D electrostatic microactuator to be used as the secondary stage actuator in HDD. The design features and actuation mechanism are demonstrated. Therefore, the performance of the microactuator is analytically and numerically studied.

Chapter 4 describes the flexure design for the bulky microactuator. According to the proposed requirements, four kinds of flexure structures are proposed: 1) straight flexure, 2) folded flexure, 3) asym-quad flexure and 4) sym-quad flexure. Both analytical and numerical methods are employed to study the flexure stiffness.

In Chapter 5, the silicon-on-glass (SOG) fabrication process of the 3-D microactuators is presented. The key technologies, such as glass etching, wafer bonding and DRIE, are highlighted and fabrication results are shown.

Chapter 6 presents the characterization of the microactuators with different types of flexures. The fabrication tolerances, such as tapered profiles and undercut, and their influence on the microactuators' performance are discussed, and modified theoretical performances are compared to the experimental results.

Chapter 7 presents the conclusions and contributions of this project, and also proposed the future work.

Chapter 2

Literature Review

This chapter gives an overview of the electrostatic microactuators, especially on the comb drive actuators, and then reviews the various flexure designs for the precise actuation. The micromachining technologies for actuator fabrication are then introduced. Lastly, the application of the microactuators in HDD is reviewed, and current proposed microactuators as the dual-stage actuators in HDD are discussed.

2.1 Development of Electrostatic Actuators

Microactuators are increasingly important as the components of complex microsystems. As introduced previously, the actuation forces are based, for example, on piezoelectric, thermo-mechanical, electrostatic or magnetic principles. The most promising and most widely used principle, however, is the application of electrostatic force for its combining versatility and simple technology. It needs neither additional elements like coils or cores, nor special materials like shape-memory-alloys or piezoelectric ceramics. Above all, the electrostatic actuation draws its force from the relation of surface to spacing and not from the relation of volume to spacing; thus, it is less affected by scaling and more favorable for VLSI actuators [23]. In general, electrostatic force mechanism can be used in both linear and rotary actuators.

2.1.1 Rotary electrostatic actuators

The first electrostatic micromotors were developed by Fan *et al* [24] and were called side-drive motors since they utilize the electrostatic force that acts between the edges of a rotor and a stator, both of which were made with polysilicon (Figure 2.1). The rotational speed of 500 rpm was relatively low because the sliding surface

generated substantial friction force even when silicon nitride film was deposited on the surface to reduce the friction. Later improvements by Mehregany *et al* [25] enabled rotational speeds of up to 15 000 rpm and continuous operation for more than a week. They reduced the clearance between the rotor and the shaft and formed three dimples under the rotor for both support and electrical contact.

Figure 2.1 Rotary side-drive electrostatic actuator

Even for improved micromotors, friction has been a major problem. One solution is to replace the sliding contact at the center axis of the device with a rolling contact. A type of micromotor, called a wobble motor [25], was designed on this principle and realized using surface micromachining. The results in two advantages for the motor: reduction of friction and higher torque at low speed. However, a good way to avoid friction is the use of elastic supports without contact. The linear parallel-plate actuator and comb drive are examples of non-contact electrostatic actuators.

2.1.2 Parallel-plate capacitors

The mechanism of the linear electrostatic actuator can be illustrated by a simple parallel-plate capacitor. A basic example is shown schematically in Figure 2.2, where an electrode is used to actuate a movable structure. There are two electrodes separated with an air gap. One electrode is usually fixed and the other is free to move.

Figure 2.2 Schematic diagram of a parallel-plate capacitor

The capacitance between the two parallel plates is given by

$$C = \frac{\varepsilon_0 A_0}{z} \quad (2.1)$$

where ε_0 is the vacuum permittivity ($\varepsilon_0 = 8.85 \times 10^{-12}$ F/m), A_0 is the area of plate, and z is the electrode separation distance. When a constant voltage, V , is applied across the electrodes, the movable structure is pulled towards the fixed one, increasing the capacitance between the electrodes, thus increasing the stored energy, W .

Since $dW = F dz$, the attracting force F between the two electrodes is [26]

$$F = \frac{dW}{dz} = \frac{d\left(\frac{1}{2}CV^2\right)}{dz} = -\frac{\varepsilon_0 A_0 V^2}{2z^2} \quad (2.2)$$

Thus, the force F is proportional to the square of applied voltage V and inversely proportional to the square of air gap z . Because of the square dependence, the narrow air gap obtains the very big electric force. Obviously, the relationship between the electrostatic force and the displacement is nonlinear. This means that the parallel-plates capacitor with displacement in the normal direction may not be an ideal choice for an actuation application. This is the main drawback of the linear-motion structures unless the movement is very small compared to the separation gap. The demand for a linear actuate/sense device leads to the known design of comb drive structure, which consists of interdigitated cantilever beams called comb fingers (or comb teeth).

2.1.3 Comb drive actuators

Among the linear actuators, comb drive structures are well researched and widely used, which were first demonstrated by William Chi-Keung, Tang [26-28]. The actuator is laterally-driven polysilicon microstructure parallel to the substrate plane. As shown schematically in Figure 2.3, a comb drive actuator consists of two sections of comb-like structures with fingers overlapping each other. One comb is suspended and free to move while the other one is fixed.

Figure 2.3 Schematic diagram of a linear comb drive structure, with suspended beams

The actuator can be thought of as a spring-mass-damper system, the damping being provided by the air below, above, and in between the movable parts. By applying a voltage across the fixed and movable comb fingers, an electrostatic force is produced which drives the mass in the lateral direction [29]. The working concept of lateral comb drive is similar to that of parallel-plates capacitor, and the comb drive may also have transverse, vertical, and rotary motion. Many researchers have engaged in the shape design [30], fabrication and operation of comb-drive actuators [31-33].

- **Lateral motion comb drive**

Figure 2.4 shows a cross section of a comb drive. The movable finger overlapping length is x , the gap is d , and the depth of the finger is t . There are two

kinds of capacitances when there is a potential difference applied between the fixed and the movable fingers: capacitance C and the parasitic capacitance C_p . C_p is due to the undesirable fringing electric field.

Figure 2.4 Outline of a comb drive displaying the presence of capacitance

According to the equations of parallel-plates capacitor, and taking into account of the parasitic capacitance, the total capacitance between the electrodes is

$$C = \frac{\epsilon t x}{d} + \frac{\epsilon t b}{L - x} = C_0 + C_p \quad (2.3)$$

To consider the changes in the lateral motion of the movable fingers, equation (2.3) is generalized as

$$\Delta C = \frac{\epsilon t (x + \Delta x)}{d} + C_p \quad (2.4)$$

Generally, the overlapping length is far larger than the finger width (i.e., $x \gg b$) and finger air gap is far smaller than the parasitic distance (i.e., $d \ll L - x$). Therefore, the parasitic capacitance C_p is far smaller than the capacitance C_0 (i.e., $C_p \ll C_0$) and C_p can be ignored [33]. From the equation (2.2), the lateral electrostatic force along n pairs of movable comb fingers is determined as

$$F_x = \frac{\partial C}{\partial x} V^2 = \frac{\partial \left(n \frac{\epsilon t (x + \Delta x)}{d} \right)}{\partial x} V^2 = \frac{n \epsilon t}{d} V^2 \quad (2.5)$$

Equation (2.5) can be used to calculate the actuation force of lateral driven comb drives. The generated electrostatic force by comb drives is generally smaller than the direct force across parallel-plates capacitors, but it exhibits a better controllable range of motion. As long as the movable fingers are precisely centered in their corresponding gaps, the lateral force components that act on each finger (in the y -direction) exactly cancel out. The lateral comb drive provides an electrostatic force that is independent of lateral position of the movable electrodes. The lateral electrostatic force increases with the decrease of air gap and increase in the number of comb fingers. The linearity of the capacitance, sensitivity and the force make lateral motion a good choice for both sensing and actuating application.

• **Transverse motion comb drive**

Figure 2.5 Schematic diagram of a comb drive for transverse motion

Besides the lateral motion in the comb drive, there are also other motions. Opposite to the lateral movement, the comb fingers may move in the transverse direction. Figure 2.5 shows the cross section view of a transverse motion comb drive. It can be considered as one of the undesired motion for a linear comb drive due to the fact that there is a tendency for the opposite finger sticking together. The working principle of transverse comb is also similar to that of parallel-plates capacitor.

In Figure 2.5(a) the movable finger is located in the center and the capacitances at both sides are equal. Under the condition of a weak stiffness ratio of a flexure, it will

result in a shift in the position of the movable fingers. This may lead to variation in the capacitances on both sides of the movable fingers, as shown in Figure 2.5(b). The differential electrostatic force in the transverse direction is expressed as:

$$F_y = F_{y1} + F_{y2} = \frac{1}{2} n \epsilon x t V^2 \left(\frac{1}{(d-y)^2} - \frac{1}{(d+y)^2} \right) \quad (2.6)$$

From the above, we can see that the transverse force involves a non-linearity with the displacement, thereby causing the pull-in or side-instability of the comb drive. Obviously, the non-linearity in force versus displacement is undesirable for actuating; and it should be avoided by controlling the voltage and maximum lateral deflection.

- **Vertical motion comb drive**

Figure 2.6 Cross-section view of the levitated comb drive

The vertical motion of comb drive has a phenomenon of comb finger levitation. This effect of levitation is mostly caused by the electrostatic repulsion due to the image charges mirrored in the ground plane. In particular, positively biased comb finger induces negative charges both on the ground plane and the movable comb finger, as shown in Figure 2.6 [26]. However, this ground plane contributes to an unbalanced electrostatic field distribution. The imbalance in the field distribution results in a net vertical force induced on the movable comb fingers, which levitates the movable structure away from the substrate.

There are two typical ways to reduce the effect of levitation. One way is to reverse the polarity of the stator parts of the comb fingers, thus resulting in an altered field distribution. Another way is to modify the ground plane so that underneath each comb finger is a strip of conductor biased at the same potential. Besides the linear motion in lateral, transverse and vertical directions, the comb drives can also have rotary motion with particular structures.

- **Rotary comb drive actuator**

Rotary comb drive is developed as an extension of the linear comb drive. There are two types of rotary comb drive actuators that are investigated. One type of the rotary comb drive is explored by Horsley *et al* [34], which is also called angular comb drive. Figure 2.7 shows the view of one quadrant of the rotary comb drive actuator.

Figure 2.7 One quadrant view of the rotary comb drive actuator

The comb drive actuator consists of N pairs of capacitive plates, half of which are used for clockwise rotation, and half for counter-clockwise rotation. For small rotation angles, θ , the plates can be considered as parallel-plates capacitors with varying gaps. Thus an electrostatic torque with applied voltage v_1 to one half of the structure is [34]:

$$\tau(v_1, \theta) = \frac{N}{4} r \epsilon_0 A \left(\frac{v_1}{x_1} \right)^2 = \frac{N}{4} r \epsilon_0 A \left(\frac{v_1}{x_n - r\theta} \right)^2 \quad (2.7)$$

where r is the distance from the center of the rotor to the center of the plate, x_n is the nominal plate gap and A is the plate area. As shown in Equation (2.7), the output torque is a nonlinear function of both voltage and rotation θ . The voltage non-linearity may be reduced by applying differential voltages to the two halves of the structure.

Another type of rotary comb drive is quite similar to the lateral comb drive, as shown in Figure 2.8. The rotary comb drive (or torsional comb drive) is designed for torsional resonant plates in which the comb fingers lie on arcs of concentric circles and can actuate or sense torsional motion about the center of these circles [35].

Figure 2.8 Schematic diagram of a rotary comb drive

The circumferential electrostatic force is

$$F_x = \frac{1}{2} \frac{\partial C}{\partial x} V^2 = - \frac{\epsilon t V^2}{2r \ln \left(1 + \frac{d}{r_i} \right)} \approx \frac{\epsilon t V^2}{2d} \quad (2.8)$$

where r_i is the outer radius of the rotor finger, d is gap between two fingers, x is the circumferential displacement of the rotor, and $x = r\theta$. Here, V is the applied voltage. The electrostatic force can be simplified to be equivalent to that of a lateral comb drive, considering the value d/r being small. However, Equation (2.8) can only be used for rough calculation as the assumption involves some errors about 2%. This

error is dependent on the number of comb fingers and it becomes significant as the number of comb fingers increases.

To conclude, by using the electrostatic mechanism, both linear and rotary actuators have been developed. Among them, comb drive structures are widely used as the actuator/resonators. In linear comb drive design, lateral-motion comb drive provides the good linearity of capacitance, sensitivity and electrostatic force for actuating and sensing application; and the electrostatic force is independent of lateral displacement. Transverse and vertical motion can be looked as underdog of linear actuator. The motion in transverse direction causes side-instability and sticking of the electrodes together. For vertical motion, the levitation phenomenon may cause undesirable effect in the actual application and should be avoided. The rotary design is for rotary resonant plates in which the comb fingers lie on arcs of concentric circles and actuate/sense rotary motion about the center of the circles.

For good linearity of the driving force and displacement, the stability of actuation and sensing application, the lateral and rotary comb drives have been developed widely in MEMS devices. This project has exploited the basic lateral comb drive structure to develop a new electrostatic microactuator for hard disk drives.

2.2 Development of Mechanical Flexures

2.2.1 Overview and design considerations of flexures

The micromechanical flexures are of great interest for a wide range of sensing and actuating applications, such as accelerometers, gyroscopes, and resonators. The flexure mechanisms rely on material elasticity to produce motion and transmit forces. These mechanisms constitute a significant body of mechanical designs and find applications in conventional mechanical field as well as emerging sensing and

actuating areas. The key advantage of flexure mechanisms lies in their simplicity and high precision in the absence of friction and backlash. There exists a considerable amount of design knowledge on flexures, which is derived from the experiences of numerous inventors and engineers [36-37].

The motion of flexure is generated due to deformation at the molecular level, resulting in two primary characteristics of flexures – smooth motion and small range of motion. Except for the limit of elasticity, flexures present few other boundaries as far as applications are concerned. Flexures have been used as bearings to provide smooth and guided motion, for example, in precision motion stages; as springs to provide preload, for example, in the brushes of a DC motor or a camera lens cap; as clamping devices, for example, the collet of a lathe; for elastic averaging as in a windshield wiper; and for energy storage, such as, in a bow or a catapult. This list encompasses applications related to the transmission of force, displacement and energy, thereby making the versatility of flexures quite evident.

Figure 2.9 Scheme of typical constraint elements

From the perspective of precision machine design, flexures may be thought as a means for providing constraints. The objective of an ideal constraining element is to provide infinite stiffness and zero displacements along certain directions, and allow infinite motion and zero stiffness along all other directions. The directions that are constrained are known as degrees of constraint (DOC), whereas the directions that are

unconstrained are referred to as degrees of freedom (DOF). While designing a machine or a mechanism with appropriate constraints, the designer faces a choice between various kinds of constraining elements, two of which are considered: ball bearings and flexures (as shown in Figure 2.9).

Clearly, ball bearings meet the definition of a constraint quite well, since they are very stiff in one direction, and provide very low resistance to motion in other directions. Nevertheless, motion in the DOF direction is associated with undesirable effects such as friction, stiction and backlash that typically arise at the interface of two surfaces. Hereby, flexures are used for precision mechanisms as the replacements of conventional ball bearings or hinges, thus eliminating friction and backlash [37-38].

The systematic flexure design has to be based on performance measures. One set of important performance measures are the DOC and DOF of a flexure mechanism. The stiffness values along the DOF and DOC of a flexure mechanism are key dynamic performance measures while designing a motion system. Apart from damping, which is often added externally, the stiffness and mass properties of the mechanism determine its dynamic characteristics. Another important performance measure in flexure mechanisms is accuracy of motion. Any deviation from the intended motion trajectory may be termed as undesired motion or parasitic error motion. Typically, designers strive to eliminate or minimize these errors by making insightful use of geometry and symmetry.

For their distinct advantages, the flexures have many applications which require accurate displacement or rotation over a limited range. The potential applications consist of actuators, sensors, precision robotics, for either macro or micro scale.

2.2.2 Development of various flexure structures

The flexures have a variety of structures and shapes to provide the precise displacement for specific applications. The simplest and most basic flexure is the cantilever beam. As shown in Figure 2.10, the cantilever beam flexure is fully constrained at one end and free at the other end. The simple motion of the flexure can be predicted by using linear beam theory [39]. Based on the classical deflection equations, the stiffness of the flexure in x and y directions are

$$k_x = \frac{EA}{L} = \frac{Etw}{L} \quad (2.9)$$

$$k_y = \frac{3EI}{L^3} = \frac{Etw^3}{4L^3} \quad (2.10)$$

where E is the Young's modulus, t is the thickness of the flexure; w and L are the flexure width and length, respectively. The stiffness ratio of the flexure is

$$\frac{k_x}{k_y} = \left(\frac{2L}{w} \right)^2 \quad (2.11)$$

Figure 2.10 Cantilever beam flexure

The simple cantilever flexure has both the linear and rotary deflection at the free end, and it is not optimal as most applications need the separate motion. The trajectory pathway of the flexure deflection generates the parasitic displacements in other directions besides the operational direction, which is actually undesirable for the actuation. Furthermore, the cantilever flexure normally has low stiffness in out-of-

plane, and has the possibility of buckling or twisting with misaligned loads. Thus, more types of flexures have been proposed to improve the simple flexure.

- **Fixed-fixed flexure**

Figure 2.11 shows the schematic of the four-end anchored fixed-fixed flexure, which is also called as clamped-clamped flexure [30]. The beam flexures are with rectangular cross sections and the shuttle mass is applied by a concentrated force. The spring constants in x and y directions and stiffness ratio are given as

Figure 2.11 Fixed-fixed beam flexure

$$k_x = \frac{Etw}{L} \quad (2.12)$$

$$k_y = \frac{4Etw^3}{L^3} \quad (2.13)$$

$$\frac{k_x}{k_y} = \left(\frac{L}{2w} \right)^2 \quad (2.14)$$

The fixed-fixed flexure has high stiffness and overcomes the rotary deflection caused by the trajectory pathway of the flexure. However, it is not suitable for large displacement applications, as the extensional axial forces develop in the beam results in a non-linear force-to-displacement relation. Nonetheless, it is useful in applications that require measurement of axial forces, for instance, in order to determine residual stresses or measure externally applied axial stresses as in sensing applications.

- **Crab-leg flexure**

In order to reduce the extensional axial force of the fixed-fixed beam flexure, the crab-leg flexure is used [30]. Figure 2.12 shows the sketch of the crab-leg flexure. The

thin segment has a second moment of inertia I_1 with length L_1 ; the thigh segment has a second moment of inertia I_2 with length L_2 .

Figure 2.12 Crab-leg flexure

The spring constants, as a result of a concentrated force on the shuttle, in the x- and y-directions are given by [30]:

$$k_x = \frac{12EI_2}{L_2^3} \left(\frac{L_1I_2 + 2L_2I_1}{2L_1I_2 + L_2I_1} \right) \quad (2.15)$$

$$k_y = \frac{24EI_1}{L_1^3} \left(\frac{L_1I_2 + L_2I_1}{L_1I_2 + 4L_2I_1} \right) \quad (2.16)$$

When the shin and thigh have the same width and thickness, the stiffness ratio can be obtained:

$$\frac{k_x}{k_y} = \frac{L_1^3}{2L_2^3} \left(\frac{L_1^2 + 6L_1I_2 + 8L_2^2}{2L_1^2 + 3L_1I_2 + L_2^2} \right) \quad (2.17)$$

Although this crab-leg flexure design increases the linear deflection region to a certain extent, a large reduction of the stiffness ratio is introduced.

- **Folded-beam flexure**

A flexure design that is less susceptible to a decrease in the stiffness ratio is the folded flexure design [26]. Figure 2.13 shows two forms of the folded-beam flexures. The beams are anchored near the centre and the trusses allow expansion or contraction of the beams along the x axis.

Figure 2.13 Folded-beam flexure

Assuming rigid trusses, the spring constant of the folded flexure design in the x - and y - directions can be found as:

$$k_x = \frac{2Etw}{L} \quad (2.18)$$

$$k_y = \frac{2Etw^3}{L^3} \quad (2.19)$$

$$\frac{k_x}{k_y} = \left(\frac{L}{w} \right)^2 \quad (2.20)$$

The folded flexure design strongly reduces the development of axial forces and exhibits a much larger linear deflection range. The stiffness ratio for small deflections is equal to the stiffness ratio of a fixed-fixed beam flexure. This design is therefore very suitable for large deflection actuators. The limitation is that the anchors of beams are near the centre of the shuttle mass and may affect the location of driving element.

- **Serpentine flexure**

A serpentine flexure is shown in Figure 2.14. Several mechanical suspension designs use serpentine flexures [26-27]. Compliant serpentine flexures can be designed with compact springs. The width of the meanders is adjusted to give the desired stiffness ratio. Residual stress and extensional axial stress are relieved through bending of the

meanders. The serpentine flexures obtain the name from the meandering snake-like pattern of the beam segments. Generally, the width of the first and last meanders is half that of other meanders [40].

Figure 2.14 Serpentine flexure

The theoretical determination process of the spring constant (stiffness) of the serpentine flexure is quite complicated, and also the expressions of the flexure stiffness are complex [41]. Because of the compliant and meandering structures, the serpentine flexure has low stiffness in all directions and provides the large deflection of the shuttle mass. The stiffness ratio of the flexure is also not significant and thus it is not suitable for the applications with simple deflection. Many other types of flexures have also been developed for various mechanical applications, such as micromotors [42], claspers and grippers [43-44].

To sum up, flexures are the key components for precision control of the motion system, and are especially important for the actuator design with high precision requirement. The mechanical flexures have many special advantages. For example, the flexures are wearing-free and friction-free, the motion is smooth and continuous in a small range, and the displacement can be accurately predicted according to the material elasticity. However, there are also some drawbacks for the flexures, such as the limitation of small displacements, weak toleration for large loads (buckling

failure), and dependence on elasticity. Flexures have very wide applications in sensing or actuation devices, and the adoption of flexures depends on the performance requirements of the applications.

2.3 MEMS Fabrication Technology

The early development of MEMS technology began in the 1960s, using the fabrication facilities developed for VLSI technology. While the electronics are fabricated using integrated circuit (IC) process sequence (e.g., CMOS, or BICMOS processes), the micromechanical components are fabricated by using compatible “micromachining” processes that selectively etch away parts of silicon wafer or add new structural layers to form the mechanical and electromechanical devices [45].

The fabrication processes to realize MEMS are a combination of conventional IC processes and specialized technologies unique to silicon micromachining. “Micromachining” is the key technology for the fabrication of solid-state MEMS devices such as microsensors and microactuators. These technologies are: bulk-micromachining, surface-micromachining [46] and micromoulding technologies including classical electroplating, the HexSil [47], and the LIGA [48-49] processes.

To fabricate the electrostatic actuators, either surface-micromachining or bulk-micromachining can be used. Surface micromachining of deposited thin films is the straight-forward application of existing IC processes, and the actuator structures are built in the deposited polysilicon thin films; while bulk micromachining combines the RIE (reactive ion etching) in bulk silicon wafers and wafer bonding technologies to create the three-dimensional structures of the actuators. Besides that, electrostatic actuators are nowadays most commonly fabricated with SOI (silicon-on-insulator) wafer, with three layers of device layer, isolators, and handle layer [50].

2.3.1 Surface micromachining

Surface-micromachining is a process whereby the micromechanical structures or devices are made entirely on the surface of the wafer without penetrating the wafer surface, i.e. an additive process. Typically, the desired mechanical structure is built up by depositing and patterning thin-film of structural and sacrificial material on the substrate surface. Usually, the polysilicon or silicon nitride is used as the structural layer to build planar structures because of the flexibility and ease of 2D design, and silicon dioxide or PSG (phosphosilicate glass) is used as the sacrificial layer for the ease of removal [51]. Many micromotor designs have been realized using surface micromachining, which leads to a multi-layer surface micromachining process [46]. Even complicated mechanical components, like beams, guide ways, bearings, hinges and locking mechanisms can be realized by choosing a suitable combination of layers and their appropriate processing.

Figure 2.15 Generic surface-micromachining process flow for a typical electrostatic rotary micromotor

Figure 2.15 illustrates a generic surface-micromachining process of electrostatic actuators [6,51]. The fabrication begins with passivation of a wafer with a layer of

Si_3N_4 on top of thermally grown SiO_2 film. (a) A thin layer of polysilicon is deposited by LPCVD (low pressure chemical vapor deposition) and then patterned which serves as electrical interconnect; (b) a thick sacrificial PSG layer is deposited by LPCVD, and then patterned with dimples by RIE; (c) the anchors of stators are then patterned; (d) a $2\mu\text{m}$ -thick polysilicon structure layer is deposited and patterned to form the stators and rotors of the actuator; (e) another PSG layer is deposited and patterned as the anchor of the center hub; (f) a third layer of polysilicon is deposited and patterned as the hub; (g) finally, the wafer is immersed in buffered HF to remove the PSG layer.

The key advantage of surface-micromachining is its compatibility with conventional IC processing. Surface-micromachined micromechanical devices are fabricated using standard IC thin-film deposition and patterning technologies without resorting to potentially contaminating chemicals or non-standard process. Another advantage of surface-micromachining is that the size of the devices is not constrained to the minimum device dimensions. Instead, surface micromachined devices can be reduced to much smaller sizes than possible with bulk micromachined components.

The major disadvantage of surface-micromachining is that it is inherently a two-dimensional planar process that limits the flexibility of the design. Surface-micromachining is a relatively new technology compared to bulk-micromachining and has not been widely employed in commercial MEMS production.

2.3.2 Bulk micromachining

Bulk-micromachining is a process that the silicon substrate is selectively removed to shape and form mechanical structures, which is actually a subtractive process, as shown in Figure 2.16. There are many ways of machining the substrate, including isotropic etching, anisotropic etching, electromechanical etching, spark machining, mechanical milling, ultrasonic milling, laser etching, plasma-phase etching, etc [52].

Figure 2.16 Typical bulk-micromachined structure

In wet bulk-micromachining, features are sculpted in the bulk of materials such as silicon, glass, and quartz by anisotropic or by isotropic wet etchants. Isotropic etchants etch in all crystallographic directions at the same rate; they usually are acidic, such as $\text{HF}/\text{HNO}_3/\text{CH}_3\text{COOH}$ (HNA), and lead to rounded isotropic features in single-crystal silicon. For anisotropic etching of single-crystal Si, several solutions, like KOH, TMAH and EDP, are commonly used. However, single-crystal Si with (100) and (110) planes is etched much faster than that with (111) planes [45].

Currently, an introduction of DRIE (deep reactive ion etching) reactor provides a powerful tool for bulk-micromachining of silicon to etch very deep trenches (up to $500\text{ }\mu\text{m}$) with nearly vertical sidewalls [53-54], independent of the crystal orientation. DRIE evolves from the need within the micromachining community for an etch process capable of anisotropically etching high aspect ratio trenches at high rate.

Figure 2.17 Profile of DRIE process exploits alternating etching and deposition steps

Figure 2.17 illustrates the fluorine based DRIE process [55]. A photoresist or silicon dioxide can be used as the mask, with the high selectivity of 50:1 to 200:1. The

etching proceeds in alternative steps of reactive ion etching in a SF_6 plasma and polymer deposition (passivation) from a C_4F_8 plasma. During the etch process, the polymer is rapidly removed from the bottom of the feature but lingers on the sidewall, protecting it from the SF_6 etchant. As a result, the silicon beneath the first cut is removed during the second etch cycle, but the top of the feature does not become wider. Eventually the polymer protecting on the sidewall is eroded.

The DRIE process allows small structures ($<2\text{ }\mu\text{m}$) to be etched with high aspect-ratios (>20) and good anisotropy ($>99\%$) [53]. The silicon etch rate of DRIE can be very fast, from few microns to more than 20 microns per minutes. So far, DRIE is the most suitable and popular anisotropic etching technology for the fabrication of electrostatic actuators, especially for the narrow-gap comb drive actuators.

An extension to this basic definition of bulk-micromachining would be the formation of a desired microstructure by utilizing the bulk of the substrate. This extension is meant to include the widespread use of wafer bonding technology [56-57]. Two or more etched wafers are bonded to create the 3D microstructures.

Figure 2.18 Schematic diagram of anodic bonding apparatus

Wafer-to-wafer bonding is categorized into direct bonding, anodic bonding, and bonding with intermediate layers [58-62]. Wafers can be directly contacted without the assistance of significant pressure or intermediate layers or fields. These bonding schemes rely on the tendency for smooth surfaces to adhere, and utilize the thermal

cycling to increase the bond strength. Anodic bonding typically performs between a sodium-bearing glass and silicon wafer by the application of an electric field at temperatures in the range of 300-450 °C. Intermediate-layer bonding includes all bonding mechanisms that require an intermediate layer to promote the wafer bond. This could include eutectic bonds, polymers, solders, or thermo-compression bonds.

Anodic bonding is used to bond silicon to Pyrex glass, as schematically shown in Figure 2.18. The contacted wafers are heated to 300-450 °C while a voltage of 200-1000 V is applied. Under these conditions, the mobile Na ions in the glass migrate away from the bonded interface, leaving behind a fixed charge in the glass that creates a high electric field across the bond interface with image charges in the silicon. Thus, a chemical bond that fuses the wafers together is occurred. The primary variables that control this process are temperature, voltage, time and surface roughness [63-64].

However, anodic bonding also builds up residual stress between bonded Si/glass chips at high temperature, since Si and glass have different coefficients of thermal expansion (CTE) [64]. The residual stress may cause minor distortion between the silicon and glass wafers, and this becomes a critical issue for IC patterning applications, while as it is acceptable for many MEMS applications. The residual stress of the bonded Si/glass wafers can be reduced by selecting suitable glass type with a CTE fit for silicon.

Anodic bonding is widely used in the fabrication of pressure sensors, accelerometers, micropumps and electrostatic devices. The anodic bond provides high yield, good hermetic seals, electrical insulation, optical transparency, and low cost; it also provides the feasibility of making multi-layer stacks to form 3-D structures.

Nowadays, SOI wafers are widely used for the fabrication of actuators as it facilitates the fabrication process [50,61]. The SOI wafers consist of two silicon layers

and one SiO₂ layer, and the fabrication technology is direct bonding [65-66]. However, SOI wafers have some drawbacks like sticking failure after release, and low yield. MEMS BESOI wafers are expensive, usually a few hundred dollars per piece.

To sum up, bulk-micromachining is a straightforward process and has been widely developed in MEMS production. Bulk-micromachining can be used to realize the 3-D microstructures with deep trenches, bulk cavities, and complex stacked multi-wafer layers. Nevertheless, it has several disadvantages. For example, the etchant chemicals commonly used in bulk-micromachining are incompatible with IC or IC fabrication equipment. For the electrostatic microactuators with high-aspect-ratio 3-D structures, bulk-micromachining is definitely the most expectative and cost-effective fabrication technologies. The deep silicon processing with DRIE and wafer bonding techniques could be employed to fabricate the electrostatic microactuators, instead of conventional surface-micromachining and SOI process.

2.4 Development of Microactuators for Hard Disk Drives

2.4.1 Background of actuators for HDD

With the development of information technologies, the data storage devices such as magnetic hard disk drives (HDD) are undergoing a high-speed revolution. Due to the demand for huge data capacity from the digital/audio media, the data storage industry is pushing for higher areal density and performance of HDD. In the last decade, the areal density of HDD increased at a rate of 60% per annum, and every eighteen months the data storage capacity offered in a given form-factor (size) is doubled while the price remains or even drops [67]. At the same time, the form-factor of HDD becomes smaller. 5.25" drives have now disappeared, and 3.5" drives even 2.5" drives are the mainstream in PC market.

This achievement has been attributed to various technological breakthroughs, including magneto-resistive (MR) sensing devices, signal processing techniques, and high density recording media. To maintain the pace, another technical leap which allows for much narrower data track width is required. Narrow data tracks allow allocating more tracks in a fixed disk radius, resulting in higher areal density. This requires high-bandwidth and high-accuracy positioning of the recording heads because the heads must follow the narrow data track with high accuracy promptly.

This trend requires improved performance of the head-positioning servo system in order to accurately maintain the selected head position along the center of the track (track-following mode) and to provide rapid movement of the head from one track to another selected track (track-seeking mode). In addition, to maintain a high level of performance at a reduced track pitch, the actuator servo system must be able to compensate disturbances caused by internal and external sources.

Figure 2.19 Photo of a modern hard disk drive, with major components annotated

A scheme of a commercial HDD is shown in Figure 2.19 [68]. A hard disk drive consists of a sealed enclosure inside of which there are one or more rigid platters that

are coated with magnetically sensitized material. The platters rotate at a constant speed and data can be written to, and read from the surface of the platters, by means of electromagnetic heads. An electromagnetic voice coil motor (VCM) is served as an actuator to move the head/slider assembly radically across the disk, which is attached to the opposite end of the suspension. Read/write elements are affixed to a ceramic slider, which is bonded to a gimbal at the end of the stainless steel suspension.

The actuator is a very important part of HDD, as changing from track to track on the disk is active movement: changing heads is an electronic function, and changing sectors involves waiting for the right sector number to spin around and come under the head (passive movement). Changing tracks means the heads must be shifted, and so making sure this movement can be done quickly and accurately is of paramount importance. The only position measurement is the position error signal (PES), which measures the head deviation from the track center by the heads.

The actuators for the head assembly come in two stages: stepper motor and VCM actuators [19,69]. The track-seeking time has greatly dropped from 50 ms of stepper motor to less than 10 ms of VCM, and the positioning accuracy has also been enhanced. However, VCM actuator still cannot keep up with the high increasing speed of areal density and access performance of HDD. VCM is actually a single actuator system and the bandwidth is limited by the mechanical resonance of the carriage, coil, and the hysteresis of the actuator's pivot bearing ball [67]. Thus, it is difficult to position a magnetic head right on a narrow data track with high accuracy.

For these reasons, it is difficult for the current VCM actuator to fulfill the high-density and high-performance trend of HDD, and novel actuators or solutions must be proposed for the revolution of the HDD.

2.4.2 Dual-stage actuator servo system

To satisfy the demand for high-bandwidth high-accuracy positioning of HDD, a dual-stage actuator servo system that utilizes MEMS fabricated actuator as the second positioning stage actuator has been proposed and developed [70-73]. The microactuator is designed to increase the bandwidth of a HDD tracking servo and pack more recording tracks on a disk. The increased tracking capability on conventional mechanical disk enables system integrators to configure hard disk drives at a very low cost per megabyte, a very large capacity at a fixed cost, or very low profile HDD for a fixed storage capacity requirement.

A scheme of a dual-stage actuator servo system for HDD is shown in Figure 2.20. There are two actuators employed in this system: the MEMS-based actuator acts as a fine positioning and high-bandwidth mechanism while the VCM continues to perform coarse positioning. Usually, the microactuator is assembled between the suspension and the slider, so that it can compensate for the resonant modes of the VCM.

Figure 2.20 Schematic of dual-stage actuator system for hard disk drives

The mechanism for the dual-stage actuator positioning can improve both track-following and track-seeking performance. The VCM is used in the first stage to generate large but coarse movement, while the microactuator is used as a secondary

stage to provide fine and fast positioning. Position measurement from the microactuator will be used in the servo loop to achieve the desired high bandwidth and tracking accuracy.

2.4.3 Types of microactuators for HDD

Currently, various types of microactuators for HDD are under development. These microactuators are classified into three groups, according to the position where they are placed and the object they will actuate.

1) The first type may be classified as a suspension-driven actuator [22,74-77]. The microactuator is located between the head suspension and the base plate, which is moved by the VCM. A slider is attached to the tip of the suspension. Figure 2.21 shows an example of a suspension-driven PZT microactuator.

Figure 2.21 A scheme of suspension-driven PZT microactuator for HDD

In this approach, conventional assembly and machining techniques are used to integrate an electromagnetic or piezoelectric actuator into a steel suspension or the head mounting plate, and they are expected to be the first to reach production. Another advantage is that this type of microactuator does not need large modifications to the shape of the suspension assembly. This type can achieve a large displacement of the head element by using the suspension length as a swinging radius. However, actuators in this type are greatly influenced by the mechanical resonance of head suspension, for the actuators are located far from the R/W elements.

2) The second type may be classified as a head-driven actuator, and Figure 2.22 shows an example of such type [78-81]. In this type, the microactuator is placed between the slider and the read/write head element.

Figure 2.22 Scheme of head-driven microactuator integrated with the slider

This type is superior in terms of achieving very high servo bandwidth because the mass of the head element is extremely small, which only slightly increases the slider payload. In theory, a large servo bandwidth can be achieved if the microactuator is located closer to the head. In practice, however, other considerations such as shock, flying height, manufacturability, and cost have a significant impact on the actuator choice and placement. The critical limitation of this approach is that the actuator fabrication process must be compatible with the head/slider fabrication process, implying that the two fabrication processes should be merged. However, the merged process can be complex and cause the low yield.

3) The third type may be classified as a slider-driven actuator, and Figure 2.20 presents the typical scheme of a slider-driven microactuator [20-21,82-86]. The microactuator is located between the slider and the gimbal of a suspension. The type of actuator is not influenced by the mechanical resonance of the head suspension, and its advantage is co-location of the actuator and sensor, so a servo bandwidth of 2 kHz plus is possible. Another advantage is that it uses a conventional slider, so there is no

influence on either the magnetic head process or the air bearing dynamics of the slider to maintain a stable low flying-height [87]. However, as these actuators are almost of the same size as sliders, it is required to overcome the problem of increase of the suspension payload and the space between disks.

Nowadays many structures of microactuators for HDD have been designed and discussed. They can also be categorized by the principles of driving forces. The forces generated between stationary and moving parts are based on piezoelectric, electromagnetic, thermal-mechanical, and electrostatic principle. Here the typical types of proposed dual-stage microactuators will be reviewed.

- **Electrostatic microactuator**

The electrostatic actuators are driven by electrostatic force. Their remarkable advantage is that they are suitable for miniaturization, easily integrated on a chip, easily controlled, and consumed little power. Electrostatic microactuator is driven essentially by voltage. It is easy to control with high switch speed, low power consumption and high efficiency. An electrostatic design is chosen for ease of fabrication – the device material need only to be conductive, rather than ferromagnetic or piezoelectric. Furthermore, electrostatic actuators allow high accuracy, capacitive measurement of displacement, and are capable of high bandwidth operation.

Many types of electrostatic microactuators for HDD have been proposed and they can be classified into two categories: linear actuators suspended by thin beams [79-82,87] and rotary actuators driven by rotating motors [83-86,88-90].

Figure 2.23 shows a linear lateral-driven electrostatic microactuator proposed by Fujitsu Ltd., which drives the head elements [78]. The assembly is fully batch fabricated with the slider body, a head element, and a comb-drive actuator. The head element can be laterally driven by the electrostatic microactuator. For the small

moving mass of head element, the actuation bandwidth can be very high (>20 kHz). Due to the integrated head/actuator/slider structure, the fabrication processes of the head/slider/actuator must be merged, which is complex and can cause low yield. Also in actuation, the contact of the head on the rotating disk surface can induce an uncontrollable state, or damage the microactuator.

Figure 2.23 Lateral-driven MEMS-based head/actuator/slider assembly proposed by Fujitsu Ltd. (a) Layout. (b) Close-up of the electrostatic microactuator.

Besides linear microactuators, an angular (or rotary) electrostatic microactuator for HDD is explored. Figure 2.24 (a) shows an angular microactuator for a dual-stage servo system of HDD proposed by IBM Corporation *et al.*, which is fabricated from molded polysilicon using the HexSil process [20,84,91]. A SEM view of the actuator device is shown in Figure 2.24 (b). The microactuator is designed to drive the slider and be capable of positioning the read/write elements of 30% pico-slider over a ± 1 - μm range, with a predicted bandwidth of 2 kHz. The microactuator consists of an outer-ring stator, and an inner-ring rotor, which is connected to anchored flexures. A rotational topology was chosen because it allows high lateral stiffness, minimizing the sensitivity of the device to shock loading in the disk plane.

However, there are some drawbacks for the slider-driven rotary microactuator. For these actuators which are almost of the same size as sliders, they are required to

overcome the problem of increasing of the suspension payload and the space between disks. In addition, they have the disadvantages of high sensitivity to contamination, high voltage, hysteretic behavior and pull-in instability under large motion.

Figure 2.24 (a) Exploded view of rotary microactuator assembly. (b) SEM of 100-um-high 2.6-mm-diameter rotary electrostatic microactuator.

• **Piezoelectric microactuator**

Piezoelectric microactuators have been proposed to actuate the head suspension or the slider of HDD. Piezoelectric materials have many excellent properties, such as high stiffness, high reaction speeds, large driving force, reproducibility of the traveled distance, and precise movements in micro- and nano- meter range.

Figure 2.25 Scheme of shear mode suspension-driven piezoelectric microactuator

There are mainly two types of PZT (lead-zirconium-titanate) microactuators that are investigated for HDD. One type is the shear-mode suspension-driven PZT microactuator, which was developed by Fujitsu Ltd. [74-76]. Two PZT bars are

bonded between the load beam and mount plate of the suspension [92-93], as shown in Figure 2.25. The PZT bars expand or contract in their lengthwise directions when a voltage potential is applied across their thickness. The motion of the PZT bars provides the second-stage servo control of slider motion in the radial direction. The major limitation of this type of design is the unreliability of PZT material, which is brittle and may have many cracks. As well, the HDD and its component level assembly processes, such as head-gimbal assembly and head stack assembly, have many steps providing enough disturbances on creating cracks on PZT bars.

Another proposed type is the so-called stacked type (multiplayer) piezoelectric microactuator which actuates the slider [21,94-95]. Figure 2.26(a) shows a scheme of designed microactuator and head assembly. The microactuator consists of a fixed part connected to a suspension, a movable part connected to a pico-slider and two beam parts that generate bending motion and displacement.

Figure 2.26 (a) Scheme of the stacked piezoelectric Microactuator and magnetic head assembly. (b) Detailed picture of the microactuator structure

Figure 2.26(b) shows the detailed structure of the piezoelectric microactuator and driving voltages. The beam parts comprise ten stacked piezoelectric layers, with each beam part contracting or expanding alternately with applied opposing-phase voltages. The difference in length between the two beams provides a bending motion to the beam parts to drive the slider in tracking direction.

However, the structure of the stacked piezoelectric microactuators is thick, and they are not suitable for thin applications. For the piezoelectric microactuators that exploit the transverse piezoelectric effect, the force transfer loss is large because the stainless steel sheet prevents these elements from expanding.

Also other types of piezoelectric actuator models have been developed [96-98]. Piezoelectric microactuators have the advantage of generating large force, but they have some difficulties in manufacturing. Another serious problem is that the reliability of the piezoelectric elements regarding displacement is unsettled.

- **Electromagnetic microactuator**

The application of electromagnetic microactuators for high-density HDD has also been studied. Compared to electrostatic actuators, electromagnetic actuators tend to use lower voltages, more power, and are sensitive to the magnetic properties of the material used. Also, it is more reliable than the piezoelectric microactuator.

Figure 2.27 (a) Scheme of an electromagnetic microactuator for dual-stage actuation of HDD. (b) A planar electromagnetic microactuator with wrap-around copper coils

In an early publication, an electromagnetic piggyback micro-gimbal secondary actuator was designed and fabricated with micromachining techniques, using wrap-around copper coils (the entire device fits inside the center coupon of the gimbal) [82,99]. This is shown in Figure 2.27. The slider is attached to the gimbal by a set of

thin planer springs. A pair of microactuators is used to move the slider in the in-plane direction, which is attached to the center alloy core.

When a voltage is applied to the coil, magnetic force is exerted on the moving core, pulling it towards the stator and resulting in an in-plane motion of the slider. This design has an advantage in that the fabrication technology is very similar to that used in making thin-film inductive heads, leveraging the expertise of the magnetic industry. However, shock loads applied in the direction of in-plane motion lead to extra-uncontrolled displacement of an actuator with low in-plane stiffness.

Fujitsu Ltd. also reported an electromagnetic slider-driven microactuator for HDD, which is constructed at the tip carriage arm and supported by a cross-shaped spring [22]. The suspension assembly attached to the head-mounting block is driven in rotary direction around a stator shaft by electromagnetic force. The entire system has a very rigid structure in the positioning system and an improved system dynamics. However, it is quite thick and not suitable for thin-plate applications in HDD.

To sum up, the application of microactuators in the head-positioning systems of HDD will be necessary as the capacity and performance in magnetic storage systems continues to grow. So far, many kinds of microactuators have been proposed, and the major types are based on electrostatic, piezoelectric, and electromagnetic principles.

Piezoelectric microactuators have the advantage of generating large force, high stiffness and fast reaction speed; but they have some difficulties in fabrication and reliability. Electromagnetic microactuators usually require long cables to make the electric coil that generate the magnetic field, and have large resistance with Joule loss, so it is not suitable for miniaturization.

Generally, the electrostatic microactuator is most promising in the high-capacity and miniaturization trend of HDD. It is relatively easily fabricated, easily integrated

on the slider, easily controlled, and consumes little power, compared to other types of actuators. The electrostatic microactuator can be used to drive the slider or head element, which can provide a high servo bandwidth. Table 2.1 lists the major characteristics of the three compared electrostatic microactuators.

Table 2.1 Major characteristics of electrostatic microactuators

	Head-driven linear micro-actuator [11,12]	Slider-driven area-efficient rotary microactuator [6,16]	Slider-driven rotary micro-actuator [14,15]
Moving mass	0.85 μg (heads)	1.6 mg (slider)	1.6 mg (slider)
Movable finger number	24	240	60
Air gap	2, 5 μm	3 μm	--
finger width	4 μm	4 μm	--
Flexure width	3 μm	3 μm	3 μm
Driving voltage	50 V	60 V	80 V
Driving force or torque	21.3 μN	0.14 $\mu\text{N}\cdot\text{m}$ (torque)	0.18 $\mu\text{N}\cdot\text{m}$ (torque)
Resonant freq. (loading slider)	34 kHz	1.37 kHz	1.7 kHz
Displacement	$\pm 0.55 \mu\text{m}$	$\pm 2 \mu\text{m}$	$\pm 1 \mu\text{m}$

According to Table 2.1, loading with the pico-slider, the resonant frequency of the flexure-based microactuator is limited to 2 kHz. The structures have 2~10 μm in thickness, air gap and the flexure width are within 3 μm , which makes the actuator structure to be thin and sensitive to fatigue. Furthermore, for the rotary microactuator, the output electrostatic torque is nonlinear to the driving voltage and rotation angle, and high driving voltage is required (typically more than 60 V). Therefore, an improved design of electrostatic microactuator with fewer shortcomings, such as low driving voltage, high bandwidth, good reliability and low cost, should be developed for the dual-stage microactuator of HDD. The major performance requirements of the microactuator for HDD are: driving voltage of 40 V or below, displacement of $\pm 0.5 \mu\text{m}$, loading resonant frequency of 2 kHz.

2.5 Chapter Summary

Microactuators are the key components of the complex microsystems to perform the physical functions. Among the various actuation principles, the most promising and widely used principle, however, is the electrostatic force due to its combining versatility and simple technology.

This chapter reviewed the development of electrostatic actuators, and especially highlighted comb drive actuators. The conventional comb drives have thin plate and high driving voltage, with problems of uncertain reliability and high cost. The flexure design for the precise actuation in MEMS applications was then investigated, and the commonly used flexures were discussed. MEMS fabrication technologies for the microactuators have been introduced. Lastly, the applications of the microactuators in HDD, which serves as the dual-stage actuator in the positioning system to meet the high-capacity and high-performance trend of HDD, have been reviewed. Various types of microactuators were discussed for their pros and cons. The need for a novel electrostatic microactuator with fewer shortcomings, high reliability and effective cost has been underscored.

Chapter 3

Design of the Microactuators

The goal of this chapter is to demonstrate the actuation mechanism and structural design of a novel developed microactuator. Basically, for a resonant microstructure, the actuating components together with the stiffness of the flexure mechanism are important in designing the resonant frequency of the microactuators. This project looks into the electrostatic actuation mechanism and focuses on its most widely used microstructure, the comb drive, which was first developed by W. C. Tang [47]. A novel slider-driven three-dimensional electrostatic microactuator for the head positioning system of hard disk drives is investigated in this project, and a suitable flexure is designed for the microactuator.

3.1 Structure Design of Electrostatic Microactuator

3.1.1 Requirements of microactuator for HDD

From the literature study of microactuators for hard disk drive applications, it can be concluded that there are some requirements for the microactuators to realize a high performance of the HDD tracking servo system [19, 29].

➤ The first requirement is the dimension stability due to the temperature influence. Typical electrostatic actuators have small inter-electrode gaps that must be strictly maintained. A change of gap width can cause a shift in the actuation characteristics or, in the worst case, a short-circuit failure. As the temperature changes, the suspension beam expands or contracts due to thermal expansion mismatch between the structure and the substrate material. Thus, the gap between the stationary electrode and the suspended electrode changes. In this case, the gap change Δd is expressed as:

$$\Delta d = \Delta T \times (\alpha_{sus} - \alpha_{sub}) \times l \quad (3.1)$$

Where ΔT is the temperature variation, α_{sus} and α_{sub} are the thermal coefficients of expansion (TCE) of the suspension beam material and substrate material, respectively, and l is the suspension length. The gap variation should be relatively small, compared to the origin gap. Therefore, a structure material with a small difference of TCE to that of substrate is preferred. A silicon substrate is usually used for its potential integration with electrical circuit.

➤ Secondly, the microactuator must be flexible in the operational direction (along the data track), but very stiff in vertical and radial directions, to prevent the excitation of resonance modes in these directions. Thus, the structural resonant frequencies in the vertical and radial directions must be much higher than that of the operational direction of motion. In addition, the z- (vertical) directional stiffness must be high enough to withstand the loading force that is applied by the suspension beam to the slider. This loading force presses the slider down to the disk surface and is necessary to maintain an adequate air bearing between the slider and the disk surface during operation. Thus, the microactuator must be a high-aspect-ratio structure with a large structural height in order to meet these requirements [100].

➤ The third requirement is large force output. Since the microactuator must drive a slider, head element or head suspension, which weighs a few milligrams, and the maximum usable area and driving voltage of the microactuator are limited, an area-efficient design is required. A high-aspect-ratio structure with a large structural height is also advantageous here, since the electrostatic force, which propels the microactuator and slider, is proportional to the structural height of the actuator's stator and rotor electrodes and inversely proportional to the gap width of electrodes.

➤ The forth requirement is the easy assembly of the microactuator. Since the microactuator is attached to the slider/suspension/head, it is necessary to establish electrical connections through the microactuator that are mechanically very flexible.

➤ The last requirement is low manufacturing cost. Since these microactuators will be used in HDD industry, the cost of manufacturing such device must be competitive. As opposed to other applications where cost is not a primary concern, sensors and actuators are used in HDD only when they are absolutely necessary; otherwise the industry cannot remain cost competitive.

The newly designed electrostatic microactuator should fulfill the requirements for HDD, and have some prominent advantages over other types of microactuators that have been reviewed in Chapter 2. Besides, the electrostatic microactuator proposed in this research work aims to achieve the following goals:

- To employ the electrostatic actuation mechanism and com-drive structures as many pairs as possible so as to enhance the electrostatic driving force.
- To be driven by a relatively low voltage in comparison with other electrostatic actuators. A typical driving voltage is less than 40 V.
- To achieve the tracking accuracy of 0.1 μm , the stroke of around $\pm 0.5 \mu\text{m}$, and the resonant frequency of the microactuator of over 1 kHz [83,84].
- To facilitate with the batch micromachining technologies.

3.1.2 Scheme of the dual-stage microactuator for HDD

In this research work, the proposed electrostatic microactuator adopts the prevailing dual-stage actuator servo system design frame and at the same time acts as

the secondary actuator, and to satisfy the high accuracy and high bandwidth requirement of the HDD positioning system of hard disk drives.

Figure 3.1 Schematic diagram of dual-stage microactuator assembly.

Figure 3.1 shows the basic scheme of dual-stage microactuator assembly for HDD. There are two actuators in the positioning system: one is the voice coil motor (VCM), which is used as a first stage to generate large but coarse and slow movement of the R/W head; the other one is the microactuator, which serves as a secondary stage to provide fine and fast positioning.

The assembly consists of a slider, a track-following electrostatic microactuator, a recording head, and a suspension arm. A magnetic read/write head is integrated on the head plate of the microfabricated slider. The silicon electrostatic microactuator with a size of millimeter designed in this project will be placed between the suspension's gimbal and slider with read/write head. The whole silicon microactuator is bonded to the glass substrate, and is a silicon-on-glass (SOG) structure instead of silicon-on-insulator (SOI) based structure. Figure 3.2 shows the schematic diagram of the 3-D electrostatic microactuator, which is supposed to be assembled in the dual-stage actuator system.

Figure 3.2 3-D schematic view of the microactuator.

3.1.3 Features of the 3-D electrostatic microactuator

Figure 3.3 Cross section of the 3D silicon microactuator schematic.

The slider-driven electrostatic microactuator acts as the second stage actuator of the dual-stage positioning system of HDD, with the comb drives utilized as the actuation components. The microactuator consists of several parallel comb drives,

several support flexures, and some fixed electrodes. The cross section of the microactuator is shown in Figure 3.3.

The features of the 3-D electrostatic microactuator can be characterized as follows:

a) Silicon-On-Glass processed 3-D bulky and high aspect ratio structure

This is the typical characteristic of the electrostatic microactuator. The fabrication of microactuator is realized on SOG process, in lieu of SOI wafer process. Therefore, the silicon structure can be very thick in comparison with commonly used SOI wafer. In general, the SOI wafer is much more expensive than silicon wafer (generally 5~8 times of the normal Si wafer price). Obviously, low cost is the very attractive advantage of the SOG microactuator. Furthermore, this design breaks through the planar design of electrostatic microactuators that uses conventional SOI wafer process and therefore, provides a good option in the development of electrostatic actuators.

In addition, the actuator structure is perpendicular to the mounting planes of the slider, with the actuation plane upstanding, whereas the generally designed comb drive actuators based on surface micromachining technologies lie down and move in the parallel plane to the mounting planes. This also makes the microactuator to be so-called three-dimensional structure.

The thickness of the comb drive structure can be more than 100 μm based on the SOG process; at the same time, the width of comb teeth are as small as 5 μm and the air gap between comb fingers are as narrow as 3~4 μm . Thus, the comb drive presents a very high aspect ratio (height-to-width ratio) structure, which reaches more than 30:1.

The proposed bulky 3-D structure has to be fabricated by using bulk micromachining technologies, typically including the deep reactive ion etching

(DRIE) to obtain the high-aspect-ratio structure, and wafer bonding technology to bond the glass substrate.

b) Symmetrical comb drives and high electrostatic driving force

The microactuator employs comb drive structure for its electrostatic actuation mechanism. The comb drives consist of two parts: fixed electrodes and movable electrodes; therefore, many parallel-plate capacitors are formed in the interdigitated field of the two electrodes. In the proposed actuator, the comb teeth of the movable electrodes are double-side shaped and symmetrically designed in each pair of comb drives, and several pair of comb drives could be arranged in parallel for the actuation.

Figure 3.4 Left position of the comb-drive actuator when actuated.

The movable comb drives can move to both sides, depending on the side of the fixed electrodes that the voltage is applied to. The alternating driving voltage can be applied on either side of the fixed electrodes, with the movable electrodes moving towards the fixed parts by the action of the electrostatic force. Figure 3.4 shows the left positions of the microactuator in operating condition. The amplitude of the movable electrodes is limited by the stoppers at both sides of the fixed electrodes.

With the alternating electrostatic and mechanical restoring force, the microactuator is actuated to and fro continuously.

As the microactuator has a bulky structure and is assembled to be upstanding, the moving plane of the comb drives and their mounting planes to the gimbal/slider are located in the two perpendicular planes. It provides a wide space to maximize the thickness of the comb drives and the number of interdigitated comb electrodes to enhance the driving force. More comb drives can be arranged if necessary; in fact, Figure 3.5 shows an example of six-comb microactuator. Therefore, the electrostatic force generated by the comb structure can be large enough to actuate the moving mass. This is a considerable improvement for electrostatic actuating devices that are generally limited by the small driving force.

Figure 3.5 Cross-section of the microactuator with six comb drives.

c) Slider-driven microactuator

The microactuator will be assembled between the suspension's gimbal and slider when integrated with magnetic R/W head; it actuates the slider to do the track seeking and track following. The slider is mounted on the bottom plane of the movable electrodes of the silicon actuator. Under the action of the electrostatic force and springs' mechanical restoring force, the movable electrodes can move to and fro constantly, in order to laterally drive the slider with read/write magnetic heads. The

advantages of the slider-driven microactuator are the compatibility with standard slider and magnetic heads as well as its influence in reducing the mechanical resonance of the suspension.

d) Low driving voltage

As for electrostatic actuators, the driving voltage has a square effect on the resulting electrostatic force, being usually larger than 50 V in the commonly SOI wafer electrostatic actuators [6, 14-16]. Thus, it will induce danger and unreliability of the operation. In this devised electrostatic microactuator, the relatively thicker and large number of comb teeth pairs contribute by a large proportion to the increase of the resulting electrostatic force; making it possible to use lower voltage to produce the needed driving force. The designed voltage is only around 30~40 V.

e) High structural robustness and good reliability

Because of the thick silicon structure that is bonded to the glass wafer, the microactuator has a large enough robustness and strength to sustain its to-and-fro movement. The driving mass (including the slider) can be mounted on the bottom plane of the upstanding movable comb electrode, which is just one part of the microactuator instead of mounting with the whole side of the comb drives; this will facilitate the assembling with slider. The silicon-on-glass process makes the comb drive structure to be very thick (typically $> 100\ \mu\text{m}$), comb finger to be $5\ \mu\text{m}$ and the air gap to be $3\sim 4\ \mu\text{m}$, hereby making the comb drive actuator have a very good structural strength and working reliability. This overcomes the fragility of comb drive structures and electrostatic pull-in instability of the general SOI processed electrostatic actuators with a typical air gap of $2\ \mu\text{m}$.

3.2 Actuation Mechanism

3.2.1 Electrostatic mechanism

The microactuator utilized interdigitated fingers (comb drive) structure and moves in lateral direction; thereby the electrostatic actuation mechanism is used. The theoretical background of comb drives is from the parallel-plate capacitor, mechanism of which was introduced in previous section 2.1.2 and 2.1.3. The electrostatic attracting force can be derived as [26-27]

$$F_e = -\frac{\epsilon_0 A_0 V^2}{2d^2} = -\frac{\epsilon_0 t V^2}{2d} \quad (3.2)$$

Thus the force F_e is proportional to the square of the excitation voltage V and inversely proportional to the separation gap d , where ϵ_0 is the vacuum permittivity ($\epsilon_0 = 8.85 \times 10^{-12}$ F/m), A_0 is the effective area of plate ($A_0 = td$). Obviously, the relationship between the electrostatic force and the separation distance is a nonlinear one. It means the parallel-plate capacitor with displacement in the normal direction may not be an ideal choice for an actuation application. This is the major drawback of these structures; unless the movement is small compared to the electrostatic gap. The demand for a linear actuate/sense device leads to the design of the electrostatic comb drive, which consists of interdigitated cantilever beams called comb fingers.

Interdigitated comb drives are widely used in micro-resonators design, providing a lateral electrostatic force that is independent of lateral positions. The comb drive actuator, as shown schematically in Figure 3.6, consists of two sections of comb-like structures with the finger overlapping each other. One comb is free to move while the other is fixed in a stationary position. The working concept of the comb is similar to that of the parallel-plate capacitor and this concept may be applied to the linear and rotary motion of comb drives.

Figure 3.6 Schematic diagram of a comb drive structure

The lateral comb drive keeps the steady separation distance (air gap) during the movement of the interdigitated fingers. From the derivation in section 2.1.3 and equation (3.2), the lateral electrostatic force of a comb drive with n pairs of movable fingers is obtained as

$$F_e = \frac{n \epsilon t}{d} V^2 \quad (3.3)$$

Equation (3.3) can be used to calculate the electrostatic actuating force of laterally driven comb drives [73]. However, the generated interdigitated field force by comb drives is generally smaller than the direct force across parallel-plate capacitors, but it exhibits a better controllable range of motion. As long as the movable fingers are precisely centered in their corresponding gaps, the lateral force that acts on each finger (in the y -direction) will exactly cancel out one another. To increase the force and obtain maximum displacement using this comb drive mechanism, a cantilever with a large number of comb teeth should be designed.

Besides the lateral movement, the comb fingers may move in the transverse and vertical directions, i.e. y - and z - directions as shown in Figure 3.6. These two kinds of motions can be considered as undesirable for a linear comb drive due to the fact that

there is a tendency for the opposite fingers to stick together, or the levitation phenomenon of comb fingers, which causes an undesirable effect in the actual application. The transverse and levitated forces involve a non-linearity with the displacement, which will cause the side-instability of the comb drive and should be avoided in linear comb drive actuators. Several measures can be taken to reduce the transverse and vertical motions, such as controlling the driving voltage and maximum lateral deflection value, or designing a high stiffness ratio flexure for the comb drive actuator.

To conclude, the linear comb drive designing has three degrees of freedom. The lateral movement of the comb provides the linearity to be good for actuating and sensing applications. Transverse and vertical motion can be only looked as underdog of linear actuator. To ensure the good linearity of driving force and displacement as well as the stability of actuation and sensing applications, the lateral comb drives are developed widely in MEMS devices.

3.2.2 Dynamics of actuation

Figure 3.7 Schematic model of spring-mass-damper system

A simplified dynamic scheme of an electrostatic actuator, which consists of a fixed stator and a flexure-suspended moving mass, is shown in Figure 3.7. Assuming that the flexure suspension constrains the shuttle motion to a single degree of freedom

(operational direction of the comb drive), the equation describing this motion is based on a simple spring-mass-damper model [101]. An external force (electrostatic driving force) will be exerted on the flexure-suspended movable comb fingers and slider that act as the proof mass m , and mass motion bends the flexure spring (denoted with the spring stiffness K) and generates a restoring force proportional to the mass displacement. At the same time, the viscous air damping c of comb drive structure also generates a force to the movement of the mass. This force is proportional to the mass velocity and is usually an energy loss mechanism. The actuator's displacement x produced by an electrostatic force F_e is modeled by using the second-order differential equation

$$m\ddot{x} + C\dot{x} + Kx = F_e \quad (3.4)$$

where m is the mass of the moving part, C is the damping coefficient, K is the stiffness of the suspension flexure, and F_e is the electrostatic actuating force. This is a second-order system, and the transfer function $G(s)$ would be achieved by using the Laplace transformation,

$$G(s) = \frac{A\omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2} \quad (3.5)$$

Where $\omega_n = \sqrt{K/m}$, $\zeta = C/(2m\omega_n)$, A is constant value, and quality factor is

$$Q = \frac{1}{2\zeta} = \frac{m\omega_n}{C} = \frac{\sqrt{mK}}{C} \quad (3.6)$$

The displacement x is in a complex form, and its frequency response will be

$$x(f) = \frac{F_e}{-(2\pi f)^2 m + j2\pi f C + K} = \frac{x_0}{1 + \frac{j}{Q} \frac{f}{f_r} - \left(\frac{f}{f_r}\right)^2} \quad (3.7)$$

where $x_0 = F_e/K$ is the amplitude. When f is zero, and f_r is given by

$$f_r = \frac{1}{2\pi} \sqrt{\frac{K}{m}} \quad (3.8)$$

The frequency response of the mass-spring-damper system can be divided into three regions, i.e., (a) below resonance frequency range; (b) resonance frequency; (c) above resonance frequency range. In the low-frequency range, the third term (stiffness term) in equation (3.4) is dominant and thus the displacement x follows Hook's law:

$$F_e - Kx = 0 \quad (3.9)$$

At the resonance frequency f_r , the amplitude becomes the largest when damping is relatively small. In the high frequency region, the first term (mass) in equation (3.7) is dominant, and the displacement amplitude x is approximately $F/(2\pi f)^2 m$, where f is the input frequency. The working region of the actuator is the low-frequency region (below resonance frequency); since the relatively bigger displacement of the actuator is preferred at the limited force and thus the smaller stiffness and resonant frequency is desired [73]. Therefore, the micro motion of the microactuator can be controlled by the driving force and stiffness of the supporting flexure. Generally, the electric force is required to be large enough to drive the functional elements such as magnetic heads and slider; at the mean time, the flexure should be stiff enough to provide a high-bandwidth servo. Thus, the two aspects must be considered to design the actuator.

3.3 Electrostatic Design of the Microactuator

According to the objective of the microactuator and fabrication feasibility, the pre-defined specifications of the electrostatic microactuator are listed in Table 3.1.

As shown in Figure 3.8, when voltage is applied to the fixed electrodes, the parallel-plate capacitors formed in the interdigitated field are charged and the movable comb fingers move towards the fixed fingers by the electrostatic attracting force. The electrostatic force of comb electrodes only drives the movable part in parallel motion as the perpendicular electric force is balanced off when the air gaps are equal.

Table 3.1 The coarse designed microactuator specifications

Microactuator size ($\mu\text{m} \times \mu\text{m} \times \mu\text{m}$)	1000 $\times$ 500 $\times$ 200
Width of comb tooth (μm)	5
Number of movable comb teeth	Around 100
Air gap between teeth (μm)	3~4
Thickness of the comb drives (μm)	100
Flexure length (μm)	Around 420
Flexure width (μm)	5
Moving mass (with femto-slider) (mg)	0.6
Stroke of the microactuator (μm)	0.5
Loaded resonance frequency (Hz)	>1 kilo

Figure 3.8 Schematic diagram of double-sided comb teeth pairs

From the Equation (3.3), for a comb drive with n pairs of movable comb teeth, the electrostatic force along comb teeth is $F_e = \frac{n \epsilon t}{d} V^2$, where ϵ is the permittivity of air, d is the gap between the fixed comb teeth and movable comb teeth, n is the number of movable comb teeth, V is the DC driving voltage, and t is the thickness of the comb structure.

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Note that the electrostatic force F_e is proportional to the aspect ratio of the inter-electrode gap t/d , and varies in a quadratic relationship to the voltage V . Therefore, the force F_e will be increased when increasing the thickness of comb drive t , narrowing the gap of inter-electrodes d , or increasing the driving voltage V . Certainly, increasing the number of comb teeth n will also increase the electrostatic force.

Under the electrostatic force, a horizontal displacement of the movable comb drives will be produced. Together with the restoring force of the support flexures, the comb drives will be moved about to drive the head/slider which is mounted in the bottom plane of the movable comb electrode.

Once the movable comb drives move towards the fixed comb drives, the only way that they can return to the central position is by the restoring force of the support flexures. The support flexures will generate a mechanical restoring force ($F_m = -Kx$), which is balanced out with the electrostatic force. In equilibrium of the microactuator, it satisfies $F_e + F_m = 0$, which is the same as the actuator's dynamic equation (3.9).

Rearrange it and we can get

$$\frac{n \epsilon t}{d} V^2 - Kx = 0 \quad (3.10)$$

Equation (3.10) is the governing equation of the electrostatic microactuator. It shows that the displacement (or the stroke) x in the parallel direction of the comb drives is dependent on both the electrostatic force F_e and the spring constant K of the flexures. Therefore, to acquire a large designed displacement of the comb drives, either the electrostatic force should be enhanced, or the stiffness (or spring constant) of the support flexures should be properly designed in order to attain the high resonant frequency at the same time. As seen from equation (3.10), the electrostatic force depends on four factors: n , t , d and V . That is, there are four solutions to enhance the electrostatic force so as to attain the designed stroke of the microactuator.

- **Thickness of comb drive (t)**

According to equation (3.10), the electrostatic force is proportional to the actuator's effective thickness, so that the thickness is preferred to be as large as possible to provide the large force. However, too large a thickness will increase the difficulty of fabrication, since the comb drive actuator has a high-aspect-ratio structure and this ratio is limited by the fabrication capability. Here the aspect-ratio refers to the height-to-width ratio of the trenches in the actuator, which is actually the effective thickness of comb drive over the comb teeth's air gap. Up to the present, the aspect-ratio in micromachining technology is around 30:1, and thus the maximum thickness of comb actuator is 120 μm , given that the air gap is less than 4 μm .

In this project, the microactuator is fabricated with the bulk silicon wafer by using Deep RIE. This provides the possibility of the comb actuator with large thickness. The effective thickness of the comb actuator designed here is 100 μm on considering the feasible air gap. Compared to the normal thickness of electrostatic actuators that are fabricated with SOI wafer (2~50 μm), the thickness of the bulk-micromachined actuator is much larger and will generate a larger electrostatic force. This is an explicit advantage for the bulk silicon wafer fabricated 3-D microactuator.

- **Numbers of comb pairs (n)**

As introduced in the characteristics of the electrostatic microactuator (section 3.1.3), the upstanding structure of the actuator provides a large space to hold as many parallel comb drives as possible. This measure can remarkably increase the number of comb capacitor pairs n , thereby enhancing the electrostatic force F_e .

For example, if the electrostatic force of the actuator with four groups of comb drives in Figure 3.3 is smaller than the design objective, we can allocate six groups or even more parallel comb drives to the microactuator, thereby increasing the

electrostatic force greatly. Figure 3.6 shows a case of the 6-group comb actuator design. In the design, 4-group combs are enough to generate the required force together with the efforts of other parameters, and the number of the comb pairs is designed to be 100.

- **Driving voltage (V)**

In general, a higher driving voltage produces a larger electrostatic force, which is very sensitive to the changes in voltage since it varies with the square of the voltage. However, the high driving voltage would also induce disadvantages and failures for the microactuators, such as pull-in instability and high-voltage risks. Currently, most of the driving voltage of electrostatic actuators exceeds 50 V, such as the actuators developed by [14-16] and [83-84]. For this 3-D microactuator, a relatively low voltage of 30~40 V, which will be more advantageous than the common electrostatic actuators, is designed.

- **Air gap (d)**

The electrostatic force of comb drive actuator greatly depends on the air gap of comb teeth, and it is critical because the force is inversely proportional to the air gap while the gap is only several micron meters. The typical values of the air gap between the fixed and movable comb teeth are 2~5 μm . Basically, the narrower is the air gap, the larger is the electrostatic force. However, if the gap is smaller than 2 μm , it will be difficult for the micromachining process to achieve such narrow and deep trenches; furthermore, it increases the failure through sticking instability. Therefore, the air gap design would be very critical and practical to the electrostatic microactuator. In the following section, we will consider several values of the air gap and compare their effect on the electrostatic force.

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As previously described in Table 3.1, it was proposed that the electrostatic microactuator model possess the specified initial conditions and detailed dimensions. The electrostatic force can be calculated by setting the following initial conditions: the effective thickness of comb drives $t=100\text{ }\mu\text{m}$, and four parallel comb drives in the microactuator, the driving voltage to be 20 V.

Firstly, we consider an air gap of the comb drives $d=4\text{ }\mu\text{m}$, with the total number of movable comb teeth $n=100$. By using equation 3.24, we can calculate the electrostatic driving force generated in the entire microactuator

$$F_e = \frac{n \epsilon t}{d} V^2 = \frac{100 \times 8.854 \times 10^{-12} \times 100 \times 10^{-6}}{4 \times 10^{-6}} \times 20^2 = 8.854\text{ }\mu\text{N}$$

If we choose the gap $d=3\text{ }\mu\text{m}$, and the movable teeth number $n=100$, then the electrostatic force is

$$F_e = \frac{n \epsilon t}{d} V^2 = \frac{100 \times 8.854 \times 10^{-12} \times 100 \times 10^{-6}}{3 \times 10^{-6}} \times 20^2 = 11.805\text{ }\mu\text{N}$$

In another case, if we consider the gap $d=2\text{ }\mu\text{m}$, and the movable comb teeth number $n=100$, the electrostatic force will be

$$F_e = \frac{n \epsilon t}{d} V^2 = \frac{100 \times 8.854 \times 10^{-12} \times 100 \times 10^{-6}}{2 \times 10^{-6}} \times 20^2 = 17.708\text{ }\mu\text{N}$$

However, for our bulky silicon microactuator, it is almost impossible to fabricate the $2\text{ }\mu\text{m}$ air gap with $100\text{ }\mu\text{m}$ deep. Therefore, the driving voltage would have to be increased to acquire the larger electrostatic force.

If we consider the driving voltage to be 30 V, and the air gap $d=3\text{ }\mu\text{m}$, then the electrostatic force will be

$$F_e = \frac{n \epsilon t}{d} V^2 = \frac{100 \times 8.854 \times 10^{-12} \times 100 \times 10^{-6}}{3 \times 10^{-6}} \times 30^2 = 26.561\text{ }\mu\text{N}$$

If the driving voltage is 40 V with air gap of $3\text{ }\mu\text{m}$, the electrostatic force will be $47.22\text{ }\mu\text{N}$, which is very large for the electrostatic microactuator applications. Table

3.2 listed the designed electrostatic force with different voltage and air gap (the number of comb pairs is 100).

Table 3.2 Designed electrostatic force as function of air gap and voltage

	Voltage = 20 V	Voltage = 30 V	Voltage = 40 V
Air gap = 3 μm	11.805 μN	26.561 μN	47.22 μN
Air gap = 4 μm	8.854 μN	19.92 μN	35.415 μN

For the fabrication feasibility and requirement of the driving force, the air gap of comb drives is designed for two dimensions: 3 μm and 4 μm . The driving voltage will be around 20 μN to 47 μN , depending on the spring stiffness and displacement requirement.

It can be seen that the electrostatic force increases when the air gap of comb fingers decreases. However, a small gap such as 2 μm may result in a high aspect ratio structure for the bulk microactuator, thus increasing the difficulty of fabrication due to the aspect ratio limit of DRIE process. In fact, we can increase the number of movable comb fingers or enhance the driving voltage slightly to increase the force, thereby producing a large stroke. Furthermore, we can consider enhancing the stroke of the actuator: to reduce the stiffness of the flexures in the operational direction.

As shown in Equation (3.10), the lateral displacement of movable comb drives is highly dependent on the stiffness of the support flexures. Therefore, the important measure to increase the moving displacement of the comb drives is to reduce the spring constant of the support flexures, in case the electrostatic force is limited. The flexure study and dimension design will be discussed in Chapter 4. Here the simple straight flexure will be used for the preliminary design and modeling of the microactuator.

3.4 Modeling of the Microactuator

3.4.1 Micromechanical flexure in the microactuator

As seen from Figure 3.3, there are three spring flexures in the microactuator to mechanically connect and support the movable electrodes. In addition, they provide the restoring force for the movable electrodes during the electrostatic actuation. The flexure’s stiffness will influence the displacement of microactuator; it also determines the resonant frequency of the microactuator.

Figure 3.9 Schematic diagram of straight-plate flexure.

Figure 3.9 shows the scheme of the straight flexure in the microactuator, with the beam width b and length L . The straight beam/plate flexures are the simplest flexures that are usually used in micromechanical structures. They are linear, small displacement flexures; moreover, they constrain motion to a linear pathway.

Generally, the flexure should be designed to meet several requirements for the microactuator application. For the support flexures, they can be distorted in either the vertical or the desired horizontal direction when forces are applied to them. As in the

case of the microactuator used for hard disk drive positioning, it moves about in parallel to the disk surface, and does not require the vibration or excitations in the vertical or z-axis direction. So the deformation in these directions should be prevented or minimized. The ideal condition is that the microactuator must be flexible in the operational direction of motion (here, along the disk track), but stiff in the vertical and z-axis directions to prevent the resonance modes in these directions, which means the flexure should have a large stiffness ratio. These requirements are mainly determined by the flexures that support the whole moving structure.

Another basic requirement is that the flexure should be stiff enough in both vertical and operational directions. Firstly, the vertical stiffness must be high enough to withstand the weight due to the movable electrodes and also the loading force that is applied to the slider by the suspension arm. This loading force presses the slider down to the disk surface and is necessary to maintain an adequate air bearing between the slider and the disk surface during operation. At the same time, the flexure should be stiff enough in vertical direction to avoid buckling failure when the large loading forces are applied. Secondly, as the stiffness in operational direction of the flexure determines the natural frequency of the microactuator, the flexure should fulfill the high bandwidth requirement in particular applications. Basically, the desired frequency of the microactuator for HDD should be more than 1 kHz loading with 20% femto-slider (mass of 0.6 milligram [102]). Using the equation $f_n = \frac{1}{2\pi} \sqrt{\frac{K_x}{m}}$, it is able to compute the stiffness requirement for the flexure in operation direction to be:

$$K_{x0} > 4 \pi^2 f_n^2 m = 23.68 \text{ N/m}$$

In the real application, the flexure should have the stiffness in the operational direction larger than K_{x0} to achieve a high working frequency when loaded with the

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slider element. The stiffness effect and dimension design of the straight flexure will be studied in the following section.

As in the 3-D microactuator, the flexures can be approximate as thin plates because of the large thickness t and small width w . The plates are fixed at one end while subjecting to an orthogonal electrostatic force F_x and a bend moment M_0 at the other end of the flexure. The electrostatic force and moment are distributed uniformly on the end plane, and no other loads; thus, the shell can be almost simplified to be straight beams and the beam deflection theory can be used to study the deflection of the thick flexures.

A concentrated electrostatic force together with a moment from the movable electrodes is applied to one end of the beam. The axial displacement along the y-axis can be obtained from the small deflection beam theory [103], while the lateral displacement along x-axis cannot be obtained from Hooke's law as the movable end of the flexures is fixed at the heavy slider mass and constrained by a momentum. By applying the force-displacement method to the flexure, we can obtain the displacement at the end 2,

$$u_2 = \frac{F_x l^3}{3EI} - \frac{M_0 l^2}{2EI} \quad (3.12)$$

$$v_2 = 0 \quad (3.13)$$

$$\theta_2 = \frac{F_x l^2}{2EI} - \frac{M_0 l}{EI} \quad (3.14)$$

where E is the Young's Modulus of elasticity, I is the moment of inertia given by

$I = \frac{1}{12} t w^3$, where t is the thickness and w is the width of the beam. As the flexures

are connected to the slider mass and restricted to only translational movement, we can set $\theta_2 = 0$. Hence, we can calculate for $M_0 = F_x l / 2$.

$$u_2 = \frac{F_x l^3}{3EI} - \frac{M_0 l^2}{2EI} = \frac{F_x l^3}{12EI} = \frac{F_x l^3}{Etw^3} \quad (3.15)$$

Therefore, the spring constants of the flexure in the x - and y - direction are

$$k_x = \frac{F_x}{u_2} = \frac{12EI}{l^3} = \frac{Etw^3}{l^3} \quad (3.16)$$

$$k_y = \frac{EA}{l} = Et \frac{w}{l} \quad (3.17)$$

Then, we can get the total stiffness of the microactuator with 3 flexures,

$$K_x = 3k_x = \frac{3Etw^3}{l^3} \quad (3.18)$$

$$K_y = 3k_y = \frac{3Etw}{l} \quad (3.19)$$

The flexure is very stiff in the y -direction for those with only extruding /contracting deflection, instead of beam bending. Thus, the stiffness ratio is

$$\frac{K_y}{K_x} = \frac{l^2}{w^2} \quad (3.20)$$

In general, the stiffness ratio of a straight beam can be very high. For example, the stiffness ratio is as large as 7056 for a beam of length 420 μm and width 5 μm . In our case, the displacement of the straight flexure is very small (around 1 μm); thus, the non-linear effects would be very small and negligible.

Because of the high stiffness, the straight flexure has quite a small deflection under the limited electrostatic forces. Using Equations (3.16)~(3.17), we can calculate the deflections of the microactuator with straight flexures in the x - and y - directions. Suppose the electrostatic force is $F_e = 19.92 \mu\text{N}$, total moving mass of the slider and electrodes to be 0.6 mg, flexure width $w = 5 \mu\text{m}$, air gap $d = 4 \mu\text{m}$ and flexure length $l = 420 \mu\text{m}$, we obtain the stiffness and deflection of the actuator with three flexures:

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$$K_x = \frac{3Et w^3}{l^3} = \frac{3 \times 130 \times 10^9 \times 100 \times 10^{-6} \times (5 \times 10^{-6})^3}{(420 \times 10^{-6})^3} = 65.8 \text{ N/m} > K_{x0}, \quad (3.21)$$

$$\delta_x = \frac{F_e}{K_x} = \frac{F_e l^3}{3Et w^3} = \frac{19.92 \times 10^{-6} \times (420 \times 10^{-6})^3}{3 \times 130 \times 10^9 \times 100 \times 10^{-6} \times (5 \times 10^{-6})^3} = 0.404 \text{ } \mu\text{m}, \quad (3.22)$$

$$\delta_y = \frac{F_y}{K_y} = \frac{F_y l}{3Et w} = \frac{0.6 \times 10^{-6} \times 9.8 \times 420 \times 10^{-6}}{3 \times 130 \times 10^9 \times 100 \times 10^{-6} \times 5 \times 10^{-6}} = 1.27 \times 10^{-5} \text{ } \mu\text{m} \quad (3.23)$$

Then, we calculate the resonant frequency of the microactuator with and without loading the magnetic heads and slider. According to the pre-designed dimensions in Table 3.1, the effective mass of the movable part is

$$m = \rho A t = 2330 \times (1000 \times 50 + 200 \times 5 \times 30 + 4 \times 400 \times 10 + 3 \times 400 \times 5) \times 10^{-12} \times 100 \times 10^{-6} \\ = 2.37 \times 10^{-8} \text{ Kg} = 0.0237 \text{ mg}$$

$$f = \frac{1}{2\pi} \sqrt{\frac{K}{m}} = \frac{1}{2\pi} \sqrt{\frac{65.8}{2.37 \times 10^{-8}}} = 8374 \text{ Hz} \quad (3.24)$$

If the microactuator loads 20% femto-slider (mass of 0.6 mg) [102], the resonant frequency of the loaded microactuator would be

$$f = \frac{1}{2\pi} \sqrt{\frac{K}{m}} = \frac{1}{2\pi} \sqrt{\frac{65.8}{0.62 \times 10^{-6}}} = 1.64 \text{ kHz} \quad (3.25)$$

Therefore, for flexure widths of 5 μm and above, the straight-flexure microactuator has enough stiffness to support the more than 1 kHz servo bandwidth of the microactuator while loaded with the 20% femto-slider.

Another issue we need to take into account is the buckling instability of the flexures. In general, a column in the form of beam, plate, shell, stick, which is relatively long, slender and carrying an axial compressive load, could possibly have a radical deflection of the axis suddenly when the axial loading has exceeded some critical value. This phenomena is called buckling instability, with the maximum load known as critical load. If the load is not released or reduced, the column will collapse

[103,104]. Obviously, the buckling of the flexure must be avoided. Figure 3.10 shows the buckling phenomena of a long column when being compressed in axial direction.

Figure 3.10 Buckling phenomena of an axial-compressed column

In this design, the three flexures in the microactuator are pre-designed to be $L=420\text{ }\mu\text{m}$ in length, $w=5\text{ }\mu\text{m}$ in width, $t=100\text{ }\mu\text{m}$ in thickness, as the flexures are fixed at the 2 ends with the end-fixity factor being $K=0.65$. According to the equation $r = \sqrt{I/A}$ [105], the radius of gyration of the rectangular cross-section flexure is $w_a/\sqrt{12}$. Hence, the slenderness ratio of the flexure is

$$\frac{KL}{r} = \frac{0.65 \times 420 \times 10^{-6} \times \sqrt{12}}{5 \times 10^{-6}} = 189.14 \quad (3.26)$$

When the Young's Modulus is $E=130\text{ GPa}$ and yield strength is $s_y=7\text{ GPa}$ [105], the transition slenderness ratio C of the flexure is

$$C = \sqrt{\frac{2\pi^2 E}{s_y}} = \sqrt{\frac{2\pi^2 \times 130 \times 10^9}{7 \times 10^9}} = 19.15 \quad (3.27)$$

As $KL/r > C$, the flexure is long column. According to the Euler formula, the critical load P_{cr} of the 3-flexure microactuator is

$$P_{cr} = \frac{\pi^2 EI}{(KL)^2} = \frac{\pi^2 \times 130 \times 10^9 \times 100 \times 10^{-6} \times (3 \times 5 \times 10^{-6})^3}{12 \times (0.65 \times 420 \times 10^{-6})^2} = 161.4 \times 10^{-3} \text{ N} \quad (3.28)$$

At this load, the flexures will begin to buckle. A safe load should be a reduced value by applying the factor of safety to the critical load. Typically, the design factor of $N = 3$ is large enough, and the allowable load $P_a = P_{cr}/3 = 53.8 \text{ mN}$.

The microactuator would usually be strong enough to support the preload force from the suspension arm when performing the assembly of the microactuator. This preload force is usually around 3 to 3.5 gram force ($\sim 34.3 \text{ mN}$) [106]. Since the preload is smaller than the allowable load P_a , the flexures are safe to prevent the buckling instability.

3.4.2 Finite element analysis of the microactuator

Finite Element method (FEM) is a computer-based numerical technique for calculating the strength and behavior of engineering structures. It can be used to calculate and analyze deflection, stress, vibration, buckling behavior and many other phenomena.

Finite element analysis is a way to deal with structures that are more complex than that with analytical equations. FEA deals with complex boundaries better than finite difference equations will, and gives answers to "real world" structural problems. It is an efficient analysis tool to simulate the responses of the microflexures in the experiments when their geometries, material properties, as well as the applied loads and the imposed constraints are given.

However, there might be some deviations between the finite element simulation results and the analytical values. One tends to suspect the curve segment as the

underlying causes for such deviation as the curve geometry relaxes the rigid structure and this method may not be suitable to express such geometry. In the design perspective, one tends to choose the FE results as the basis for design analysis.

In this project, finite element calculation is performed using a commercial finite element package ANSYS [107]. ANSYS is a multi-purpose finite element modeling package for numerically solving a wide variety of problems in mechanical structures, thermal processes, computational fluid dynamics, and electromagnetic fields. ANSYS provides a rich graphics capability that can be used to display the results of an analysis on a high-resolution graphics workstation. In the design of the microactuators, the static analysis and modal analysis would be used to study the deflection and resonant frequency of the microactuators. Also, the stress result would be used to verify the mechanical performance of the microactuators.

A typical ANSYS analysis has three distinct steps:

- Build the model (Pre-processing).
 - Define keypoints/lines/areas/volumes
 - Define element type and material/geometric properties
 - Mesh lines/areas/volumes as required
- Apply loads and obtain the solution (Solving).
 - Specify the boundary conditions (loads, constraints)
 - Solve the resulting set of equations.
- Review the results (Post-processing).
 - Lists of nodal displacements, Element forces and moments
 - Deflection plots
 - Resonant frequency and mode shape
 - Stress contour diagrams

- **Pre-processing**

Pre-processing is the process of building the finite element model, which includes the creation of entity-model according to specified dimensions, element-type and material-property defining, meshing, and the model modification.

The finite element model used in the analysis is a simplified model of the electrostatic microactuator. The microactuator consists of hundreds of interdigitated comb teeth. The movable teeth move steadily to generate the electrostatic force for driving the flexure and slider part. Therefore the comb teeth can be omitted by applying the equivalent force to the movable slider, and this will simplify the model and greatly save the computational time while solving the finite element model. Figure 3.11 shows the simplified model of the microactuator.

Figure 3.11 Finite element model of the microactuator in ANSYS

3-Dimensional 10-nodes tetrahedral structural solid element SOLID92 is chosen due to its ease of meshing. The 3D tetrahedral solid element has three degrees of freedom per node. At each of its 10 nodes is a translation in the nodal x, y, and z direction. The material used for this micro-flexure device is bulk silicon. According to the MEMS material database [108], each property varies due to process variability as

each fabrication laboratories has its own process control. The material properties used here are listed in Table 3.3:

Table 3.3 Properties of Silicon

Young's Modulus, E	Density, ρ	Poisson ratio, ν
130 GPa	2330 Kg / m ³	0.2

The next step is to describe the geometrical dimensions for generating the finite element model. The dimensions of flexures are dependent on the specifications of microactuator, which has been designed and calculated in the previous section. Some variations of dimension occur in the flexures; thus, we can examine the stiffness variation of the flexures correspondingly. The 3-D model is then meshed by using element Solid92, with the total number of elements being 16205.

• **Applying loads and solving**

After building the 3D model and meshing with the 3D finite elements, constraints and loads will be applied to the model; thus, the computer can solve the simultaneous set of equations that the finite element method generates.

The flexures are constrained at one end, and its deflection is dependent on the electrostatic driving force in horizontal direction and a weight of the head/slider in vertical direction. The value of electrostatic force has been calculated previously. We take the value of the electrostatic force to be 19.92 μN (for a driving voltage of 30 V and air gap of 4 μm), and suppose the weight of the slider (moving mass) is 0.6 mg force, which will be exerted to the bottom plane of the slider uniformly.

• **Viewing results and discussion**

Figure 3.12 X-directional displacement of the microactuator with straight flexures

Firstly, the static analysis was carried out and the displacement result was viewed. Figure 3.12 shows the deflection of the microactuator with 5 μm -width flexures. The displacement in the x -direction (operational direction) calculated by ANSYS is 0.403 μm , while the displacement calculated by analytical equations is 0.404 μm . The difference of only 0.33% between the two results suggests that the equations are accurate enough to calculate the displacement of the microactuator in the real model.

The stress distribution for the flexures is also determined in the FE simulation, and the localized stress concentration region developed during the deformation is contoured. Using a static analysis and Von Mises stress criterion, the post-processing was carried out to plot the deformation contour of the flexure. Figure 3.13 shows the FEM Von Mises stress contour of straight-flexure microactuator. The maximum stress is 0.406×10^{-5} MPa, i.e. 40.6 Pa, which is very small compared to the yield stress of silicon material (around 7 GPa).

Figure 3.13 Contour of Von Mises stress in the straight-flexure microactuator

As expected, it can be seen that the stress concentration region is near the joint area of the flexure end as expected. The high stress experienced by the flexure is undesirable as prolonged usage may cause fatigue and results in a reliability problem. Thus flexure should not be too thin in order to avoid the large localized stress concentration.

Secondly, the modal analysis was performed to study the mode shape and resonant frequency of the microactuator. Referring to electrostatic actuator design in Figure 3.3, there are four pairs of comb teeth and they can be simplified as a slider mass. The equivalent mass of comb teeth should be added to the slider part in the microactuator model for modal analysis. The meshed 3-D model was constrained at the top plate, and number of extracted modes was set to 6 in the frequency range of 0 to 1 MHz. Table 3.4 lists the modes and their respective resonant frequencies of the equivalent microactuator model for a flexure width of 5 μm .

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Table 3.4 Extracted modes and resonant frequencies

Mode	1	2	3	4	5	6
Resonant frequency, Hz	8,385	67,645	166,672	217,817	217,940	219,085

Figure 3.14 First mode shape of the microactuator model at 8.385 kHz (x-direction)

Figure 3.15 Second mode shape of the microactuator at 67.645 kHz (torsion mode)

Figure 3.14 and 3.15 show the first two mode shapes of the equivalent electrostatic microactuator model. The FEM results show that the primary resonant frequency is 8.385 kHz, and it is very close to the theoretical result of 8.374 kHz, with an error of 0.12%. It proves that the equivalent model of the microactuator in the modal analysis is very accurate in calculating the resonant frequencies.

The first mode at resonant frequency of 8.385 kHz is the primary resonance mode, and the microactuator will work in horizontal direction below the first resonant frequency. The other modes show the mixed torsion and bending resonance, which are normally not useful. These resonance modes greatly depend on the flexure stiffness. Obviously, the resonant frequency of the microactuator can be enhanced by increasing the flexure width.

3.5 Chapter Summary

Being one of the most important actuating mechanisms, electrostatic actuation has been widely studied and developed. The motivation of the microactuator is to serve as the secondary stage actuator to perform the high-accuracy and high-bandwidth positioning of magnetic heads in the HDD, so as to fulfill the high-density and high-performance requirements of the HDD industry. This chapter has presented the design of the 3-D electrostatic microactuator that is bulk-micromachined with silicon-on-glass (SOG) technology. The design structure, actuating mechanism, and modeling of the electrostatic microactuator have been demonstrated subsequently.

Unlike the well-known and widely used planar configuration of comb drive applications, we have developed the high-aspect-ratio thick comb drives for the microactuator, which is fabricated in bulky silicon wafer and arranged to be vertical standing other than the normally planar configuration. A series of parallel comb drives

can be arranged in the microactuator to greatly enhance the electrostatic force at a low driving voltage. Furthermore, the bulky structure provides a good strength for the microactuator, and good reliability for the fabrication and operation process.

The dynamics and actuation of the microactuator have been studied, and the basic flexure has also been designed. The straight flexure is selected for the pre-design of the microactuator, and its stiffness effect has been studied. After that, finite element analysis was carried out and the results supported the analytical results. With an air gap of 4 μm and flexure width of 5 μm , the microactuator has a primary resonant frequency of 1.64 kHz loaded with the 20% femto-slider. By applying 30 V voltage, the microactuator has achieved a displacement of 0.403 μm . The results show that the SOG 3-D microactuator has a high-bandwidth and high-resolution actuation, and therefore, can be used as the secondary actuator in HDD or similar applications.

Chapter 4

Design and Analysis of the Flexures

In this chapter, several types of flexures used for the 3-D bulky microactuators are discussed. According to the stiffness requirements of the mechanical flexures, four types of flexures are studied: straight flexure, folded flexure, asymmetric-quad flexure and symmetric-quad flexure. Both the theoretical and finite element analysis are used to study the deflection and stiffness of the flexures. The suitable flexures and their optimized dimensions have been designed.

4.1 Requirements of the Flexures

Micromechanical flexures are important parts of the actuator or sensor system, such as resonators, accelerometers, and gyroscopes. By supporting the whole moving system, the main function of the micromachined flexures is to provide proper guidance for the sensing or actuating motion. Flexures are very important parts for movable structures as well as functional components, since they determine the system stiffness and dynamic performance of the micromachines.

As the electrostatic microactuator is designed to operate at high resonant frequencies of more than 1 kHz, it is desired that the microactuator moves only in the operational direction. Several requirements need to be satisfied for these flexures:

(1) A flexure should provide a good rectilinear motion to prevent the proof mass in following the flexure trajectory pathway. At the same time, the flexure should constrain the tendencies of two comb fingers from touching one another to prevent any unnecessary shock out. This ensures stability in the proof mass path of motion. Figure 4.1 illustrates a representation of the non-linear motion of a beam flexure.

Figure 4.1 (a) Schematic diagram of a non-linear motion of a beam flexure (b) Schematic diagram of a comb drive with a clamped-clamped flexure

The rectilinear motion is a general requirement for the flexures. The flexure design for the electrostatic microactuator has greatly reduced the non-linear motion as several flexures are arrayed in parallel to reduce the trajectory motion. In fact, the undesired vertical displacement is just a few nanometers and can be ignored here, since the horizontal displacement of the flexure in the designed microactuator is only $1\text{ }\mu\text{m}$.

(2) A flexure should have a high resonant frequency for the high-bandwidth application as well as moderate compliance to provide the sufficient movement to drive functional elements. Intrinsically, the flexure design will determine the system stiffness and the resonant frequency. There is a trade-off between the flexibility and the resonant frequency for the movable system. However, in some sensing and actuating applications, a resonant frequency of several kilohertz or higher is required, which requires the stiffness of the flexure in working direction to be very high. Furthermore, flexures are generally required to be stiff in the orthogonal direction in order to sustain the large proof mass without causing undesirable deflection. Therefore, a high stiffness for the flexure in these applications is required.

(3) It is desirable that the flexure have a large stiffness ratio, which means that the flexure should be compliant in the desired moving direction and very stiff in the

orthogonal direction. This is expressed as k_y/k_x in most cases, where k_y is the stiffness coefficient in the orthogonal direction and k_x is the stiffness coefficient in the desired operational direction. This is a general requirement of the flexure design, and is especially important for those systems that need constraint in the undesired movements. For those systems that already have a flexure, little change of the system stiffness is desirable after introducing a new flexure. With a suspension flexure already in the actuator for HDD, the incoming microactuator with flexures is expected to be only sensitive in the horizontal direction but stiff in the vertical direction. Therefore, the new flexure will not greatly influence the stiffness of the suspension; otherwise it will make the servo control to be more complicated.

In addition, the reliability of the flexure suspension also needs to be considered in order to reduce the stress concentration on the anchor and sharp corners in the flexure. The variation of stress concentration around the anchor will affect the Q-factor of the resonator [109]. Thick structures also require the flexures to be substantial and firm enough to sustain the bulk movable structures.

Due to the different requirements of the sensing/actuating applications, the design of flexures can vary widely. Flexures such as folded beam design [27, 109], crab-leg design [30, 41], clamp-clamp beam design [51], and spiral-spring design [47] have been investigated. As for the three-dimensional electrostatic microactuator, the flexure design should satisfy the requirements above with several types of flexures analyzed and compared in this chapter: the straight flexure, folded flexure, asymmetric-quad flexure, and symmetric-quad flexure. According to the requirements of high stiffness effect and high stiffness ratio, suitable flexures would be chosen and designed for the microactuator.

4.2 Analysis of the Different Types of Flexures

The spring constant (stiffness) of the different types of flexures were derived. The flexures for the microactuators are actually thin plates because of the large thickness t but small flexure width w . However, the force and movement of the plates in different thickness locations are quite uniform, and the small-displacement beam theory can be used to analyze the deflection of the flexures under the applied force.

4.2.1 Straight flexure

Figure 4.2 shows the schematic view of the clamped straight flexure, with the stiffness studied previously in Section 3.4.

Figure 4.2 Schematic diagram of a straight-plate flexure

The straight-beam flexure is fixed at the end 1 and the concentrated force F_x , F_y and M , are applied to the other end 2. At the end 2, the displacement u_2 , v_2 , θ_2 are

$$u_2 = \frac{4F_x l^3}{Etw^3} - \frac{6Ml^2}{Etw^3} \quad (4.1)$$

$$v_2 = \frac{F_y l}{Etw} \quad (4.2)$$

$$0 = \frac{6F_x l^2}{Et w^3} - \frac{12Ml}{Et w^3} \quad (4.3)$$

Here u_2 , v_2 , θ_2 represent the displacement in x -, y - direction and rotation angle of the flexure. Solving the three equations, the deflection of the flexure can be derived,

$$u_2 = \frac{F_x l^3}{Et w^3}, \quad v_2 = \frac{F_y l}{Et w}, \quad \theta_2 = 0 \quad (4.4)$$

The spring constants in x - and y -direction are derived as:

$$k_x = \frac{F_x}{u_2} = Et \frac{w^3}{l^3} \quad (4.5)$$

$$k_y = \frac{F_y}{v_2} = Et \frac{w}{l} \quad (4.6)$$

$$\frac{k_y}{k_x} = \frac{l^2}{w^2} \quad (4.7)$$

The straight flexure is very stiff in the y - direction because there is only very tiny elongation or compression, instead of beam bending. In general, the stiffness ratio of a straight flexure can be very high. For example, the stiffness ratio of 6,400 can be obtained for a beam of length 400 μm and width 5 μm .

The stiffness of the straight flexure is very high, and the force-to-displacement relationship is linear for small-displacement applications. However, for long and slender beams or plate structures that carry an axial compressive load, it tends to fail by the elastic instability, or buckling.

4.2.2 Folded flexure

Unlike the folded-beam flexures proposed by W.C. Tang, which has similar stiffness with the straight flexure [110,111], the folded flexure discussed here is very flexible and has small spring constant in both horizontal and vertical directions. The folded flexure would be a good choice for the one-end clamped and one-end free flexure design, which is similar to the serpentine flexure proposed by Gary Fedder

[41]. The difference lies in the constrained locations, where the serpentine flexure is constrained in the folded direction, instead of the vertical direction.

Figure 4.3 Schematic diagram of folded-plate flexure

A schematic diagram of the folded-plate flexure is shown in Figure 4.3. The folded flexure comprises three upright thin plates and two transverse conjunction plates. It is fixed at one end of the upright plate to the fixed electrodes, while the other end of the flexure is integrated into the movable electrodes. In considering the balance of the forces, the flexures should be symmetrically distributed in the microactuator; there are four such folded flexures in the system.

Generally, the folded flexure is more flexible than the straight flexure in both horizontal and vertical directions, because there are series-wound straight plates in the folded flexure, which has the same effect as the stiffness of straight flexure by increasing the flexure length.

To analyze the deflection of the folded flexure, the whole flexure can be treated as multi-segment structure: three vertical and two traverse beam units with each beam segment treated as one-end fixed and one-end free beam. Figure 4.4 shows the basic

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flexure segment and the deflection diagram, with one-end fully constrained, and the other-end constrained rotation only, i.e., $\theta_6=0$.

Figure 4.4 Schematic diagram of basic segment and deflection diagram of the folded-beam flexure. The dashed line is the beam after deformation

The total displacement at the free end of the flexure unit, u_6 , v_6 , θ_6 can be obtained by combining the displacement of each segment. Here, only the displacement resulting from bending is considered in the analysis. Deformation from shear, beam elongation, and beam shortening are neglected.

With the step-by-step analysis of the force and moment equilibrium of each segment, the displacement equations of the free-end segment can be obtained. Considering the force F_x , F_y are given, and the moment M , displacement at the end u_6 , v_6 as variables for the constrained rotation, the equilibrium equations at the free end can be achieved as the following (see Appendix A for derivation process):

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$$F_x \cdot \left(\frac{8l_0^3}{3EI_a} + \frac{l_a^3}{EI_a} + \frac{4l_a^2l_0}{EI_a} + \frac{6l_al_0^2}{EI_a} + \frac{l_a^2l_b}{EI_b} \right) - F_y \cdot \left(\frac{(l_a + 4l_0)l_b^2}{2EI_b} + \frac{3l_a^2l_b}{2EI_a} + \frac{3l_0^2l_b}{EI_a} + \frac{5l_0l_al_b}{EI_a} \right) + M \cdot \left(-\frac{(l_a + 2l_0)l_b}{EI_b} - \frac{2(l_0 + l_a)^2}{EI_a} + \frac{l_a^2}{2EI_a} \right) = u_6 \quad (4.8)$$

$$F_x \cdot \left(\frac{3l_0^2l_b}{EI_a} + \frac{3l_a^2l_b}{2EI_a} + \frac{5l_0l_al_b}{EI_a} + \frac{(l_a + 4l_0)l_b^2}{2EI_b} \right) - F_y \cdot \left(\frac{8l_b^3}{3EI_b} + \frac{(5l_a + 4l_0)l_b^2}{EI_a} \right) + M \cdot \left(-\frac{(3l_a + 2l_0)l_b}{EI_a} - \frac{2l_b^2}{EI_b} \right) = v_6 \quad (4.9)$$

$$F_x \cdot \left(\frac{2l_0^2}{EI_a} + \frac{3l_a^2}{2EI_a} + \frac{4l_0l_a}{EI_a} + \frac{(l_a + 2l_0)l_b}{EI_b} \right) - F_y \cdot \left(\frac{2l_b^2}{EI_b} + \frac{(3l_a + 2l_0)l_b}{EI_a} \right) + M \cdot \left(-\frac{(3l_a + 2l_0)}{EI_a} - \frac{2l_b}{EI_b} \right) = 0 \quad (4.10)$$

Solving the equations (4) ~ (6),

$$u_6 = (F_x(3l_a^3I_b + 8l_0^3I_b + 6l_a^2l_0I_b + 12l_al_0^2I_b + 6l_a^2l_bI_a - 24l_0l_al_bI_a - 24l_0^2l_bI_a) + F_y(-12l_0^2l_bI_b - 12l_0l_al_bI_b + 6l_al_b^2I_a) / (12EI_aI_b) \quad (4.11)$$

$$v_6 = (F_x(6l_0^2l_bI_b + 6l_0l_al_bI_b - 3l_al_b^2I_a) - F_y(4l_b^3I_a + 12l_al_b^2I_b + 12l_0l_b^2I_b)) / (6EI_aI_b) \quad (4.12)$$

$$M = F_x \left(l_0 + \frac{1}{2}l_a \right) - F_y l_b \quad (4.13)$$

Thus, the stiffness K_x of the whole flexure can be derived by

$$K_x = \frac{F_x}{u_6} = (F_x \cdot 12EI_aI_b) / (F_x(3l_a^3I_b + 8l_0^3I_b + 6l_a^2l_0I_b + 12l_al_0^2I_b + 6l_a^2l_bI_a - 24l_0l_al_bI_a - 24l_0^2l_bI_a) + F_y(-12l_0^2l_bI_b - 12l_0l_al_bI_b + 6l_al_b^2I_a)) \quad (4.14)$$

$$= \frac{12EI_a}{l_a^3 \left(\frac{3 + 8r_1^3 + 6r_1 + 12r_1^2 + 6r_2\alpha}{-24r_1r_2\alpha - 24r_1^2r_2\alpha} \right) + \frac{F_y}{F_x} l_a^3 (-12r_1^2r_2 - 12r_1r_2 + 6r_2^2\alpha)}$$

Where $\alpha = I_a/I_b = (w_a/w_b)^3$, $r_1 = l_0/l_a$, $r_2 = l_b/l_a$.

Under the conditions of the folded flexures, $l_a \gg l_0$ and $l_a \gg l_b$, i.e., $r_1 \ll 1$ and

$r_2 \ll 1$, equation (4.14) can be simplified and approximated as

$$K_x = \frac{12EI_a}{l_a^3(3 + 6r_1 + 6r_2\alpha) + \frac{F_y}{F_x}l_a^3(-12r_1r_2 + 6r_2^2\alpha)} \quad (4.15)$$

In the limiting condition of $w_a < w_b$, $F_y = 0$, the stiffness can be further simplified to

$$K_x = \frac{1}{3} \frac{Et w_a^3}{l_a^3} \quad (4.16)$$

The stiffness K_x is soft and less than 1/3 of the straight flexure stiffness.

The flexure stiffness in the y - direction is also simplified as

$$K_y = \frac{6EI_a}{l_a^3(4r_2^3\alpha + 12r_2^2 + 12r_1r_2^2) - \frac{F_x}{F_y}l_a^3(6r_1^2r_2 + 6r_1r_2 - 3r_2^2\alpha)} \quad (4.17)$$

If $F_x = 0$ and $w_a < w_b$, equation (4.17) can be approximated as

$$K_y = \frac{6EI_a}{l_a^3 12r_2^2} = \frac{Et w_a^3}{24l_a l_b^2} \quad (4.18)$$

Equations (4.15) and (4.17) show that K_x and K_y are relevant to the flexure width w_a , w_b , and also relative to the folded length l_0 , l_b and l_a . Equations (4.16) and (4.18) also show that the stiffness of the folded flexure is much smaller than that of straight flexure and are more suitable for the application of large displacement resonators. However, the exact influence of the flexure parameter on the stiffness cannot be seen directly from the complicated equations. In the next section, we will study the influence of the parameters by using finite element method.

4.2.3 Asymmetric-quad flexure

Different from the open serpentine structure of the folded flexure, the asymmetric-quad flexure has a closed-loop structure. The flexure also looks like two L-shape plates that are interlapped together, resulting in the asymmetric quadrate-loop structure with two extended ends at the corners. Figure 4.5 shows the schematic of the asymmetric-quad plate flexure.

Figure 4.5 Schematic diagram of the asymmetric-quad flexure

To analyze the deflection of the asymmetric-quad flexure, the whole flexure can be treated as a multi-segmented structure: three vertical and two traverse beam units. Each beam segment is treated as a one-end fixed and one-end free beam structure. Figure 4.6 shows the basic flexure segment and the deflection diagram, with one-end fully constrained and the other-end constrained in the rotational direction, i.e., $\theta_6=0$.

Figure 4.6 Basic segment and deflection diagram of the asymmetric-quad flexure.

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The total displacement at the free end of the flexure unit, u_6 , v_6 , θ_6 can be obtained by combining the displacement of each segment with solving the complicated force-displacement equations. Here, only the displacement coming from bending will be considered in the analysis; deformation from shear, beam elongation, and beam shortening will be ignored. With the analysis of the force and moment equilibrium of each segment step by step, the equations of the free-end segment can be obtained. Considering the force F_x , F_y are given, and the moment M , displacement at the end u_6 , v_6 as variables while rotation is constrained, the equilibrium equations at the free end can be achieved as (see Appendix B for the derivation process):

$$F_x \left(\frac{l_0^3}{3EI_a} + \frac{l_0^2 l_a}{2EI_a} + \frac{l_0 l_b (l_a + l_0)}{EI_b} \right) - F_y \frac{l_0 l_b^2}{2EI_b} - M \left(\frac{l_0^2}{2EI_a} + \frac{l_0 l_b}{EI_b} \right) - F_{x4} \left(\frac{2l_0^3}{3EI_a} + \frac{l_0^2 l_b}{EI_b} \right) + F_{y4} \left(\frac{l_0^2 l_b}{2EI_a} + \frac{l_0 l_b^2}{2EI_b} \right) - M_4 \left(\frac{l_0^2}{EI_a} + \frac{l_0 l_b}{EI_b} \right) = 0 \quad (4.19)$$

$$F_x \frac{l_b^2 (l_0 + l_a)}{2EI_b} - F_y \frac{l_b^3}{3EI_b} - M \frac{l_b^2}{2EI_b} - F_{x4} \left(\frac{l_0^2 l_b}{2EI_a} + \frac{l_0 l_b^2}{2EI_b} \right) + F_{y4} \left(\frac{l_0 l_b^2}{EI_a} + \frac{2l_b^3}{3EI_b} \right) - M_4 \left(\frac{l_0 l_b}{EI_a} + \frac{l_b^2}{EI_b} \right) = 0 \quad (4.20)$$

$$F_x \left(\frac{l_0^2}{2EI_a} + \frac{l_0 l_a}{EI_a} + \frac{l_b (l_a + l_0)}{EI_b} \right) - F_y \frac{l_b^2}{2EI_b} - M \left(\frac{l_0}{EI_a} + \frac{l_b}{EI_b} \right) - F_{x4} \left(\frac{l_0^2}{EI_a} + \frac{l_0 l_b}{EI_b} \right) + F_{y4} \left(\frac{l_0 l_b}{EI_a} + \frac{l_b^2}{EI_b} \right) - M_4 \left(\frac{2l_0}{EI_a} + \frac{2l_b}{EI_b} \right) = 0 \quad (4.21)$$

$$F_x \left(\frac{7l_0^3}{6EI_a} + \frac{l_0 l_a^2}{2EI_a} + \frac{l_a l_0 (l_a + l_0)}{EI_a} + \frac{3l_a^2 (l_a + l_0)}{2EI_a} \right) - F_y \left(\frac{3l_a^2 l_b}{2EI_a} + \frac{l_b l_a l_0}{EI_a} \right) - M \left(\frac{2l_a^2}{EI_a} + \frac{l_a l_0}{EI_a} \right) + F_{x4} \left(\frac{l_0^3}{3EI_a} + \frac{l_0^2 l_a}{2EI_a} \right) - F_{y4} \left(\frac{l_a^2 l_b}{2EI_a} + \frac{l_0 l_a l_b}{EI_a} + \frac{l_a l_b^2}{2EI_b} \right) + M_4 \left(\frac{l_0^2}{2EI_a} + \frac{l_a l_0}{EI_a} + \frac{l_a l_b}{EI_b} \right) = u_6 \quad (4.22)$$

$$F_x \left(\frac{l_a^2 l_b}{2EI_a} + \frac{l_a l_b (l_a + l_0)}{EI_a} \right) - F_y \frac{l_a l_b^2}{EI_a} - M \frac{l_a l_b}{EI_a} + F_{x4} \frac{l_0^2 l_b}{2EI_a} - F_{y4} \left(\frac{l_b^3}{3EI_b} + \frac{l_0 l_b^2}{EI_a} \right) + M_4 \left(\frac{l_b^2}{EI_b} + \frac{l_0 l_b}{EI_a} \right) = v_6 \quad (4.23)$$

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$$\begin{aligned}
& F_x \left(\frac{l_0^2}{EI_a} + \frac{l_a(l_a + l_0)}{EI_a} \right) - F_y \frac{l_a l_b}{EI_b} - M \frac{2l_a}{EI_a} + F_{x4} \frac{l_0^2}{2EI_a} - F_{y4} \left(\frac{l_b^2}{2EI_b} + \frac{l_0 l_b}{EI_a} \right) \\
& + M_4 \left(\frac{l_0}{EI_a} + \frac{l_b}{EI_b} \right) = 0
\end{aligned} \quad (4.24)$$

By solving equations (4.19) – (4.24), the displacement u_6 , v_6 and moment M at the end point 6 of the asymmetric-quad flexure can be obtained. Since there are many independent parameters such as w_a , w_b , l_0 , l_b , l_a and loadings, the mathematical computing software, Maple, was used for the symbolic equation derivation. The displacement solutions were presented in Appendix B for their lengthy outlook. However, they can hardly be further simplified without pre-defined approximations as most of the parameters are independent.

The complicated solutions of u_6 , v_6 can be simplified by assuming that $r_1 \ll 1$, $r_2 \ll 1$, where $r_1 = l_0/l_a$, $r_2 = l_b/l_a$, $\alpha = I_a/I_b = (w_a/w_b)^3$. Neglecting higher order terms in the polynomial solutions, u_6 and v_6 could be simplified as

$$u_6 = l_a^3 \left(F_x (3r_1 + 36r_1^2) + F_y (18r_1 r_2 \alpha + 6r_2^2 \alpha - 72r_1^2 r_2) \right) / (24EI_a (r_2 \alpha + 3r_1)) \quad (4.25)$$

$$v_6 = l_b l_a^2 \left(F_x (18r_1^2 + 18r_1^3 + 6r_1 r_2 \alpha) + F_y (r_2^2 \alpha^2 - 36r_1^2 r_2) \right) / (12EI_a (3r_1 + r_2 \alpha)) \quad (4.26)$$

The stiffness of the flexure can be calculated by

$$K_x = \frac{24EI_a}{l_a^3} \frac{r_2 \alpha + 3r_1}{(36r_1^2 + 3r_1) + \frac{F_y}{F_x} (18r_1 r_2 \alpha + 6r_2^2 \alpha - 72r_1^2 r_2)} \quad (4.27)$$

$$K_y = \frac{12EI_a}{l_b l_a^2} \frac{3r_1 + r_2 \alpha}{-\frac{F_x}{F_y} (18r_1^2 + 6r_1 r_2 \alpha + 18r_1^3) + (36r_1^2 r_2 - r_2^2 \alpha^2)} \quad (4.28)$$

In the limiting condition of $F_y = 0$ and $w_a < w_b$, K_x can be approximated by

$$K_x \approx \frac{24EI_a}{l_a^3} = \frac{2Et w_a^3}{l_a^3} \quad (4.29)$$

Similarly, if $F_x = 0$, equation (4.28) can be simplified

$$K_y = \frac{12EI_a}{l_b l_a^2} \frac{3r_1}{36r_1^2 r_2} = \frac{Et w_a^3}{12 l_b^2 l_0} \tag{4.30}$$

Compared to the straight flexure and folded-plate flexure, the asymmetric-quad flexure is stiffer in the x - direction, which is attributed to the overlapped close structure. However, the asymmetric-quad flexure has smaller stiffness in the y - direction than that of the straight flexure, on comparing the equations (4.6) and (4.30). Thus, the stiffness ratio of asymmetric-quad flexure is smaller than that of the straight flexure, although it has higher stiffness in the x - direction.

Later, finite element method will be used to study the exact deflections and stiffness of the asymmetric-quad flexure and to study the variation of stiffness with different flexure dimensions.

4.2.4 Symmetric-quad flexure

Figure 4.7 Free-body diagram of the symmetric-quad flexure

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Similar to the asymmetric-quad flexure, the symmetric quadrate-frame flexure is further studied. The only difference in the asymmetric-quad flexure is that the extended ends are located in the middle of the quad-loop such that the flexure is symmetrical. The symmetric-quad flexure is considered to be stiffer than the straight-beam flexure and have a higher stiffness ratio. Because of the symmetric quadrate-loop structure, the symmetric-quad flexure is more durable in sustaining both horizontal and vertical loads than the straight flexure. Also, it has a higher vertical stiffness than that of the asymmetric-quad flexure.

Figure 4.7 shows the deformation diagram of the symmetric-quad flexure. Similarly, only the bending deformation of the flexure is considered in the analytical calculation. To analyze the deflection of the flexure, each flexure unit will be divided into several beam segments, with each beam segment being treated as one-end fixed and one-end free beam. The force and equilibrium equations of the flexure under given forces of F_x , F_y , and unknown moment M are derived as follows (see Appendix C for the derivation process):

$$F_x \cdot \left(\frac{l_a l_0^2}{2EI_a} + \frac{l_0^3}{3EI_a} + \frac{l_0^2 l_b + l_0 l_a l_b}{2EI_b} \right) + F_y \cdot \left(\frac{l_0 l_b^2}{8EI_b} + \frac{l_0^2 l_b}{4EI_a} \right) - M \cdot \left(\frac{l_0 l_b}{2EI_b} + \frac{l_0^2}{2EI_a} \right) - F_{x7} \cdot \left(\frac{l_0^2 l_b}{EI_b} + \frac{2l_0^3}{3EI_a} \right) - F_{y7} \cdot \frac{l_0 l_b^2}{8EI_b} - M_7 \cdot \left(\frac{l_0 l_b}{EI_b} + \frac{l_0^2}{EI_a} \right) = 0 \quad (4.31)$$

$$F_x \cdot \left(-\frac{l_0^2 l_b}{4EI_a} - \frac{l_0 l_b^2}{8EI_b} - \frac{l_0 l_a l_b}{2EI_a} \right) + F_y \cdot \left(-\frac{l_b^3}{12EI_b} - \frac{l_b^2 l_0}{4EI_a} \right) + M \cdot \frac{l_0 l_b}{2EI_a} + F_{y7} \cdot \left(\frac{l_b^2 l_0}{2EI_a} + \frac{3l_b^3}{16EI_b} \right) - M_7 \cdot \frac{l_b^2}{4EI_b} = 0 \quad (4.32)$$

$$F_x \cdot \left(\frac{l_a l_0}{EI_a} + \frac{l_0^2}{2EI_a} + \frac{l_0 l_b + l_a l_b}{2EI_b} \right) + F_y \cdot \frac{l_0 l_b}{2EI_a} - M \cdot \left(\frac{l_b}{EI_b} + \frac{l_0}{EI_a} \right) - F_{x7} \cdot \left(\frac{l_0 l_b}{EI_b} + \frac{l_0^2}{EI_a} \right) + F_{y7} \cdot \frac{l_b^2}{8EI_b} - M_7 \cdot \left(\frac{2l_b}{EI_b} + \frac{2l_0}{EI_a} \right) = 0 \quad (4.33)$$

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$$\begin{aligned}
& F_x \cdot \left(\frac{l_0^3}{3EI_a} + \frac{8l_a^3}{3EI_a} + \frac{2l_a l_0^2}{EI_a} + \frac{4l_0 l_a^2}{EI_a} + \frac{l_b(l_0 + l_a)^2}{2EI_b} \right) + F_y \cdot \left(\frac{l_0 l_b^2}{8EI_b} + \frac{l_0^2 l_b}{4EI_a} + \frac{l_0 l_a l_b}{2EI_a} \right) \\
& + M \cdot \left(-\frac{l_0^2}{2EI_a} - \frac{2l_a^2}{EI_a} - \frac{2l_0 l_a}{EI_a} - \frac{l_0 l_b + 2l_a l_b}{2EI_b} \right) + F_{x7} \cdot \left(-\frac{l_0^3}{3EI_a} - \frac{l_0^2 l_b}{2EI_b} - \frac{l_0 l_a l_b}{2EI_b} - \frac{l_0^2 l_a}{2EI_a} \right) \\
& + F_{y7} \cdot \left(-\frac{l_0^2 l_b}{4EI_a} - \frac{l_0 l_a l_b}{2EI_a} - \frac{l_b^2 l_a}{8EI_b} - \frac{l_b^2 l_0}{4EI_b} \right) + M_7 \cdot \left(-\frac{l_a l_0}{EI_a} - \frac{l_0^2}{2EI_a} - \frac{l_0 l_b + 2l_a l_b}{2EI_b} \right) = u_8
\end{aligned} \tag{4.34}$$

$$\begin{aligned}
& F_x \cdot \left(-\frac{l_b l_0^2}{4EI_a} - \frac{l_0 l_b^2}{8EI_b} - \frac{l_0 l_a l_b}{2EI_a} \right) + F_y \cdot \left(-\frac{l_0 l_b^2}{4EI_a} - \frac{l_b^3}{12EI_b} \right) + M \cdot \frac{l_0 l_b}{2EI_a} \\
& + F_{x7} \cdot \left(\frac{l_0 l_b^2}{8EI_b} + \frac{l_0^2 l_b}{4EI_a} \right) + F_{y7} \cdot \left(\frac{l_0 l_b^2}{4EI_a} + \frac{5l_b^3}{48EI_b} \right) + M_7 \cdot \frac{l_0 l_b}{2EI_a} = v_8
\end{aligned} \tag{4.35}$$

$$\begin{aligned}
& F_x \cdot \left(\frac{2l_a l_0 + 2l_a^2}{EI_a} + \frac{l_0^2}{2EI_a} + \frac{l_0 l_b + l_a l_b}{2EI_b} \right) + F_y \cdot \frac{l_0 l_b}{2EI_a} - M \cdot \left(\frac{l_b}{EI_b} + \frac{l_0}{EI_a} + \frac{2l_a}{EI_a} \right) \\
& - F_{x7} \cdot \left(\frac{l_0 l_b}{2EI_b} + \frac{l_0^2}{2EI_a} \right) - F_{y7} \cdot \left(\frac{l_0 l_b}{2EI_a} + \frac{l_b^2}{8EI_b} \right) - M_7 \cdot \left(\frac{l_b}{EI_b} + \frac{l_0}{EI_a} \right) = 0
\end{aligned} \tag{4.36}$$

By solving the six equations (4.31) ~ (4.36), the deflection of the symmetric-quad flexure u_8 and v_8 can be derived and the results were presented in Appendix C for their lengthy outlook.

The displacement solutions of u_8 , v_8 can also be simplified by assuming that $r_1 < 1$, $r_2 < 1$, where $r_1 = l_0/l_a$, $r_2 = l_b/l_a$, $\alpha = I_a/I_b = (w_a/w_b)^3$. Thus the high-order items in the polynomial solutions can be ignored. After simplification, the solutions u_8 , v_8 are simplified to

$$u_8 = l_a^3 \left(F_x (12r_2\alpha + 64r_1 + 616r_1r_2\alpha) + F_y (24r_1r_2^2\alpha - 6r_2^3\alpha^2) \right) / (12EI_a (560r_1r_2\alpha + 128r_1 + 24r_2\alpha)) \tag{4.37}$$

$$v_8 = -l_b^2 l_a \left(F_x (288r_1 + 1152r_1^2) + F_y (128r_2^2\alpha + 1056r_1r_2) \right) / (72EI_b (560r_1r_2\alpha + 128r_1 + 24r_2\alpha)) \tag{4.38}$$

Hence, the stiffness of the flexure K_x and K_y can be found,

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$$K_x = \frac{12EI_a}{l_a^3} \frac{560r_1r_2\alpha + 128r_1 + 24r_2\alpha}{(12r_1\alpha + 64r_1 + 616r_1r_2\alpha) + \frac{F_y}{F_x} (24r_1r_2^2\alpha - 6r_2^3\alpha^2)} \quad (4.39)$$

$$K_y = \frac{72EI_b}{l_b^2 l_a} \frac{560r_1r_2\alpha + 128r_1 + 24r_2\alpha}{\frac{F_x}{F_y} (288r_1 + 1152r_1^2) + (128r_2^2\alpha + 1056r_1r_2)} \quad (4.40)$$

If $F_y=0$ and $w_a < w_b$, the stiffness K_x can be further simplified and approximated as

$$K_x = \frac{12EI_a}{l_a^3} \frac{128r_1}{64r_1} = \frac{2Et w_a^3}{l_a^3} \quad (4.41)$$

If $F_x=0$ and $w_a < w_b$, the stiffness K_y can be simplified as

$$K_y = \frac{72EI_b}{l_b^2 l_a} \frac{128r_1}{1056r_1r_2} = \frac{8Et w_b^3}{11l_b^3} \quad (4.42)$$

On comparing the stiffness K_x equations (4.5), (4.29) and (4.41), we found

$$K_x (\text{symmetric-quad}) \approx K_x (\text{asymmetric-quad}); K_x (\text{symmetric-quad}) > K_x (\text{straight})$$

Similarly, on comparing the K_y equations (4.6), (4.30) and (4.42), we found

$$K_y (\text{symmetric-quad}) > K_y (\text{asymmetric-quad}); K_y (\text{symmetric-quad}) > K_y (\text{straight})$$

The approximated stiffness ratio of the symmetric-quad flexure is

$$\frac{K_y}{K_x} = \frac{8Et w_b^3}{11l_b^3} \frac{l_a^3}{2Et w_a^3} = \frac{8}{22} \frac{l_a^3}{l_b^3} \frac{w_b^3}{w_a^3} > \frac{l_a^2}{w_a^2} \quad (4.43)$$

Compared to equation (4.7), the stiffness ratio of the symmetric-quad flexure is larger than that of the straight flexure. In addition, the symmetric-quad flexure has a higher stiffness in the x - direction. Therefore, the symmetric-quad flexure has a larger stiffness K_x than that of the straight flexure, which is similar to the asymmetric-quad flexure; furthermore, it has larger stiffness ratio K_y/K_x than that of the straight flexure, which is much higher than that of the asymmetric-quad flexure. The symmetric-quad flexure is supposed to be more suitable for the applications of actuators with high

stiffness and stiffness ratio. For instance, it can be used for the SOG microactuators to provide high resonant frequency and high stiffness ratio.

Since the analytical stiffness solutions of the different types of flexures are very complicated and the approximated equations are not accurate enough to describe the influence of the flexure parameters on their stiffness. The finite element method (FEM) is used to analyze the deflection of the flexures to study their stiffness effect and stiffness ratios. Different parameters of the flexures will be investigated and their influences on the stiffness of the flexures will be established.

4.3 Finite Element Analysis for the Flexure Design

4.3.1 Modeling of the flexures

In this study, finite element analysis (FEA) is performed by using commercially available software ANSYS. As the flexures are designed to generate small deflection of the microactuator (flexures), a static linear analysis can be used to study the strain and to obtain the stiffness of the flexures. Modal analysis is then used to study the resonant frequency of the flexures. FEA provides us with explicit results that aid the comparison and selection of a suitable flexure for the microactuator. Also the suitable dimensions of the flexures can be compared and verified.

A two-dimensional beam element BEAM3 was selected for the finite element model of the thick-plate flexures for easy and simple meshing, since the flexures have uniform shape and thickness along the z - direction. BEAM3 is a uniaxial element equipped with tension, compression, and bending capabilities. The element has three degrees of freedom at each node: translations in the nodal x - and y - directions and rotation about the nodal z -axis. Another optional element type for the thick and bulky flexures is the 3D structural solid element SOLID92. The 3D 10-nodes tetrahedral

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solid element also has three degrees of freedom per node, but it takes more computational time due to the large number of nodes and elements. In fact, the flexure models with BEAM3 and SOLID92 elements showed very little difference in the deflection results. Figure 4.8 shows the 2D model of the symmetric-quad flexure.

Figure 4.8 Beam3 2D model of the symmetric-quad flexure

The material used for this micro-flexure device is bulk silicon. The material properties used here are exactly the same as in Chapter 3: Young's Modulus $E = 130$ GPa, density $\rho = 2330 \text{ Kg/m}^3$, and Poisson ratio $\nu = 0.2$.

After the finite element model has been meshed, the constraints and loads are applied to the flexures. The flexure is constrained at one end and the other end is subjected to the electrostatic force and a weight of the functional element (head/slider), which is attached to the flexures. We take the value of $10 \text{ } \mu\text{N}$ for the electrostatic force, and suppose the weight of the magnetic head /femto-slider

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(moving mass) to be 1.5 milligram, which will be uniformly exerted to three (for straight flexure) or four (for other types) flexures.

Table 4.1 lists the static analysis results of the straight flexure and symmetric-quad flexure with 2D and 3D element models. The FEA results with 2D BEAM3 models showed very small deviation from those with 3D SOLID92 models. Therefore, BEAM3 element is selected for the finite element analysis of the different types of flexures, given the 2D model accuracy and also for the computing time. Here, $l_a=380\text{ }\mu\text{m}$, $l_0=20\text{ }\mu\text{m}$, $l_b=20\text{ }\mu\text{m}$, $w_a=5\text{ }\mu\text{m}$, $w_b=10\text{ }\mu\text{m}$, $F_x=10\text{ }\mu\text{N}$.

Table 4.1 Comparison of the 2D model, 3D model and analytical results of the flexures' x - deflection

		Straight flexure	Symmetric-quad flexure
Analytical result u_x		0.45592 μm	0.29058 μm
BEAM3 2D model	u_x result	0.45592 μm	0.29042 μm
	Deviation	0	0.055%
SOLID92 3D model	u_x result	0.45488 μm	0.25886 μm
	Deviation	0.228%	10.92%

4.3.2 FEA results and discussion

After modeling and solving the flexures models, the deflection and stiffness of the flexures can be obtained. With the FEM, the flexures with the required high stiffness and stiffness ratio can be found. Also, the influence of the flexure parameters on the flexure's stiffness can also be studied.

According to the analytical equations of the flexures, the variables that can influence the flexure stiffness are the flexure width w_a , transverse flexure width (joint height) w_b , transverse flexure length (flexure gap) l_b , overlapped length l_a and extended flexure length l_0 . Here l_0 and l_a are related by the equation of $l_a + 2l_0 = L$,

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where L is the flexure total length and dependent on the design of the microactuators. Here, we evaluate the flexure dimension for the actual application of the electrostatic microactuators. Thus, the range of the flexure parameters can be pre-defined as:

$3\text{ }\mu\text{m} < w_a < 7\text{ }\mu\text{m}$, $5\text{ }\mu\text{m} < w_b < 20\text{ }\mu\text{m}$, $5\text{ }\mu\text{m} < l_b < 30\text{ }\mu\text{m}$, $100\text{ }\mu\text{m} < l_a < 400\text{ }\mu\text{m}$.

Based on the analytical equations, the total flexure length L certainly has significant influence on the flexure stiffness. The stiffness will decrease with the increase of the flexure length. However, since the size of the microactuators has already been pre-defined, the total length of the flexures is fixed at $420\text{ }\mu\text{m}$. In the next section, the different variables that affect the flexure’s stiffness, such as w_a , w_b , l_a and l_b , would be studied by FEA.

• Influence of flexure width w_a

Table 4.2 Influence of the flexure width w_a on the flexure’s stiffness.

		$w_a=3\text{ }\mu\text{m}$	$w_a=4\text{ }\mu\text{m}$	$w_a=5\text{ }\mu\text{m}$	$w_a=6\text{ }\mu\text{m}$	$w_a=7\text{ }\mu\text{m}$
Straight flexure	$u(F_x \text{ only}), \mu\text{m}$	2.1108	0.8905	0.4559	0.2638	0.1662
	$K_x, \text{N/m}$	4.74	11.23	21.93	37.91	60.17
	$v(F_y \text{ only}), \text{nm}$	0.1077	0.08077	0.0646	0.05385	0.04615
	K_y/K_x	19598.88	11025.13	7057.58	4898.79	3601.30
Folded flexure	$u(F_x \text{ only}), \mu\text{m}$	6.031	2.5443	1.3031	0.7539	0.4748
	$K_x, \text{N/m}$	1.66	3.93	7.67	13.26	21.06
	$v(F_y \text{ only}), \text{nm}$	28.58	12.16	6.29	3.689	2.359
	K_y/K_x	211.02	209.24	207.17	204.36	201.27
Asymmetric-quad flexure	$u(F_x \text{ only}), \mu\text{m}$	1.2658	0.5340	0.27341	0.1582	0.09964
	$K_x, \text{N/m}$	7.90	18.73	36.57	63.21	100.36
	$v(F_y \text{ only}), \text{nm}$	0.283	0.138	0.0828	0.0566	0.0421
	K_y/K_x	4472.79	3869.56	3302.05	2796.63	2366.75
Symmetric-quad flexure	$u(F_x \text{ only}), \mu\text{m}$	1.2675	0.5347	0.27377	0.1584	0.09977
	$K_x, \text{N/m}$	7.89	18.70	36.53	63.13	100.23
	$v(F_y \text{ only}), \text{nm}$	0.07039	0.04821	0.0368	0.02995	0.02528
	K_y/K_x	18006.82	11091.06	7439.41	5288.81	3946.59

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Firstly, the influence of flexure width w_a on the flexure stiffness is studied. The flexures that were investigated are straight flexure, folded flexure, asymmetric-quad flexure and symmetric-quad flexure. For ease of study, the transverse flexure width is set to be equal to vertical flexure width, i.e., $w_b=w_a$. The flexures deflection at the end point in terms of the different flexure width has been displayed in Table 4.2, with the initial conditions: $L=420\text{ }\mu\text{m}$, $l_a=400\text{ }\mu\text{m}$, $l_b=10\text{ }\mu\text{m}$, $w_b=w_a$, $F_x=10\text{ }\mu\text{N}$ or $F_y=10\text{ }\mu\text{N}$.

Figure 4.9 Influence of the flexure width on the flexure stiffness

Figure 4.10 Influence of the flexure width on the stiffness ratio of the flexure

Figure 4.9 plots the comparison graph of the stiffness K_x of the four different types of flexures with variation of the flexure width w_d . It is observed that the larger flexure width w_d obtains the higher stiffness K_x of the flexure. Furthermore, it is seen that the symmetric-quad flexure and asymmetric-quad flexure almost have the same stiffness K_x due to their similar structure and they present the largest stiffness among the four flexures; the straight flexure presents moderate stiffness, and the folded flexure has the smallest stiffness under the same conditions and flexure width.

Since the natural frequency (or resonant frequency) of the flexure is significantly dependent on the stiffness, a higher stiffness of the flexures will impose a higher resonant frequency. Therefore, the symmetric-quad flexure and asymmetric-quad flexure will present the largest resonant frequency for the actuators.

Figure 4.10 shows the stiffness ratios K_y/K_x of the different types of flexures with variation of flexure width w_d . The comparison of the results shows that the symmetric-quad flexure has the largest stiffness ratio K_y/K_x among all. The straight flexure presents a smaller stiffness ratio while the asymmetric-quad flexure and folded flexure have much smaller stiffness ratio under the same conditions. The stiffness ratio is an important parameter to describe the stiffness preference of the flexures in particular applications. For instance, some applications require the flexure to be only flexible in the operational direction, but very stiff, exhibiting small deflection in other directions. This objective can be described as the large stiffness ratio. The FEA results show that the symmetric-quad flexure presents the smallest deflection in the vertical direction while providing the same deflection in operational direction as the other types of flexures.

- **Influence of transverse flexure width w_b**

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Except for the straight flexure, the other three types of flexures have both transverse and vertical parts. The transverse flexure width w_b also influences the flexure stiffness. Table 4.3 lists the FEM result of the influence of w_b on the flexure's stiffness, which consist of the folded flexure, asymmetric-quad flexure and symmetric-quad flexure, with initial conditions: $L=420\text{ }\mu\text{m}$, $l_a=400\text{ }\mu\text{m}$, $l_b=10\text{ }\mu\text{m}$, $w_a=5\text{ }\mu\text{m}$, $F_x=10\text{ }\mu\text{N}$ or $F_y=10\text{ }\mu\text{N}$.

Table 4.3 Influence of the transverse flexure width w_b on the flexure's stiffness

		$w_b = 5\text{ }\mu\text{m}$	$w_b = 8\text{ }\mu\text{m}$	$w_b = 10\text{ }\mu\text{m}$	$w_b = 15\text{ }\mu\text{m}$	$w_b = 20\text{ }\mu\text{m}$
Folded flexure	$u(F_x\text{ only}),\text{ }\mu\text{m}$	1.3031	1.2584	1.2514	1.2462	1.2449
	$K_x,\text{ N/m}$	7.67	7.95	7.99	8.02	8.03
	$v(F_y\text{ only}),\text{ nm}$	6.29	6.257	6.251	6.247	6.246
	K_y/K_x	207.17	201.12	200.19	199.49	199.31
Asymmetric-quad flexure	$u(F_x\text{ only}),\text{ }\mu\text{m}$	0.27341	0.26259	0.26084	0.2596	0.2592
	$K_x,\text{ N/m}$	36.58	38.08	38.34	38.52	38.58
	$v(F_y\text{ only}),\text{ nm}$	0.08285	0.07376	0.0723	0.07123	0.07096
	K_y/K_x	3300.06	3560.06	3607.75	3644.53	3652.76
Symmetric-quad flexure	$u(F_x\text{ only}),\text{ }\mu\text{m}$	0.27377	0.2626	0.26085	0.25955	0.2592
	$K_x,\text{ N/m}$	36.53	38.08	38.34	38.53	38.58
	$v(F_y\text{ only}),\text{ nm}$	0.0368	0.03459	0.03423	0.03396	0.03389
	K_y/K_x	7439.41	7591.78	7620.51	7642.82	7648.27

Figure 4.11 Influence of the transverse flexure width on the flexure stiffness

Figure 4.12 Influence of the transverse flexure width on flexure stiffness ratio

Figures 4.11 and 4.12 show the influence of the transverse flexure width w_b on the flexure stiffness K_x and stiffness ratio K_y/K_x for folded flexure, asymmetric-quad flexure and symmetric-quad flexure.

The FEA results show that the variation of the transverse flexure width w_b has minimum influence on the flexure stiffness and stiffness ratio. With the increase of the transverse flexure width, the stiffness K_x and stiffness ratio K_y/K_x will slightly increase. Furthermore, the stiffness tends to be steady when the transverse flexure width w_b reaches 10 μm or above. Therefore, the optimal transverse flexure width w_b of the flexures should be 10 μm . In addition, the sym-quad flexure and asymmetric-quad flexure have the largest stiffness K_x . Furthermore, the symmetric-quad flexure is stiffer as it presents a higher stiffness ratio than that of the asymmetric-quad flexure.

• **Influence of overlapped length l_a**

The flexure total length L and overlapped length l_a also have influence on the flexure stiffness. The total length L depends on the specifications of the actuators, and here we set $L=420 \mu\text{m}$ in the FEA. Since the overlapped length l_a is part of the flexure

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and has the relationship of $l_a + 2l_0 = L$, we can study the influence of the overlapped length on the flexures stiffness and find the optimized value to obtain the large flexure stiffness. Here, $L=420\text{ }\mu\text{m}$, $l_b=10\text{ }\mu\text{m}$, $w_b=2w_a=10\text{ }\mu\text{m}$, $F_x=10\text{ }\mu\text{N}$ or $F_y=10\text{ }\mu\text{N}$.

Table 4.4 Influence of the flexure width w_a on the flexure’s stiffness.

		$l_a=100\text{ }\mu\text{m}$	$l_a=150\text{ }\mu\text{m}$	$l_a=200\text{ }\mu\text{m}$	$l_a=250\text{ }\mu\text{m}$	$l_a=300\text{ }\mu\text{m}$	$l_a=350\text{ }\mu\text{m}$	$l_a=400\text{ }\mu\text{m}$
Folded flexure	$u(F_x\text{ only}),\text{ }\mu\text{m}$	0.4689	0.4987	0.5562	0.6511	0.7924	0.9895	1.2514
	$K_x, \text{N/m}$	21.33	20.05	17.98	15.36	12.62	10.11	7.99
	$v(F_y\text{ only}),\text{ nm}$	3.943	4.328	4.711	5.095	5.480	5.866	6.251
	K_y/K_x	118.92	115.23	118.06	127.79	144.59	168.68	200.19
Asymmetric-quad flexure	$u(F_x\text{ only}),\text{ }\mu\text{m}$	0.4529	0.4458	0.4318	0.4086	0.3739	0.3254	0.2608
	$K_x, \text{N/m}$	22.08	22.43	23.16	24.47	26.75	30.73	38.34
	$v(F_y\text{ only}),\text{ nm}$	0.6492	0.5531	0.4569	0.3608	0.2646	0.1685	0.0723
	K_y/K_x	697.6	806.0	945.1	1132.5	1413.1	1931.2	3607.2
Symmetric-quad flexure	$u(F_x\text{ only}),\text{ }\mu\text{m}$	0.4531	0.4458	0.4317	0.4086	0.3739	0.3254	0.2609
	$K_x, \text{N/m}$	22.07	22.43	23.16	24.47	26.75	30.73	38.33
	$v(F_y\text{ only}),\text{ nm}$	0.0573	0.0535	0.0496	0.0458	0.0419	0.0381	0.0342
	K_y/K_x	7907.5	8338.9	8701.9	8927.2	8919.4	8545.2	7620.5

Figure 4.13 Influence of the overlapped flexure length l_a on flexure stiffness K_x

In Figure 4.13, the influence of the overlapped length l_a on the flexures stiffness K_x is shown. As it can be seen from the FEA results, the stiffness K_x of the asymmetric-quad flexure and symmetric-quad flexure increases with the increase of the overlapped flexure length l_b , while the folded flexure has an inverse relationship. Thus, a larger overlapped flexure length l_a is preferred for the large stiffness K_x of the asymmetric-quad flexure and symmetric-quad flexure.

Figure 4.14 Influence of the overlapped flexure length on flexure stiffness ratio K_y/K_x

Figure 4.14 shows the influence of the overlapped flexure length on the stiffness ratio. It is found that the symmetric-quad flexure has the largest stiffness ratio K_y/K_x among the three types of flexures, and the stiffness ratio is slightly influenced by the variation of the overlapped length l_a . However, the asymmetric-quad flexure is significantly affected and presents larger stiffness ratio with larger l_a .

Therefore, consideration both the stiffness K_x and stiffness ratio K_y/K_x , the symmetric-quad flexure is preferred to have the overlapped length l_a of 350~400 μm in the case of the flexure total length of 420 μm ; the asymmetric-quad flexure is also preferred to have the overlapped length of 400 μm to obtain the high stiffness effect.

- **Influence of transverse flexure length l_b**

As it can be seen from the analytical equations of the flexure’s stiffness, the stiffness is also affected by the transverse flexure length l_b . Table 4.5 displays the FEA result of the deflection and stiffness of the three types of flexures, i.e., folded flexure, asymmetric-quad flexure and symmetric-quad flexure, with variation of the transverse flexure length l_b . The initial conditions of the flexures are: $L=420\text{ }\mu\text{m}$, $l_a=400\text{ }\mu\text{m}$, $w_b=2w_a=10\text{ }\mu\text{m}$, $F_x=10\text{ }\mu\text{N}$ or $F_y=10\text{ }\mu\text{N}$.

Table 4.5 Influence of the transverse flexure length l_b on the flexure’s stiffness

		$l_b=5\text{ }\mu\text{m}$	$l_b=10\text{ }\mu\text{m}$	$l_b=15\text{ }\mu\text{m}$	$l_b=20\text{ }\mu\text{m}$	$l_b=30\text{ }\mu\text{m}$	$l_b=40\text{ }\mu\text{m}$
Folded flexure	$u(F_x\text{ only}),\text{ }\mu\text{m}$	1.2473	1.2514	1.2547	1.2584	1.2658	1.2732
	$K_x,\text{ N/m}$	8.02	7.99	7.97	7.95	7.90	7.85
	$v(F_y\text{ only}),\text{ nm}$	1.702	6.251	13.83	24.46	0.054.85	97.47
	K_y/K_x	732.84	200.19	90.72	51.45	23.08	13.06
Asymmetric-quad flexure	$u(F_x\text{ only}),\text{ }\mu\text{m}$	0.2599	0.2608	0.2618	0.2627	0.2645	0.2663
	$K_x,\text{ N/m}$	38.48	38.34	38.20	38.07	37.81	37.55
	$v(F_y\text{ only}),\text{ nm}$	0.04327	0.0723	0.1221	0.1938	0.4074	0.6075
	K_y/K_x	6006.47	3607.19	2144.14	1355.52	649.24	438.35
Symmetric-quad flexure	$u(F_x\text{ only}),\text{ }\mu\text{m}$	0.2599	0.26085	0.2618	0.2627	0.2646	0.2664
	$K_x,\text{ N/m}$	38.48	38.34	38.20	38.07	37.79	37.54
	$v(F_y\text{ only}),\text{ nm}$	0.03389	0.03423	0.03514	0.03691	0.04416	0.05823
	K_y/K_x	7668.93	7620.51	7450.19	7117.31	5991.85	4574.96

Figure 4.15 shows the FEM results of the stiffness K_x with variation of the transverse flexure length l_b . The results indicate that the length of the transverse flexure has very small influence on the flexure stiffness K_x , and smaller l_b presents slightly higher K_x . Thus, the transverse flexure length l_b is preferred to be around 5~10 μm . Figure 4.16 shows the influence of the transverse flexure length on the flexure stiffness ratio K_y/K_x . It is seen that the stiffness ratio of all the three types of flexures decreases when the transverse flexure length increases. Hereby, the smaller l_b of less than 10 μm is preferred to the larger stiffness ratio.

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Considering the fabrication limit of high-aspect-ratio structures, the width of transverse flexure l_b of symmetric-quad and asymmetric-quad flexure are designed to be $10\text{ }\mu\text{m}$ in order to obtain the high stiffness K_x and high stiffness ratio K_y/K_x .

Figure 4.15 Influence of the transverse flexure length l_b on flexure stiffness K_x

Figure 4.16 Influence of the transverse flexure length l_b on stiffness ratio K_y/K_x

To conclude, with the FEM, the influences of the flexure parameters on the four types of flexure stiffness have been studied and compared. The flexure width w_a is found to have the largest influence on the flexure stiffness K_x , K_y and stiffness ratio K_y/K_x , for the four types of discussed flexures: straight flexure, folded flexure, asymmetric-quad flexure and symmetric-quad flexure. The increase of the flexure

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width w_a will increase the stiffness K_x and K_y but decrease the stiffness ratio K_y/K_x . Thus, a moderate w_a of 5~6 μm is preferred to present the optimized stiffness effect.

The stiffness of straight flexure is influenced by the flexure length and flexure width. Other than the straight flexure, the latter three types of flexure are also influenced by the transverse flexure width w_b , transverse flexure length l_b and overlapped flexure length l_a . The transverse flexure width w_b shows little influence on the stiffness K_x and stiffness ratio K_y/K_x , and the stiffness tends to be steady when w_b is 10 μm or above. Thus, the transverse flexure width w_b of 10 μm can be designed.

The transverse flexure length l_b also shows little influence on the stiffness K_x , but it presents large reverse influence on the stiffness ratio; thus, an appropriate small l_b is preferred to obtain the tradeoff of large stiffness and large stiffness ratio. In order to maintain a high stiffness ratio and also for the ease of the fabrication, l_b of the asymmetric-quad flexure and symmetric-quad flexure can be designed to be 10 μm .

The overlapped length l_a has a relatively large influence on the stiffness K_x , i.e., the larger l_a obtains the larger K_x . At the same time, l_a has a significant positive influence on the stiffness ratio of the asymmetric-quad flexure, while the stiffness ratio of symmetric-quad flexure is relatively little influenced by l_a , and the ratio starts to decrease a lot when l_a is more than 400 μm . Therefore, the overlapped length of the asymmetric-quad and symmetric-quad flexures should be around 350~400 μm .

Among the four types of flexures, the asymmetric-quad and symmetric-quad flexures are found to have the largest stiffness K_x under the same initial conditions; furthermore, the symmetric-quad flexure has the largest stiffness ratio K_y/K_x . Therefore, the symmetric-quad flexure has the largest stiffness effect. The design of the different flexure dimensions and performance are briefly listed in Table 4.6. The driving voltage of the comb microactuator is 30 or 40 V, and the air gap is 3 μm .

Table 4.6 Design of the flexures parameters and actuator performance using FEM

	Straight- flexure actuator	Folded- flexure actuator	Asymmetric- quad flexure actuator	Symmetric- quad flexure actuator
Flexure total length L (μm)	420	420	420	420
Flexure width w_a (μm)	5	5	5	5
Transverse flexure width w_b (μm)	N.A.	10	10	10
Overlapped flexure length l_a (μm)	N.A.	380	380	380
Transverse flexure length l_b (μm)	N.A.	10	10	10
Flexure stiffness K_x (N/m)	21.93	8.79	38.84	38.84
Actuator stiffness (N/m)	65.8	35.16	155.36	155.36
Stiffness ratio K_p/K_x	7057.58	200.19	3607.75	7620.51
Displacement at 30 V (μm)	0.40	0.76	0.17	0.17
Displacement at 40 V (μm)	0.72	1.34	0.30	0.30
Natural freq. (kHz)	8.39	6.28	14.53	14.53
Picoslider-loaded natural freq. (kHz)	1.02	0.71	1.56	1.56
Femtoslider-loaded natural freq. (kHz)	1.67	1.16	2.54	2.54

4.4 Comparison of Analytical Method and Finite Element Method

Although FEM is a very attractive analytical method in calculating the stress and deformation of complex geometrical structures, it only presents specific results under pre-defined boundary conditions. Since FEM cannot systematically describe the relationship between the output and input parameters, it is a limited analytical method. The classical analytical method is used to represent the link between output and input by using formulas, but the analytical theory has to more or less assume or admit some simplification of the structure deformation, especially for the complicated geometrical structures; this means that there may be some error in comparison with the realistic

deformation. Therefore, FEM can be used for the analysis of complex structures, and it is an important supplement to analytical method.

However, there are some differences between the analytical method and FEM. We have compared the analytical results and FEM results to study the differences between them. With reference to Table 4.1, the error between the analytical results and FEM results is less than 1% by using a 2D element model when a single force (F_x) is applied. Here, the symmetric-quad flexure with BEAM3 element model was selected as the example to verify the difference with the analytical model. The flexure parameters such as flexure width w_a and transverse flexure width w_b are studied by using the modified equations of deflection v .

Firstly, the influence of flexure width w_a of the sym-quad flexure with FEM and analytical method have been compared. The analytical result of the flexure deflection is based on Equation (4.40) ~ (4.41), with the boundary conditions the same as those of FEM. The results are shown in Table 4.7, with the initial conditions: $L=420\text{ }\mu\text{m}$, $l_a=400\text{ }\mu\text{m}$, $l_b=10\text{ }\mu\text{m}$, $w_b=2w_a$, $F_x=10\text{ }\mu\text{N}$ or $F_y=10\text{ }\mu\text{N}$.

Table 4.7 Comparison of FEM and analytical method for symmetric-quad flexure, with variation of flexure width w_a .

		$w_a=3\text{ }\mu\text{m}$	$w_a=4\text{ }\mu\text{m}$	$w_a=5\text{ }\mu\text{m}$	$w_a=6\text{ }\mu\text{m}$	$w_a=7\text{ }\mu\text{m}$
FEM	$u(F_x\text{ only}),\text{ }\mu\text{m}$	1.2675	0.5347	0.27377	0.1584	0.09977
	$v(F_y\text{ only}),\text{ nm}$	0.07039	0.04821	0.0368	0.02995	0.02528
	K_y/K_x	18006.82	11091.06	7439.41	5288.81	3946.59
Before revising Eqn. v	$u(F_x\text{ only}),\text{ }\mu\text{m}$	1.2675	0.5347	0.27379	0.1584	0.09978
	$v(F_y\text{ only}),\text{ nm}$	0.03625	0.01529	0.007829	0.004531	0.002853
	K_y/K_x	34965.52	34970.57	34971.26	34959.17	34973.71

As seen in Table 4.7, FEM and analytical results exhibit very little difference in the flexure deflection u . However, a big difference between FEM and analytical

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results is shown in the deflection v . This difference is believed from the analytical equations. For the analytical equations, we assumed that there is no beam elongation or shortening in the flexure deformation, and only considered the bending of the beams; thus, all the flexures' deflections are approximate and simplified results. This is the major reason for the difference between analytical and FEM results. Since the beam elongation or shortening only happens along the beam length (y -axis) when carried on an axial force, only the analytical deflection v is affected by the neglect, while the bending deflection u orthogonal to the beam length (x -axis) is hardly influenced. This explains why FEM and analytical equations present almost the same deflection u but show a difference in deflection v .

To eliminate the difference of deflection v between the analytical equations and FEM, the analytical equations should be modified by including the deformation of beam elongation and shortening along the beam length.

As seen in Figure 4.7, there are four vertical beam segments that induce the beam elongation when the symmetric-quad flexure is applied by a vertical force F_y . The total beam elongation v_e of the symmetric-quad flexure can be considered as the sum of the beam elongation of four vertical segments, which can be calculated as

$$v_e = \frac{F_y l_0}{Etw_a} + \frac{F_y l_0}{Etw_a} + \frac{F_y l_a}{2Etw_a} = \frac{F_y (4l_0 + l_a)}{2Etw_a} \quad (4.44)$$

Therefore, the total deflection v_8 of the symmetric-quad flexure in Equation (C.109), Appendix C can be revised by including the beam elongation v_e . In that case, the equation will be

$$v_g' = v_g - v_e$$
$$= -l_b^2 l_a (F_x (288r_1 - 18r_1r_2^4\alpha^4 - 36r_1^2r_2^3\alpha^3 + 3312r_1^2r_2\alpha + 912r_1^2r_2^2\alpha^2 + 24r_2\alpha + 216r_1r_2^3\alpha^3 + 102r_2^2\alpha^2 + 1308r_1r_2\alpha + 18r_2^4\alpha^4 + 1152r_1^2 + 1362r_1r_2^2\alpha^2 + 96r_1^2r_2^3\alpha^3) + F_y (15r_2^5\alpha^4 + 195r_1r_2^4\alpha^3 + 155r_2^4\alpha^3 + 1712r_1r_2^3\alpha^2 + 128r_2^2\alpha + 1056r_1r_2 + 3716r_1r_2^2\alpha + 412r_2^3\alpha^2)) / (72EI_b (15r_2^4\alpha^4 + 86r_2^3\alpha^3 + 95r_2^2\alpha^2 + 536r_1r_2^2\alpha^2 + 128r_1 + 96r_1r_2^3\alpha^3 + 560r_1r_2\alpha + 24r_2\alpha)) - F_y (4l_0 + l_a) / (2Etw_a)$$

(4.45)

Based on the revised equations of deflection v , the analytical deflections of symmetric-quad flexure have been re-calculated, and the results are compared to the FEM and prior analytical results. Table 4.8 lists the comparison result of the symmetric-quad flexure’s deflection, with the initial conditions: $L=420\text{ }\mu\text{m}$, $l_a=400\text{ }\mu\text{m}$, $l_b=10\text{ }\mu\text{m}$, $w_b=2w_a$, applied by $F_x=10\text{ }\mu\text{N}$ or $F_y=10\text{ }\mu\text{N}$.

Table 4.8 Comparison of the symmetric-quad flexure deflection with FEM and analytical methods

		$w_a=3\text{ }\mu\text{m}$	$w_a=4\text{ }\mu\text{m}$	$w_a=5\text{ }\mu\text{m}$	$w_a=6\text{ }\mu\text{m}$	$w_a=7\text{ }\mu\text{m}$
FEM	$u(F_x\text{ only}),\text{ }\mu\text{m}$	1.2076	0.5095	0.2609	0.1509	0.09506
	$v(F_y\text{ only}),\text{ nm}$	0.05819	0.04306	0.03423	0.02843	0.02432
	K_y/K_x	20752.7	11832.3	7622.0	5307.8	3908.7
Before revising Eqn. v	$u(F_x\text{ only}),\text{ }\mu\text{m}$	1.2078	0.5095	0.2609	0.1509	0.09508
	$v(F_y\text{ only}),\text{ nm}$	0.04844	0.02044	0.01046	0.006055	0.003813
	K_y/K_x	249339.4	249266.1	249426.4	249215.5	249357.5
After revising Eqn. v	$u(F_x\text{ only}),\text{ }\mu\text{m}$	1.2078	0.5095	0.2609	0.1509	0.09508
	$v(F_y\text{ only}),\text{ nm}$	0.06126	0.04435	0.03489	0.02881	0.02456
	K_y/K_x	19716.0	11488.2	7477.8	5237.8	3871.3
	Error of v	5.27%	2.99%	1.93%	1.34%	0.98%

The deflection v (in y -direction) of the symmetric-quad flexure with different flexure width w_a is plotted in Figure 4.17. The revised results of the calculated deflection v show a good matching to the FEM results of deflection v ; the initial calculation of deflection v is much lower as it does not include the deformation of beam elongation.

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The results show that the revised analytical deflection of the flexure with beam elongation is very close to FEM deflection. It also means that the revised equations are well-expressed for the deflection of the flexure.

Figure 4.17 Comparison of the symmetric-quad flexure deflection v vs. flexure width

With including the elongation and shortening of the vertical beam segments under the vertical load, the error of vertical deflection v to the FEM deflection v is around 5%. The deviation tends to decrease when flexure width increases. This is because a bigger flexure width has higher stiffness and reduces the deflections in all directions. Since we only considered the elongation of four vertical beam segments when applied by a vertical force, there is definitely some elongation of transverse beam segments. The elongation of transverse beam segments tends to be smaller as the flexure width increases; thus, its influence on the total deflection of the flexure tends to be smaller, i.e., the error tends to be smaller as the flexure width increases.

Figure 4.18 shows the comparison result of stiffness ratio of the sym-quad flexure with variation of the flexure width. Here, the revised calculated results of stiffness ratio also match very well with the FEM results. The results also prove that higher flexure width will reduce the stiffness ratio; thus, moderate flexure width is preferred.

Figure 4.18 Comparison of the symmetric-quad flexure stiffness ratio vs. flexure width

Similarly, the comparison is also done with the study of transverse flexure width w_b and the results are shown in Table 4.9, with the initial conditions: $L=420\text{ }\mu\text{m}$, $l_a=400\text{ }\mu\text{m}$, $l_b=10\text{ }\mu\text{m}$, $w_a=5\text{ }\mu\text{m}$, $F_x=10\text{ }\mu\text{N}$ or $F_y=10\text{ }\mu\text{N}$.

Table 4.9 Comparison of the FEM results and analytical results with variation of the transverse flexure width w_b

		$w_b=5\text{ }\mu\text{m}$	$w_b=8\text{ }\mu\text{m}$	$w_b=10\text{ }\mu\text{m}$	$w_b=15\text{ }\mu\text{m}$	$w_b=20\text{ }\mu\text{m}$
FEM	$u(F_x\text{ only}), \mu\text{m}$	0.27377	0.2626	0.26085	0.25955	0.2592
	$v(F_y\text{ only}), \text{nm}$	0.0368	0.03459	0.03423	0.03396	0.03389
Before revising Eqn. v	$u(F_x\text{ only}), \mu\text{m}$	0.2738	0.2627	0.2609	0.2596	0.2592
	$v(F_y\text{ only}), \text{nm}$	0.007829	0.003202	0.000105	0.000312	0.00013
After revising Eqn. v	$u(F_x\text{ only}), \mu\text{m}$	0.2738	0.2627	0.2609	0.2596	0.2592
	$v(F_y\text{ only}), \text{nm}$	0.04167	0.03587	0.03489	0.03416	0.03398
	Error of v	13.23%	3.70%	1.93%	0.59%	0.27%

Figure 4.19 plots the deflection v of the symmetric-quad flexure as the function of the transverse flexure width w_b , by using FEM and analytical equations. The analytical results of deflection v without considering the beam elongation are much

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smaller than the FEM deflection results; however, the revised analytical deflection v with beam elongation almost matches with the FEM results. It can be seen that the error between the revised analytical deflection and FEM result is less than 2% when the transverse flexure width reaches 10 μm or more, suggesting that the revised analytical results of the flexure are true and acceptable.

Figure 4.19 Comparison of the symmetric-quad flexure’s deflection v versus transverse flexure width

In the same way, the revised analytical equations of the sym-quad flexure can be used to study the stiffness influence of l_a and l_b . Furthermore, the analytical equations can be modified for the folded flexure and asymmetric-quad flexure, and the analytical deflection results would be very close to the FEM results by including the beam elongation. Therefore, the analytical equations with the beam elongation /shortening can well represent the deflection of the flexures, and can be used to analytically study the flexures’ deflection and stiffness.

4.5 Chapter Summary

To realize the high bandwidth and fine actuating range applications of the microactuator, the mechanical flexures should have proper high stiffness in the

operational direction K_x and have high stiffness ratio K_y/K_x to prevent or minimize deflection in the perpendicular direction. As the flexures are usually loaded with x - and y - direction forces, and the shear deformation in the z -direction is much smaller than other deformation, only K_x and K_y are considered in the flexure design. Several types of flexures, including straight flexure, folded flexure, asymmetric-quad flexure, and symmetric-quad flexure, have been studied for their stiffness effect.

Analytical and finite element methods have been used to derive the deflection and stiffness of the flexures. Four major influencing factors on the flexure stiffness, i.e., the flexure width w_a , transverse flexure width w_b , overlapped flexure width l_a and transverse flexure length l_b , have been studied and their dimensions have been optimized to obtain the high stiffness and high stiffness ratio. Among the four types of flexures, the asymmetric-quad flexure and symmetric-quad flexure have the largest stiffness K_x under the same conditions; furthermore, the symmetric-quad flexure has the largest stiffness ratio K_y/K_x . The straight flexure shows moderate stiffness. The folded flexure is most flexible and suitable for the large deflection applications. After FEM analysis, the design of the different flexure dimensions and actuator performance are listed in Table 4.6.

Lastly, the FEM and analytical results have been compared. Since the derivation of analytical solution of the flexure deflection is complex and has admitted simplification, there is remarkable difference between the analytical and FEM results. By including the beam elongation and/or shortening, the analytical equations of the flexure deformation are modified. The revised analytical deformation result is verified to be very close to the FEM result.

Chapter 5

Fabrication of the Microactuators

This chapter describes fabrication method and materials for the prototypes of the microactuators. There are two methods have been explored: silicon-on-glass (SOG) wafer process and single-silicon wafer with thick oxidation layer process. The SOG fabrication processes are highlighted and the good fabrication results are shown.

The electrostatic microactuator is featured with thick and high-aspect-ratio structure, and fabricated with bulk silicon wafer. Since the electrostatic actuators have both movable and fixed structures, the actuators are conventionally fabricated by using silicon-on-insulator (SOI) wafer [50,112], which has an insulator layer of silicon dioxide and structural silicon layer (device layer) on top of the substrate silicon (handle layer). Figure 5.1 shows the scheme of the SOI wafer. The comb structures are fabricated in the device layer and supported by handle layer, and the silicon dioxide underneath the movable electrodes is released to provide moving space.

Figure 5.1 Schematic diagram of silicon on insulator (SOI) wafer

There are several kinds of SOI wafers with different fabrication technologies. One of the most widely used SOI wafer in the industry is using Smart-Cut process from SOITEC Company [50]. Based on the technical principles of ion implantation and wafer bonding, and like an atomic scalpel, Smart-Cut allows transfer of very thin layers of single-crystalline material onto a mechanical support. The cost of SOITEC

wafers is well controlled and suitable for high volume IC industries. Thicker film (device layer) of 1.5 μm above requires the epitaxial deposition after smart-cut process, and this significantly increased cost. SIMOX (Separation by Ion Implantation of Oxygen) is another fabrication technology of SOI wafers, but is much expensive than Smart-Cut process, and is effectively used for the high-temperature sensors [65]. BESOI (Bonded and Etchback SOI) is a typical type of SOI wafers especially for MEMS applications with different requirements of film thickness (1-50 μm). BESOI wafers used the selectively chemical etchback thinning technology after initial bonding and grinding process [65-66]. The etchback has single or double etchstop formed by B^+/P^+ implant and epitaxy, and offers the possibility of a wide range of the buried oxide thickness as well as the silicon layer thickness. These BESOI wafers are normally very costly comparing to other SOI wafers. In addition, most SOI wafers have the same disadvantage of low release yield of buried oxide. On consideration of the robustness requirement of the 3D microactuators, the cost-effective SOG wafers were selected for the fabrication process. Another motivation of the SOG wafer is that SOG process was currently not well developed for the electrostatic actuators.

In this project, we developed the SOG fabrication process of the electrostatic microactuators by using wafer bonding technology. To realize the prototypes of the microactuator, a number of MEMS fabrication techniques are employed. These techniques consist of photolithography, wet etching, thin film deposition, wafer bonding and deep reactive ion etching (DRIE), which have been introduced in Chapter 2.

The SOG fabrication process for creating high-aspect-ratio microactuators is presented in section 5.1. The photo-mask design and some of fabrication results using SOG process are presented in section 5.2.

5.1 Silicon on Glass Fabrication Process

For the fabrication of 3-D electrostatic microactuators, we developed SOG process to replace the expensive SOI process. Glass is a convenient substrate material of the electrostatic actuator as it provides not only very good isolation between different electrodes, but also good mechanical support. In addition, it facilitates bulk micromachining. Another advantage of glass wafer is the low cost compared to SOI wafer. Thus, wafer bonding is one of the key technologies in the fabrication of SOG microactuator. The electrostatic microactuator has a three-dimensional structure of 100 μm in thickness, with an aspect ratio of more than 30:1. The typical anisotropic bulking micromachining technology – deep reactive ion etching (DRIE) is utilized.

There are several key processes involved in the fabrication of the microactuator:

- Photolithography. It is the elementary step for all the pattern transferring processes for further etching and deposition processes.
- Thin-film deposition. The metal electrodes, Cr and Au, are deposited on a glass substrate successively. Magnetic sputtering technique is used to for the deposition process. Other thin-film material, such as amorphous silicon, is deposited by chemical vapor deposition as the mask to etch glass.
- Wet etching. This process is used to etch the glass substrate and metal electrodes. The glass is etched in concentrated hydrofluoric solution to define the dicing frame of the actuator devices.
- Wafer bonding. It is used to bond a silicon wafer to glass wafer. As there are electrodes between the silicon and glass wafer, we have studied two types of bonding techniques: eutectic bonding and anodic bonding.
- Deep reactive ion etching (DRIE). It is used to fabricate the high-aspect-ratio structure and is one of the most critical techniques for the fabrication process.

Based on the two kinds of wafer bonding techniques, we proposed two sets of the process flow for the fabrication of SOG microactuators.

5.1.1 Eutectic bonding SOG process flow

Low temperature eutectic bonding has been carried out to bond silicon wafer to the glass substrate with an intermediate Cr/Au layer. A thick Cr/Au layer (30/500 nm) is deposited and patterned on the glass substrate; this acts as not only electrode pads but also the intermediate eutectic bonding layer. This kind of bonding process requires that the surface condition of bonding wafers be very clean.

The process flow is summarized in a series of cross-section diagrams in Figure 5.2. (a) Firstly, a Pyrex glass wafer is cleaned and deposited with 30/500 nm Cr/Au by using sputtering machine; (b) lithography on the Cr/Au layer is then carried out to define the electrode pattern; (c) wet etching of Au and Cr in the respective etchant solution to determine the electrodes that is also the bonding layer; (d) depositing and patterning the glass etching mask layer on the glass wafer [113~115]; (e) double-side wet etching of the glass substrate in HF solution of 200 μm deep so as to define the dicing line; (f) lithography on the silicon wafer (thickness of 450 μm) to define the backside cavity pattern; (g) deep etching of the 100 μm cavity on the backside of silicon wafer to create the moving space of the microactuator; (h) eutectic bonding of the silicon wafer to the glass substrate with intermediate Au layer at 380 °C. The wafer bonding process should be performed with great caution, and the bonding surfaces of the two wafers must be very clean; (i) wet etching in KOH solution to thin the silicon wafer to 200 μm ; (j) lithography on the silicon wafer to define the pattern of comb drive actuators; (k) deep RIE on the topside silicon wafer of 110 μm depth to achieve the high-aspect-ratio comb drive actuators, and dice the glass substrate to obtain the individual microactuator devices.

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(a) Deposition of Cr/Au layer on the glass substrate;

(b) Photolithography and patterning on the photoresist layer;

(c) Wet etching of Au, Cr respectively to obtain the electrodes;

(d) Glass etching masking and lithography on the glass substrate;

(e) Wet etching the glass substrate to get the dicing lines;

(f) Lithography and patterning on silicon backside;

(g) Deep RIE on the backside silicon wafer to get the cavities;

(h) Eutectic bonding the silicon to glass with the intermediate of Au;

(i) Thinning the silicon wafer from topside using wet etching;

(j) Lithography and patterning with the topside silicon etching mask;

(k) Deep RIE of the silicon wafer to form the structure of microactuator.

Figure 5.2 Proposed process flow of the microactuator based on eutectic bonding

5.1.2 Anodic bonding SOG process flow

Other than eutectic bonding, the anodic bonding process was also developed for the SOG fabrication of the microactuators. Anodic bonding is a well established process in MEMS industry which generally has very high yield with bonding strength. Anodic bonding is usually used to bond silicon wafer to glass wafer. The process requires a high quality surface. However, the electrodes would be built on the glass

substrate and bonded in between the two wafers; thus, the thick electrode pads should be avoided in the bonding by pre-etching some cavities in silicon wafer.

The proposed fabrication process of the microactuator based on anodic bonding is presented in Figure 5.3. (a) Firstly, a Pyrex glass is patterned with photoresist; (b) Ti/Pt of 20 nm/30 nm is sputtered on the glass wafer; (c) release the photoresist (lift-off) to form the conduction electrodes; (d) patterning of the Cr/Au layer of 50 nm/0.25 μm , and then lift-off to form the bonding pads on the glass wafer; (e) deposition and patterning of glass etching mask on top and bottom side of the glass wafer [113]; (f) double-side wet etching of the glass substrate in HF solution, forming the through hole; (g) deep etching for the 100 μm cavity on the backside of silicon wafer to create the moving space of the microactuator; (h) anodic bonding of the silicon wafer to the glass at 400 $^{\circ}\text{C}$. The wafer bonding process should be performed with great caution, and the bonding surfaces of the two wafers must be very clean. During this bonding process, the major portion of silicon is bonded to the glass substrate. At the same time, the small portion of Pt/Ti layer on the glass is diffused into the silicon to form the ohmic contact for the electric conduction of the electrodes; (i) lithography and patterning on the topside silicon for DRIE etching of actuator structures; (j) deep RIE on the topside silicon wafer of 110 μm depth to achieve the high-aspect-ratio comb drive actuators, and then, final release and dicing to obtain the individual devices.

(a) Photolithography on the glass substrate;

(b) Deposition of 50 nm Pt/Ti on the Pyrex glass wafer;

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(c) Release of the photoresist to get the Ti/Pt electrode pads;

(d) Deposition and patterning of 300 nm Cr/Au on the glass wafer;

(e) Deposition and patterning of the glass etching mask layer;

(f) Double-sided glass etching in HF to define dicing lines;

(g) Deep RIE silicon cavity etching (backside 100 μm deep);

(h) Anodic bonding of silicon to the Pyrex glass;

(i) Lithography of the microactuator's structures;

(j) Deep RIE silicon structures of the microactuator and dicing.

Figure 5.3 Proposed process flow of the microactuator based on anodic bonding

5.2 Mask Design, Fabrication Highlights and Results

5.2.1 Mask design for the microactuators

Once the process flows for fabricating the microactuator are designed, the next step is to design the photo mask for the lithography. Since the photolithography process transfers copies of a pattern from a mask onto the surface of a solid material (wafer), via the photoresist polymer, the mask design process is critical.

Based on the previous structural design and proposed fabrication flow, the eutectic and anodic bonding required four and five layers of masks, respectively. Mask layout software L-Edit is used to design the photo-masks [116]. As the microactuator comprises silicon and glass wafers, with double side patterning of the silicon wafer, four masks are required for eutectic bonding process flow. They are presented in different layers: one layer for the electrode pad on the glass substrate, one layer for dicing frames on the glass, one for backside cavity pattern of the silicon, and the final layer for the topside microactuator pattern. For the anodic bonding process flow, one extra mask for the thin Ti/Pt layer is needed.

Figure 5.4 presents a cell structure of the microactuator with four layers for eutectic bonding process. Each cell will be fabricated for one actuator prototype. Several types of flexures that have been investigated in Chapter 4, such as straight flexure, folded flexure, and asymmetric-quad flexure, are plotted in different cells.

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The close-up of flexures is shown in Figure 5.5. Figure 5.6 shows a cell design of the microactuator for the anodic bonding process with five mask layers designed for the anodic bonding process, with the sym-quad flexures.

Figure 5.4 One cell of the microactuator mask for eutectic bonding process flow

Figure 5.5 Close-up of different cell of the microactuator with different flexures

Figure 5.6 One cell view of the microactuator with symmetric-quad flexures for anodic bonding process

The completed mask designs were sent to the mask manufacturing company to produce soda/lime glass masks, generally made of chrome layer which can be recycled many times during the photolithography process.

5.2.2 Fabrication highlight and results

- **Deposition and patterning the electrodes of Cr/Au, Ti/Pt**

This process is used to fabricate the electrode pads on the glass substrate, which will interconnect the silicon electrodes after wafer bonding. The eutectic bonding process needs only the Cr/Au layer with thickness around 0.5 μm , as shown in Figure 5.2 (a)-(c). Firstly, a thin-film of Cr/Au (50 nm/450 nm) is deposited on the glass substrate by using DC magnetron sputtering machine. A 50 nm chrome layer is deposited on the glass wafer as an adhesive layer before depositing the gold layer. The thick gold layer acts as the bonding layer and wiring pads. Au is selected because of its good compatibility capability with silicon wafer during eutectic bonding.

Sputtering is preferred over evaporation in thin-film deposition due to the faster deposition rate, good uniformity, and excellent adhesion to substrate. The glass substrate with Cr/Au layer is then patterned with AZ 9260 photoresist by using photolithography, followed by wet etching of the Au and Cr respectively. After releasing the photoresist, the electrode pad structure is realized on the glass substrate. Figure 5.7 shows the photo of the Cr/Au electrode pad on the glass substrate.

Figure 5.7 Top view of the electrode pad deposited on the glass substrate

Two different layers of electrodes are designed for the anodic bonding process, i.e. very thin interconnection layer of Ti/Pt and thick wiring pad layer of Cr/Au. The process steps for the electrodes are shown in Figure 5.3 (a)-(d). The thickness of Ti and Pt are 20 nm and 30 nm respectively. These layers are patterned using lift-off process since platinum is very difficult to etch. Firstly, the thick photoresist (AZ 9260) is lithographed on the glass with Ti/Pt photomask, and then Ti and Pt metal film are sputtered on the patterned wafer. The resist is released by soaking it in acetone solution, creating the Ti/Pt electrodes on the substrate. The thick layer of Cr/Au (~ 300 nm) is sputtered and wet etched on top of the first Ti/Pt electrode layer. Figure 5.8 shows the picture of the two layered electrodes on the glass substrate.

Figure 5.8 Picture of the two layers of electrode pads on glass wafer

- **Selection of glass material**

In MEMS devices fabrication, glass is the second most widely used material, after silicon. Remarkable properties such as high chemical resistance, high heat resistance, high electrical isolation, biocompatibility, large optical transition range and low optical absorption are desired from the device material. Glasses used in MEMS need to meet several important requirements: micromachinable using patterning and etching process, bondability to silicon, and similar coefficient of thermal expansion (CTE) to silicon to avoid thermal stress.

There are three major groups of techniques used for glass etching: mechanical, dry and wet. Mechanical drilling or powder blasting cannot generate smooth surfaces and is difficult to control etching depth. Dry etching techniques include plasma and laser etching of glass, which are expensive process. In addition, the plasma etching rates are very low. Among all the available etching process, the wet chemical etching using hydrofluoric-based solution remains one of the most low-cost and effective solutions.

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Glass is a “mixture” of oxide, and the composition and concentration of these oxides gives the main properties. Due to different compositions, the glass etching rate is different. Some oxides in glass, such as CaO , MgO or Al_2O_3 give insoluble products in HF solution. These insoluble products are deposited on the generated surfaces and act as masking layers. As a result, after etching, the surface becomes rough and, in time, the etching rate decreases. For Hoya SD2 a large amount of Al_2O_3 – (about 20%) is present, while in the composition of soda lime, CaO (8.8%), MgO (4%) can be found [115,117,118]. In the composition of Corning 7740 the insoluble products is only 2% (Al_2O_3) [115]. For this reason, deep wet etching of glass is recommended for glass with low concentration of oxides that give insoluble products in HF. This is why our experiments focus more on Pyrex glass (Corning7740).

Figure 5.9 Coefficients of thermal expansion variance of various glass materials compared with silicon material

Another important property of glass is the bondability to silicon. Any type of plain glass can be anodic bonded to silicon wafer. After wafer bonding the sandwich of silicon and glass will be cool down from around 400 °C to room temperature. Whenever the coefficients of thermal expansion (CTE) of silicon and glass do not match, cooling causes residual stress. At the moment the main glasses for anodic

bonding are: Corning (Pyrex) 7740, Corning 7070, Borofloat 33 and Hoya SD1 and SD2 [117]. The relationship of the different CTE between several standard materials and silicon in microelectronics is shown in Figure 5.9 [118-119].

The high resistivity of Corning 7070 glass is a significant advantage during anodic bonding in vacuum, when an increased voltage is required. Corning 7740 glass, which is the most popular, ensures the smallest warping and waving of bonded sandwich, SD2 Hoya glass enables the lowest thermal shock stresses after bonding, and the sealing process can be performed at a higher temperature.

- **Process limitations of glass wet etching**

Pinholes and notching defects on the edges are the main defects that limit the wet etching process of the glass. Figure 5.10 shows an example of glass etching defects in HF solution after removal of the Cr/Au mask. These defects, which are illustrated in Figure 5.11, are the result of the interaction between the etchant and masking material. The main factors that contribute to defect generation can be summarized as: residual stress in the masking layer [120]; type of stress (tensile or compressive) [120]; stress gradients (for multilayer mask) [121]; hydrophilicity/hydrophobicity of the mask surface [121].

Figure 5.10 Optical image of 100µm deep channel etched in glass using a Cr/Au mask

The residual stress type has a strong influence on the defect generation. The small defects on the mask surface, which were generated during depositions, generate small microchannels in the masking layer having tensile stress. If the surface of the layer is hydrophilic, the etching solution is sucked into this microchannels, and in a short time, pinholes will start to form. For this reason, a masking layer with compressive stress (general characteristic of amorphous silicon or polysilicon) is preferred.

Figure 5.11 (a) Generation of pinholes due to tensile stress or stress gradient (creep); (b) generation of the notch defects

The notching defects on the edges are a characteristic of metal masks (Cr/Au or Cr/Cu) having tensile stress. These defects are generated by the stress gradient (the stress in the Cr layer can be 800 MPa ~ 1 GPa) [121] - an example is presented in Figure 5.12 (a) - or by the breakage of the edge of the mask [120] (Figure 5.12 (b)). During isotropic etching process, due to the undercut effect, the edge of the mask becomes a freestanding structure.

Figure 5.12 (a) Optical image showing the notching defects on the edges generated by stress gradient; (b) the broken mask during the wet etching due to the residual stress

Hydrophilicity or hydrophobicity of the surface also plays an important role. A small defect (micro-creep) on a hydrophobic surface will be very difficult to be filled with the etching solution. Silicon surface presents a hydrophobic characteristic, and for this reason, amorphous silicon or polysilicon masks are preferred over Cr/Au mask. The hydrophilicity of the surface can be changed by retaining and hard baking the photoresist mask. During the baking process, the photoresist fills the micro-creeps of the α -Si or Cr/Au mask. By hard baking the layer, the adhesion of the photoresist at the microcreep walls increases and its removal by the etching solution becomes very difficult. In our case, we used AZ7220 (from Clariant). The baking process for AZ7220 was performed on a hotplate at 120 °C for 30 minutes.

- **High etch rate of glass wet etching**

In wet etching process of glass, materials used as masking layers, such as silicon, silicon carbide or gold, are inert in the HF-based etchants. In such cases, the etching process is limited by the defects of the masking layer and the penetration of the etchant through these defects. For this reason, a fast etch rate of glass will lead to a deeper etching while the defect generation or mask decay will maintain at the same rate each time. The etch rate of Corning 7740 in HF 49% solution is around 7 $\mu\text{m}/\text{min}$. The main factor that affects the etch rate is the concentration of HF solution (maximal concentration being desired for a very deep wet etching process). A decreasing of HF concentration from 49% to 40% is equivalent to a decrease of the etch rate by 60% [122]. Annealing of the glass [122], usually at temperatures around 600 °C, can increase the etch rate due to the redistribution of oxides (some oxides are etched faster) but also generates a rough etching surface. Warming the solution can increase the etch rate of the glass material [122] but for safety reasons, this method is not recommended (increasing the temperature increases the quantity of HF vapors).

- **Masking material for wet etching of glass**

There are three major groups of masking materials for glass wet etching: photoresist, metal and silicon. Photoresist is very often used as mask layer [123-124], but its area of application is limited (20-30 μm). Another commonly used metal mask is Cr/Au [124-125], where Cr layer is used to improve the adhesion of gold to glass. Another material very commonly used for glass etching is silicon, deposited by different methods: PECVD amorphous silicon [114,124], LPCVD polysilicon [124], silicon carbide [126] or even anodically bonded bulk silicon.

- a) Photoresist masking layer

Photoresist is mainly used for silicon oxide etching in diluted HF (BHF) solutions. In our experiment, we used photoresist AZ7220. In highly concentrated HF solutions, the quality of the photoresist mask was very poor. The maximum etching time – around 3 minutes (equivalent with an etching depth of 25 μm on non-annealed wafers) – was achieved after the photoresist was hard-baked at 120 °C for 30 minutes. A huge isotropy was noted (5:1) even when an adhesive promoter (HDMS) was used. After etching a longer time, the photoresist mask would peel off. The technique can be used in application where no more than 25 μm -deep etching is required.

- b) Cr/Au

One of the most commonly used metal masks is Cr/Au because Au is an inert material in HF solutions while Cr is used as adhesive layer. As mentioned before, the high value of tensile stress and the creeps in the mask layer limits the etching depth. The measured residual stress in the sputtered Cr/Au layer was 250-300 MPa tensile. Usually, the Cr/Au layer can be used for etching of 50-100 μm deep glass patterns. Increasing the Au thickness will increase the time for the solution to penetrate through the defects, but the final results are not significantly improved. A solution to improve

the quality of Au layer is to make successive Au depositions [122], with each deposition being followed by a relaxation time. Using this method the etching time was increased up to 40 minutes for around 250 μm etch depth in Pyrex glass [122].

c) PECVD amorphous silicon ($\alpha\text{-Si}$)

Silicon is an inert material in HF-based solutions. It also has the advantage of being a hydrophobic material. Hence, the penetration of etchant through the small impurities of the mask is relatively difficult. The $\alpha\text{-Si}$ layer presents the advantage of deposition at low temperatures (300 $^{\circ}\text{C}$ for PECVD deposition), but as we analyzed in [122], the high value of compressive stress induced in this layer (400-600 MPa) limits its application up to 20 minutes (150 μm deep etch). The annealing of the mask layer can reduce the stress value and improve the performance (up to 30 minutes – equivalent of 200 μm [114]). The isotropy of the etching was 1:1.2.

The influence of stress is presented in Figure 5.13, where wet etching of glass was performed using the same $\alpha\text{-Si}$ mask, but this was annealed at 400 $^{\circ}\text{C}$ at different times, resulting in a different residual stress in the mask layer (600 MPa compressive to 100 MPa tensile). The evolution of the stress value in an $\alpha\text{-Si}$ layer with the annealing time is presented in Figure 5.14.

Figure 5.13 Optical image showing intersections of two channels etched in glass for 20 minutes (150 μm in depth) in 49% HF solution using $\alpha\text{-Si}$ mask with different stress values: a) 600 MPa, b) 300 MPa, c) 100 MPa, d) 100 MPa (tensile).

Figure 5.14 Annealing effect at 400 °C on the residual stress of PECVD α -Si

We also optimized the PECVD process to achieve the very low stress α -Si layer [127]. At this process condition, the deposited α -Si presents a low compressive stress of 10-20 MPa and this can be used for the very deep or through-hole etching of glass with combination of photoresist mask [128].

d) LPCVD Polysilicon

The test layer - 0.8 μm thick - was deposited in a Tystar furnace at 590 °C. The resulting stress in the layer was 50 MPa compressive. The resistance of the mask in HF solution was 30-35 minutes. The isotropy of the etching was very good – almost 1:1. Similar results obtained with the low stress PECVD α -Si and LPCVD polysilicon masking layer underline the importance of stress value in the process of wet etching of glass. The main disadvantage of the LPCVD polysilicon layer relies in its relatively high deposition temperature, usually between 590 °C and 620 °C.

e) Amorphous SiC

SiC can be another material that can be used in deep wet etching of glass. Previous work [126, 129] reported that amorphous SiC as mask is almost an inert material in HF solution for wet etching of glass. Moreover, its hydrophobic surface avoids an easy penetration of the solution through the small defects of the masking layer. Meanwhile, the stress can be easily tuned near to zero by annealing [129] or by

RF power optimization of deposition process [130]. The low-stress SiC layer is another good mask material for glass deep wet etching.

- **Wet etching process of glass**

This process aims to etch the grooves in the glass wafer in order to define the dicing boundary of the microactuator device. Corning Pyrex 7740 glass with thickness of 700 μm is used for the process, and etching depth of glass is 200 ~ 400 μm . As discussed previously, wet etching of glass is chosen for its very fast, simple process and inexpensive cost. The etchant used is highly concentrated 49% HF solution.

As studied in previously sections, masking material is very important for the glass wet etching due to the particular etching process limitations. The photoresist is the cheapest solution for rapid wet etching of glass, but the photoresist can only sustain a few minutes in HF solution before it starts to peel off. Another commonly used mask, Cr/Au layer, can withstand the aggressive HF acid to yield an etching depth of 100-150 μm before exhibiting significant pinholes or defects on the surface. Therefore, low-stress α -Si plus photoresist layers are chosen as the masking material in the glass deep wet etching. Figure 5.15 shows the comparison of through-hole glass etching results with metal mask and low-stress α -Si mask. [113-114].

Figure 5.15 (a) Pin-hole defects on Cr/Au mask with etched through glass in HF, (b) good surface condition of low stress α -Si mask with etched through glass in HF

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To achieve an etching depth of more than 200 μm glass grooves as the dicing lines, the low-stress α -Si mask is preferred. There are two kinds of PECVD deposition methods to obtain the low stress layer. One is to deposit the normal-stress PECVD α -Si layer and anneal to reduce the residual stress. The Technics PECVD system was used to deposit the α -Si layer, and the RF power used is 120 W at 300 $^{\circ}\text{C}$. The residual stress of amorphous silicon is normally above 500 MPa. After annealing at 400 $^{\circ}\text{C}$ for four hours, the residual stress can be reduced to be around 100 MPa.

Figure 5.16 (a) 100 μm deep glass etching with α -silicon, (b) 300 μm deep glass etching with low stress α -silicon mask

Another method that we have tried is the optimized PECVD deposition of α -Si layer [113,127]. By optimizing the deposition conditions of STS PECVD reactor at low temperature of 200 $^{\circ}\text{C}$ and low-frequency generator power of 120 W, the residual stress of deposited α -Si thin film is remarkably reduced to 10~20 MPa [127]. The low-stress α -Si layer is naturally very good for the deep wet etching of glass wafer.

The α -Si mask was then lithographed with AZ7220 phototresist and etched in a Technics RIE system, using $\text{CF}_4/\text{O}_2 = 9/1$ sccm at 140 mTorr with power of 350W. After mask patterning, the glass is wet etched in 49% HF solution with a magnet stirrer. Figure 5.16 (a) shows the excellent result of glass etching with 100 μm depth, while Figure 5.16 (b) shows the 300 μm etching with the PECVD α -silicon mask. To

get deeper etching of the grooves, the double-side masking layer can be used to etch both sides of glass. For example, the 400 μm deep grooves can be achieved by 200 μm deep etching on both sides. This can be achieved without pinholes and notch defect in glass wafer [127].

- **Eutectic bonding of silicon to glass**

When the glass was etched and silicon wafer was patterned with cavities, the next step was to bond the silicon wafer to glass substrate. Glass substrate was selected because it can provide good isolation between the different electrodes and a good mechanical strength to support the movement of the microactuator. Given the existence of electrode pads on the glass wafer, eutectic bonding was firstly used to bond the two wafers with gold as an intermediate layer. The gold layer was pre-deposited on the glass and worked as not only the electrodes but also the intermediate layer for the eutectic bonding process.

Eutectic bonding process happens at a temperature of 363°C, which is lower than the melting point of glass. Au is the most commonly used intermediate material in this kind of bonding [131-132]. The bonding stress is comparable to that of fusion bonding. The eutectic bonding should be carried through very carefully because many factors such as temperature, bonding pressure, and even native oxides can prevent bonding from taking place.

The eutectic bonding was carried out with Karl Suss WB6, at a temperature of 380 °C with pressure of 2000 mbar. The eutectic bonding result is shown in Figure 5.17, and most of the area was successfully bonded. However, the bonding process did not show high yield and the bonding results were not stable or even worse no eutectic bonding happened in some areas. The reason is that the surfaces of the two wafers are critical to the particles or native oxide, which results in the weak bonding

between the wafers. Therefore, the low yield and bonding reliability are the main problems for the eutectic bonding.

Figure 5.17 Eutectic bonding of Si to glass with Au intermediate layer

The main advantage of Si-Au eutectic bonding is its technological compatibility with IC process and a low temperature process that does not destroy the metallization. On the other hand, the biggest drawback of this method of bonding is weak wetting through gold of residual and native oxides, which occur on the silicon surface. Intrinsic oxides, which evolve during bonding in the air, need to be “broken” using the method of mashing or rubbing, in order to ensure direct contact between silicon and gold [64,133]. This mechanical operation is quite brutal and can destroy the fragile, 3-D MEMS structures fabricated on the bonded wafers. The native oxide on silicon surface can be also removed by RIE or BHF etching [131,133]. However, it is still difficult to completely prevent the oxide during the eutectic bonding process and thus affect the bonding quality. In our experiments, the eutectic bonding results were not good and there were quite a number of bubbles in the bonded wafers. The oxide layer should be the reason of our low-yield eutectic bonding.

- **Anodic bonding of silicon to glass**

Anodic bonding was also employed to bond the silicon to glass wafers. Anodic bonding is a method that hermetically and permanently joins glass to silicon without the use of any adhesives. The bonding temperature is in the range of 300-450 °C (depending on the glass type) at which the alkali-metal ions (sodium ions) in the glass become mobile. The process requires a good surface quality with roughness of less than 0.1 μm . The components are brought into contact and a high voltage (up to >1000 V) is applied across them. This causes the alkali cations to migrate from the interface, resulting in a depletion layer with high electric field strength. The resulting electrostatic attraction brings the silicon and glass into intimate contact. Further current flow of the oxygen anions from the glass to the silicon results in an anodic reaction at the interface and the glass becomes bonded to the silicon with a permanent chemical bond.

However, anodic bonding also builds up residual stress between bonded Si/glass chips at high temperature, since Si and glass have different coefficients of thermal expansion (CTE). The residual stress may cause minor distortion between the silicon and glass wafers, and this becomes a critical issue for IC patterning applications, while as it is acceptable for many MEMS applications. The residual stress of the bonded Si/glass wafers can be reduced by annealing or selecting suitable glass type with a CTE fit for silicon.

From Figure 5.9, the difference of CTE between glass and silicon is very small when the temperature is not more than 310 °C. when temperature is below 260 °C, the compressive stress is induced, which easily leads to bending or buckling of thin membranes structures; higher temperature than 260 °C, the residual stress turns to be tensile stress, which is preferable as it only reduces the mechanical response and

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introduces a soft non-linearity [119]. In order to make sure that bending or buckling will not happen, a sufficient large margin must be provided towards higher temperature, saying more 360 °C is preferred in anodic bonding.

In this project, electrodes are fabricated on the glass substrate, making it difficult to perform anodic bonding. However, the anodic bonding has a roughness tolerance of 0.1 μm . The electrodes can be designed with two layers: contacting anodic bonding layer of less than 0.1 μm and wiring electrode layer of 0.3~0.5 μm . The fabrication of the electrodes is shown in Figure 5.3 (a)~(d). The thin layer of Ti/Pt with thickness of 20/30 nm is deposited on glass by using lift-off process. Since the contact layer of Ti/Pt is only 50 nm thick, the anodic bonding would happen between the patterned glass and silicon wafer as well as on the bare glass to silicon wafer [134]. In addition, the Ti/Pt layer acts as the electrode interconnection. As the bonding side of silicon was pre-etched with cavities, the thick Cr/Au would not have contact with silicon.

Figure 5.12 Anodic bonding of Si to glass wafer with intermediate layer

The anodic bonding was carried out with Karl Suss wafer bonder at the temperature of 400 °C and negative voltage of 1000 V. The central bonding occurred

with voltage of 2000 V for two minutes, and area bonding of 1000 V lasted around fifteen minutes. Figures 5.12 and 5.13 showed the anodic bonding result, which has a very good yield and strong bonding stress. By using the multi layer electrodes, the anodic bonding process was preferred in the project.

Figure 5.13 close up of the anodic bonding between silicon to glass with electrodes

- **Deep RIE of the silicon structures**

As the microactuator has high aspect ratio (depth-to-width ratio) structures, the typical anisotropic dry etching technology – deep reactive ion etching (DRIE) – was used to realize the bulk-micromachining. DRIE is a powerful technique with distinct advantages, such as high selectivity to photoresist or silicon dioxide (50:1 to 200:1), and high aspect ratio (height-to-width ratio can be more than 30:1) etching to form substantially deep trenches (up to 500 μm) with vertical sidewalls [53,54,135,136].

The STS inductively coupled plasma (ICP) multiplex system based on the fluorine chemical etching has been used in this fabrication process, as illustrated in Figure 2.16 [55]. For very deep etching, a combination of photoresist and oxide are

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required. The etching proceeds in alternative steps of reactive ion etching in a SF_6 plasma and polymer deposition (passivation) from a C_4F_8 plasma.

For DRIE process, the etch rate, profile, and anisotropy greatly depend on the chamber pressure, gas flow rate, cycle time, and applied electrode power (bias power). For instance, pressure increases the formation of atomic fluorine and other radicals, relative to that of ions, and, thus, promotes isotropic profiles. It also reduces the maximum ion energy. Applied electrode power increases the self-bias and the ion directionality and hence etching anisotropy. Finally, increasing the SF_6 flow rate increases the removal rate of etching products and significantly affects the etching rate. Change of cycle time of etching and passivation also affect the anisotropy. Hence, etching rate, anisotropy, and uniformity have a noticeable dependence on pressure, applied power, and flow rates [137~139]. The ultra-high anisotropy vertical profiles can be achieved by the optimization of these process parameters, such as pressure, gas flow rate, and electrode power.

In the SOG fabrication of microactuators, DRIE is used to etch backside cavities and topside comb actuators, as shown in Figure 5.3 (g) and (j). As the microactuator needs to be bonded with a glass substrate, the silicon wafer needs to be double side patterned and etched: the topside silicon is etched to get the comb drive actuators, while the backside silicon is etched to form the cavities (100 μm in depth) for the movement space of the microactuator. The silicon wafer used for the SOG process is 200- μm thick, therefore, the remaining silicon for fabrication the topside electrodes has a thickness of around 100 μm . In the design of the electrostatic microactuator, the air gap is 3 μm or 4 μm and comb teeth is 5 μm wide; therefore, the aspect-ratio of trenches is 25:1 to 33:1. This ultra high-aspect-ratio etching can be achieved only by DRIE as the aspect ratio of anisotropic wet etching by KOH is quite small. The recipe

of the Multiplexed ICP DRIE for the high-aspect-ratio etching was developed, based on our previous study on the high anisotropy deep silicon etching [140~144].

Figure 5.14 SEM photo of a whole view of SOG fabricated microactuator

Figure 5.15 SEM partial view of the microactuator with folded-plate flexures

Figure 5.16 Close-up view of the combs with high aspect ratio (more than 30:1)

Figure 5.17 Close-up of the asymmetric-quad flexures and curvy flexures

Figure 5.14 shows the plan view of a whole microactuator device fabricated on glass substrate by using anodic bonding. The microactuator is 1.2 mm by 0.8 mm in size (including the metal pads), and 0.8 mm in thickness (0.2 mm of silicon structure). The microactuator was cut by using the dicing saw. Figure 5.15 presents the SEM photos of the microactuator structures with folded flexures. Figure 5.16 in turn shows the close-up of the comb electrodes in the microactuator, with designed air gap of 4 μm and 3 μm. Figure 5.17 shows the close up of the microactuators with asym-quad flexures and curvy flexures. In these photos, we see that the comb structures have very deep trenches with high anisotropy and good uniformity. The width of comb electrode is measured to be $4.8 \pm 0.2 \mu\text{m}$ and the air gap is $3.2 \pm 0.2 \mu\text{m}$ or $4.2 \pm 0.2 \mu\text{m}$

in width for different microactuator prototypes, and the depth of the comb structure is 100 μm . Thus, the structure has a high aspect ratio of more than 30:1.

The anodic bonding and DRIE based SOG fabrication process of the electrostatic microactuators has proved a good yield and feasibility. The fabrication yield before the last DRIE on SOG wafer was 95% through the whole wafer, however, the last DRIE process generate a relatively significant drop of yield to be around 60~70%. This is due to the systematic process limitation of DRIE lag and increased cooling challenge of SOG wafer. With the SOG fabrication process in this project, the aspect ratio of the comb drive microstructures can be optimized to be very high (30:1). However, some of the critical dimensions may change and deviate to the designed values due to the very deep silicon etching. The small difference of the dimensions would probably influence the performance of the microactuators, which will be discussed in the next chapter.

5.3 Chapter Summary

The prototype fabrication of the microactuators was realized by using SOG wafer process, with wafer bonding and high-anisotropy DRIE techniques. The anodic bonding process was succeeded to bond the glass and silicon wafer with intermediate electrode layer. The glass etching technologies were studied and the selection of glass material and masking material were highlighted. The DRIE on the SOG wafer was another critical technique, and the process was optimized to get the very high aspect ratio 3-D microstructures with relatively high yield. The feasibilities have been proved by the fabrication results.

Chapter 6

Characterization and Discussions

In this chapter, the characterization of the MEMS microactuators would be described and discussed. Generally, the characterization of dynamic micro-devices includes static testing and dynamic testing. Firstly, the test setup for the static and dynamic characterization of the microactuators is described. The experimental results are then presented according to the different types of flexure structures. The difference between the measurement and theoretical result is discussed and studied. The fabrication process generated some geometrical tolerances, and their influences on the performance of the microactuators are discussed in the chapter.

The electrostatic microactuators have the in-plane motion and their static characteristics were performed with an optical microscope using its image capturing function. The dynamic behavior was then characterized with particular instruments, the Micro Motion Analyzer MMA-300 from Polytec PI [145], which can measure both the out-of-plane and in-plane motion of the microstructures.

6.1 Static Testing

6.1.1 Testing experiment set-up

Static testing involves the measurement of the displacement (or deflection) of the microactuators at different applied voltage. This test was conducted using the Alessi Rel-4800 Probe Station in the Micromachines Centre in NTU.

The equipments used for the static testing are the Alessi Probe Station equipped with a microscope, CCD Camera, function generator, power amplifier, and a

workstation with Image Pro Express Software. Figure 6.1 shows the scheme of the experiment set-up.

Figure 6.1 Scheme of the static testing experiment set-up for electrostatic microactuators

Figure 6.2 Image of the probe test of microactuator with straight flexures

After setting up the equipments, the microactuator was put in the probe station. The probes were contacted onto the electrode pads of the test specimen, as indicated in Figure 6.2 (the probes can be applied on one pair of fixed and movable electrodes).

Subsequently, the magnification lens and the position of the test specimen were adjusted until the bottom left corner of the test specimen could be seen in the workstation monitor. The image was captured using the workstation and saved as the initial static image (See Figure 6.3). The DC voltage at the function generator was then adjusted and applied to the probes. The image of microactuator after excitation was then captured (as shown in Figure 6.3). The deflection of the microactuator under applied voltage was obtained by measuring the gap between the stopper and the movable electrode, and then compared to the gap in the initial position. There might have been some error induced in the displacement reading since the vision-capture measurement method is subjective.

Figure 6.3 Cross-section diagram with probe station (a) Initial position of the microactuator (b) position of an excited microactuator

By varying the applied voltage, the microactuator had a series of values, which were obtained by measuring the displacements of the microactuator with reference to the initial position. The range of the applied DC voltage to the electrode pads was between 10 V and 70 V.

6.1.2 Static testing result

This test is done with three types of microactuator specimens: the straight flexure, folded flexure and asymmetric-quad flexure. A few specimens of microactuators with

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the symmetric-quad flexures were fabricated but tested to be malfunctioned. In fact, the symmetric-quad flexure and asymmetric-quad flexure have very similar stiffness in operational direction and frequency response. The measured results of static characterization of the microactuators are tabulated in Table 6.1. The theoretical results are listed in Table 6.2. The three types of tested microactuators had a designed thickness of 100 μm and comb air gap of 3 μm . Based on equations (3.4) and (3.10) in Chapter 3, the theoretical results are calculated. The electrostatic force varies with the DC voltage:

$$F_e = \frac{n\epsilon t}{d}V^2 = \frac{100 \times 8.854 \times 10^{-12} \times 100 \times 10^{-6}}{3 \times 10^{-6}} \times V^2 = 0.0295V^2 \mu\text{N}$$

(6.1)

Since there are two stoppers in the microactuators to limit the amplitude of the movable sliding part, the maximum displacement of the microactuator is equal to the gap of 3 μm . Since the CCD optical measurement might have some random errors, and the displacement responses under different voltages might have hysteric effect, several repeat of measurements was done to obtain the average values. The error-bars of measurement results were presented in Figure 6.4-6.6.

Table 6.1 Measured displacement results of the microactuators with different flexures

Voltage (V)	Measured Average Displacements (m)		
	Straight Flexure	Folded Flexure	Asymmetric-quad Flexure
10	0.1	0.12	0
15	0.15	0.2	0.04
20	0.3	0.39	0.05
25	0.41	0.58	0.08
30	0.53	0.9	0.15
35	0.71	1.15	0.17
40	0.94	1.73	0.26
45	1.21	2.12	0.27
50	1.51	2.63	0.36
55	1.82	2.94	0.41
60	2.17	2.96	0.54
65	2.47	2.86	0.59
70	2.85	2.93	0.73

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Table 6.2 Theoretical displacement results of the microactuators

Voltage (V)	Theoretical Displacements (m)		
	Straight Flexure	Folded Flexure	Asymmetric-quad Flexure
10	0.04	0.08	0.02
15	0.10	0.19	0.04
20	0.18	0.34	0.08
25	0.28	0.52	0.12
30	0.40	0.75	0.17
35	0.55	1.03	0.23
40	0.72	1.34	0.30
45	0.91	1.70	0.38
50	1.12	2.10	0.47
55	1.35	2.54	0.57
60	1.61	3.02 (3.0)	0.68
65	1.89	3.54 (3.0)	0.80
70	2.19	4.11 (3.0)	0.93
K	65.8	35.16	155.4

Figure 6.4 Displacement versus voltage of microactuator with straight flexure

As seen from the measurement results, when the voltage is 40 V, the deflections are 0.94 μm , 1.73 μm and 0.26 μm for the microactuators with straight, folded and asymmetric-quad flexures, respectively. The folded-flexure microactuator is the most flexible, and the displacement is close to the maximum when the voltage exceeds 55 V due to the gap limit of stoppers. The graph of displacement versus the driving

voltage is plotted in Figure 6.4 ~ 6.6 for the microactuators with different types of flexure designs.

Figure 6.5 Displacement versus voltage of microactuator with folded-flexure

Figure 6.6 Displacement versus voltage of microactuator with asymmetric-quad flexure

6.2 Dynamic Testing

6.2.1 Testing equipment set-up

The dynamic testing involves the testing of the frequency response and the resonance amplitude, and was conducted by using the Polytec Micro Motion Analyzer (MMA-300). The Polytec MMA-300 comprises a combination of microscope

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scanning vibrometer MSV-300 and a planar motion analyzer PMA-300 that uses laser doppler vibrometer (LDV) and stroboscopic video microscopy to measure both the out-of-plane and in-plane motion respectively.

The experimental set-up for the dynamic testing includes Polytec MMA-300 with workstation, probes, function generator and amplifier. Figure 6.7 shows the scheme of the dynamic testing set-up. This testing procedure is much simpler than the static test as the software tabulates the results automatically. On the other hand, the user could alter the test settings using the workstation. The attached data management system also allows the output to be presented in bode diagrams and compiled the test results.

Figure 6.7 Scheme of dynamic testing set-up for electrostatic microactuators

The procedure of the measurement is simple but quite time-consuming. After setting up the equipments, the test specimen (microactuator) is placed in the MMA-300, attaching the probes onto the electrode pads of the specimen. The magnification is adjusted so that the bottom left corner or right corner of the sample is seen in the workstation. The electrodes are then connected to the function generator with the power on. The parameters of the sweeping frequency can then be inputted in the workstation to start sampling the vibration positions of the microactuator. A sinusoidal AC voltage of 10 V with a DC bias of 5 V is applied with a frequency scanning increment of 50 Hz. The snap shots taken are set to be 30 times per period.

The dynamic analysis takes about 1 hour for a frequency range of 10 kHz. If the curve required is very smooth, the frequency step should be decreased and snapshots should be increased a lot. However, the analysis time have to be remarkably increased.

6.2.2 Dynamic testing results

Three types of specimens, namely the microactuators with straight, folded and the asymmetric-quad flexures were tested with Polytec MMA-300 for the dynamic analysis. The dynamic tests were conducted with an AC voltage of 10 V and DC bias of 5 V. With a rough frequency sweeping, the resonance of the microactuators with straight, folded and asymmetric-quad flexures were observed to occur around 7 kHz, 5.6 kHz, 15.5 kHz, respectively. Thus, the fine frequency sweeping range for the three microactuators are 3.5 k ~ 9.5 kHz, 4.8 k ~ 6.8 kHz and 12 k ~ 19 kHz, respectively. The results contained the deflection magnitude (in μm) and phase (in degree) at each frequency step. Figures 6.8, 6.9 and 6.10 show the frequency response of the microactuators with the three different types of flexures.

As seen from the fine range of frequency sweeping, the resonant frequency of the microactuators with straight flexure, folded flexure and asymmetric-quad flexure are 7.2 kHz, 5.89 kHz and 15.85 kHz, respectively. At the resonance frequency range, the phase of the microactuators drops significantly by 180° .

Figure 6.8 Frequency response of the microactuator with straight flexure

Figure 6.9 Frequency response of the microactuator with folded flexure

Figure 6.10 Frequency response of the microactuator with asymmetric-quad flexure

From the bode graph of frequency response, the Q factor of the microactuators can be estimated; it describes the sharpness of the system's response. Quality factor is a ratio of the energy stored per cycle vs. the energy dissipated in a cycle. In general, the higher the Q factor, the higher the absolute frequency and accuracy capability of the resonator would be. The Q factor is calculated based on the below equation:

$$Q = \frac{f_r}{f_2 - f_1} \quad (6.2)$$

where f_r is the resonance frequency, f_1 and f_2 are the lower and higher frequency at which the magnitude is 3 dB lower than the magnitude at the resonance frequency. This method can be used for the rough calculation of the quality factor. The reason is that the calculation is based on the amplitude in dB, and is affected greatly by the accuracy of the measured amplitude of the microactuators. The Q factor will be greatly underestimated, if the highest amplitude is not measured correctly at the resonant frequency.

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Table 6.3 Quality factor of the microactuators with different flexures

Flexure	f_l (Hz)	f_r (Hz)	f_2 (Hz)	Q -Factor
Straight flexure	7170	7200	7300	55.4
Folded flexure	5630	5890	6190	10.5
Asmy-quad flexure (10V)	15740	15850	15930	83.4

Based on the bode graph of measurement results, the quality factor for the three types of flexures is tabulated in Table 6.3. This 3 dB drop estimation method is a very rough method for the estimation of the Q factor. Another method for the more accurate estimation of the Q factors is based on the curve fitting with the dynamic models of the microactuator system.

6.2.3 Curve fitting for Q factors

To verify the accuracy of Q factors of the microactuators from the measurement result, the curve fitting of dynamic model to the experimental result with a suitable damping coefficient was carried out. The curve fitting was realized by using the Curve Fitting Toolbox of MATLAB software.

The Curve Fitting Toolbox uses the method of least squares for fitting data. The fitting process requires a model that relates the response data to the predicted data with one or more coefficients. The result of the fitting process is an estimate of the "true" but unknown coefficients of the model. Here, the dynamic model of the microactuator was employed (described in section 3.2.2). For the microactuator, the transfer function is

$$G(s) = \frac{A\omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2}$$

(6.3)

where $\omega_n = \sqrt{K/m}$, $\zeta = C/(2m\omega_n)$, A is constant value. Thus, the quality factor is

$$Q = \frac{1}{2\zeta} = \frac{m\omega_n}{C} = \frac{\sqrt{mK}}{C} \quad (6.4)$$

The electrostatic microactuators typically can be expressed by a second-order dynamic system, with only two coefficients, ω_n and ζ , needed to be fitted for the system. From the experimental results, we can obtain the primary resonant frequency of the microactuator. Thus, only the damping coefficient ζ is required to fit the dynamic model to the experimental results.

To obtain the closest damping coefficient estimates, the least square error method minimizes the summed square of residuals. The residual for the data point i is defined as the difference between the measured amplitude response value Amp_exp and the fitted response value Amp and is identified as the error associated with the data. The expression of $Amp(i)$ and the sum square of residuals are given as

$$Amp(i) = \frac{\omega_n^2}{\sqrt{(\omega_n^2 - (2\pi f_i)^2)^2 + (4\pi\omega_n^2 f_i \zeta)^2}} \quad (6.5)$$

$$Sum_Error = \sum_{i=1}^n (Amp - Amp_exp)^2 \quad (6.6)$$

where n is the number of data points included in the fit and Sum_Error is the sum of squares error estimate.

To minimize the influence of initial conditions like the excitation voltage of the measurement, all the amplitude data from the experiments and dynamic model have been processed with normalization. Damping coefficient is scanned in the range of (0, 0.1) in 1000 steps, and the closest damping coefficient that best fits the dynamic model to experimental results was found.

Figure 6.11 shows the sum of square error with variation of damping coefficient from 0 to 0.1 and the least sum of square error between the experimental results and dynamic model when the damping ζ being 0.0074. With this damping coefficient, the

simulated frequency response curve is plotted and fitted to the measured results. Figure 6.12 shows the curve fitting of the microactuator with straight flexures (damping coefficient $\zeta=0.0074$). The graph shows a good matching of the results.

Figure 6.11 The sum of square residual errors with variation of damping coefficient for the microactuator with straight flexures

Figure 6.12 Curve fitting of dynamic model to experimental result for the microactuator with straight flexures, $\zeta=0.0074$

Therefore, the Q factor of the microactuator with straight flexures from the measurement results should be

$$Q = 1/2\zeta = 67.57 \tag{6.7}$$

Figure 6.13 Curve fitting for the actuator with asymmetric-quad flexures, at $\zeta=0.0051$

Figure 6.14 Curve fitting for the microactuator with folded flexures, at $\zeta=0.0461$

With the same method of the least sum of square residual error, the damping coefficient of the actuator with asymmetric-quad flexures can be obtained as $\zeta=0.0051$. The fitting curve is shown in Figure 6.13 with the quality factor being

$$Q = 1/2\zeta = 98.04 \quad (6.8)$$

For the microactuator with folded flexures, the experimental damping coefficient is 0.0461 and Figure 6.14 shows the fitting result. Thus, the quality factor is

$$Q = 1/2\zeta = 10.85 \quad (6.9)$$

The curve fitting aids to obtain the quality factors of microactuator from the dynamic measurement result. The quality factors of the microactuators from theoretical calculation are required to evaluate the experiment results.

6.3 Discussions

6.3.1 Fabrication tolerance and their influence

It was observed from Figures 6.4 - 6.6 that the displacement of the microactuators increases with increasing driving voltage. However, a mismatch of the result curves is shown in the graphs: for the microactuators with straight flexures and folded flexures, the measured displacement results were a bit larger than the theoretical values, while the reverse condition appears for the microactuator with asymmetric-quad flexures. There are a number of reasons for the difference.

Firstly, some incidental errors might have occurred during testing. The most possible cause is the introduction of external vibrations from nearby equipments or the ambient environment of the probe station. This could in turn induce certain amount of accidental displacement of the actuators. For example, when the image is captured using the CCD camera, the display appears to be jerky with external

excitation. Surely, the human errors in capturing and reading images during the static measurement also contribute to the total errors. In addition, dust or etched by-products (re-deposited particles) in the comb electrodes would also affect the movement of the microactuator.

Another reason concerns the systematic error. The mismatch is possibly caused by the theoretical calculation result with the incorrect boundary conditions. For example, the initial designed dimensions of the device are based on the theoretical results; however, after fabrication process, the actual dimensions of the device might differ from theoretical ones. Some of the dimensions of the device, such as flexure width and comb air gap, are critical to the mechanical performance of the microactuators, which we did not take into account in the theoretical calculation. As a result, the errors would definitely be induced between the measured results and calculated results based on the initial design.

The variation of the dimensions is usually generated in micro-fabrication process, such as the deep silicon etching process. For the 3-D electrostatic microactuators, the comb structures have very high depth-to-width aspect ratio ($> 30:1$), with an etching depth of 100 μm . This causes tremendous difficulty in maintaining the design dimensions and vertical profiles. The variation of dimensions and profile is also called fabrication tolerance, which is usually caused by the limitation and/or drawback of specific process and equipments such as DRIE process.

For DRIE process, the etch rate, profile and anisotropy greatly depends on the chamber pressure, gas flow rate, cycle time and applied electrode power (bias power) [137~139]. For instance, pressure affects the formation of fluorine atoms and other radicals, relative to that of ions (a critical factor promoting isotropic profile). It also reduces the maximum ion energy. Applied electrode power increases the self-bias and

the ion directionality and hence etching anisotropy. Finally, increasing the SF_6 flow rate increases the removal rate of etching products, which significantly affects the etching rate. The change of cycle time of etching and passivation also affect the anisotropy. The ultra vertical profiles can be achieved by the optimization of these process parameters.

However, in most conditions, the fabrication tolerance is inevitable. It changes the critical dimensions of the microactuator devices, especially for the long time etching. The DRIE tolerance influences the performance of the electrostatic microactuators, in terms of electrostatic force, flexure stiffness, displacement, air damping and quality factor, thereby inducing the difference between the measured and calculated performance of the microactuators. The DRIE tolerance consists of two aspects: (a) the profile tolerance, (b) undercut tolerance. The profile tolerance is discussed first.

(a) Profile tolerance and their effect

In general, there are two kinds of profiles after silicon DRIE: (a) positive sloped profile, where the etched trench is closing; (b) negative sloped profile, where the trench is opening. Figure 6.15 demonstrates the two kinds of tapered etching profiles.

Figure 6.15 The schematic diagrams of the two kinds of tapered profiles after deep RIE, (a) positive profile, (b) negative profile

The following discusses the influence of the sloped profiles on the actuator's performance. The comb drive microactuator with straight flexures was taken as an example to calculate the electrostatic force, flexure stiffness, and displacement according to the equations in Chapter 3.

- **Electrostatic force**

Firstly, the positive etching profile will be studied. For negative (closing) profile, the trench in between the neighbored comb fingers decreases with the etching depth, as shown in Figure 6.16. The tapered angle of the positive profile is θ , the initial beam width is w , the initial trench gap is d , the comb finger length is L and the comb thickness is t .

Figure 6.16 Schematic model of positive etching profile

According to the equations of electrostatic force in the comb drives, $dF = \frac{1}{2} \frac{\partial C}{\partial x} V^2$, we can calculate the electrostatic force of the comb finger with tapered angle θ . Assuming that C is the capacitance of the beam, V is the applied voltage, and the x -axis is the moving direction of the comb fingers.

$$dA = dz(L + dx) \quad (6.10)$$

$$dC = \frac{\epsilon A}{d} = \frac{\epsilon \cdot dz(L + dx)}{d - 2z \cdot \tan \theta} \quad (6.11)$$

$$dF = \frac{1}{2} \frac{\partial C}{\partial x} V^2 = \frac{1}{2} \frac{\varepsilon \cdot dz \cdot dx}{(d - 2z \cdot \tan \theta) dx} V^2 = \frac{1}{2} \frac{\varepsilon V^2}{d - 2z \cdot \tan \theta} dz \quad (6.12)$$

$$\begin{aligned} F_x &= 2 \int_0^t \frac{1}{2} \frac{\varepsilon V^2}{d - 2z \cdot \tan \theta} dz = \varepsilon V^2 \int_0^t \frac{1}{d - 2z \cdot \tan \theta} dz \\ &= -\frac{\varepsilon V^2}{2 \tan \theta} \int_0^t \frac{1}{d - 2z \cdot \tan \theta} d(d - 2z \cdot \tan \theta) = -\frac{\varepsilon V^2}{2 \tan \theta} \ln(d - 2z \cdot \tan \theta) \Big|_0^t \\ &= -\frac{\varepsilon V^2}{2 \tan \theta} \ln\left(1 - \frac{t}{d} \cdot 2 \tan \theta\right) = \frac{\varepsilon V^2}{2 \tan(-\theta)} \ln\left(1 + \frac{t}{d} \cdot 2 \tan(-\theta)\right) \end{aligned} \quad (6.13)$$

Equation (6.13) describes the electrostatic force of the comb finger with sidewall angle θ . It is seen that when $\theta \rightarrow 0$, $F_x(0) = \varepsilon t V^2 / d$, which is exactly the same as comb drive with ideal vertical sidewall profiles. Given the conditions of the microactuator, $t = 100 \mu\text{m}$, $d = 3 \mu\text{m}$, $w = 5 \mu\text{m}$, we can get the relative change of the electrostatic force,

$$\frac{F_x(\theta)}{F_x(0)} = \frac{\ln\left(1 - \frac{t}{d} \cdot 2 \tan \theta\right)}{-\frac{t}{d} \cdot 2 \tan \theta} = \frac{\ln(1 - 66.7 \tan \theta)}{-66.7 \tan \theta} \quad (6.14)$$

When $\theta = 0.85^\circ$, $F_x(0.85^\circ) \rightarrow +\infty$. In the actual condition of $t = 100 \mu\text{m}$ and $d = 3 \mu\text{m}$, it means that the air gap at the bottom is closed and the trench is not etched through when the sloped angle θ is more than 0.85° . In this case, the comb fingers connect each other and the comb drive actuator cannot work.

If the etching profile of the comb beam is negatively sloped (gap opening), the trench between the comb beams increases with the etching depth. The slope angle θ of negative profile is just opposite to that of positive profile, and the electrostatic force can be calculated accordingly.

$$\begin{aligned} F_x &= 2 \int_0^t \frac{1}{2} \frac{\varepsilon V^2}{d + 2z \cdot \tan \theta} dz = \varepsilon V^2 \int_0^t \frac{1}{d + 2z \cdot \tan \theta} dz \\ &= \frac{\varepsilon V^2}{2 \tan \theta} \int_0^t \frac{1}{d + 2z \cdot \tan \theta} d(d + 2z \cdot \tan \theta) \\ &= \frac{\varepsilon V^2}{2 \tan \theta} \ln(d + 2z \cdot \tan \theta) \Big|_0^t = \frac{\varepsilon V^2}{2 \tan \theta} \ln\left(1 + \frac{t}{d} \cdot 2 \tan \theta\right) \end{aligned} \quad (6.15)$$

Equation (6.15) is almost the same as equation (6.13) with only a difference in polarity of the slope angle θ . When $\theta \rightarrow 0$, $F_x(0) = \frac{\varepsilon \cdot t \cdot V^2}{d}$, which is also as same as the comb drive with vertical sidewall profile. Given $t=100 \mu\text{m}$, $d=3 \mu\text{m}$, $w=5 \mu\text{m}$, we can get the relative electrostatic force with variation of the negative profile angle θ ,

$$\frac{F_x(\theta)}{F_x(0)} = \frac{\ln\left(1 + \frac{t}{d} \cdot 2 \tan \theta\right)}{\frac{t}{d} \cdot 2 \tan \theta} = \frac{\ln(1 + 66.7 \tan \theta)}{66.7 \tan \theta} \quad (6.16)$$

In fact, the sloped angle of negative profile also has a limit. The bottom width of the comb-finger electrode decreases to zero when $\theta=2.86^\circ$, under the conditions of $t=100 \mu\text{m}$. Therefore, the negatively sloped angle should be ideally less than 2.86° .

As seen from equation (6.13) and (6.15), the comb drive actuators with positive and negative profiles share the same electrostatic force expression with only a difference in direction of tapered angle θ . Thus, the electrostatic force with the two kinds of tapered profiles can be described within the same graph (Figure 6.17). According to the figure, the angle of positive profile is set to be positive, and the angle of negative profile is shown as negative.

Figure 6.17 The relative electrostatic force against slope angle of comb electrodes

It can be seen that the electrostatic force decreases with an increase in the negative profile angle, but the electrostatic force will increase with the profile angle in the positive etching profile. Furthermore, the electrostatic force tends to be infinite when the angle θ reaches some value which is dependent on the designed air gap.

- **Flexure stiffness**

Besides the electrostatic force, the stiffness of the flexure has also changed since the profile of the flexure has changed during the etching process and this induces a change of the inertial momentum of the flexure beam. Figure 6.18 shows the SEM image of the flexure with negative profile measured from the broken pieces of the comb actuators.

Figure 6.18 (a) SEM image of broken piece of the actuator (b) close-up of the flexure with negative etching profile

Generally, with the conditions of our ICP-RIE machine, the trench with a small gap of less than 5 μm will generate positively tapered profile, and the gap at the trench bottom tends to close; whereas for the trench with wide gap of more than 20 μm , the etching process tends to generate negatively sloped profile [140,146]. In other words, the flexures profile will be negatively tapered since the surrounding trenches of the flexure beams are usually larger than 20 μm . Therefore, the small-trench comb

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fingers and wide-trench flexures could have different tapered profiles under the same etching conditions. In fact, this phenomenon is also observed in previous studies on ICP silicon etching with the different trench widths [135,138,139,147].

Figure 6.19 shows a scheme of the flexure beam with positive sloped profile. Assuming that the width of the flexure to be w , slope angle to be ϕ (flexure profile ϕ might be different from the comb finger slope θ), and cross-section area to be A , the inertial momentum I can be calculated by using integration,

Figure 6.19 The inertia momentum model of flexure beam with negative profile

$$\frac{1}{2} A_1 = \frac{1}{2} \left(x + x - \frac{t}{2} \tan \theta \right) \cdot \frac{t}{2}; \quad (6.17)$$

$$\frac{1}{2} A_2 = \frac{1}{2} \left(x + x + \frac{t}{2} \tan \theta \right) \cdot \frac{t}{2}; \quad (6.18)$$

$$A = A_1 + A_2 = \left(x + x - \frac{t}{2} \tan \phi \right) \cdot \frac{t}{2} + \left(x + x + \frac{t}{2} \tan \phi \right) \cdot \frac{t}{2} = 2x \cdot t; \quad (6.19)$$

$$I = \int_A x^2 dA = \int_0^{\frac{w}{2} - \frac{t}{2} \tan \phi} 2t \cdot x^2 dx = \frac{2t}{3} x^3 \Big|_0^{\frac{w}{2} - \frac{t}{2} \tan \phi} = \frac{t}{12} (w - t \cdot \tan \phi)^3 \quad (6.20)$$

When $\phi \rightarrow 0$, $I = tw^3/12$, which is the inertia momentum of rectangular beam.

Then, the stiffness in x-direction of the straight flexure K_x is calculated as

$$K_x = \frac{12EI}{L^3} = \frac{Et}{L^3} (w - t \cdot \tan \phi)^3 \quad (6.21)$$

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where E is the Young's modulus of the silicon material, w is the width of the flexure beam, $K_x(0)$ is the stiffness of flexure with vertical profile. Under the conditions of flexure thickness $t=100\text{ }\mu\text{m}$, width $w=5\text{ }\mu\text{m}$, the relative stiffness is

$$\frac{K_x(\phi)}{K_x(0)} = \frac{(w - t \cdot \tan \phi)^3}{w^3} = \frac{(5 - 100 \tan \phi)^3}{125} \quad (6.22)$$

Similarly, if the etching profile of the flexure beam is positive, the stiffness of the flexure can be expressed as

$$K_x = \frac{12EI}{L^3} = \frac{Et}{L^3} (w + t \cdot \tan \phi)^3 \quad (6.23)$$

The relative stiffness under the conditions of $t=100\mu\text{m}$ and $w=5\mu\text{m}$ can be obtained,

$$\frac{K_x(\phi)}{K_x(0)} = \frac{(w + t \cdot \tan \phi)^3}{w^3} = \frac{(5 + 100 \tan \phi)^3}{125} \quad (6.24)$$

Figure 6.20 Relative flexure's stiffness as the function of slope angle

From equations (6.21) and (6.23), the flexure stiffness K_x is the same for both the negative and positive profiles, only differing in the direction of the angle ϕ . Therefore,

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the relative flexure stiffness as a function of the two kinds of sloped etching profiles can also be illustrated in the same graph, as shown in Figure 6.20. In the figure, the slope angle ϕ ranges from -1.5° to 1.5° with the positive profile showing a positive angle.

In the positive profile, the flexure stiffness increases with the increase of the slope angle ϕ . When $\phi=0.1^\circ$, $\frac{K_x(0.1^\circ)}{K_x(0)} = 1.108$; when $\phi=0.5^\circ$, $\frac{K_x(0.5^\circ)}{K_x(0)} = 1.620$; when $\phi=1.0^\circ$, $\frac{K_x(1^\circ)}{K_x(0)} = 2.455$. However, in the negative profile, the flexure stiffness would decrease when the slope angle ϕ increases. When $\phi=0.1^\circ$, $\frac{K_x(0.1^\circ)}{K_x(0)} = 0.899$; when $\phi=0.5^\circ$, $\frac{K_x(0.5^\circ)}{K_x(0)} = 0.562$; when $\phi=1.0^\circ$, $\frac{K_x(1^\circ)}{K_x(0)} = 0.276$.

- **Displacement of the microactuator**

Therefore, the displacement of the microactuator with negative etching profile of comb fingers and flexures can be expressed as

$$\begin{aligned}\delta_x &= \frac{F_x}{K_x} = \frac{n \cdot \epsilon V^2}{2 \tan \theta} \ln \left(1 + \frac{t}{d} \cdot 2 \tan \theta \right) \cdot \frac{L^3}{Et(w - t \cdot \tan \phi)^3} \\ &= \frac{n \cdot \epsilon V^2 \cdot L^3}{Et(w - t \cdot \tan \phi)^3 \cdot 2 \tan \theta} \cdot \ln \left(1 + \frac{t}{d} \cdot 2 \tan \theta \right)\end{aligned}\quad (6.25)$$

The displacement of the microactuator is a function of both the comb-finger's slope angle, θ and flexure's slope angle, ϕ . Thus, the relative change of the displacement in terms of profile angle θ and ϕ can be obtained as,

$$\frac{\delta_x(\theta, \phi)}{\delta_x(0)} = \frac{d}{t \cdot 2 \tan \theta} \cdot \frac{w^3}{(w - t \cdot \tan \phi)^3} \ln \left(1 + \frac{t}{d} \cdot 2 \tan \theta \right) \quad (6.26)$$

where $\delta_x(0)$ is the displacement of the microactuator with ideal vertical sidewall profile.

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If the etching profiles of the comb fingers and flexures are positive, then the displacement and relative displacement will be

$$\delta_x = \frac{F_x}{K_x} = \frac{n \cdot \epsilon V^2 \cdot L^3}{Et(w + t \cdot \tan \phi)^3 \cdot 2 \tan \theta} \cdot \ln \left(1 - \frac{t}{d} \cdot 2 \tan \theta \right) \quad (6.27)$$

$$\frac{\delta_x(\theta, \phi)}{\delta_x(0)} = -\frac{d}{t \cdot 2 \tan \theta} \cdot \frac{w^3}{(w + t \cdot \tan \phi)^3} \ln \left(1 - \frac{t}{d} \cdot 2 \tan \theta \right) \quad (6.28)$$

Under the conditions of $t=100 \mu\text{m}$, $d=3 \mu\text{m}$, $w=5 \mu\text{m}$, the relative displacement of the actuator with positive profiles can be obtained:

$$\frac{\delta_x(\theta)}{\delta_x(0)} = \frac{\ln(1 - 66.7 \tan \theta)}{-66.7 \tan \theta} \cdot \frac{125}{(5 + 100 \tan \phi)^3} \quad (6.29)$$

Figure 6.21 Relative displacement as function of slope angle of flexures and combs

Since the displacement of the actuator has two independent profile variables after deep etching, the curve of relative displacement of the microactuator with the slope angles is plotted in a 3D graph as shown in Figure 6.21. The profile angle θ of the comb fingers and angle ϕ of flexures are in the range of -1° to 1° .

The graph shows that a combination of negatively tapered flexure profile and positively tapered comb profile generates the largest displacement of the comb microactuator. For example, the relative displacement of the microactuator would be as large as 8.560 when the flexure tapered angle, $\phi = -1.0^\circ$ and comb tapered angle, $\theta = 1.0^\circ$. The displacement tends to be infinite when the angle exceeds 1.0° and it is because when the comb gap is closing to zero in positive profile, the electrostatic force tends to be infinite theoretically. On the other hand, the positively sloped flexure profile and negatively sloped comb-fingers profile will decrease the displacement of the actuator.

- **Quality factor**

From the analysis above, the deep RIE tolerance in both negative and positive profiles has significant influence on the actuator's electrostatic force, stiffness effect, and displacement. As a result of these output changes, the microactuator's dynamic performance will also be influenced, such as the resonant frequency, air damping and quality factor of the actuator. According to equation $f_r = \frac{1}{2\pi} \sqrt{K/m}$, the resonant frequency increases with an increase of the flexure stiffness, with assumption that the mass change in the tapered profiles is negligible. In fact, the movable shuttle mass has one-order magnitude relationship with slope angle, while the stiffness has three-order magnitude relationship with the angle. Thus, the slope angles have greater impact on the actuator's stiffness than on the mass. Here, the influence of the slope angle on the actuator's quality factor would be discussed.

The air damping in the laterally driven comb actuators is slide film damping and is presented in three fluid layers: underneath the comb drive, above the comb drive,

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and between the comb fingers (sidewall). These air damping would cause three parts of energy loss of the microactuator, Q_d , Q_∞ and Q_c , which are expressed as [148~150],

$$\frac{1}{Q} = \frac{1}{Q_d} + \frac{1}{Q_\infty} + \frac{1}{Q_c} = \frac{C_D}{\sqrt{mK}} = \frac{\mu}{\sqrt{mK}} \left(\frac{A_q}{d} + \frac{A_q}{\delta} + \frac{A_c}{d_c} \right) \quad (6.30)$$

where C_D is the total air damping, μ , being the absolute viscosity of air, d , the gap between the comb drive and substrate, δ , the characteristic distance, d_c , the air gap between the comb fingers, A_q and A_c , the top effective area of the plate and the overlapped sidewall area of the comb fingers, respectively. For the SOG fabricated comb drive actuator, the gap between the comb drive and substrate d and characteristic distance δ are far bigger than the com air gap d_c . Therefore, the damping between the sidewalls of the comb fingers is the dominant part of the air damping of the comb actuator. Furthermore, the etching profile has no influence on the effective top area of the comb drive. Thus the total air damping is $C_D = C + C_0 \cong C$, where C_0 denotes the minor and constant damping part. Therefore, the rough quality factor with negatively tapered profiles can be derived as

$$Q(\theta, \phi) \approx \frac{\sqrt{mK}}{C} = \frac{\sqrt{m \cdot 3Et(w - t \cdot \tan \phi)^3 / L^3}}{2n\varepsilon \cdot L \cdot \ln \left(1 + \frac{t}{d} \cdot 2 \tan \theta \right) / 2 \tan \theta} = \frac{2 \tan \theta \sqrt{m \cdot 3Et(w - t \cdot \tan \phi)^3 / L^3}}{2n\varepsilon \cdot L^2 \cdot \ln \left(1 + \frac{t}{d} \cdot 2 \tan \theta \right)} \quad (6.31)$$

where n is the number of movable comb fingers. Thus, the relative quality factor as the function of negative profiles is

$$\frac{Q(\theta, \phi)}{Q(0)} = \frac{t \cdot 2 \tan \theta \sqrt{(w - t \cdot \tan \phi)^3}}{d \sqrt{w^3} \ln \left(1 + \frac{t}{d} \cdot 2 \tan \theta \right)} \quad (6.32)$$

If the slope profiles of comb fingers and flexures are positive, the quality factor and relative quality factor would be

$$Q(\theta, \phi) \approx \frac{\sqrt{mK}}{C} = \frac{-2 \tan \theta \sqrt{m \cdot 3Et(w + t \cdot \tan \phi)^3 / L}}{2n\epsilon \cdot L^2 \cdot \ln\left(1 - \frac{t}{d} \cdot 2 \tan \theta\right)} \quad (6.33)$$

$$\frac{Q(\theta)}{Q(0)} = \frac{-t \cdot 2 \tan \theta \sqrt{(w + t \cdot \tan \phi)^3}}{d \sqrt{w^3} \ln\left(1 - \frac{t}{d} \cdot 2 \tan \theta\right)} \quad (6.34)$$

Figure 6.22 Relative Q-factor as function of tapered profiles of comb fingers and flexures

The formulas above can be used to roughly evaluate the variation of quality factor in terms of the sloped etching angle, on considering that other conditions are maintained constant or with only minor change. Figure 6.22 shows the relative quality factor in terms of the different sloped profiles of the flexures and comb fingers, with the conditions of $t=100 \mu\text{m}$, $d=3 \mu\text{m}$, $w=5 \mu\text{m}$, slope angles from -1° to 1° . It is found that the combination of the flexure with a negative profile and comb fingers with a positive profile induces the great drop in the quality factor of the comb actuator. For instance, the Q factor is only 50.1% of the original Q with vertical profiles if comb tapered angle $\theta=0.5^\circ$ and flexure tapered angle $\phi=-0.5^\circ$. However, the reverse applies

too. The relative quality factor will be 1.61 times that of the vertical profiles, if the comb tapered angle $\theta = -0.5^\circ$ and flexure tapered angle $\phi = 0.5^\circ$.

(b) Undercut effect

Figure 6.23 (a) undercut effect of isotropic etching, with undercut $L \approx H$; (b) undercut effect of anisotropic etching, with undercut $L \ll H$.

Besides the profile tolerance, undercut is another common phenomenon introduced by DRIE process. Generally, the fluoride electrochemical dry etching of silicon is isotropic with the etching rate in vertical and transverse directions being almost the same. Figure 6.23 (a) shows the profile of isotropic etching with SF_6 , and the undercut L is almost equal to the etching depth H . Anisotropic DRIE was developed to obtain the high-aspect-ratio and vertical profile microstructures by introducing the alternation of C_4F_8 passivation process and SF_6 etching process. However, the anisotropic DRIE cannot completely eliminate the undercut effect, and this phenomenon becomes to be worse when the SF_6 etching starts as the first cycle. Figure 6.23(b) shows the scheme of the anisotropic etching profile with a much smaller undercut L compared to etching depth.

Figure 6.24 (a) shows the undercut image of anisotropic etching before removing the photoresist, which has an obvious change of the trench width. Figure 6.24 (b) shows the close up of DRIE comb drive actuators after removal of the photoresist.

The width of comb electrodes is observed to be around 4.7~4.8 μm . In general, the undercut at the top of the trench appears more serious when the etching depth is deeper. For example, 18 μm undercut is observed for a 275 μm deep etch, and 8 μm undercut is observed for a 125 μm deep etch [151].

Figure 6.24 (a) DRIE undercut in the high-aspect-ratio structure before removing the mask layer; (b) Close-up of the comb electrodes with DRIE undercut after removing mask layer

With the undercut of the comb electrodes and flexure beams being Δd and Δw , the actual air gap becomes $d + 2\Delta d$, and the flexure width turns to $w + 2\Delta w$. Since the displacement of the microactuator with design dimensions is given as

$$\delta_x = \frac{F_x}{K_x} = \frac{n \cdot \epsilon V^2 L^3}{E d w^3} \quad (6.35)$$

The relative electrostatic force with the comb electrodes and flexure undercut of Δd and Δw will be

$$\frac{\delta_x(\Delta)}{\delta_x} = \frac{d w^3}{(d + 2\Delta d)(w + 2\Delta w)^3} \quad (6.36)$$

If the fabricated microactuators have both the profile and undercut tolerances, the relative displacement can be derived by combining equations (6.28) and (6.36),

$$\frac{\delta_x(\theta, \phi, \Delta)}{\delta_x(0)} = -\frac{d}{t \cdot 2 \tan \theta} \cdot \frac{w^3}{(w - 2\Delta w + t \cdot \tan \phi)^3} \ln \left(1 - \frac{t}{d + 2\Delta d} \cdot 2 \tan \theta \right) \quad (6.37)$$

Similarly, the relative Q factor with major inter-electrodes damping can be obtained

$$\frac{Q(\theta, \phi, \Delta)}{Q(0)} = \frac{-t \cdot 2 \tan \theta \sqrt{(w - 2\Delta w + t \cdot \tan \phi)^3}}{d \sqrt{w^3} \ln \left(1 - \frac{t}{d + 2\Delta d} \cdot 2 \tan \theta \right)} \quad (6.38)$$

In conclusion, the influence of the fabrication tolerance on the performance of the electrostatic microactuators has been studied with the analytical equations being derived. These equations are very meaningful in the prediction and evaluation of the actual performance of the fabricated electrostatic microactuators.

6.3.2 Dimension characterization and modified calculation results

The critical dimensions of the fabricated microactuators are measured with an optical microscope and SEM, and the tolerance of the dimensions has been observed. Figures 5.16 and 6.25 show the plan view and side elevation close-up of the SOG fabricated comb drive actuator, respectively.

Figure 6.25 Close-up of the comb drive microactuator, showing the positive etching profile of the narrow trenches

Figure 6.26 Backside view of the microactuator, showing the closing gap at the bottom

Figure 6.27 (a) 0.05 μm undercut of the comb electrodes (b) 0.15 μm undercut of the comb electrodes

The comb electrodes have very deep etched narrow trenches (with aspect-ratio of more than 30:1) and these comb electrodes have shown the positive slope profile. The air gap and narrow trenches decrease with the increase in depth which also can be seen in Figure 6.25. The positive etching profile results in the narrowed gap at the bottom as shown in Figure 6.26. However, as discussed in the previous section

(Figure 6.18), the flexures present the negative etching profile as the neighboring trenches are very large. Besides the profile tolerance, the microactuators also have undercut as has been shown in Figure 6.24 (b) and Figure 6.27.

Table 6.4 Measured dimensions of the SOG fabricated comb drive microactuators

Thickness of comb drive structure		100 μm
Gap between movable comb to glass substrate		100 μm
Comb electrode width	Designed	5 μm
	Measured (top)	4.7~4.9 μm
	Measured (bottom)	5.2~5.4 μm
Comb air gap	Designed	3 μm
	Measured (top)	3.1~3.3 μm
	Measured (bottom)	2.6~2.8 μm
Flexure width	Designed	5 μm
	Measured (top)	4.7~4.9
	Measured (bottom)	4.2~4.4 μm

By measuring the broken pieces of the comb fingers, the fabricated dimensions of the comb drive actuators are characterized and the results are listed in Table 6.4. Based on the measured dimensions, the slope angles of comb electrodes and flexures are

$$\theta = \tan^{-1}((5.3 - 4.8)/(2 \times 100)) = \tan^{-1}(0.0025) \cong 0.15^\circ \quad (6.39)$$

$$\phi = \tan^{-1}((4.3 - 4.8)/(2 \times 100)) = \tan^{-1}(-0.0025) \cong -0.15^\circ \quad (6.40)$$

The average undercuts of the comb electrodes Δd and flexure, Δw are around 0.1 μm . Therefore, by using the equation derived previously, the relative displacement and Q factor of the microactuator with straight flexures can be computed as

$$\frac{\delta_x(0.15^\circ, -0.15^\circ, 0.1, 0.1)}{\delta_x(0)} = 1.35 \quad (6.41)$$

$$\frac{Q(0.15^\circ, -0.15^\circ, 0.125, 0.125)}{Q(0)} = 0.85 \quad (6.42)$$

Figure 6.28 Static performance of the microactuator with straight flexures, and comparison of modified calculation with measured critical dimensions

Figure 6.29 Static performance of the microactuator with folded flexures, and comparison of modified calculation with measured critical dimensions.

With the modified calculation of the microactuator’s performance, the displacement versus the DC driving voltage can be plotted again, as displayed in Figure 6.28. The figure shows that the measured displacements of the microactuator

with straight flexures match the modified calculation displacements based on the measured dimensions. Similarly, the graph of displacement versus driving voltage of the microactuator with folded flexures can be plotted, as shown in Figure 6.29. The modified calculation of the static performance result shows a good match to the measured result of the microactuator with folded flexures.

However, the measured displacements of microactuator with asymmetric-quad flexures were smaller than the theoretical calculation result (shown in Figure 6.6). The measured dimensions show that the quad flexures have positive profiles due to the close-loop structure of the flexure, with the flexure width at the bottom increasing to around 5.6 μm . The comb-finger electrodes show the positive profile which is similar to the other kinds of actuators. Therefore, it can be seen that the stiffness of the asymmetric-quad flexure has increased by a large percentage. Figure 6.30 shows the graph of displacement versus voltage, with comparison of the measured and calculated results. The modified calculation result with actual dimensions shows a good match to the measured curve.

Figure 6.30 Displacement Vs voltage of the microactuator with asymmetric-quad flexures, with comparison of measured results and calculated result

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Besides the errors seen in the static performance, the resonant frequencies of the microactuators between the measured and calculated results are also found to have discrepancies. Table 6.5 summarizes the theoretical simulated (FEM) and measured resonance frequency of the microactuators with straight, folded, and asymmetric-quad flexures. Obviously, the fabrication tolerance of the microactuators had the most contribution to the errors between the measured and theoretical calculated dynamic performance of the devices. In addition, the air damping and quality factors are also affected by the fabrication tolerance.

Table 6.5 Comparison of theoretical and measured resonance frequency

Flexure type	Measured resonance frequency (kHz)	Theoretical resonance frequency (kHz)	Corrected resonance frequency (kHz)
Straight flexure	7.2	8.4	7.16
Folded flexure	5.89	6.28	5.75
Asymmetric-quad flexure	15.85	14.53	16.11

The above experimental performance of the microactuators are characterized without loading the driving components, slider and magnetic read/write heads. As introduced in Chapter 3, the slider and magnetic heads will be mounted onto the movable plate of the SOG microactuator. There are two kinds of commercially available miniaturized slider in the HDD market, one being the 30% form-factor pico slider with a mass of 1.6 mg, the other one being the 20% femto slider with a mass of 0.6 mg [102,152]. The mass of the magnetic heads is generally neglected as it is less than 1% of the slider mass. The moving mass of the comb actuator is calculated to be 22.9 μ g, which is also negligible in comparing to the slider mass.

Therefore, the electromechanical performance of the microactuators loading with slider and magnetic heads can be calculated respectively. According to the deflection

equations in Chapter 3, the displacement of the microactuators is independent of the shuttle mass; thus, the displacements of the microactuators loaded with the sliders are the same as those without loading. However, the resonant frequency of the loaded microactuators would be significantly affected by the mass. Table 6.6 lists the calculated performance of the different types of flexure microactuators loading with sliders.

Table 6.6 Comparison of calculated performance of the microactuators with different types of flexures, with loading the sliders

Flexure type	Measured displacement at 40 V	Calculated displacement at 40 V	Measured f_r w/o loading	Calculated f_r with 1.6mg pico slider	Calculated f_r with 0.6mg femto slider
Straight flexure	0.94 μm	0.96 μm	7.2 kHz	0.87 kHz	1.41 kHz
Folded flexure	1.73 μm	1.61 μm	5.89 kHz	0.70 kHz	1.14 kHz
Asymmetric-quad flexure	0.26 μm	0.24 μm	15.85 kHz	1.89 kHz	3.09 kHz

6.3.3 Air damping analysis and Q-factor estimation

Air damping is related to the surface area of the moving parts and it is very important for micromechanical devices and systems in determining their dynamic performance. Air damping is expected to be reduced to a minimum for a high Q factor in many mechanical systems. The basic mechanisms of air damping for micromechanical structures are squeeze-film air damping and slide-film air damping [153]. In contrast to the vertically driven devices, in which squeeze-film damping is the major source of energy dissipation, viscous drag of the ambient air fluid (slide-film air damping) is the dominant dissipative source in laterally driven structures such as comb drive actuators. Air damping in laterally moving structures has been investigated with two kinds of models: Couette flow where the effective distance is much larger than the distance between the substrate and moving plate [111,154] and

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Stokes flows for more general conditions [148, 153]. In fact, Couette flow is steady Stokes flow between two infinite parallel plates with all the edge effects being ignored, or very thin air film of Stokes flow [155].

In order to investigate the basic features of slide-film damping in lateral driven actuators, a simplified mechanical model is considered: an infinite long plate, immersed in an incompressible viscous fluid moving in a lateral direction at a constant distance from the substrate. The schematic model is shown in Figure 6.31.

Figure 6.31 Slide-film air damping models in terms of velocity profiles, u . (a) Couette flow model, (b) Stokes flow model (one-dimensional)

The characteristic distance of the lateral driven microstructures is defined as

$$\delta = \sqrt{\frac{2\mu}{\rho\omega}} \quad (6.43)$$

where μ is the absolute viscosity of air (or dynamic viscosity), ρ is the density of air, and ω is the natural frequency of the microactuator. For example, the characteristic distance of the microactuator with straight flexures is given as

$$\delta = \sqrt{\frac{2\mu}{\rho\omega}} = \sqrt{\frac{2 \times 1.8 \times 10^{-5}}{1.229 \times 2\pi \times 7200}} = 25.4 \mu\text{m} \quad (6.44)$$

As the microactuators are fabricated with silicon-on-glass wafers and the gap d between the movable comb electrodes and glass substrate is $100 \mu\text{m}$, which is larger

than the characteristic distance δ . Thus, the air damping in the SOG fabricated microactuators models more like the Stokes flow. There are several kinds of models for the air damping in electrostatic actuators which have been studied. One of the widely accepted models was proposed by Y.H. Cho *et al* at University of California, Berkeley [148,149,156].

The air damping involved in the laterally driven microactuators consists of several components: (a) air damping underneath the movable comb plate (inter-plate gap damping), (b) air damping above the movable comb plate, (c) air damping between the sidewall of movable comb electrodes and fixed electrodes (inter-electrode damping). The overall Q factor due to the slide-film damping can be described as:

$$\frac{1}{Q} = \frac{1}{Q_d} + \frac{1}{Q_\delta} + \frac{1}{Q_c} \quad (6.45)$$

Q_d , Q_δ and Q_c are the energy losses due to slide-film damping underneath the comb drive plate, above the top surface of the comb drive plate and between the movable and fixed comb fingers, respectively.

The microactuator with straight flexures was taken as the example during the calculation. Assuming the effective mass of the system is m , and the related the effective area is A_m :

$$m = \rho t A_m = \rho t \left(A_p + \frac{12}{35} A_b + \frac{1}{4} A_t \right) \quad (6.46)$$

where t is the thickness of the movable comb electrodes and A_p , A_b , A_t the top area of the plate (including comb fingers), the supporting flexure beams and the outer connecting truss. For the straight flexure, A_t is zero.

The resonant frequency of the microactuator with straight flexures is given as

$$f_r = \frac{1}{2\pi} \sqrt{\frac{K}{m}} = \frac{\omega}{2\pi} \quad (6.47)$$

where m and ω are the effective mass and natural frequency of the microactuator, respectively. In this case, K is the stiffness of microactuator with three straight flexures.

The Couette flow is the simplest slide film damping model and is the thin and steady Stokes flow with all the edge effects being neglected. The quality factor Q_{cd} due to the Couette flow slide film damping underneath the comb plate is given by Schmitt and Howe as in the equation [157]

$$\frac{1}{Q_{cd}} = \frac{\mu A_q}{d \sqrt{mK}} = \frac{\mu A_q}{d m \omega} \quad (6.48)$$

where A_q is the damping-related effective area of the system, given by

$$A_q = A_p + 0.5 (A_t + A_b) \quad (6.49)$$

The quality factor Q_{sd} due to Stokes flow damping underneath the comb plate can be expressed as

$$\frac{1}{Q_{sd}} = \frac{\mu A_q \beta}{m \omega} \left(\frac{\sinh 2\beta d + \sin 2\beta d}{\cosh 2\beta d - \cos 2\beta d} \right) \quad (6.50)$$

where $\beta = 1/\delta$.

The $Q_{s\delta}$ due to the Stokes flow damping above the comb plate is given as

$$\frac{1}{Q_{s\delta}} = \frac{\mu A_q \beta}{m \omega} \quad (6.51)$$

The quality factor related to damping between the comb fingers is given as

$$\frac{1}{Q_c} = \frac{\mu A_c}{d_c m \omega} \quad (6.52)$$

where A_c is the overlapped area of the comb fingers, and d_c is the comb air gap.

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Therefore, the quality factor due to slide film damping can be rewritten as

$$\frac{1}{Q} = 2\zeta = \frac{\mu}{m\omega} \left(\frac{A_q}{\delta} \left(\frac{\sinh 2\beta d + \sin 2\beta d}{\cosh 2\beta d - \cos 2\beta d} \right) + \frac{A_q}{\delta} + \frac{A_c}{d_c} \right) \quad (6.53)$$

For the microactuator with straight flexures, the average air gap d_c is 3 μm , and the distance d between the moving plate and substrate is 100 μm . The viscosity of air μ is $1.8 \times 10^{-5} \text{ N}\cdot\text{s}/\text{m}^2$ and the density of air ρ is $1.229 \text{ kg}/\text{m}^3$. From the measurement, the resonant frequency of the actuator with straight flexures is 7200 Hz. Therefore, we can have

$$A_p = (1000 \times 50 + 50 \times 4 \times 30 \times 5 + 4 \times 400 \times 10) \times 10^{-12} = 9.6 \times 10^{-8} \text{ m}^2 \quad (6.54)$$

$$A_b = 3 \times 5 \times 400 \times 10^{-12} = 6 \times 10^{-9} \text{ m}^2 \quad (6.55)$$

$$A_M = A_p + \frac{12}{35} A_b = 9.81 \times 10^{-8} \text{ m}^2 \quad (6.56)$$

$$A_q = A_p + 0.5 A_b = 9.9 \times 10^{-8} \text{ m}^2 \quad (6.57)$$

$$A_c = 100 \times 4 \times 10 \times 100 \times 10^{-12} \text{ m}^2 = 4 \times 10^{-7} \text{ m}^2 \quad (6.58)$$

$$m = \rho h A_M = 2330 \times 100 \times 10^{-6} \times 9.81 \times 10^{-8} = 2.29 \times 10^{-8} \text{ kg} \quad (6.59)$$

$$\begin{aligned} \frac{1}{Q} = 2\zeta &= \frac{1}{Q_d} + \frac{1}{Q_s} + \frac{1}{Q_c} = \frac{\mu}{m\omega} \left(\frac{A_q}{\delta} \left(\frac{\sinh 2\beta d + \sin 2\beta d}{\cosh 2\beta d - \cos 2\beta d} \right) + \frac{A_q}{\delta} + \frac{A_c}{d_c} \right) \\ &= \frac{1.8 \times 10^{-5}}{2.29 \times 10^{-8} \times 2\pi \times 7200} \left(\frac{9.9 \times 10^{-8}}{25.4 \times 10^{-6}} \left(\frac{\sinh 7.87 + \sin 7.87}{\cosh 7.87 - \cos 7.87} \right) + \frac{9.9 \times 10^{-8}}{25.4 \times 10^{-6}} + \frac{4 \times 10^{-7}}{3 \times 10^{-6}} \right) \\ &= 0.01737 \times (3.9 \times 10^{-3} + 3.898 \times 10^{-3} + 1.333 \times 10^{-1}) = 0.00245 \end{aligned} \quad (6.60)$$

$$Q = 408.16 \text{ (} f = 7200 \text{ Hz)} \quad (6.61)$$

However, the experimental Q factor from dynamic testing is found to be 67.57, which is much smaller than the theoretical calculated Q value. There are two aspects accounting for the mismatch of Q factors: a) the theoretical model under-estimated the damping effect in the SOG comb-drive microactuators and b) the experimental

Characterization and Discussions

Chapter 6

maximum amplitude of the resonance was not measured correctly or was underestimated.

Firstly, the errors in theoretical model have been reviewed. Besides the air damping model proposed by Cho *et al*, another damping model for the electrostatic comb actuators has been studied by Bao [153] and the equations of the air damping was given as

$$2\zeta = \frac{l}{Q} = \frac{\mu}{m\omega} \left(\frac{A_q}{d} + \frac{A_q}{\delta} + \frac{A_c}{d_c} + \frac{32}{3} l \right) \quad (6.62)$$

where l is the characteristic dimension of the moving structure that can be taken as half the width of the plate. In this model, the first three damping components are the same as those proposed by Cho *et al*; in addition, it includes the air drag on the moving plate in the general conditions. With this air drag component involved in our device, the total damping equation (6.53) can be modified as

$$\begin{aligned} \frac{l}{Q} &= \frac{\mu}{m\omega} \left(\frac{A_q}{\delta} \left(\frac{\sinh 2\beta i + \sin 2\beta i}{\cosh 2\beta o - \cos 2\beta o} \right) + \frac{A_q}{\delta} + \frac{A_c}{d_c} + \frac{32}{3} l \right) \\ &= \frac{1.8 \times 10^{-5}}{2.29 \times 10^{-8} \times 2\pi \times 7200} \left(\frac{9.9 \times 10^{-8}}{25.4 \times 10^{-6}} \left(\frac{\sinh 7.87 + \sin 7.87}{\cosh 7.87 - \cos 7.87} \right) + \frac{9.9 \times 10^{-8}}{25.4 \times 10^{-6}} \right. \\ &\quad \left. + \frac{4 \times 10^{-7}}{3 \times 10^{-6}} + \frac{32}{3} \times 0.5 \times 10^{-3} \right) \quad (6.63) \\ &= 0.01737 \times (3.9 \times 10^{-3} + 3.898 \times 10^{-3} + 1.333 \times 10^{-1} + 0.533 \times 10^{-2}) = 0.00254 \end{aligned}$$

$$Q = 393.2 \quad (6.64)$$

This Q value is still far bigger than the experimental Q value of 67.57, indicating that air damping is greatly underestimated. Similarly, the estimated Q factors of the microactuators with folded flexures and asym-quad flexures have been calculated with the values 323.4 and 843.2 being much larger than the experimental values.

The possible under-estimated air damping in the theoretical model of the comb drive actuators has been reviewed. Equations (6.53) and (6.63) only apply to the one-

dimensional Stokes flow or Couette flow with the infinite plates assumed in the governing equations. Thus, all the finite-size edge effects of the air flow surrounding the plates have been neglected [149,156]. However, our microactuator structures have a finite plate length and area with the additional damping introduced due to the edge effects. The extensive 3-D numerical analysis is required to evaluate the edge effects, finite-size effects, and pressure gradient of the oscillating structures, even for the simple structures such as beams and plates. Therefore, until recently, it was considered impractical to numerically solve the 3-D Stokes flow's equation for such complicated structures like the comb drive arrays. It was believed that the edge effects would be a major source of the discrepancies of 20% between the measured and calculated Q factors.

The study of the 3-D Stokes model with FastStokes Program by Ye *et al* had shown that the difference of the drag force obtained between the 3-D model and two 1-D models (Couette flow and 1-D Stokes flow) is mostly due to the finite-size effects of the oscillating microstructures which are attributed from the side and top of the resonating structures [158,159]. Their case study showed that the 1-D Stokes model has over-estimated the value of Q factor by a factor of two. In addition, their 3-D simulation analysis is thought to well predict the drag force in the comb structures with an error of 10%. The other possible sources of errors are the inaccuracies of the calculation parameters, such as the structure, geometry, air viscosity and air density. These variations could easily contribute to 10% of the absolute errors.

Therefore, by using the 3-D Stokes model and considering all the possible theoretical sources of error, the estimated Q factor in Equation (6.53) and (6.64) of SOG fabricated actuators may have up to 70% error. It means that the modified estimated Q value of straight- flexure microactuator should be around 117.9.

However, this Q factor is still larger than the experimental Q value. The discrepancies must partially come from the inaccuracies of the experimental results.

As discussed in Section 6.2.2, the Q factors are greatly affected by the maximum amplitude of the microactuators at resonance that have been measured in the dynamic testing. There are two possible sources of errors for the inaccurate measurement of the maximum amplitude. Firstly, the maximum amplitude of the microactuators at resonance may exceed $3\text{ }\mu\text{m}$ and cannot be measured since the microactuators have a gap limit of $3\text{ }\mu\text{m}$ between the stopper and movable mass. Secondly, the scanning frequency step may be too big; the very accurate resonant frequency was not found out. Thus, the maximum amplitude did not appear during the measurement. In our dynamic testing, the scanning step of frequency is set to be 10 Hz in the fine scanning range which is actually not fine enough. Thus, most probably, the value of the maximum amplitude was not measured correctly and the quality factor was greatly under-estimated.

To enhance the quality factors of the microactuators, the device should be packaged and tested in vacuum conditions, so as to greatly decrease the air damping in the oscillating microstructures. In addition, the characterization should be performed with good isolation of the ambient vibration. The frequency scanning step should also be fine enough to guarantee the accuracy of the experimental results.

6.4 Chapter Summary

The SOG fabricated microactuators with different types of flexures have been characterized. The device characterization includes the testing of both the static and dynamic performance. The experimental testing showed that with a driving voltage of 40 V, the microactuators with straight flexures, folded flexures and asymmetric-quad

flexures have the displacement of 0.94 μm , 1.73 μm and 0.26 μm , respectively. They have the primary resonant frequency of 7.2 kHz, 5.89 kHz and 15.85 kHz, respectively. The calculated and experimental results of the microactuators with different types of flexures were summarized in Table 6.5 and 6.6.

Further, the experimental performance of the microactuators has shown the discrepancies to the theoretical calculated performance which is based on the originally designed geometry. The characterization indicates that the dimensions of microactuators have deviated from the initially designed dimensions after the fabrication process; this deviation is called fabrication tolerance. Based on the study of the etching profile tolerance and undercut tolerance, their influences on the performance of the comb-drive microactuators have been calculated. The modified calculated performance of the microactuators showed a good match to the experimental results.

The quality factors of the fabricated microactuators have also been characterized, revealing much lower values than those of the theoretical ones estimated by using the classical slide-film damping model. The great discrepancy is basically caused by the under-estimation of air damping using one-dimensional Stokes flow or Couette flow models in the high-aspect-ratio microactuators. Other errors are believed to be from the inaccurate measurement of the maximum amplitude of the microactuators at resonance. The measures to enhance the quality factors of the microactuators have been suggested.

Chapter 7

Conclusions and Future Work

7.1 Conclusions

This thesis has focused on the design, fabrication and characterization of the high-aspect-ratio electrostatic microactuators with vertical configuration. The bulky 3-D comb-drive electrostatic microactuator was designed with high-bandwidth, high-accuracy actuation, good reliability, and low driving voltage. One of their orientated applications is the dual-stage actuator in hard disk drives (HDD). Work has also been done to explore several types of flexure design for the bulky microactuators using both the analytical and numerical methods. The microactuators were fabricated using the silicon-on-glass (SOG) process by using bulk-micromachining technologies, in place of the conventional SOI process. Furthermore, characterization of the fabricated microactuators has been done to verify the theoretical works. By utilizing the theoretical analysis, the performance of the fabricated actuators was well studied.

The proposed linear electrostatic microactuators in this thesis had ultra high-aspect-ratio (depth-to-width ratio of 30:1) 3-D microstructures which were fabricated in 200- μm -thick silicon wafer. The bulk structures enabled a unique design of the electrostatic microactuator, i.e., a vertical configuration of comb-drive working plane to the mounting plane of driving components, in lieu of the conventional parallel or planar arrangement of comb drives to their mounting plane. The vertical arrangement of comb drives provided enough space to arrange several parallel bi-directional comb drives and as a result, remarkably increased the number of comb electrodes. This greatly enhanced the electrostatic driving force under low voltage input. Besides

contributing a larger electric force, the ultra thick microstructures also endowed the microactuators with good mechanical strength and reliability.

Due to the high stiffness effect requirements, the design of micromechanical flexures for the 3-D microactuators has been explored. By using the beam theories, several types of flexures were studied: the straight flexure, folded flexure, symmetric-quad flexure and asymmetric-quad flexure. Due to the limitations of analytical method, numerical method (FEM) was used to compare the stiffness of the four types of flexures. The symmetric-quad flexure was found to have the largest stiffness and stiffness ratio over the discussed flexures. Suitable dimensions of the symmetric-quad and straight flexures have been designed for the 3-D electrostatic microactuators.

The fabrication of the electrostatic microactuators was realized, using the SOG process, instead of the commonly employed SOI wafer process. The SOG wafer was realized through the anodic bonding of glass to silicon wafer with 0.1- μm intermediate layer of metal. Glass wet etching technologies were studied and the masking materials in HF-based wet etching were optimized. High anisotropy deep reactive ion etching (DRIE) technique was developed to obtain the high aspect ratio of the bulk comb drive microactuators.

The performance of the fabricated SOG electrostatic microactuators was characterized. With a driving voltage of 40 V, the microactuators with straight flexures, folded flexures and asymmetric-quad flexures have the displacement of 0.94 μm , 1.73 μm and 0.26 μm , respectively. They have the primary resonant frequency of 7.2 kHz, 5.89 kHz and 15.85 kHz, respectively. The characterization of the microactuators showed various fabrication tolerances of the critical dimensions and their influence on the performance of the microactuators has been studied. On consideration of the influence of the various fabrication tolerances, the modified

calculated performance of the microactuators showed a good match to the experimental results.

7.2 Future Work

As reviewed in Chapter 2, it was believed that the secondary microactuator would be introduced in the high-bandwidth and high-accuracy positioning system of the HDD in the very near future so as to meet the high-capacity and high performance revolutions of the data storage device. The developed SOG microactuators in this thesis would be the good prototypes of such secondary-stage actuators. However, there is still a long way for the microactuators to go before entering into the factory and realizing their commercialization in HDD. There are yet many technical challenges to be solved before these microactuators can be introduced in HDD industry. For instance, the assembly and compatibility to the current VCM actuator, the influence on the head-disk interface (flying height), close-loop servo control, reliability test, failure analysis and control. More research work would be needed to investigate and solve these bottlenecks.

The thesis focused on the design, fabrication and characterization of the novel electrostatic microactuators fabricated with SOG wafers. The proposed possible application of the microactuator is as the dual-stage microactuator for HDD, and the assembly of the microactuator into HDD is a very important part of followed-up future study on this microactuator. We have already considered the feasible assembly process, and also provided some conceptual solutions for the foreseeable problems. The microactuator is to be assembled between the HDD suspension and 20%-size femto-slider (size of 0.85mm×0.7mm×0.23mm) [160]. As shown in Figure 7.1, the microactuator has a very thick glass substrate. The top side of glass substrate with the

Conclusions and Future Work

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surface area of 0.6mm by 2mm will be easily assembled to the end of suspension gimbal. The assembly between the microactuator and femto-slider is more challenging, since the assembly plate of the movable silicon actuator is around 100 μm in width and 1000 μm in length. The assembly plate to the femto-slider is a bit small but still manageable. The bonding glue could be some kind of super glue. Two wires will be used for the electrical interconnect to the metal pads on the glass substrate by soldering. If there is not enough space on the metal pads for soldering, the metal pads pattern will be modified and fabricated again. In this case, the wire bonding will be used to interconnect the different small electrode pads to the big wiring pad.

Figure 7.1 Assembly scheme of the actuator with the HDD suspension and slider

In general, the assembly of the actuator into the HDD will consist of two stages: a laboratory testing stage and a manufactory stage. The assembly process might require some minor or major modifications of the SOG microactuators to fit for the HDD packaging. This also explains the reason why there is a general rule that the

fabrication cost of the device is just one tenth of the packaging and testing cost. In this project, the possible high cost of assembly process comes from two aspects, one is resulted from the device damage during glue assembly, and it could be improved by using a controllable micro motion manipulator stage. Another aspect is from the wiring interconnect, and this can be modified by the redesign of the wiring and electrode pads. In fact, the assembly technique for the microactuator and HDD slider has been reported previously by several research groups, such as Fujitsu, IBM and UC Berkeley [73,78,85]. They managed to assemble the rotary microactuator to the slider body. However, the assembly force between the SOG actuator and HDD slider might be different from common SOI actuators, and it may cause the buckling failure of the flexure structures in the actuators. Therefore, the specially designed high stiffness flexures for the SOG microactuator can prevent the buckling, and this also helps to reduce the assembly cost.

As for the 3-D high-aspect-ratio microactuator itself, a novel design of comb-drive microactuator was proposed and fabricated with bulk silicon-on-glass wafers. The fabrication technologies have been integrated and optimized to obtain the ultra high-aspect-ratio microstructures, which is very meaningful for the study and application of the SOG fabrication process.

The high stiffness of the symmetric-quad and asymmetric-quad flexures has great potentials in the sensors and actuators applications. For example, these flexures can be used in the high-G high-sensitivity accelerometers and gyroscopes [161].

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Appendix A

Analytical Derivation of Folded Flexure Deflection

To analyze the deflection of the folded flexure, the whole flexure can be treated as multi-segment structure: three vertical and two traverse beam units with each beam segment treated as one-end fixed and one-end free beam. Figure A.1 shows the basic flexure segment and the deflection diagram, with one-end fully constrained, and the other-end constrained rotation only, i.e., $\theta_6=0$.

Figure A.1 Schematic diagram of basic segment and deflection diagram of the folded-beam flexure. The dashed line is the beam after deformation

The total displacement at the free end of the flexure unit, u_6 , v_6 , θ_6 can be obtained by combining the displacement of each segment. Here, only the displacement resulting

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from bending is considered in the analysis. Deformation from shear, beam elongation, and beam shortening are neglected.

Figure A.2 Free-body diagram of the folded flexure. (a) segment 12, (b) segment 23, (c) segment 34, (d) segment 45, (e) segment 56

According the free-body diagram, shown in Figure A.2, the force and moment equilibrium equations are

$$\sum F_x = 0: F_{xN} = F_x \quad (\text{A.1})$$

$$\sum F_y = 0: F_{yN} = F_y \quad (\text{A.2})$$

$$\sum M_{(0)} = 0: F_x(l_a + 2l_0) - F_y 2l_b - M + M_1 = 0 \quad (\text{A.3})$$

Thus,

$$M_1 = M - F_x(l_a + 2l_0) + F_y 2l_b \quad (\text{A.4})$$

At the end point 2 of beam segment 12,

$$\sum F_x = 0: F_{x2} = F_x \quad (\text{A.5})$$

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$$\sum F_y = 0: F_{y2} = F_y \quad (\text{A.6})$$

$$\sum M_{(1)} = 0: F_x(l_0 + l_a) - M_2 + M_1 = 0 \quad (\text{A.7})$$

Thus,

$$M_2 = M - F_x l_0 + F_y 2l_b \quad (\text{A.8})$$

The deflection at the end of beam segment 12 is

$$u_2 = \frac{F_x(l_0 + l_a)^3}{3EI_a} - \frac{M_2(l_0 + l_a)^2}{2EI_a} \quad (\text{A.9})$$

$$v_2 = 0 \quad (\text{A.10})$$

$$\theta_2 = \frac{F_x(l_0 + l_a)^2}{2EI_a} - \frac{M_2(l_0 + l_a)}{EI_a} \quad (\text{A.11})$$

At the end point 3 of beam segment 23,

$$\sum F_x = 0: F_{x3} = F_x \quad (\text{A.12})$$

$$\sum F_y = 0: F_{y3} = F_y \quad (\text{A.13})$$

$$\sum M_{(2)} = 0: M_2 - M_3 - F_{y3}l_b = 0 \quad (\text{A.14})$$

Thus,

$$M_3 = M - F_x l_0 + F_y l_b \quad (\text{A.15})$$

The relative deflection at the end point 3 of beam segment 23 is

$$u_{3r} = 0 \quad (\text{A.16})$$

$$v_{3r} = -\frac{F_y l_b^3}{3EI_b} - \frac{M_3 l_b^2}{2EI_b} \quad (\text{A.17})$$

$$\theta_{3r} = -\frac{F_y l_b^2}{2EI_b} - \frac{M_3 l_b}{EI_b} \quad (\text{A.18})$$

The rigid-body displacement of the beam segment 23 is given by

$$u_{3b} = u_2 \quad (\text{A.19})$$

$$v_{3b} = v_2 + l_b \theta_2 \quad (\text{A.20})$$

$$\theta_{3b} = \theta_2 \quad (\text{A.21})$$

Therefore, the absolute displacement of beam segment 23 is given by

$$u_3 = u_{3r} + u_{3b} \quad (\text{A.22})$$

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$$v_3 = v_{3r} + v_{3b} \quad (\text{A.23})$$

$$\theta_3 = \theta_{3r} + \theta_{3b} \quad (\text{A.24})$$

Combining the equations (A.16) ~ (A.24), we get

$$u_3 = \frac{F_x(l_0 + l_a)^3}{3EI_a} - \frac{M_2(l_0 + l_a)^2}{2EI_a} \quad (\text{A.25})$$

$$v_3 = \frac{F_x(l_0 + l_a)^2 l_b}{2EI_a} - \frac{M_2(l_0 + l_a)l_b}{EI_a} - \frac{F_y l_b^3}{3EI_b} - \frac{M_3 l_b^2}{2EI_b} \quad (\text{A.26})$$

$$\theta_3 = \frac{F_x(l_0 + l_a)^2}{2EI_a} - \frac{M_2(l_0 + l_a)}{EI_a} - \frac{F_y l_b^2}{2EI_b} - \frac{M_3 l_b}{EI_b} \quad (\text{A.27})$$

At the end point 4 of beam segment 34,

$$\sum F_x = 0: F_{x4} = F_x \quad (\text{A.28})$$

$$\sum F_y = 0: F_{y4} = F_y \quad (\text{A.29})$$

$$\sum M_{(3)} = 0: M_3 - M_4 - F_{x4}l_a = 0 \quad (\text{A.30})$$

Thus,

$$M_4 = M - F_x(l_a + l_0) + F_y l_b \quad (\text{A.31})$$

The relative deflection at the end point 4 of beam segment 34 is

$$u_{4r} = \frac{F_x l_a^3}{3EI_a} + \frac{M_4 l_a^2}{2EI_a} \quad (\text{A.32})$$

$$v_{4r} = 0 \quad (\text{A.33})$$

$$\theta_{4r} = -\frac{F_x l_a^2}{2EI_a} - \frac{M_4 l_a}{EI_a} \quad (\text{A.34})$$

The rigid-body displacement of the beam segment 23 is given by

$$u_{4b} = u_3 - l_a \theta_3 \quad (\text{A.35})$$

$$v_{4b} = v_3 \quad (\text{A.36})$$

$$\theta_{4b} = \theta_3 \quad (\text{A.37})$$

Therefore, the absolute displacement of beam segment 34 is given by

$$u_4 = u_{4r} + u_{4b} \quad (\text{A.38})$$

$$v_4 = v_{4r} + v_{4b} \quad (\text{A.39})$$

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$$\theta_4 = \theta_{4r} + \theta_{4b} \quad (\text{A.40})$$

Substituting equations (A.32) ~ (A.37) into equations (A.38) ~ (A.40), we get

$$u_4 = \frac{F_x(l_0 + l_a)^3}{3EI_a} - \frac{M_2(l_0 + l_a)^2}{2EI_a} - \frac{F_x(l_0 + l_a)^2 l_a}{2EI_a} + \frac{M_2(l_0 + l_a)l_a}{EI_a} - \frac{F_y l_b^2 l_a}{2EI_b} + \frac{M_3 l_b l_a}{EI_b} + \frac{F_x l_a^3}{3EI_a} + \frac{M_4 l_a^2}{2EI_a} \quad (\text{A.41})$$

$$v_3 = \frac{F_x(l_0 + l_a)^2 l_b}{2EI_a} - \frac{M_2(l_0 + l_a)l_b}{EI_a} - \frac{F_y l_b^3}{3EI_b} - \frac{M_3 l_b^2}{2EI_b} \quad (\text{A.42})$$

$$\theta_4 = \frac{F_x(l_0 + l_a)^2}{2EI_a} - \frac{M_2(l_0 + l_a)}{EI_a} - \frac{F_y l_b^2}{2EI_b} - \frac{M_3 l_b}{EI_b} - \frac{F_x l_a^2}{2EI_a} - \frac{M_4 l_a}{EI_a} \quad (\text{A.43})$$

At the end point 5 of beam segment 45,

$$\sum F_x = 0: F_{x5} = F_x \quad (\text{A.44})$$

$$\sum F_y = 0: F_{y5} = F_y \quad (\text{A.45})$$

$$\sum M_{(4)} = 0: M_4 - M_5 - F_y l_b = 0 \quad (\text{A.46})$$

Thus,

$$M_5 = M - F_x(l_a + l_0) \quad (\text{A.47})$$

The relative deflection at the end point 5 of beam segment 45 is

$$u_{5r} = 0 \quad (\text{A.48})$$

$$v_{5r} = -\frac{F_y l_b^3}{3EI_b} - \frac{M_5 l_b^2}{2EI_b} \quad (\text{A.49})$$

$$\theta_{5r} = -\frac{F_y l_b^2}{2EI_b} - \frac{M_5 l_b}{EI_b} \quad (\text{A.50})$$

The rigid-body displacement of the beam segment 23 is given by

$$u_{5b} = u_4 \quad (\text{A.51})$$

$$v_{5b} = v_4 + l_b \theta_4 \quad (\text{A.52})$$

$$\theta_{5b} = \theta_4 \quad (\text{A.53})$$

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Therefore, the absolute displacement of beam segment 34 is given by

$$u_5 = u_{5r} + u_{5b} = u_4 \quad (\text{A.54})$$

$$v_5 = v_{5r} + v_{5b} = v_4 + l_b \theta_4 - \frac{F_y l_b^3}{3EI_b} - \frac{M_5 l_b^2}{2EI_b} \quad (\text{A.55})$$

$$\theta_5 = \theta_{5r} + \theta_{5b} = \theta_4 - \frac{F_y l_b^2}{2EI_b} - \frac{M_5 l_b}{EI_b} \quad (\text{A.56})$$

At the end point 6 of beam segment 56, the loads are F_x , F_y and M . The relative deflection of beam segment 56 is

$$u_{6r} = \frac{F_x (l_0 + l_a)^3}{3EI_a} - \frac{M (l_0 + l_a)^2}{2EI_a} \quad (\text{A.57})$$

$$v_{6r} = 0 \quad (\text{A.58})$$

$$\theta_{6r} = \frac{F_x (l_0 + l_a)^2}{2EI_a} - \frac{M (l_0 + l_a)}{EI_a} \quad (\text{A.59})$$

The rigid-body displacement of the beam segment 23 is given by

$$u_{6b} = u_5 + (l_a + l_0) \theta_5 \quad (\text{A.60})$$

$$v_{6b} = v_5 \quad (\text{A.61})$$

$$\theta_{6b} = \theta_5 \quad (\text{A.62})$$

Therefore, the absolute displacement of beam segment 56 is given by

$$u_6 = u_{6r} + u_{6b} = u_5 + (l_0 + l_a) \theta_5 + \frac{F_x (l_0 + l_a)^3}{3EI_a} - \frac{M (l_0 + l_a)^2}{2EI_a} \quad (\text{A.63})$$

$$v_6 = v_{6r} + v_{6b} = v_5 \quad (\text{A.64})$$

$$\theta_6 = \theta_{6r} + \theta_{6b} = \theta_5 + \frac{F_x (l_0 + l_a)^2}{2EI_a} - \frac{M (l_0 + l_a)}{EI_a} \quad (\text{A.65})$$

Substituting the equations into equation (A.63) ~ (A.65), we get the deflection at the end point 6 of the folded flexure as

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$$\begin{aligned}
 u_6 = & F_x \left(\frac{8l_0^3}{3EI_a} + \frac{l_a^3}{EI_a} + \frac{4l_a^2l_0}{EI_a} + \frac{6l_al_0^2}{EI_a} + \frac{l_a^2l_b}{EI_b} \right) - F_y \left(\frac{(l_a + 4l_0)l_b^2}{2EI_b} + \frac{3l_a^2l_b}{2EI_a} + \frac{3l_0^2l_b}{EI_a} \right. \\
 & \left. + \frac{5l_0l_al_b}{EI_a} \right) + M \left(-\frac{(l_a + 2l_0)l_b}{EI_b} - \frac{2(l_0 + l_a)^2}{EI_a} + \frac{l_a^2}{2EI_a} \right)
 \end{aligned}
 \tag{A.66}$$

$$\begin{aligned}
 v_6 = & F_x \left(\frac{3l_0^2l_b}{EI_a} + \frac{3l_a^2l_b}{2EI_a} + \frac{5l_0l_al_b}{EI_a} + \frac{(l_a + 4l_0)l_b^2}{2EI_b} \right) - F_y \left(\frac{8l_b^3}{3EI_b} + \frac{(5l_a + 4l_0)l_b^2}{EI_a} \right) \\
 & + M \left(-\frac{(3l_a + 2l_0)l_b}{EI_a} - \frac{2l_b^2}{EI_b} \right)
 \end{aligned}
 \tag{A.67}$$

$$\begin{aligned}
 \theta_6 = & F_x \left(\frac{2l_0^2}{EI_a} + \frac{3l_a^2}{2EI_a} + \frac{4l_0l_a}{EI_a} + \frac{(l_a + 2l_0)l_b}{EI_b} \right) - F_y \left(\frac{2l_b^2}{EI_b} + \frac{(3l_a + 2l_0)l_b}{EI_a} \right) \\
 & + M \left(-\frac{(3l_a + 2l_0)}{EI_a} - \frac{2l_b}{EI_b} \right)
 \end{aligned}
 \tag{A.68}$$

Appendix B

Analytical Derivation of asymmetric-quad flexure deflection

To analyze the deflection of the asymmetric-quad flexure, the whole flexure can be treated as a multi-segmented structure: three vertical and two traverse beam units. Each beam segment can be treated as a one-end fixed and one-end free beam structure. Figure B.1 shows the basic flexure segment and the deflection diagram, with one-end fully constrained and the other-end constrained in the rotational direction, i.e., $\theta_6=0$.

Figure B.1 Schematic diagram of basic segment and deflection diagram of the asymmetric-quad flexure. The dashed line is the beam after deformation

The total displacement at the free end of the flexure unit, u_6 , v_6 , θ_6 can be obtained by combining the displacement of each segment with solving the complicated force-displacement equations. Here, only the displacement coming from

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bending will be considered in the analysis; deformation from shear, beam elongation, and beam shortening will be ignored.

Figure B.2 Free-body diagram of the asymmetric-quad flexure. (a) segment 12, (b) segment 23, (c) segment 34, (d) segment 25, (e) segment 54, (f) segment 46

For the whole flexure, there are three external loads: F_x , F_y , and M . For the whole flexure, the force and moment equilibrium equation at the end point 1 are

$$\sum F_x = 0: F_{xN} = F_x \quad (\text{B.1})$$

$$\sum F_y = 0: F_{yN} = F_y \quad (\text{B.2})$$

$$\sum M_{(1)} = 0: F_x(l_a + 2l_0) - F_y l_b - M + M_1 = 0 \quad (\text{B.3})$$

Thus,

$$M_1 = M - F_x(l_a + 2l_0) + F_y l_b \quad (\text{B.4})$$

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According to the free-body diagram in Figure B.2, the force and moment equilibrium equations at the end point 2 of beam segment 12 are,

$$\sum F_x = 0: F_{x2} = F_x \quad (B.5)$$

$$\sum F_y = 0: F_{y2} = F_y \quad (B.6)$$

$$\sum M_{(1)} = 0: F_x l_0 - M_2 + M_1 = 0 \quad (B.7)$$

Thus,

$$M_2 = M - F_x(l_0 + l_a) + F_y l_b \quad (B.8)$$

The displacements at end point 2 are then obtained

$$u_2 = \frac{F_x l_0^3}{3EI_a} - \frac{M_2 l_0^2}{2EI_a} \quad (B.9)$$

$$v_2 = 0 \quad (B.10)$$

$$\theta_2 = \frac{F_x l_0^2}{2EI_a} - \frac{M_2 l_0}{EI_a} \quad (B.11)$$

As the asymmetric-quad flexure has a loop structure with two joints at the point 2 and point 4, the displacements at point 4 with beam segment 34 and segment 54 should be the same. At the end point 3 with beam segment 34,

$$\sum F_x = 0: F_{x3} = F_x \quad (B.11)$$

$$\sum F_y = 0: F_{y3} = F_y \quad (B.12)$$

$$\sum M_{(3)} = 0: M_4 - F_{y4} l_b - M_3 = 0 \quad (B.13)$$

Thus,

$$M_3 = M_4 - F_{y4} l_b \quad (B.14)$$

The displacements at point 3 of beam segment 23 are then given as

$$u_{3r} = \frac{F_{x4} l_a^3}{3EI_a} + \frac{M_3 l_a^2}{2EI_a} \quad (B.15)$$

$$v_{3r} = 0 \quad (B.15)$$

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$$\theta_{3r} = \frac{F_{x4}l_a^2}{2EI_a} - \frac{M_3l_a}{EI_a} \quad (B.16)$$

The rigid-body displacements at the point 3 are

$$u_{3b} = u_2 + l_a\theta_2 \quad (B.17)$$

$$v_{3b} = v_2 \quad (B.18)$$

$$\theta_{3b} = \theta_2 \quad (B.19)$$

Therefore, the absolute displacements at point 3 are

$$u_3 = u_{3r} + u_{3b} = \frac{F_{x4}l_a^3}{3EI_a} + \frac{M_3l_a^2}{2EI_a} + \frac{F_{x4}l_0^3}{3EI_a} - \frac{M_2l_0^2}{2EI_a} + \frac{F_{x4}l_0^2l_a}{2EI_a} - \frac{M_2l_0l_a}{EI_a} \quad (B.20)$$

$$v_3 = v_{3r} + v_{3b} = 0 \quad (B.21)$$

$$\theta_3 = \theta_{3r} + \theta_{3b} = \frac{F_{x4}l_a^2}{2EI_a} - \frac{M_3l_a}{EI_a} + \frac{F_{x4}l_0^2}{2EI_a} - \frac{M_2l_0}{EI_a} \quad (B.22)$$

The displacements at the end point 4 of beam segment 34 are

$$u_{4r} = 0 \quad (B.23)$$

$$v_{4r} = -\frac{F_{y4}l_b^3}{3EI_b} + \frac{M_4l_b^2}{2EI_b} \quad (B.24)$$

$$\theta_{4r} = -\frac{F_{y4}l_b^2}{2EI_b} + \frac{M_4l_b}{EI_b} \quad (B.25)$$

The rigid-body displacements at the end point 4 of beam segment 34 are

$$u_{4b} = u_3 \quad (B.26)$$

$$v_{4b} = v_3 + l_b\theta_3 \quad (B.27)$$

$$\theta_{4b} = \theta_3 \quad (B.28)$$

Therefore, the displacements at point 4 from the left beam segment 34 can be obtained as

$$u_{4L} = u_{4r} + u_{4b} = \frac{F_{x4}l_a^3}{3EI_a} + \frac{M_3l_a^2}{2EI_a} + \frac{F_{x4}l_0^3}{3EI_a} - \frac{M_2l_0^2}{2EI_a} + \frac{F_{x4}l_0^2l_a}{2EI_a} - \frac{M_2l_0l_a}{EI_a} \quad (B.29)$$

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$$v_{4L} = v_{4r} + v_{4b} = -\frac{F_{y4}l_b^3}{3EI_b} + \frac{M_4l_b^2}{2EI_b} + \frac{F_{x4}l_a^2l_b}{2EI_a} - \frac{M_3l_al_b}{EI_a} + \frac{F_xl_0^2l_b}{2EI_a} - \frac{M_2l_0l_b}{EI_a} \quad (B.30)$$

$$\theta_{4L} = \theta_{4r} + \theta_{4b} = -\frac{F_{y4}l_b^2}{2EI_b} + \frac{M_4l_b}{EI_b} + \frac{F_{x4}l_a^2}{2EI_a} - \frac{M_3l_a}{EI_a} + \frac{F_xl_0^2}{2EI_a} - \frac{M_2l_0}{EI_a} \quad (B.31)$$

In the same way, the displacements at the end point 4 of right beam segment 54 can also be obtained. The force and moment equilibrium equations of beam segment 56 at the end point 5 are

$$\sum F_x = 0: F_{x5} = F_x - F_{x4} \quad (B.32)$$

$$\sum F_y = 0: F_{y5} = F_y - F_{y4} \quad (B.33)$$

$$\sum M_{(5)} = 0: F_x(l_a + l_0) - F_{x4}l_a - M - M_4 + M_5 = 0 \quad (B.34)$$

Thus,

$$M_5 = M + M_4 + F_{x4}l_a - F_x(l_a + l_0) \quad (B.35)$$

The displacements at the end point 5 of beam segment 25 are

$$u_{5r} = 0 \quad (B.36)$$

$$v_{5r} = -\frac{F_{y5}l_b^3}{3EI_b} - \frac{M_5l_b^2}{2EI_b} \quad (B.37)$$

$$\theta_{5r} = -\frac{F_{y5}l_b^2}{2EI_b} - \frac{M_5l_b}{EI_b} \quad (B.38)$$

The rigid-body displacements at the end point 5 are

$$u_{5b} = u_2 \quad (B.39)$$

$$v_{5b} = v_2 + l_b\theta_2 \quad (B.40)$$

$$\theta_{5b} = \theta_2 \quad (B.41)$$

Thus, the absolute displacements at the end point 5 are

$$u_5 = u_{5r} + u_{5b} = \frac{F_xl_0^3}{3EI_a} - \frac{M_2l_0^2}{2EI_a} \quad (B.42)$$

$$v_5 = v_{5r} + v_{5b} = -\frac{F_{y5}l_b^3}{3EI_b} - \frac{M_5l_b^2}{2EI_b} + \frac{F_xl_0^2l_b}{2EI_a} - \frac{M_2l_0l_b}{EI_a} \quad (B.43)$$

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$$\theta_5 = \theta_{5r} + \theta_{5b} = -\frac{F_{y5}l_b^2}{2EI_b} - \frac{M_5l_b}{EI_b} + \frac{F_xl_0^2}{2EI_a} - \frac{M_2l_0}{EI_a} \quad (B.44)$$

For the beam segment 54, the force and moment equilibrium equations are

$$\sum F_x = 0: F_{x4}' = F_x - F_{x4} \quad (B.45)$$

$$\sum F_y = 0: F_{y4}' = F_y - F_{y4} \quad (B.46)$$

$$\sum M_{(5)} = 0: F_xl_0 - M - M_4 + M_4' = 0 \quad (B.47)$$

Thus,

$$M_4' = M + M_4 - F_xl_0 \quad (B.48)$$

The displacements at the end point 4 of beam segment 54 are

$$u_{4r} = \frac{F_{x4}'l_a^3}{3EI_a} - \frac{M_4'l_a^2}{2EI_a} \quad (B.49)$$

$$v_{4r} = 0 \quad (B.50)$$

$$\theta_{4r} = \frac{F_{x4}'l_a^2}{2EI_a} - \frac{M_4'l_a}{EI_a} \quad (B.51)$$

The rigid-body displacements of the end point 4 of beam segment 54 are

$$u_{4b} = u_5 + l_a\theta_5 \quad (B.52)$$

$$v_{4b} = v_5 \quad (B.53)$$

$$\theta_{4b} = \theta_5 \quad (B.54)$$

Therefore, the displacements at the end point 4 of right beam segment 54 are given as

$$\begin{aligned} u_{4R} &= u_{4r} + u_{4b} \\ &= \frac{F_{x4}'l_a^3}{3EI_a} - \frac{M_4'l_a^2}{2EI_a} + \frac{F_xl_0^3}{3EI_a} - \frac{M_2l_0^2l_a}{2EI_a} - \frac{F_{y5}l_b^2l_a}{2EI_b} - \frac{M_5l_b l_a}{EI_b} + \frac{F_xl_0^2l_a}{2EI_a} - \frac{M_2l_0l_a}{EI_a} \end{aligned} \quad (B.55)$$

$$v_{4R} = v_{4r} + v_{4b} = -\frac{F_{y5}l_b^3}{3EI_b} - \frac{M_5l_b^2}{2EI_b} + \frac{F_xl_0^2l_b}{2EI_a} - \frac{M_2l_0l_b}{EI_a} \quad (B.56)$$

$$\theta_{4R} = \theta_{4r} + \theta_{4b} = \frac{F_{x4}'l_a^2}{2EI_a} - \frac{M_4'l_a}{EI_a} - \frac{F_{y5}l_b^2}{2EI_b} - \frac{M_5l_b}{EI_b} + \frac{F_xl_0^2}{2EI_a} - \frac{M_2l_0}{EI_a} \quad (B.57)$$

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Since the displacements calculated from beam segment 34 and beam segment 54 should be the same, we can get the following equilibrium equations

$$u_{4L} = u_{4R} \quad (\text{B.58})$$

$$v_{4L} = v_{4R} \quad (\text{B.59})$$

$$\theta_{4L} = \theta_{4R} \quad (\text{B.60})$$

i.e.,

$$\begin{aligned} \frac{F_{x4}l_a^3}{3EI_a} + \frac{M_3l_a^2}{2EI_a} + \frac{F_xl_0^3}{3EI_a} - \frac{M_2l_0^2}{2EI_a} + \frac{F_xl_0^2l_a}{2EI_a} - \frac{M_2l_0l_a}{EI_a} &= \frac{F_{x4}l_a^3}{3EI_a} - \frac{M_4l_a^2}{2EI_a} \\ + \frac{F_xl_0^3}{3EI_a} - \frac{M_2l_0^2l_a}{2EI_a} - \frac{F_{y5}l_b^2l_a}{2EI_b} - \frac{M_5l_b l_a}{EI_b} + \frac{F_xl_0^2l_a}{2EI_a} - \frac{M_2l_0l_a}{EI_a} \end{aligned} \quad (\text{B.61})$$

$$\begin{aligned} -\frac{F_{y4}l_b^3}{3EI_b} + \frac{M_4l_b^2}{2EI_b} + \frac{F_{x4}l_a^2l_b}{2EI_a} - \frac{M_3l_a l_b}{EI_a} + \frac{F_xl_0^2l_b}{2EI_a} - \frac{M_2l_0l_b}{EI_a} &= -\frac{F_{y5}l_b^3}{3EI_b} \\ -\frac{M_5l_b^2}{2EI_b} + \frac{F_xl_0^2l_b}{2EI_a} - \frac{M_2l_0l_b}{EI_a} \end{aligned} \quad (\text{B.62})$$

$$\begin{aligned} -\frac{F_{y4}l_b^2}{2EI_b} + \frac{M_4l_b}{EI_b} + \frac{F_{x4}l_a^2}{2EI_a} - \frac{M_3l_a}{EI_a} + \frac{F_xl_0^2}{2EI_a} - \frac{M_2l_0}{EI_a} &= \frac{F_{x4}l_a^2}{2EI_a} - \frac{M_4l_a}{EI_a} \\ -\frac{F_{y5}l_b^2}{2EI_b} - \frac{M_5l_b}{EI_b} + \frac{F_xl_0^2}{2EI_a} - \frac{M_2l_0}{EI_a} \end{aligned} \quad (\text{B.63})$$

By substituting the previous equations into equations (B.61) ~ (B.63) and simplifying, equations (B.61) ~ (B.63) can be re-written as

$$\begin{aligned} F_x \left(\frac{l_0^3}{3EI_a} + \frac{l_0^2l_a}{2EI_a} + \frac{l_0l_b(l_a + l_0)}{EI_b} \right) - F_y \frac{l_0l_b^2}{2EI_b} - M \left(\frac{l_0^2}{2EI_a} + \frac{l_0l_b}{EI_b} \right) - F_{x4} \left(\frac{2l_0^3}{3EI_a} + \frac{l_0^2l_b}{EI_b} \right) \\ + F_{y4} \left(\frac{l_0^2l_b}{2EI_a} + \frac{l_0l_b^2}{2EI_b} \right) - M_4 \left(\frac{l_0^2}{EI_a} + \frac{l_0l_b}{EI_b} \right) = 0 \end{aligned} \quad (\text{B.64})$$

$$\begin{aligned} F_x \frac{l_b^2(l_0 + l_a)}{2EI_b} - F_y \frac{l_b^3}{3EI_b} - M \frac{l_b^2}{2EI_b} - F_{x4} \left(\frac{l_0^2l_b}{2EI_a} + \frac{l_0l_b^2}{2EI_b} \right) + F_{y4} \left(\frac{l_0l_b^2}{EI_a} + \frac{2l_b^3}{3EI_b} \right) \\ - M_4 \left(\frac{l_0l_b}{EI_a} + \frac{l_b^2}{EI_b} \right) = 0 \end{aligned} \quad (\text{B.65})$$

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$$\begin{aligned}
& F_x \left(\frac{l_0^2}{2EI_a} + \frac{l_0 l_a}{EI_a} + \frac{l_b(l_a + l_0)}{EI_b} \right) - F_y \frac{l_b^2}{2EI_b} - M \left(\frac{l_0}{EI_a} + \frac{l_b}{EI_b} \right) - F_{x4} \left(\frac{l_0^2}{EI_a} + \frac{l_0 l_b}{EI_b} \right) \\
& + F_{y4} \left(\frac{l_0 l_b}{EI_a} + \frac{l_b^2}{EI_b} \right) - M_4 \left(\frac{2l_0}{EI_a} + \frac{2l_b}{EI_b} \right) = 0
\end{aligned} \tag{B.66}$$

By solving equations (B.64) ~ (B.66), F_{x4} , F_{y4} and M_4 can be obtained.

The displacements at the end point 6 of beam segment 56 are then given as

$$u_{6r} = \frac{F_x l_0^3}{3EI_a} - \frac{M l_0^2}{2EI_a} \tag{B.67}$$

$$v_{6r} = 0 \tag{B.68}$$

$$\theta_{6r} = \frac{F_x l_0^2}{2EI_a} - \frac{M l_0}{EI_a} \tag{B.69}$$

The rigid-body deflection of beam segment 56 can be derived following either segment 54 or segment 34. For instance, the rigid-body displacements of beam segment 56 following the segment 54 are

$$u_{6b} = u_{4R} + l_0 \theta_{4R} \tag{B.70}$$

$$v_{6b} = v_{4R} \tag{B.71}$$

$$\theta_{6b} = \theta_{4R} \tag{B.72}$$

Therefore, the displacements at the end point 6 of beam segment 56 are given as

$$u_6 = u_{6r} + u_{6b} = \frac{F_x l_0^3}{3EI_a} - \frac{M l_0^2}{2EI_a} + u_{4R} + l_0 \theta_{4R} \tag{B.73}$$

$$v_6 = v_{6r} + v_{6b} = v_{4R} \tag{B.74}$$

$$\theta_6 = \theta_{6r} + \theta_{6b} = \frac{F_x l_0^2}{2EI_a} - \frac{M l_0}{EI_a} + \theta_{4R} \tag{B.75}$$

By simplifying, equations (B.73) ~ (B.75) are re-written as

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$$u_6 = F_x \left(\frac{7l_0^3}{6EI_a} + \frac{l_0 l_a^2}{2EI_a} + \frac{l_a l_0 (l_a + l_0)}{EI_a} + \frac{3l_a^2 (l_a + l_0)}{2EI_a} \right) - F_y \left(\frac{3l_a^2 l_b}{2EI_a} + \frac{l_b l_a l_0}{EI_a} \right) - M \left(\frac{2l_a^2}{EI_a} + \frac{l_a l_0}{EI_a} \right) \\ + F_{x4} \left(\frac{l_0^3}{3EI_a} + \frac{l_0^2 l_a}{2EI_a} \right) - F_{y4} \left(\frac{l_a^2 l_b}{2EI_a} + \frac{l_0 l_a l_b}{EI_a} + \frac{l_a l_b^2}{2EI_b} \right) + M_4 \left(\frac{l_0^2}{2EI_a} + \frac{l_a l_0}{EI_a} + \frac{l_a l_b}{EI_b} \right) \quad (\text{B.76})$$

$$v_6 = F_x \left(\frac{l_a^2 l_b}{2EI_a} + \frac{l_a l_b (l_a + l_0)}{EI_a} \right) - F_y \frac{l_a l_b^2}{EI_a} - M \frac{l_a l_b}{EI_a} + F_{x4} \frac{l_0^2 l_b}{2EI_a} - F_{y4} \left(\frac{l_b^3}{3EI_b} + \frac{l_0 l_b^2}{EI_a} \right) \\ + M_4 \left(\frac{l_b^2}{EI_b} + \frac{l_0 l_b}{EI_a} \right) \quad (\text{B.77})$$

$$\theta_6 = F_x \left(\frac{l_0^2}{EI_a} + \frac{l_a (l_a + l_0)}{EI_a} \right) - F_y \frac{l_a l_b}{EI_b} - M \frac{2l_a}{EI_a} + F_{x4} \frac{l_0^2}{2EI_a} - F_{y4} \left(\frac{l_b^2}{2EI_b} + \frac{l_0 l_b}{EI_a} \right) \\ + M_4 \left(\frac{l_0}{EI_a} + \frac{l_b}{EI_b} \right) \quad (\text{B.78})$$

By substituting the expressions of F_{x4} , F_{y4} and M_4 into equations (B.76) ~ (B.78), the displacements u_6 , v_6 and θ_6 at the end point 6 can be obtained.

$$u_6 = l_a^3 (F_x (4r_2^4 \alpha^3 + 24r_1 r_2^3 \alpha^3 + r_2 \alpha + 48r_1^4 r_2^3 \alpha^3 + 88r_1^3 r_2^3 \alpha^3 + 3r_1 + 48r_1^4 + 67r_1 r_2^2 \alpha^2 + 72r_1^3 \\ + 208r_1^4 r_2 \alpha + 328r_1^3 r_2 \alpha + 36r_1^2 + 34r_1 r_2 \alpha + 5r_2^2 \alpha^2 + 208r_1^4 r_2^2 \alpha^2 + 60r_1^2 r_2^3 \alpha^3 + 180r_1^2 r_2 \alpha \\ + 204r_1^2 r_2^2 \alpha^2 + 344r_1^3 r_2^2 \alpha^2) + F_y (12r_1 r_2^4 \alpha^4 + 60r_1^2 r_2^4 \alpha^4 + 204r_1^2 r_2^3 \alpha^3 + 3r_2^4 \alpha^3 + 18r_1 r_2 \alpha \\ - 114r_1^2 r_2^4 \alpha^3 + 84r_1 r_2^2 \alpha^2 - 126r_1^3 r_2^4 \alpha^3 + 78r_1 r_2^3 \alpha^3 - 15r_1 r_2^4 \alpha^3 + 258r_1^3 r_2^3 \alpha^3 - 492r_1^3 r_2^3 \alpha^2 \\ - 108r_1^3 r_2 - 54r_1 r_2^3 \alpha^2 + 72r_1^3 r_2^4 \alpha^4 + 6r_2^2 \alpha + 9r_2^3 \alpha^2 + 54r_1^3 r_2 \alpha - 396r_1^2 r_2^3 \alpha^2 - 72r_1^2 r_2 \\ + 240r_1^3 r_2^2 \alpha^2 - 474r_1^3 r_2^2 \alpha + 180r_1^2 r_2^2 \alpha^2 + 36r_1^2 r_2 \alpha - 27r_1 r_2^2 \alpha - 354r_1^2 r_2^2 \alpha) / (24EI_a (r_2 \alpha \\ + 13r_1 r_2^2 \alpha^2 + 3r_1 + 3r_1 r_2^3 \alpha^3 + r_2^3 \alpha^3 + 13r_1 r_2 \alpha + 2r_2^2 \alpha^2)) \quad (\text{B.79})$$

$$v_6 = l_b l_a^2 (F_x (18r_1^2 + 18r_1^3 + 18r_1^3 r_2^2 \alpha^2 + 66r_1^2 r_2 \alpha + 60r_1^3 r_2 \alpha + 6r_1 r_2 \alpha + 24r_1^2 r_2^2 \alpha^2 \\ + 6r_1 r_2^2 \alpha^2) + F_y (18r_1^2 r_2^3 \alpha^3 + 21r_1 r_2^3 \alpha^3 + 60r_1^2 r_2^2 \alpha^2 + 3r_1 r_2^4 \alpha^3 + 18r_1^2 r_2 \alpha - 18r_1 r_2^2 \alpha + r_2^4 \alpha^3 \\ - 36r_1^2 r_2^3 \alpha^2 + r_2^2 \alpha^2 - 120r_1^2 r_2^2 \alpha - 17r_1 r_2^3 \alpha^2 - 36r_1^2 r_2 + 15r_1 r_2^2 \alpha^2)) / (12EI_a (3r_1 + r_2 \alpha \\ + 3r_1 r_2^2 \alpha^2 + r_2^2 \alpha^2 + 10r_1 r_2 \alpha)) \quad (\text{B.80})$$

$$M_6 = l_a (F_x (6r_1^2 + 3r_1 + r_2 \alpha + r_2^2 \alpha^2 + 20r_1^2 r_2 \alpha + 12r_1 r_2 \alpha + 5r_1 r_2^2 \alpha^2 + 6r_1^2 r_2^2 \alpha^2) + F_y (-r_2^2 \alpha \\ - 3r_1 r_2^3 \alpha^3 - r_2^3 \alpha^2 - 3r_1 r_2 \alpha - 10r_1 r_2^2 \alpha^2)) / (6r_1 r_2^2 \alpha^2 + 6r_1 + 2r_2^2 \alpha^2 + 2r_2 \alpha + 20r_1 r_2 \alpha) \quad (\text{B.81})$$

where $\alpha = I_a / I_b = (w_a / w_b)^3$, $r_1 = l_0 / l_a$, $r_2 = l_b / l_a$.

Appendix C

Analytical Derivation of the Symmetric-Quad flexure Deflection

Similar to the asymmetric-quad flexure, the symmetric quadrate-frame flexure is further studied. The only difference in the asymmetric-quad flexure is that the extended ends are located in the middle of the quad-loop such that the flexure is symmetrical.

Figure C.1 Basic segment and deflection diagram of the asymmetric-quad flexure. The dashed line is the beam after deformation

Figure C.1 shows the deformation diagram of the symmetric-quad flexure. Similarly, only the bending deformation of the flexure is considered in the analytical calculation. To analyze the deflection of the flexure, the flexure will be divided into several beam segments, with each beam segment being treated as one-end fixed and one-end free beam. The rotation at the free end is constrained for the actuator

applications. The free-body diagram of the symmetric-quad flexure is shown in Figure C.2.

Figure C.2 Free-body diagram of the symmetric-quad flexure. (a) segment 12, (b) segment 23, (c) segment 34, (d) segment 47, (e) segment 25, (f) segment 56, (g) segment 67, (h) segment 78

Since there is a loop structure with two joints at the end point 2 and point 7 in the symmetric-quad flexure, the flexure can be split into two halves, as shown in Figure C.3. The displacements at the end point 7 from the beam segment 67 and segment 47 should be the same.

Figure C.3 Two halves segments diagram of the symmetric-quad flexure

At the fully constrained point 1, the force and moment equilibrium equations for the whole flexure are

$$\sum F_x = 0: F_{xN} = F_x \quad (C.1)$$

$$\sum F_y = 0: F_{yN} = F_y \quad (C.2)$$

$$\sum M_{(1)} = 0: M_1 + F_x(l_a + 2l_0) - M = 0 \quad (C.3)$$

Thus,

$$M_1 = M - F_x(l_a + 2l_0) \quad (C.4)$$

For the beam segment 12, the moment equilibrium equation is

$$\sum F_x = 0: F_{x2} = F_x \quad (C.5)$$

$$\sum F_y = 0: F_{y2} = F_y \quad (C.6)$$

$$\sum M_{(1)} = 0: M_1 + F_x l_0 - M_2 = 0 \quad (C.7)$$

i.e.,

$$M_2 = M - F_x(l_a + l_0) \quad (C.8)$$

The displacements at the end point 2 of beam segment 12 are

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$$u_2 = \frac{F_x l_0^3}{3EI_a} - \frac{M_2 l_0^2}{2EI_a} \quad (C.9)$$

$$v_2 = 0 \quad (C.10)$$

$$\theta_2 = \frac{F_x l_0^2}{2EI_a} - \frac{M_2 l_0}{EI_a} \quad (C.11)$$

At the end point 2 of beam segment 23, the force and equilibrium equations are

$$\sum F_x = 0: F_{x3} = F_x - F_{x7} \quad (C.12)$$

$$\sum F_y = 0: F_{y3} = F_y - F_{y7} \quad (C.13)$$

$$\sum M_{(3)} = 0: F_x(l_a + l_0) + F_y \frac{l_b}{2} - M - F_{x7}l_a - F_{y7} \frac{l_b}{2} - M_7 + M_3 = 0 \quad (C.14)$$

Thus,

$$M_3 = M + M_7 - F_x(l_a + l_0) - F_y \frac{l_b}{2} - F_{x7}l_0 + F_{y7} \frac{l_b}{2} \quad (C.15)$$

The displacements at the end point 3 of beam segment 23 are

$$u_{3r} = 0 \quad (C.16)$$

$$v_{3r} = -\frac{F_y \left(\frac{l_b}{2}\right)^3}{3EI_b} - \frac{M_3 \left(\frac{l_b}{2}\right)^2}{2EI_b} \quad (C.17)$$

$$\theta_{3r} = -\frac{F_y \left(\frac{l_b}{2}\right)^2}{2EI_b} - \frac{M_3 \frac{l_b}{2}}{EI_b} \quad (C.18)$$

The rigid-body displacements at the end point 3 are

$$u_{3b} = u_2 \quad (C.19)$$

$$v_{3b} = v_2 + \frac{l_b}{2} \theta_2 \quad (C.20)$$

$$\theta_{3b} = \theta_2 \quad (C.21)$$

Therefore, the absolute displacements at the end point 3 can be obtained

$$u_3 = u_{3r} + u_{3b} = \frac{F_x l_0^3}{3EI_a} - \frac{M_2 l_0^2}{2EI_a} \quad (C.22)$$

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$$v_3 = v_{3r} + v_{3b} = -\frac{F_y \left(\frac{l_b}{2}\right)^3}{3EI_b} - \frac{M_3 \left(\frac{l_b}{2}\right)^2}{2EI_b} + \frac{F_x l_0^2 \frac{l_b}{2}}{2EI_a} - \frac{M_2 l_0 \frac{l_b}{2}}{EI_a} \quad (C.23)$$

$$\theta_3 = \theta_{3r} + \theta_{3b} = -\frac{F_y \left(\frac{l_b}{2}\right)^2}{2EI_b} - \frac{M_3 \frac{l_b}{2}}{EI_b} + \frac{F_x l_0^2}{2EI_a} - \frac{M_2 l_0}{EI_a} \quad (C.24)$$

For the beam segment 348, the force and moment equilibrium equations are

$$\sum F_x = 0: F_{x3} = F_x - F_{x7} \quad (C.25)$$

$$\sum F_y = 0: F_{y3} = F_y - F_{y7} \quad (C.26)$$

$$\sum M_{(3)} = 0: F_{x4} l_0 - M_4 + M_3 = 0 \quad (C.27)$$

Thus,

$$M_4 = M + M_7 - F_x l_a - F_y \frac{l_b}{2} + F_{y7} \frac{l_b}{2} \quad (C.28)$$

The displacements at the end point 4 of beam segment 34 are

$$u_{4r} = \frac{F_{x3} l_a^3}{3EI_a} - \frac{M_4 l_a^2}{2EI_a} \quad (C.29)$$

$$v_{4r} = 0 \quad (C.30)$$

$$\theta_{4r} = \frac{F_{x3} l_a^2}{2EI_a} - \frac{M_4 l_a}{EI_a} \quad (C.31)$$

The rigid-body displacements at the end point 4 are

$$u_{4b} = u_3 + l_a \theta_3 \quad (C.32)$$

$$v_{4b} = v_3 \quad (C.33)$$

$$\theta_{4b} = \theta_3 \quad (C.34)$$

Therefore, the absolute displacements at the end point 4 are

$$\begin{aligned} u_4 = u_{4r} + u_{4b} &= \frac{F_{x3} l_a^3}{3EI_a} - \frac{M_4 l_a^2}{2EI_a} + \frac{F_x l_0^3}{3EI_a} - \frac{M_2 l_0^2}{2EI_a} - \frac{F_y \left(\frac{l_b}{2}\right)^2 l_a}{2EI_b} - \frac{M_3 \frac{l_b}{2} l_a}{EI_b} \\ &+ \frac{F_x l_0^2 l_a}{2EI_a} - \frac{M_2 l_0 l_a}{EI_a} \end{aligned} \quad (C.35)$$

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$$v_4 = v_{4r} + v_{4b} = -\frac{F_y \left(\frac{l_b}{2}\right)^3}{3EI_b} - \frac{M_3 \left(\frac{l_b}{2}\right)^2}{2EI_b} + \frac{F_x l_0^2 \frac{l_b}{2}}{2EI_a} - \frac{M_2 l_0 \frac{l_b}{2}}{EI_a} \quad (C.36)$$

$$\theta_4 = \theta_{4r} + \theta_{4b} = \frac{F_{x3} l_a^2}{2EI_a} - \frac{M_4 l_a}{EI_a} - \frac{F_y \left(\frac{l_b}{2}\right)^2}{2EI_b} - \frac{M_3 \frac{l_b}{2}}{EI_b} + \frac{F_x l_0^2}{2EI_a} - \frac{M_2 l_0}{EI_a} \quad (C.37)$$

At the end point 7 of beam segment 47, the force and moment equilibrium equations are

$$\sum F_x = 0: F_{x7}' = F_x - F_{x7} \quad (C.38)$$

$$\sum F_y = 0: F_{y7}' = F_y - F_{y7} \quad (C.39)$$

$$\sum M_{(7)} = 0: F_{y7}' \frac{l_b}{2} + M_4 - M_7' = 0 \quad (C.40)$$

Thus,

$$M_7' = M + M_7 - F_x l_0 \quad (C.41)$$

The displacements at the end point 7 of beam segments 47 are

$$u_{7r} = 0 \quad (C.42)$$

$$v_{7r} = -\frac{F_{y7}' \left(\frac{l_b}{2}\right)^3}{3EI_b} - \frac{M_7' \left(\frac{l_b}{2}\right)^2}{2EI_b} \quad (C.43)$$

$$\theta_{7r} = -\frac{F_{y7}' \left(\frac{l_b}{2}\right)^2}{2EI_b} - \frac{M_7' \left(\frac{l_b}{2}\right)}{EI_b} \quad (C.44)$$

The rigid-body displacements at the end point 7 are

$$u_{7b} = u_4 \quad (C.45)$$

$$v_{7b} = v_4 - \frac{l_b}{2} \theta_4 \quad (C.46)$$

$$\theta_{7b} = \theta_4 \quad (C.47)$$

Therefore, the absolute displacements at the end point 7 of right beam segment 47 are

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$$u_{7R} = u_{7r} + u_{7b} = \frac{F_{x3}l_a^3}{3EI_a} - \frac{M_4l_a^2}{2EI_a} + \frac{F_xl_0^3}{3EI_a} - \frac{M_2l_0^2}{2EI_a} - \frac{F_y\left(\frac{l_b}{2}\right)^2 l_a}{2EI_b} - \frac{M_3\frac{l_b}{2}l_a}{EI_b} + \frac{F_xl_0^2 l_a}{2EI_a} - \frac{M_2l_0l_a}{EI_a} \quad (C.48)$$

$$v_{7R} = v_{7r} + v_{7b} = -\frac{F_{y7}\left(\frac{l_b}{2}\right)^3}{3EI_b} - \frac{M_7\left(\frac{l_b}{2}\right)^2}{2EI_b} - \frac{F_y\left(\frac{l_b}{2}\right)^3}{3EI_b} - \frac{M_3\left(\frac{l_b}{2}\right)^2}{2EI_b} + \frac{F_xl_0^2\frac{l_b}{2}}{2EI_a} - \frac{M_2l_0\frac{l_b}{2}}{EI_a} - \frac{F_{x3}l_a^2\frac{l_b}{2}}{2EI_a} - \frac{M_4l_a\frac{l_b}{2}}{EI_a} - \frac{F_y\left(\frac{l_b}{2}\right)^3}{2EI_b} - \frac{M_3\left(\frac{l_b}{2}\right)^2}{EI_b} + \frac{F_xl_0^2\frac{l_b}{2}}{2EI_a} - \frac{M_2l_0\frac{l_b}{2}}{EI_a} \quad (C.49)$$

$$\theta_{7R} = \theta_{7r} + \theta_{7b} = -\frac{F_{y7}\left(\frac{l_b}{2}\right)^2}{2EI_b} - \frac{M_7\left(\frac{l_b}{2}\right)}{EI_b} + \frac{F_{x3}l_a^2}{2EI_a} - \frac{M_4l_a}{EI_a} - \frac{F_y\left(\frac{l_b}{2}\right)^2}{2EI_b} - \frac{M_3\frac{l_b}{2}}{EI_b} + \frac{F_xl_0^2}{2EI_a} - \frac{M_2l_0}{EI_a} \quad (C.50)$$

For the left part beam segments 257, the force and moment equilibrium equations at the end point 2 are

$$\sum F_x = 0: F_{x2}' = F_{x7} \quad (C.51)$$

$$\sum F_y = 0: F_{y2}' = F_{y7} \quad (C.52)$$

$$\sum M_{(2)} = 0: F_{x7}l_a + M_7 - M_2' = 0 \quad (C.53)$$

Thus,

$$M_2' = M_7 + F_{x7}l_a \quad (C.54)$$

Then, for the beam segment 25, the force and moment equilibrium equations are

$$\sum F_x = 0: F_{x5} = F_{x2}' \quad (C.55)$$

$$\sum F_y = 0: F_{y5} = F_{y2}' \quad (C.56)$$

$$\sum M_{(5)} = 0: F_{y2}'\frac{l_b}{2} + M_5 - M_2' = 0 \quad (C.57)$$

Thus,

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$$M_5 = M_7 + F_{x7}l_0 - F_{y7}\left(\frac{l_b}{2}\right) \quad (C.58)$$

The displacements at the end point 5 of beam segment 25 are

$$u_{5r} = 0 \quad (C.59)$$

$$v_{5r} = -\frac{F_{y7}\left(\frac{l_b}{2}\right)^3}{3EI_b} - \frac{M_5\left(\frac{l_b}{2}\right)^2}{2EI_b} \quad (C.60)$$

$$\theta_{5r} = -\frac{F_{y7}\left(\frac{l_b}{2}\right)^2}{2EI_b} - \frac{M_5\left(\frac{l_b}{2}\right)}{EI_b} \quad (C.61)$$

The rigid-body displacements at the end point 5 are

$$u_{5b} = u_2 \quad (C.62)$$

$$v_{5b} = v_2 - \frac{l_b}{2}\theta_2 \quad (C.63)$$

$$\theta_{5b} = \theta_2 \quad (C.64)$$

Therefore, the absolute displacements at the end point 5 of beam segment 25 are

$$u_5 = u_{5r} + u_{5b} = \frac{F_x l_0^3}{3EI_a} - \frac{M_2 l_0^2}{2EI_a} \quad (C.65)$$

$$v_5 = v_{5r} + v_{5b} = -\frac{F_{y7}\left(\frac{l_b}{2}\right)^3}{3EI_b} - \frac{M_5\left(\frac{l_b}{2}\right)^2}{2EI_b} - \frac{F_x l_0^2 \frac{l_b}{2}}{2EI_a} + \frac{M_2 l_0 \frac{l_b}{2}}{EI_a} \quad (C.66)$$

$$\theta_5 = \theta_{5r} + \theta_{5b} = -\frac{F_{y7}\left(\frac{l_b}{2}\right)^2}{2EI_b} - \frac{M_5\left(\frac{l_b}{2}\right)}{EI_b} + \frac{F_x l_0^2}{2EI_a} - \frac{M_2 l_0}{EI_a} \quad (C.67)$$

For the beam segment 67, the force and moment equilibrium equations are

$$\sum F_x = 0: F_{x6} = F_{x7} \quad (C.68)$$

$$\sum F_y = 0: F_{y6} = F_{y7} \quad (C.69)$$

$$\sum M_{(5)} = 0: F_{x7}l_a + M_6 - M_5 = 0 \quad (C.70)$$

Thus,

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$$M_6 = M_7 - F_{y7} \left(\frac{l_b}{2} \right) \quad (C.71)$$

The displacements at the end point 6 of beam segment 56 are

$$u_{6r} = \frac{F_{x7} l_a^3}{3EI_a} + \frac{M_6 l_a^2}{2EI_a} \quad (C.72)$$

$$v_{6r} = 0 \quad (C.73)$$

$$\theta_{6r} = \frac{F_{x7} l_a^2}{2EI_a} + \frac{M_6 l_a}{EI_a} \quad (C.74)$$

The rigid-body displacements at the end point 6 are

$$u_{6b} = u_5 + l_a \theta_5 \quad (C.75)$$

$$v_{6b} = v_5 \quad (C.76)$$

$$\theta_{6b} = \theta_5 \quad (C.77)$$

Therefore, the absolute displacements at the end point 6 are

$$u_6 = u_{6r} + u_{6b} = \frac{F_{x7} l_a^3}{3EI_a} + \frac{M_6 l_a^2}{2EI_a} + \frac{F_{x7} l_0^3}{3EI_a} - \frac{M_2 l_0^2}{2EI_a} - \frac{F_{y7} \left(\frac{l_b}{2} \right)^2 l_a}{2EI_b} - \frac{M_5 \left(\frac{l_b}{2} \right) l_a}{EI_b} + \frac{F_{x7} l_0^2 l_a}{2EI_a} - \frac{M_2 l_0 l_a}{EI_a} \quad (C.78)$$

$$v_6 = v_{6r} + v_{6b} = -\frac{F_{y7} \left(\frac{l_b}{2} \right)^3}{3EI_b} - \frac{M_5 \left(\frac{l_b}{2} \right)^2}{2EI_b} - \frac{F_{x7} l_0^2 \frac{l_b}{2}}{2EI_a} + \frac{M_2 l_0 \frac{l_b}{2}}{EI_a} \quad (C.79)$$

$$\theta_6 = \theta_{6r} + \theta_{6b} = \frac{F_{x7} l_a^2}{2EI_a} + \frac{M_6 l_a}{EI_a} - \frac{F_{y7} \left(\frac{l_b}{2} \right)^2}{2EI_b} - \frac{M_5 \left(\frac{l_b}{2} \right)}{EI_b} + \frac{F_{x7} l_0^2}{2EI_a} - \frac{M_2 l_0}{EI_a} \quad (C.80)$$

The displacements at the end point 7 of the beam segment 67 are

$$u_{7r} = 0 \quad (C.81)$$

$$v_{7r} = -\frac{F_{y7} \left(\frac{l_b}{2} \right)^3}{3EI_b} + \frac{M_7 \left(\frac{l_b}{2} \right)^2}{2EI_b} \quad (C.82)$$

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$$\theta_{7r} = -\frac{F_{y7}\left(\frac{l_b}{2}\right)^2}{2EI_b} + \frac{M_7\left(\frac{l_b}{2}\right)}{EI_b} \quad (C.83)$$

The rigid-body displacements of the beam segment 67 are

$$u_{7b} = u_6 \quad (C.84)$$

$$v_{7b} = v_6 + \frac{l_b}{2}\theta_6 \quad (C.85)$$

$$\theta_{7b} = \theta_6 \quad (C.86)$$

Therefore, the absolute displacements at the end point 7 of left beam segment 67 are

$$u_{7L} = u_{7r} + u_{7b} = \frac{F_{x7}l_a^3}{3EI_a} + \frac{M_6l_a^2}{2EI_a} + \frac{F_{x7}l_0^3}{3EI_a} - \frac{M_2l_0^2}{2EI_a} - \frac{F_{y7}\left(\frac{l_b}{2}\right)^2 l_a}{2EI_b} - \frac{M_5\left(\frac{l_b}{2}\right) l_a}{EI_b} + \frac{F_{x7}l_0^2 l_a}{2EI_a} - \frac{M_2l_0 l_a}{EI_a} \quad (C.87)$$

$$v_{7L} = v_{7r} + v_{7b} = -\frac{F_{y7}\left(\frac{l_b}{2}\right)^3}{3EI_b} + \frac{M_7\left(\frac{l_b}{2}\right)^2}{2EI_b} - \frac{F_{y7}\left(\frac{l_b}{2}\right)^3}{3EI_b} - \frac{M_5\left(\frac{l_b}{2}\right)^2}{2EI_b} - \frac{F_{x7}l_0^2 \frac{l_b}{2}}{2EI_a} + \frac{M_2l_0 \frac{l_b}{2}}{EI_a} + \frac{F_{x7}l_a^2 \frac{l_b}{2}}{2EI_a} + \frac{M_6l_a \frac{l_b}{2}}{EI_a} - \frac{F_{y7}\left(\frac{l_b}{2}\right)^3}{2EI_b} - \frac{M_5\left(\frac{l_b}{2}\right)^2}{EI_b} + \frac{F_{x7}l_0^2 \frac{l_b}{2}}{2EI_a} - \frac{M_2l_0 \frac{l_b}{2}}{EI_a} \quad (C.88)$$

$$\theta_{7L} = \theta_{7r} + \theta_{7b} = -\frac{F_{y7}\left(\frac{l_b}{2}\right)^2}{2EI_b} + \frac{M_7\left(\frac{l_b}{2}\right)}{EI_b} + \frac{F_{x7}l_a^2}{2EI_a} + \frac{M_6l_a}{EI_a} - \frac{F_{y7}\left(\frac{l_b}{2}\right)^2}{2EI_b} - \frac{M_5\left(\frac{l_b}{2}\right)}{EI_b} + \frac{F_{x7}l_0^2}{2EI_a} - \frac{M_2l_0}{EI_a} \quad (C.89)$$

To fit the continuous conditions of the beam segment 67 and segment 47, there are the following equilibrium equations

$$u_{7L} = u_{7R} \quad (C.90)$$

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$$v_{7L} = v_{7R} \quad (C.91)$$

$$\theta_{7L} = \theta_{7R} \quad (C.92)$$

After substitution of the variables and simplification, equations (C.90) ~ (C.92) can be re-written as

$$\begin{aligned} F_x \cdot \left(\frac{l_a l_0^2}{2EI_a} + \frac{l_0^3}{3EI_a} + \frac{l_0^2 l_b + l_0 l_a l_b}{2EI_b} \right) + F_y \cdot \left(\frac{l_0 l_b^2}{8EI_b} + \frac{l_0^2 l_b}{4EI_a} \right) - M \cdot \left(\frac{l_0 l_b}{2EI_b} + \frac{l_0^2}{2EI_a} \right) \\ - F_{x7} \cdot \left(\frac{l_0^2 l_b}{EI_b} + \frac{2l_0^3}{3EI_a} \right) - F_{y7} \cdot \frac{l_0 l_b^2}{8EI_b} - M_7 \cdot \left(\frac{l_0 l_b}{EI_b} + \frac{l_0^2}{EI_a} \right) = 0 \end{aligned} \quad (C.93)$$

$$\begin{aligned} F_x \cdot \left(-\frac{l_0^2 l_b}{4EI_a} - \frac{l_0 l_b^2}{8EI_b} - \frac{l_0 l_a l_b}{2EI_a} \right) + F_y \cdot \left(-\frac{l_b^3}{12EI_b} - \frac{l_b^2 l_0}{4EI_a} \right) + M \cdot \frac{l_0 l_b}{2EI_a} \\ + F_{y7} \cdot \left(\frac{l_b^2 l_0}{2EI_a} + \frac{3l_b^3}{16EI_b} \right) - M_7 \cdot \frac{l_b^2}{4EI_b} = 0 \end{aligned} \quad (C.94)$$

$$\begin{aligned} F_x \cdot \left(\frac{l_a l_0}{EI_a} + \frac{l_0^2}{2EI_a} + \frac{l_0 l_b + l_a l_b}{2EI_b} \right) + F_y \cdot \frac{l_0 l_b}{2EI_a} - M \cdot \left(\frac{l_b}{EI_b} + \frac{l_0}{EI_a} \right) - F_{x7} \cdot \left(\frac{l_0 l_b}{EI_b} + \frac{l_0^2}{EI_a} \right) \\ + F_{y7} \cdot \frac{l_b^2}{8EI_b} - M_7 \cdot \left(\frac{2l_b}{EI_b} + \frac{2l_0}{EI_a} \right) = 0 \end{aligned} \quad (C.95)$$

By solving the equations (C.93) ~ (C.95), the expressions of F_{x7} , F_{y7} and M_7 can be obtained.

The displacements at the end point 8 of beam segment 78 are

$$u_{8r} = \frac{F_x l_0^3}{3EI_a} - \frac{M l_0^2}{2EI_a} \quad (C.96)$$

$$v_{8r} = 0 \quad (C.97)$$

$$\theta_{8r} = \frac{F_x l_a^2}{2EI_a} - \frac{M l_a}{EI_a} \quad (C.98)$$

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The rigid-body displacements at the end point 8 can be derived following either left segment 67 or right segment 47. For instance, the rigid-body displacements of beam segment 78 following the right segment 47 are

$$u_{8b} = u_{7R} + l_0 \theta_{7R} \quad (C.99)$$

$$v_{8b} = v_{7R} \quad (C.100)$$

$$\theta_{8b} = \theta_{7R} \quad (C.101)$$

Therefore, the absolute displacements at the end point 8 can be obtained as

$$\begin{aligned} u_8 = u_{8r} + u_{8b} = & \frac{F_x l_0^3}{3EI_a} - \frac{M l_0^2}{2EI_a} + \frac{F_{x3} l_a^3}{3EI_a} - \frac{M_4 l_a^2}{2EI_a} + \frac{F_x l_0^3}{3EI_a} - \frac{M_2 l_0^2}{2EI_a} - \frac{F_y \left(\frac{l_b}{2}\right)^2 l_a}{2EI_b} \\ & - \frac{M_3 \frac{l_b}{2} l_a}{EI_b} + \frac{F_x l_0^2 l_a}{2EI_a} - \frac{M_2 l_0 l_a}{EI_a} - \frac{F_{y7} \left(\frac{l_b}{2}\right)^2 l_0}{2EI_b} - \frac{M_7 \left(\frac{l_b}{2}\right) l_0}{EI_b} + \frac{F_{x3} l_a^2 l_0}{2EI_a} \\ & - \frac{M_4 l_a l_0}{EI_a} - \frac{F_y \left(\frac{l_b}{2}\right)^2 l_0}{2EI_b} - \frac{M_3 l_0 \frac{l_b}{2}}{EI_b} + \frac{F_x l_0^3}{2EI_a} - \frac{M_2 l_0^2}{EI_a} \end{aligned} \quad (C.102)$$

$$\begin{aligned} v_8 = v_{8r} + v_{8b} = & -\frac{F_{y7} \left(\frac{l_b}{2}\right)^3}{3EI_b} - \frac{M_7 \left(\frac{l_b}{2}\right)^2}{2EI_b} - \frac{F_y \left(\frac{l_b}{2}\right)^3}{3EI_b} - \frac{M_3 \left(\frac{l_b}{2}\right)^2}{2EI_b} + \frac{F_x l_0^2 \frac{l_b}{2}}{2EI_a} \\ & - \frac{M_2 l_0 \frac{l_b}{2}}{EI_a} - \frac{F_{x3} l_a^2 \frac{l_b}{2}}{2EI_a} - \frac{M_4 l_a \frac{l_b}{2}}{EI_a} - \frac{F_y \left(\frac{l_b}{2}\right)^3}{2EI_b} - \frac{M_3 \left(\frac{l_b}{2}\right)^2}{EI_b} + \frac{F_x l_0^2 \frac{l_b}{2}}{2EI_a} - \frac{M_2 l_0 \frac{l_b}{2}}{EI_a} \end{aligned} \quad (C.103)$$

$$\begin{aligned} \theta_8 = \theta_{8r} + \theta_{8b} = & \frac{F_x l_a^2}{2EI_a} - \frac{M l_a}{EI_a} - \frac{F_{y7} \left(\frac{l_b}{2}\right)^2}{2EI_b} - \frac{M_7 \left(\frac{l_b}{2}\right)}{EI_b} + \frac{F_{x3} l_a^2}{2EI_a} - \frac{M_4 l_a}{EI_a} \\ & - \frac{F_y \left(\frac{l_b}{2}\right)^2}{2EI_b} - \frac{M_3 \frac{l_b}{2}}{EI_b} + \frac{F_x l_0^2}{2EI_a} - \frac{M_2 l_0}{EI_a} \end{aligned} \quad (C.104)$$

After simplification, equations (C.102) ~ (C.104) can be re-written as

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$$\begin{aligned}
& F_x \cdot \left(\frac{l_0^3}{3EI_a} + \frac{8l_a^3}{3EI_a} + \frac{2l_al_0^2}{EI_a} + \frac{4l_0l_a^2}{EI_a} + \frac{l_b(l_0+l_a)^2}{2EI_b} \right) + F_y \cdot \left(\frac{l_0l_b^2}{8EI_b} + \frac{l_0^2l_b}{4EI_a} + \frac{l_0l_al_b}{2EI_a} \right) \\
& + M \cdot \left(-\frac{l_0^2}{2EI_a} - \frac{2l_a^2}{EI_a} - \frac{2l_0l_a}{EI_a} - \frac{l_0l_b+2l_al_b}{2EI_b} \right) + F_{x7} \cdot \left(-\frac{l_0^3}{3EI_a} - \frac{l_0^2l_b}{2EI_b} - \frac{l_0l_al_b}{2EI_b} - \frac{l_0^2l_a}{2EI_a} \right) \\
& + F_{y7} \cdot \left(-\frac{l_0^2l_b}{4EI_a} - \frac{l_0l_al_b}{2EI_a} - \frac{l_b^2l_a}{8EI_b} - \frac{l_b^2l_0}{4EI_b} \right) + M_7 \cdot \left(-\frac{l_al_0}{EI_a} - \frac{l_0^2}{2EI_a} - \frac{l_0l_b+2l_al_b}{2EI_b} \right) = u_8
\end{aligned}
\tag{C.105}$$

$$\begin{aligned}
& F_x \cdot \left(-\frac{l_b l_0^2}{4EI_a} - \frac{l_0 l_b^2}{8EI_b} - \frac{l_0 l_a l_b}{2EI_a} \right) + F_y \cdot \left(-\frac{l_0 l_b^2}{4EI_a} - \frac{l_b^3}{12EI_b} \right) + M \cdot \frac{l_0 l_b}{2EI_a} \\
& + F_{x7} \cdot \left(\frac{l_0 l_b^2}{8EI_b} + \frac{l_0^2 l_b}{4EI_a} \right) + F_{y7} \cdot \left(\frac{l_0 l_b^2}{4EI_a} + \frac{5l_b^3}{48EI_b} \right) + M_7 \cdot \frac{l_0 l_b}{2EI_a} = v_8
\end{aligned}
\tag{C.106}$$

$$\begin{aligned}
& F_x \cdot \left(\frac{2l_al_0+2l_a^2}{EI_a} + \frac{l_0^2}{2EI_a} + \frac{l_0l_b+l_al_b}{2EI_b} \right) + F_y \cdot \frac{l_0l_b}{2EI_a} - M \cdot \left(\frac{l_b}{EI_b} + \frac{l_0}{EI_a} + \frac{2l_a}{EI_a} \right) \\
& - F_{x7} \cdot \left(\frac{l_0l_b}{2EI_b} + \frac{l_0^2}{2EI_a} \right) - F_{y7} \cdot \left(\frac{l_0l_b}{2EI_a} + \frac{l_b^2}{8EI_b} \right) - M_7 \cdot \left(\frac{l_b}{EI_b} + \frac{l_0}{EI_a} \right) = \theta_8
\end{aligned}
\tag{C.107}$$

By substituting the expressions of F_{x7} , F_{y7} and M_7 into equations (C.105) ~ (C.107), the deflection u_6 , v_6 can be obtained.

$$\begin{aligned}
u_8 = & l_a^3 (F_x (79r_2^2 \alpha^2 + 163r_2^3 \alpha^3 + 12r_2 \alpha + 76r_1^4 r_2^3 \alpha^3 + 64r_1 + 120r_1^3 r_2^4 \alpha^4 + 3r_1 r_2^5 \alpha^5 + 432r_1^2 r_2^4 \alpha^4 \\
& + 1840r_1^3 r_2^3 \alpha^3 + 4288r_1^4 r_2^2 \alpha^2 + 282r_1 r_2^4 \alpha^4 + 2484r_1^2 r_2^3 \alpha^3 + 7192r_1^3 r_2^2 \alpha^2 + 4480r_1^4 r_2 \alpha + 616r_1 r_2 \alpha \\
& + 1678r_1 r_2^2 \alpha^2 + 3840r_1^2 r_2 \alpha + 105r_2^4 \alpha^4 + 1380r_1 r_2^3 \alpha^3 + 5196r_1^2 r_2^2 \alpha^2 + 6912r_1^3 r_2 \alpha + 1024r_1^4 + 9r_2^5 \alpha^5 \\
& + 1536r_1^3) + F_y (27r_1 r_2^5 \alpha^4 - 18r_2^5 \alpha^4 + 63r_1 r_2^4 \alpha^3 - 6r_2^3 \alpha^2 + 24r_1 r_2^2 \alpha + 90r_1 r_2^3 \alpha^2 - 24r_2^4 \alpha^3)) \\
& / (12EI_a (15r_2^4 \alpha^4 + 86r_2^3 \alpha^3 + 95r_2^2 \alpha^2 + 536r_1 r_2^2 \alpha^2 + 128r_1 + 96r_1 r_2^3 \alpha^3 + 560r_1 r_2 \alpha + 24r_2 \alpha))
\end{aligned}
\tag{C.108}$$

$$\begin{aligned}
v_8 = & -l_b^2 l_a (F_x (288r_1 - 18r_1 r_2^4 \alpha^4 - 36r_1^2 r_2^3 \alpha^3 + 3312r_1^2 r_2 \alpha + 912r_1^2 r_2^2 \alpha^2 + 24r_2 \alpha + 18r_2^4 \alpha^4 \\
& + 216r_1 r_2^3 \alpha^3 + 102r_2^2 \alpha^2 + 1308r_1 r_2 \alpha + 1152r_1^2 + 1362r_1 r_2^2 \alpha^2 + 96r_1^2 r_2^3 \alpha^3) + F_y (15r_2^5 \alpha^4 \\
& + 195r_1 r_2^4 \alpha^3 + 155r_2^4 \alpha^3 + 1712r_1 r_2^3 \alpha^2 + 128r_2^2 \alpha + 1056r_1 r_2 + 3716r_1 r_2^2 \alpha + 412r_2^3 \alpha^2)) \\
& / (72EI_b (15r_2^4 \alpha^4 + 86r_2^3 \alpha^3 + 95r_2^2 \alpha^2 + 536r_1 r_2^2 \alpha^2 + 128r_1 + 96r_1 r_2^3 \alpha^3 + 560r_1 r_2 \alpha + 24r_2 \alpha))
\end{aligned}$$

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(C.109)

$$\begin{aligned}
M_8 = & l_a (F_x (3456r_1r_2\alpha + 36r_2^4\alpha^4 + 144r_1r_2^4\alpha^4 + 768r_1 + 96r_1r_2\alpha + 360r_2^2\alpha^2 + 1164r_1r_2^3\alpha^3 \\
& + 1536r_1^2 + 300r_2^3\alpha^3 + 1152r_1^2r_2^3\alpha^3 + 3708r_1r_2^2\alpha^2 + 6720r_1^2r_2\alpha + 6432r_1^2r_2^2\alpha^2) \\
& - F_y (112r_2^2\alpha + 430r_2^3\alpha^2 + 315r_2^4\alpha^3 + 45r_2^5\alpha^4)) / (12(15r_2^4\alpha^4 + 86r_2^3\alpha^3 + 95r_2^2\alpha^2 + 128r_1 \\
& + 560r_1r_2\alpha + 96r_1r_2^3\alpha^3 + 536r_1r_2^2\alpha^2 + 24r_2\alpha))
\end{aligned}$$

(C.110)

where $\alpha = I_a/I_b = (w_a/w_b)^3$, $r_1 = l_0/l_a$, $r_2 = l_b/l_a$.

Appendix D Process recipe details

D.1 Photolithography

D.1.1 Photoresist type: AZ7220 (Clariant)

- Spinning: 500 rpm for 10 seconds (uniforming), 3000 rpm for 45 seconds (thinning)
- Photoresist thickness: 2.2 μm
- Baking: 100 °C 90 seconds for pre-exposure baking, 110 °C 1 minute for post-baking, 120 °C 30 minute for hard-baking (optional)
- Aligner machine: Karl Suss MA6
- Contact type: soft contact
- Exposure time: 3.4 seconds
- Development: 50 sec in MIF300 developer, then rinse in DI water and spin-dry

D.1.2 Photoresist type: AZ9260 (Clariant)

- Spinning: 500 rpm for 10 seconds (uniforming), 3000 rpm for 45 seconds (thinning)
- Photoresist thickness: 8 μm
- Baking: 110 °C 4 min for pre-exposure baking, no post-baking, 120 °C 30 min hard baking (optional)
- Aligner machine: Karl Suss MA6
- Contact type: soft contact
- Exposure time: 20 seconds
- Development: 60~70 sec in AZ400K developer, then rinse in DI water and spin-dry

D.2 Deposition and patterning the electrodes of Cr/Au, Ti/Pt

- Machine: DC/RF sputtering system
- Deposition pressure: 2 mtorr
- Gas: Ar 20 sccm
- Deposition temperature: room temperature ($\sim 25^\circ\text{C}$)
- Deposition power: DC 200 W
- Deposition rate: 10 nm/min for Cr, 25 nm/min for Au, 10 nm/min for Ti, 15 nm/min for Pt

D.3 Deposition of amorphous silicon for glass etching mask

- Machine: Technics PECVD
Temperature: 300°C ; power: 120 W; gas: silane (SiH_4) and Ar
Measured film stress: 500–600 MPa compressive; can be reduced to around 200 MPa by annealing at 400°C for 4 hours
- Machine: STS Pro PECVD
Temperature: 200°C ; power: 120 W at 384 kHz generator; gas: SiH_4 and Ar;
pressure: 800 mtorr
Measured film stress: ~ 20 MPa compressive

D.4 RIE etching of amorphous silicon

- Machine: Technics RIE system
- Chamber pressure: 140 mtorr
- Gas: CH_4 9 sccm and O_2 1 sccm
- Temperature: room temperature
- Power: 350 W

D.5 Wet etching of glass wafer

- Glass material: Corning pyrex 7740
- Etching solution: 49% hydrofluoric acid (HF)
- Temperature: room temperature
- Etching rate: 6~7 $\mu\text{m}/\text{min}$

D.6 Eutectic bonding of silicon to glass wafers

- Machine: Karl Suss WB6 (and BA6 for pre-alignment)
- Temperature: 380°C
- Pressure: 2000 mbar
- Bonding time: 15 min
- Bonding yield: 85% bonded, but with a few distinct bubbles

D.7 Anodic bonding of silicon to glass wafers

- Machine: Karl Suss WB6 (and BA6 for pre-alignment)
- Temperature: 360-400°C
- Pressure: 2000 mbar
- Voltage: 2000 V for center bond 2 min, 1000 V for area bond 10 min
- Bonding time: 12 min
- Bonding yield: 99% bonded, no distinct bubbles observed

D.8 DRIE of silicon

- Machine: STS ICP Multiplex system
- Gas: SF_6 100 sccm, O_2 13 sccm, C_4F_8 130 sccm
- Cycle time: etching 14 sec and passivation 8 sec (starting with passivation) or

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etching 8 sec and passivation 5 sec (starting with passivation)

- Pressure: 5 mtorr
- Power: Coil power 800 W, platen power 10~12 W, multiple 10×
- Etch rate: 2~3 μm for 14s/8s (etching /passivation time), 1.4~1.8 μm for 8s/5s (etching /passivation time), which is dependent on the exposed area and feature size

Appendix E

Photographs of the Fabricated Microactuators

Figure F.1 SEM whole view of the silicon-on-glass (SOG) electrostatic microactuator. The dimensions of silicon actuator are: width of 1200 μm , length of 500 μm and thickness of 200 μm . The glass die size has the dimensions of 1500 μm in width, 800 μm in length, and 700 μm in thickness.

Figure F.2 Top view of the SOG microactuator, showing the electrode pads.

Figure F.3 Side view of the SOG microactuator. The free-standing come-drive structure has a thickness of 100~110 μm , and the Si base thickness is 200 μm .

Figure F.4 Close up of high-aspect-ratio comb drive structure, with trench width of 4 μm and trench depth of 110 μm (aspect ratio of 27.5)

Figure F.5 Close up of the comb drive structure, with trench gap of 3 μm , trench depth of 100 μm (aspect ratio of around 33:1)

Figure F. 6 Portion view of the microactuator with straight flexures

Figure F.7 Portion view of the microactuator with asymmetric-quad flexures

Appendix F

List of Publication

Journal paper:

1. B. Chen, J. Miao, F. Tay, Fabrication and Characterization of DRIE-Micromachined Electrostatic Microactuators for Hard Disk Drives. *Microsystem Technologies*, Volume 13, No. 1, 2007, pp. 11-19.
2. B. Chen and J. Miao, Influence of deep RIE tolerances on comb-drive actuators performance. *Journal of Physics D: Applied Physics*, Volume 40, No. 4, 2007, pp. 970-976.
3. C. Iliescu, B. Chen and J. Miao, A review on wet etching of pyrex glass. *Sensors and Actuators A*, 2007, Submitted.
4. Y. Ong, B. Chen, F. Tay, C. Iliescu, Process analysis and optimization on PECVD amorphous silicon on glass substrate. *Journal of Physics: Conference Series*, Volume 34, 2006, pp.812-817.
5. B. Chen, J. Miao, C. Lim, F. Tay and C. Iliescu, Dynamics Behaviors of High-G MEMS Accelerometer Incorporated with Novel Micro-Flexures. *Int. Journal of Software Engineering and Knowledge Engineering*, May, 2005, pp. 225~230.
6. B. Chen, J. Miao and H. Zhu, Analysis and Comparison of Flexure Structures Used in 3D Electrostatic Microactuators for Hard Disk Drives. *Journal of Computational Engineering Science*, Volume 4 No. 2, June 2003, pp. 227~230.
7. H. Zhu, J. Miao, B. Chen, Z. Wang and W. Zhu, Membrane Microcantilever Arrays Fabrication with PZT Thin Films for Nanorange Movement. *Microsystem Technologies*, Volume 12, No. 8, May 2005, pp. 1121-1126.
8. H. Zhu, J. Miao, B. Chen, Z. Wang and W. Zhu, "Micromachining Process of Piezoelectric Microcantilever Using PZT Thin Film". *Journal of Computational Engineering Science*, Volume 4, No.3, September 2003, pp. 509-513.

Conference paper:

1. C. Iliescu, B. Chen and J. Miao, Deep Wet Etching-Through of 1 mm Pyrex Glass Wafer for Microfluidics Applications. 20th IEEE Conference on Micro Electro Mechanical Systems (MEMS07), 21-25 Jan, 2007, Kobe, Japan, pp. 1579.
2. B. Chen, J. Miao and F. Tay, Study of DRIE tolerance of comb-drive actuators and their influence. In *Proceedings of International Microelectromechanical systems Conference*, May 2006.

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3. C. Iliescu, B. Chen, F. Tay, G. Xu and J. Miao, Characterization of deep wet etching of glass. In *Proceedings of SPIE*, Australia, 2005, 6037, 60370A.
4. Francis E.H. Tay, C. Iliescu, G. Xu, B. Chen, and M. Ayram, A new fabrication process of silicon nanotips – NERCOM. In *Proceedings of SPIE*, Australia, 2005, 6036, 603608.
5. B. Chen, J. Miao, S. Wang, F. Tay, A. Ong, High-aspect-ratio SOG Electrostatic Microactuators and Their Applications in Hard Disk Drives. In *Proceedings of ICMAT'05*, Singapore, 2005, pp. 321~324.
6. C. Iliescu, F. Tay, G. Xu, J. Miao and B. Chen, SU-8 Wafer-to-wafer Bonding By Using Contact Imprinting. In *Proceedings of ICMAT'05*, Singapore, 2005, pp.99~102.
7. B. Chen, J. Miao and H. Zhu, Bulk-Micromachined Electrostatic Microactuators for Hard Disk Drives. In *Proceedings of the Design, Test, Integration and Packaging of MEMS/MOEMS 2004*, Switzerland, May 2004, pp.361~364.
8. B. Chen and J. Miao, Electrostatic Microactuators Fabricated using Deep RIE and Wafer Bonding for Hard Disk Drives. In *Proceedings of 7th International Conference on Solid-State and Integrated-Circuit Technology*, Volume 3, Nov 2004, pp. 1868-1871.
9. B. Chen, J. Miao and H. Zhu, Development of 3-D Electrostatic Microactuators for Hard Disk Drives. In *Proceedings of JSME-IIP/ASME-ISPS Joint Conference on Micromechatronics and Precision Equipment'03*, Japan, June 2003.
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