Design of a microgripper for manipulation of micro-parts and biological cells

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Design of a microgripper for manipulation of micro-parts and biological cells

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School of Mechanical & Aerospace Engineering

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ABSTRACT

Precision control is a very important aspect in many areas of engineering. In the field of biotechnology, there is an increasing need for high precision sensing, positioning and manipulation of cells and other biological entities. In the field of medicine, positional accuracy greater than what human hands are capable of achieving is required in some surgeries. The introduction of a highly precision manipulator will assist in biological research on micro-sized cells, helping to save time and manpower costs. The use of micro robots for handling dangerous or unknown biological cells or drugs avoids human contact and the risk of contaminations.

Grasping and manipulating micro-objects is also required for a wide range of important applications such as the assembly of micro-components to obtain micro-systems in electronics industry. The manipulation of objects of the size ranging between 1 mm and 1 μ m would present a challenge to researchers. The possible applications are the handling of micro-parts of complex shapes in restricted environments with difficult access, in vacuum environments or cleanrooms, where adhesive or vacuum gripping is not suitable. Thus there is a demand for an inexpensive gripper with properties like loss- and backlash-free motion transmission, small size, special materials and integration of sensory properties.

An effective mechanical micromanipulator should possess the ability to grasp steadily objects with different shape, high positioning accuracy and large number of degrees of freedom. The manipulators for biological applications should be able to work in liquid solution and sterile environment, accurately control grasping forces in order to

avoid any damage to the delicate biological structures and small size compatible with the extremely small size of the biological elements which are a few mm in diameter.

In this study, the design, fabrication and practical tests of the monolithic compliant microgripper have been presented. To design the model for the microgripper, the geometry design and the material stresses are considered through the finite element analysis. For simulation of the complex physical properties of the microgripper, a model has been developed and implemented in a program for coupled finite element simulation. The presented simulation model has been used to study in detail profiles of von Mises stresses, deformation and contact forces. The results on displacement characteristics have been confirmed by experiments.

The microgripper is suitable for micro-robotic tasks with high demands on size and work output. Possible applications are the handling of micro-parts of complex shape in restricted environments with difficult access, in vacuum environments or clean rooms, where adhesive or vacuum gripping is not suitable. The microgripper presented in this study has several advantages. Due to its monolithic flexural design, the microgripper can be fabricated in a simple process and may be scaled easily to very small dimensions. Due to its optimized geometry, the use of the piezoelectric actuation is possible which allows for very fine precisioning up to the nano-scale.

The microgrippers were designed, fabricated and characterized to be used as end-effectors for micro-assembly applications. Based on the design results, two distinctive prototype designs were developed. The first design was a microgripper prototype manufactured from spring steel and aluminum using the micro-EDM wire cutting

technique with mechanical amplification of piezoelectric actuation. The second design was a relatively small scale microgripper manufactured from silicon using MEMS machining. Fabrications of both designs were successfully demonstrated.

For the wire electro-discharge machined microgripper prototype of $32 \times 35 \times 3 \text{ mm}^3$ in size, a maximum stroke of $180 \mu\text{m}$ and a maximum mechanical ratio of 3 are achieved. The maximum stress in the microgripper device keeps well below critical stress values for fatigue.

The simulation and experimental results have proven the good performance and feasibility of the proposed microgripper. Especially in case of gripping and manipulating objects such as micro-parts and biomaterials, the proposed microgripper can be very useful for this type of micromanipulation tasks. Micromanipulation tests have been carried out in the laboratory. Results confirm the potential application of the proposed microgripper with piezoelectric actuation in handling objects in the micro-scale environment.

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CHAPTER 1

INTRODUCTION

1.1 BACKGROUND

Gripping and manipulating micro-objects is required for a wide range of important applications such as the assembly of micro-parts to obtain miniature systems or surgery and research in biology and biotechnology. The manipulation of objects of the size ranging between 1 mm and 1 μm would present a challenge to researchers. First of all, the physics of micromanipulation is quite different from that of macro manipulation. The most important difference is that the surface adhesion forces become stronger than the inertial mass forces as the size of the objects decreases, meaning that the weight of the micro-objects is negligible compared to the adhesion forces. The parts interactions in a micro environment are dominated by adhesive forces such as van der Waals, electrostatic and surface tension, making it difficult to release parts during micromanipulation tasks. The problems and challenges of dealing with the adhesive forces between micro-objects have been discussed in some papers [1, 2].

Conventional glass micropipettes, which use a suction method for handling micro-objects, have been used successfully in micromanipulation of biological samples such as cells. Under the microscope, the micropipettes have been used not only as a micro-handling device but also as a micro-injector, micro-hook, micro-spring and micro-cutter. Progress has been made in micro-electromechanical systems in the

construction of microgrippers in order to create new devices that have wide possibility for novel manipulation techniques instead of the glass micropipettes [3].

The microgripper is one of the key elements in micro-robotics and micro-assembly technologies for handling and manipulating micro-objects without damage. An essential component of all microgrippers is the actuator, which provides the required grasping motions and generates applied force to make the device operate as a gripper or tweezers. Various prototypes of microgrippers of different actuation methods have been developed, including electrothermal actuators [4 – 8], electrostatic actuators [9 – 14], piezoelectric actuators [15, 16], electromagnetic actuators [17] and shape memory alloy actuators [18, 19]. Each method has its own characteristics, which depend on the design and fabrication process.

1.2 RESEARCH OBJECTIVES

The main objective of this research is to investigate and evaluate a microgripper manipulator that is suitable for grasping and handling of industrial or electronics micro-parts or even a biological cell. The design is made up of a monolithic compliant mechanism that uses parallel motion to perform the gripping tasks. Piezoelectric actuation is considered as a suitable actuation mode to drive the compliant mechanism. In addition, the contact problems that exist between the microgripper and the grasped target object are investigated and the calculations will be verified with experimental results.

1.3 RESEARCH SCOPE

The scope of this research is to develop a simple and relatively inexpensive microgripper system that incorporates its own actuation function, for performing grasping and handling solutions for micro-parts and biological cells. The two characterization methodologies in the design of this microgripper are based on the use of compliant mechanism to form the microgripper and the utilization of a piezoelectric actuator to drive it.

1.4 OVERVIEW OF THE THESIS

Chapter one gives a brief introduction of the microgripper background and research objective. In order to trace the development of the microgripper designs that have been investigated and fabricated by other researchers, some representative and current research work in the area of micromanipulation grippers is reviewed in chapter two. Compliant mechanisms, micro-actuators and micro-sensors are also discussed in chapter two. The detailed theoretical design and calculations of the compliant mechanism, actuation mode and contact problems are presented in chapter three. This is followed by the finite element analysis simulation in chapter four. Chapters five and six present the prototype system of the microgripper using the MEMS method and wire electrode discharge machining method respectively, along with the performance testings. Finally, chapter seven gives the summary and conclusions for this research.

CHAPTER 2

LITERATURE REVIEW

2.1 MICROMANIPULATION GRIPPER

2.1.1 Introduction to microgripper

An effective mechanical micromanipulator should possess three main characteristics which are the ability to grasp steadily objects with different shape, high positioning accuracy and large number of degrees of freedom. The manipulators for biological applications should have the following characteristics to work in liquid solution and sterile environment, accurately control grasping forces in order to avoid any damage to the delicate biological structures and small size compatible with the extremely small size of the biological elements which are a few micrometers in diameter [20].

2.1.2 Components of microgripper

For the micromanipulation tasks the end-effector essentially consists of the microgripper, micro-actuators and micro-sensors. The microgripper, although relatively simple in geometry and kinematic structure, poses a number of technological problems [20].

The microgripper will be of a simple planar degree of freedom configuration and the focus is towards the design of the mechanical structure, the fabrication technology and the selection of the actuation principle. In order to minimize the size of the

microgripper, the structure has to be fabricated in a single piece and with elastic flexure hinges. The selection of the actuation principle for the microgripper is also very important and critical as the actuators suitable for miniature devices are not easily available [20].

The possible applications are the handling of micro-parts of complex shapes in restricted environments with difficult access, in vacuum environments or cleanrooms, where adhesive or vacuum gripping is not suitable.

2.1.3 Existing microgripper designs

It is not possible to use conventional grippers in micro-systems because of design and geometrical size. A step reduction in size of those systems is not realizable due to technological limits. Thus there is a demand for an inexpensive gripper with properties like loss- and backlash-free motion transmission, small size, special materials and integration of sensory properties. Several microfabrication technologies have been used in commercial and research industries to manufacture the microgrippers. The following examples will show some designs and fabrication methods of existing microgripper systems.

- i) The LIGA technique has been used for the batch production of metallic or plastic microstructures with micron details and large aspect ratio. The process has been used for the design and fabrication of metallic microgrippers [21 – 23]. The metallic microgrippers are fabricated on silicon wafers using SU8-based UV-LIGA and nickel electroplating technologies. The electroplated microgrippers are

subsequently released by removal of either a thermally grown SiO_2 or a spin-coated SU8 release layer. The UV-LIGA technology to fabricate high-aspect-ratio microstructures has been used to make 200 μm thick electroplated nickel microgrippers as shown in Figure 2.1 [24].

Figure 2.1: The electroplated nickel microgripper by UV-LIGA technique [24].

ii) Microgrippers have been made from wafers of photosensitive glass by using micromachining technology. The glasses are microstructurable with a modified photolithographic process. The glass microstructures are used as solid-state hinges actuated by a piezoelectric ceramic monomorph [25].

Figure 2.2: Solid state hinges of the microgripper and the various designs [25].

Figures 2.2 and 2.3 show a microgripper based on microstructured glass. The special mechanism consists of microstructured elastic solid state hinges and lever arms. These elements transmit the gripping force and gripping motion and realize a high distance ratio.

Figure 2.3: Scheme of the microgripper [25].

Another microgripper with parallel movement of the gripping arms using the piezoelectric actuation principle was presented. The microgripper was fabricated by means of a UV-lithographic process and chemical wet etching technology from microstructurable photosensitive glass [26]. The gripping movement is shown in Figure 2.4.

Figure 2.4: (a) Rotational movement of the gripping arms,
(b) Parallel movement of the gripping arms [26].

Figure 2.5 shows schematically the developed microgripper that realizes a parallel movement of the gripping arms. Figure 2.6 shows a modification of the microgripper with two piezoactuators integrated in the kinetic structure. This design has an additional positioning ability.

Figure 2.5: Scheme of the parallel microgripper [26].

Figure 2.6: Scheme of the parallel microgripper with additional positioning ability [26].

Figure 2.7: The positioning process of the gripped object [26].

Figure 2.7 shows the different positioning of the gripped object. The realization of the elements with micrometer dimensions such as flexure hinges, compliant microlevers and parallelogram micromechanism demonstrates the excellent properties of the glass microfabrication technique for development of new micromechanical components as given in Figure 2.8.

Figure 2.8: Elements of the microgripper mechanism [26].

iii) A microgripper system which consists of a monolithic shape memory alloy (SMA) device is presented in a previous work [27]. Figure 2.9 shows a scheme of the mechanical gripper, which comprises a linear actuator and a gearing mechanism with five joints. This scheme is implemented in a completely monolithic SMA gripping device. The structure and operation principle of the gripping device are shown in Figure 2.9.

Figure 2.9: Mechanical Gripper: scheme and monolithic SMA gripping device in open and closed condition, respectively [27].

Figure 2.10: SMA gripper system assembly [27].

Shape memory alloys have great advantages such as large deformation (up to 10% for a single crystal), strong recovery force and a high work density ($2.7 \times 10^7 \text{ Jm}^{-3}$). The SMA gripper is micromachined by laser cutting of TiNi sheets of $50 - 230 \mu\text{m}$ thickness. The microgripper is then mounted onto a ceramic substrate by adhesive bonding [32]. The SMA microgripper is illustrated in Figures 2.10 and 2.11.

Figure 2.11: SMA microgripper system [27].

iv) SU8 shows a wide elastic range without plastic deformation like other epoxies. With a much smaller Young's modulus than silicon or metals typically used for compliant structures, SU8 mechanical structures can be designed softer and with less critical cross sections. It is used to fabricate high-aspect-ratio microstructures of thickness between 300 and 400 μm [28]. Some examples of microgrippers made of SU8 are shown in Figure 2.12.

Figure 2.12: SU8 microgripper with SMA actuator and pneumatic actuators [28].

v) An example of a commercially available microgripper system is the Zyvex patented microgripper systems with active micromachined silicon structures for micro- and nanoscale research, development, and production applications. With up to

50 μm of tip actuation available, these grippers are capable of manipulating micro components with features as large as 500 μm , to below 1 μm [29]. A diagram of the Zyvex silicon microgripper is shown in Figure 2.13.

Figure 2.13: Zyvex microgripper with 36 μm opening powered position [29].

2.2 MEMS micro-fabrication [30, 31]

The IC industry has been the technology platform that has driven MEMS. This is shown in both the bulk silicon and polysilicon efforts that have been the main support of MEMS devices. The MEMS community has made significant advances in the area of deep etching bulk silicon and in surface micromachining with polysilicon. MEMS have driven the silicon community into understanding the mechanical and electrical properties of silicon structures.

The advantages of the MEMS technology include miniature size, low power consumption, very high precision in manufacture, and the potential for low cost through batch fabrication. These key advantages like low cost per function provide great benefits in fabrication of medical MEMS devices. Small size allows in vivo

implants and surgical procedures to be performed in confined spaces. Experimental work has been done using silicon scissors for surgery and silicon neural probes to study neuron activity. The small size of MEMS structures also makes them ideal for manipulating biological samples at the microscopic level.

Micro-fabrication uses a sequence of process steps, in which some are taken from semiconductor fabrication technology and some developed specifically for micromachining, assembled together in a given order to produce a physical structure. In a micro-fabrication process, a substrate is utilized and a device is built either out of its bulk material or on its surface. The former is referred to as bulk micromachining, whereas the latter is known as surface micromachining.

Silicon is the most common material in micro-fabrication due to its role in the fabrication of integrated circuits. It comes in a single crystal wafer form with typical diameters of 75 – 200 mm and thicknesses of 0.25 – 1.0 mm. In addition to its excellent electrical properties, silicon also possesses outstanding mechanical properties, enabling the design of micromechanical structures. There exists a wide range of ways to micro-machine silicon, and the ability to do this in combination with integrated-circuit fabrication leads to the potential to form monolithically integrated microsystems.

Although the range of micromachining processes for glass is less extensive than for silicon, glass provides some unique features, such as optical transparency. Glass wafers are available in many different compositions and sizes. Two important examples are fused silica wafers and borosilicate wafers. Fused silica wafers are pure

amorphous silicon dioxide (SiO_2). They can withstand high temperatures and are optically transparent down to short wavelengths. Borosilicate wafers such as Pyrex (Corning 7740) are much less expensive than fused silica and can be less expensive than silicon.

2.2.1 Bulk silicon micro-machining

Bulk micromachining refers to the fabrication process of single crystal silicon through anisotropic etching technique. High aspect ratio silicon micro-mechanism structures can be formed in this way. The pre-treatment lithography process is carried out before the etching of the silicon wafer. The bulk micromachining is capable of creating desired etch profiles based on anisotropic etching characteristics of the single crystal silicon [31].

2.2.2 Wet cleaning in wafer preparation

Wet cleaning is a process of contaminants removal from the wafer surface in the liquid-phase which is the common cleaning method in semiconductor manufacturing. Wet cleaning chemistries are selected to form soluble compounds of surface contaminants that are often enhanced by ultrasonic agitation. It is then followed by de-ionized water rinse and spin dry cycle [31].

2.2.3 Patterning process

The concept of lithography in MEMS is basically transferring of the pattern on the mask to a photosensitive material by selective exposure to a radiation source such as light. A photosensitive material is a material that experiences a change in its physical properties when exposed to a radiation source. By selective exposure of a photosensitive material to radiation (i.e. by masking some of the radiation) the pattern of the radiation on the material is transferred to the material exposed due to the differences in the properties of the exposed and unexposed regions [31].

2.2.4 Etching methods [32]

Dry etching is a process used in the removal of material, typically a masked pattern of semiconductor material. The material is exposed to a bombardment of ions which dislodge portions of the material from the exposed surface. There are three different classes of dry etching techniques that are being practiced which are reactive ionized etching (RIE), sputter etching, and vapor phase etching. Unlike with many of the wet chemical etchants used in wet etching, the dry etching process typically etches directionally or anisotropically.

Dry etching is normally used in conjunction with photolithographic techniques in a bid to attack certain areas of a semiconductor surface so that recesses can be formed in material, for example, contact holes or via holes or to otherwise remove portions of semiconductor layers where predominantly vertical sides are desired.

Dry etching is attributed with better etch resolution and fine vertical sidewalls for deep etching. A new variant of dry etching is deep reactive-ion etching (DRIE). This technique, introduced in the past 5 years, is revolutionizing micro-fabrication. Although the technology is quite expensive, it has the capability to make very deep and narrow structures in silicon. DRIE is a dry etch process that can be used to etch deeply into a silicon wafer with vertical sidewalls feature. The etching direction is independent of the crystallographic orientation of the silicon wafer. The etching process relies on a high-density plasma source and an alternating process of etching and protective polymer deposition. This etching process can achieve aspect ratio up to 200:1. The commonly used masking layers for DRIE are photoresist and silicon dioxide.

RIE is the most common dry etching technique in micromachining applications. During the etching process, ions are accelerated towards the substrate material to be etched and chemically react with the surface of the material, forming a gaseous product. RIE is an anisotropic etching technique and thus the desired etch profile is not limited by the crystallographic directions of the silicon wafer.

During RIE, the substrate is to be placed inside a reactor whereby several gases are introduced. The plasma is struck in the gas mixture using an RF power source, breaking the gas molecules into ions. The ions are accelerated towards, and are to react at the surface of the material which is being etched. This will form another gaseous material. The process is known as the chemical part of reactive ion etching. There is also a physical part which is similar in nature to the sputtering deposition process. It is important to know that high-energy ions can knock atoms out of the

material to be etched without a chemical reaction. Although, it is very complex to develop dry etch processes that balance chemical and physical etching, since there are many parameters to be taken care of, it is still possible to carry out dry etching.

Deep RIE (DRIE) is special subclass of RIE which is fast becoming popular in its application. It is the technique to be applied in this project. In DRIE, etch depths of hundreds of microns can be achieved with almost vertical sidewalls. A formation of a polymer on the surface of the substrate is caused by the first gas, and the second gas composition etches the substrate. The polymer is immediately sputtered away by the physical part of the etching, but only on the horizontal surfaces. The polymer dissolves very slowly in the chemical part of the etching, so it builds up on the sidewalls and protects them from etching. As a result, etching aspect ratios of 50 to 1 can be achieved. The process can easily be used to etch completely through a silicon substrate, and etch rates are expected to be 3 to 4 times higher than wet etching.

On the other hand, this technology is expensive to run compared to wet etching. Dry etching is widely recommended for those who are concerned with feature resolution in thin film structures or vertical sidewalls for deep etchings in the substrate. Dry etching is an enabling technology, which comes at a sometimes high cost.

In lithography for micromachining, the photosensitive material used is typically a photoresist (also called resist, other photosensitive polymers are also used). When the resist is exposed to a radiation source of a specific wavelength, the chemical resistance of that resist to the developer solution changes. For a positive photoresist, the exposed region is the unwanted portion and will be removed in the developing

process. The exposure of the UV light changes the chemical structure of the resist so that it becomes more soluble and removable while immersing in the developer solution. The positive resist can achieve a high resolution up to 0.5 μm .

2.3 COMPLIANT MECHANISM DESIGN

A mechanism is a mechanical device used to transfer or transform motion, force or energy. Compliant mechanism is a kind of mechanism whereby elastic deformation is the source for motion and force transformation as compared to motion by joints in conventional mechanisms. Pseudo-rigid models are used in the dimensional synthesis of compliant mechanisms with lumped compliance. The compliant mechanism is modeled as a rigid link mechanism where the link compliance is represented using linear springs and the joint compliance is represented using torsional springs [33].

2.3.1 Introduction to Compliant Mechanism

A compliant mechanism is a single-piece flexible structure that delivers the desired motion by undergoing elastic deformation. It is jointless and requires no assembly, thus reducing the problems due to wear, backlash, lubrication and noise. It can be designed for any desired input-output force-displacement characteristics including specified volume to weight ratio, stiffness and natural frequency constraints. In microelectromechanical systems, a compliant design that needs no assembly is not only an advantage but necessary as well. In the design of precision instruments, stackup of errors and clearances in conventional mechanisms affect precision and therefore compliant mechanisms are the only solution. Compliant mechanisms can be

built in one piece which lowers the number of fabrication steps [33]. Figure 2.14 shows some examples of compliant mechanism designs.

Figure 2.14: Examples of some micro compliant mechanism [33].

The actuation methods such as piezoelectric, electrostatic, and shape memory alloys seem to exhibit certain limitations in terms of power density, stroke length and bandwidth. Piezoceramic actuators and SMA actuators are commonly employed in many smart structure applications. However they are not capable of generating adequate actuating force or displacements needed for most practical applications. A disadvantage of piezoelectric materials is the need for high voltage electronics to power them. The disadvantage of the SMA is the rate at which the metal can be cooled and is limited to a few Hz. This limited cooling rate of the SMA constrains the bandwidth of the mechanism. In summary, piezoelectric actuators have high bandwidth but low strain, while SMA actuators have relatively higher strain but extremely low bandwidth. There are ways to amplify the displacement by stacking multiple piezoactuators in different configurations. However such arrangements are

cumbersome and require large voltages. A simple and effective way to modify the force-displacement characteristics is to integrate a piezoactuator with a compliant transmission device [33].

Implementing smart materials with compliant mechanisms leads to systems with actuation functionality built into the structure. Such structures distribute the actuation energy derived from a single remotely located actuator to the application surface. Compliant mechanisms can amplify the stroke of smart actuators and allow remote location of the actuator from the sensitive application environment [33].

The requirement is to establish a kinematically functional design that generates the desired output motion when subjected to prescribed input forces using topological synthesis. The size and shape of individual elements can be optimized with imposed constraints. Performance constraints include minimizing the energy loss in the mechanism, obtaining desired motion or force amplification, and non-buckling of the elements under the action of applied forces and external loads [33].

2.3.2 Advantages of compliant mechanism [34]

Compliant mechanisms have special features which make them useful in many applications. Compliant mechanisms can be built in one piece as modern manufacture technologies make it possible to build mechanisms in one piece by etching. It will reduce the total number of parts required to accomplished a specified task and realize a significant reduction in weight by using compliant mechanisms over their rigid-body counterparts. Compliant mechanisms are desirable because there is no backlash

in the transfer of motion. They require no assembly or much less assembly work than rigid-body mechanisms. Compliant mechanisms also have smaller number of joints such as pin and sliding joints. These results in less wear and need for lubrication. Vibration and noise cause by the turning and sliding of joints of rigid-body mechanisms may also be reduced in many applications. Since compliant mechanisms rely on the deflection of flexible members, energy is stored in the form of strain energy in the flexible members. This stored energy is similar to the potential energy in a deflected spring and the effects of the springs may be integrated into a compliant mechanism design.

Probably the most important advantage of compliant mechanisms is the ease in which they are miniaturized. Simple microstructures, actuators and sensors are widely used and many other microelectromechanical systems (MEMS) show great promise. The reduction in the total number of parts and joints offered by compliant mechanisms is a significant advantage in the fabrication of micro mechanisms. Compliant micro mechanisms may be fabricated using technology and materials similar to those used in the fabrication of integrated circuits. Moreover, a variety of motions are possible with compliant mechanisms as there is no limitation on how a properly shaped and sized continuum of material should deform. The flexible nature of the components of compliant mechanisms provides the capability of absorbing overloads without immediate structural failure. As flexure is permitted in compliant mechanisms, they can be equipped with non-mechanical actuators such as thermal, fluid pressure, electrostatic, magnetic, piezoelectric and shape memory alloy. In addition, compliant mechanisms can significantly benefit the field of smart structures as they provide a

simple and cost-effective means to generate desired shape changes without excessive quantity of actuators.

2.3.3 Challenges of compliant mechanisms [34]

There are also several challenges and disadvantages of compliant mechanisms for some applications. Perhaps the greatest challenge is the relative difficulty in analyzing and designing of compliant mechanisms. Fatigue analysis is vital for compliant mechanisms. Flexural pivots are essential parts of deformation and transfer of force and energy. Flexural pivots are relatively thinner and smaller than other main rigid parts of the mechanisms. Since compliant members are often loaded cyclically, it is important to design those members such that they will have sufficient fatigue life to perform their functions well. Compliant mechanism also cannot be too complicated in shape as it will bring more difficulty in manufacturing.

2.3.4 Design of a compliant gripper mechanism [35]

It is required to find a systematic method for design of a compliant mechanism starting from functional specifications to derive the topology (configuration) of a compliant mechanism given the desired input and output forces and displacements. The design will be based on linear elastic models. Geometric non-linearities due to large deformation and the dynamic response characteristics will not be taken into account during the synthesis stage.

The goal of structural optimization is to design the topology, size and shape of the mechanism that best performs given specific design requirements. The criteria for a compliant transmission design are as followed:

- Required kinematic motion with both magnitude and direction
- Desired mechanical and geometry advantage
- Material properties
- Stress limitations
- Buckling instabilities
- Actuator characteristics

For example an actuator provides a 2 μm displacement and generates 40 μN force and the desired output is 20 μm displacement against a resisting force of 3 μN . This design will require a 10:1 compliant motion amplifier which can be designed systematically by employing the energy formulation. The main advantage of the formulation is that conservation of energy applies for any mechanical system. A piezoelectric actuator model can be included in the system and electrical energy needed to drive the actuator can be calculated as the input work. Due to stored strain energy a compliant mechanism can only achieve a maximum of 80% efficiency.

Design of compliant mechanisms is previously be done by trial and error methods. A better design tool called topological optimization is used to design mechanical structure with high performance and low weight. This method allows the user to specify the mechanical or geometrical advantages of the compliant mechanisms and the topologies obtained require only little intervention before transfer to the fabrication process. The topology optimization problem can be defined as to distribute

a given amount of material within a design domain for a prescribed mechanical or geometrical advantage such that the mechanical efficiency is maximized. The optimization problem is formulated as minimization of least squares errors in obtaining the prescribed elastic behaviour and is solved using sequential linear programming method (SLP). SLP is the solving of a nonlinear program by a sequence of linear approximations and using linear programming to solve each one. The linear approximations are usually done by using the first-order Taylor expansion.

The energy in compliant mechanisms is no longer conserved between the input and output point because of the strain energy stored in the flexible parts of the mechanisms. The relationship between the input and output of the compliant mechanism is described in Figure 2.15.

Figure 2.15: The input and output relationship for the compliant mechanism.

To establish the optimal topology of a mechanism, the design domain is meshed using ground structure of frame elements as shown in Figure 2.16.

Figure 2.16: Example of design domain meshed by ground frame elements structure [35].

Figure 2.17: (a) Truss element ground structure for topology optimization, (b) Possible topology after removing some elements [35].

Figure 2.18: (a) Initial design, (b) Optimal solution, (c) Verification [35].

Figures 2.17 and 2.18 show the use of truss elements to design and optimize the compliant mechanism. Sequential linear programming is also another method to optimize the design of the compliant mechanism by defining the input and output values of the desired design domain as illustrated in Figure 2.19.

Figure 2.19: Optimized compliant mechanisms with design domain and input and output load cases [36].

2.3.5 Flexure hinges design [37]

The microgripper mechanism design includes flexure hinges as these hinges can provide small angular motion without friction and backlash. While simple in shape and operation, the flexures are mathematically complex to compute. The flexure hinge is basically a mechanical member that is compliant in bending about an axis, known as the input or sensitive axis, but rigid about the cross axes.

The single-axis flexure is usually made of rectangular or circular cross-section area. The rectangular cross-sectioned flexure will be investigated as it is easier to fabricate for this case. An example of a flexure hinge is shown in Figure 2.20.

Figure 2.20: The typical structure of flexural hinge.

Using the basic bending moment equation [37]:

$$M_z(x) = EI_z(x) \frac{d^2 y}{dx^2}$$

$$\frac{d^2 y}{dx^2} = \frac{M_z(x)}{EI_z(x)} \quad (2.1)$$

whereby $M_z(x)$ is the bending moment about the z-axis and $I_z(x)$ is the moment of inertia about the z-axis. By integrating the above equation, the angular deflection α_z about the z-axis can be calculated as:

$$\alpha_z = \frac{dy}{dx} = \int \frac{M_z(x)}{EI_z(x)} dx \quad (2.2)$$

Figure 2.21: Parameters of a notch hinge.

With reference from Figure 2.21, moment of inertia about the z-axis is

$$I_z = \frac{bh^3}{12} = \frac{b[2(R + \frac{t}{2} - R \cos \theta)]^3}{12} = \frac{8b(R + \frac{t}{2} - R \cos \theta)^3}{12} \quad (2.3)$$

where $\frac{h}{2} = R + \frac{t}{2} - R \cos \theta$

Thus from equation (2.2)

$$\alpha_z = \frac{dy}{dx} = \int \frac{M_z(x)}{EI_z(x)} dx = \int_b \frac{[M_z + F_y(R \sin \theta_m - R \sin \theta)]R \cos \theta}{E \frac{2b}{3} [R + (\frac{t}{2}) - R \cos \theta]^3} d\theta \quad (2.4)$$

as given by *Paros and Weisbord* [37].

The compliance of the flexure can be given as:

$$C_r = \frac{\alpha_z}{M_z} \approx \frac{9\pi R^{\frac{1}{2}}}{2Ebt^{\frac{5}{2}}} \quad (2.5)$$

on the assumption that $\frac{t}{2R} \ll 1$ and $t \ll h$.

The bending stiffness, also known as the spring rate, of the flexure is the reciprocal of the compliance and is thus given by [37]:

$$K = \frac{2Ebt^{\frac{5}{2}}}{9\pi R^{\frac{1}{2}}} \quad (2.6)$$

The relative thickness is defined as: $\beta = \frac{t}{2R}$

The maximum stress $\sigma_{\max}$ occurs at the surface point of the hinge section with the minimum thickness which is represented by t as shown in Figure 2.22.

Figure 2.22: Maximum bending stress at a circular notch hinge.

By using the basic bending moment equation [38],

$$\sigma_{\max} = \frac{M_{\max} c}{I} = \frac{K \alpha_{\max} \left(\frac{t}{2}\right)}{\frac{bt^3}{12}} = \frac{6K \alpha_{\max}}{bt^2} \quad (2.7)$$

where c is defined as the distance from the neutral line to the point of interest, $M_{\max}$ is the moment and $\alpha_{\max}$ is the angle of rotation that occurs during the maximum stress.

Another theoretical option for calculating the compliance is given by Tseytlin [38].

$$\text{For thin circular notch hinges: } C_r = 4 \times \frac{1 + [1 + 0.1986(\frac{2R}{t})]^{\frac{1}{2}}}{Eh(\frac{t}{2})^2} \quad (2.8)$$

$$\text{For thick notch hinges: } C_r = 4 \times \frac{1 + [1 + 0.5573(\frac{2R}{t})]^{\frac{1}{2}}}{2Eh(\frac{t}{2})^2} \quad (2.9)$$

$$\text{For intermediate notch hinges: } C_r = 4 \times \frac{1 + [1 + 0.373(\frac{2R}{t})]^{\frac{1}{2}}}{1.45Eh(\frac{t}{2})^2} \quad (2.10)$$

2.3.6 Pseudo-rigid-body model

In order to analyze systems that undergo large, non-linear deflections like compliant mechanism systems, the pseudo-rigid-body model is used. The pseudo-rigid-body model is a simplified method to model the deflection of flexible members using rigid-body components that have equivalent force-deflection characteristics. Rigid-link mechanism theory may then be used to analyze the compliant mechanism. In this way the pseudo-rigid-body model is the bridge that connects the rigid-body mechanism

theory and compliant mechanism theory as shown in Figure 2.23. The method is particular useful in the design of compliant mechanisms [38].

Figure 2.23: Pseudo-rigid-body model as a bridge that connects the rigid-body mechanism theory and compliant mechanism theory.

For each flexible segment, the pseudo-rigid-body model predicts the deflection path and force-deflection relationships. The motion is modeled by rigid-links attached at pin joints. Spring constants are added to the model to accurately predict the force-deflection relationships of the compliant segments.

Figure 2.24: The parallel links mechanism to be investigated with pseudo-rigid-body model.

The Figure 2.24 shows the link of the parallel linkage system with small-length flexural pivots at both its ends. Consider the deflected cantilever beam with two segments: one short and flexible and the other longer and rigid as shown in Figure 2.25.

Figure 2.25: Small-length flexural pivot [35].

The small segment is called a small-length flexural pivot with length $l \ll L$. Since the flexible section is much shorter than the rigid section, the motion of the system may be modeled as two rigid links joined at a pin joint called the characteristic pivot. The characteristic pivot is located at the centre of the flexural segment as shown in Figure 2.26. This is an accurate assumption as the deflection occurs at the flexible segment in which the length is much smaller than that of the rigid segment.

Figure 2.26: Pseudo-rigid-body model for small-length flexural pivot [35].

The x and y coordinates of the beam end, b and a respectively, are approximated as:

$$a = \frac{l}{2} + (L + \frac{l}{2}) \cos \theta \quad (2.11)$$

and

$$b = (L + \frac{l}{2}) \sin \theta \quad (2.12)$$

The beam's resistance to deflection is modeled using a torsional spring with spring constant K . The torque required to deflect the torsional spring through an angle θ is given by:

$$T = K\theta \quad (2.13)$$

The bending angle of the beam can be written as:

$$\theta = \frac{Ml}{EI} \quad (2.14)$$

Since torque is equal to the moment, that is, $T = M$,

$$K = \frac{EI}{l} \quad (2.15)$$

Figure 2.26 also illustrates the small-length flexural pivot beam with a force F at the free end. It is useful to view the force in terms of the components of the force that are normal and tangential to the path of the beam end. The component tangent to the path (normal to the rigid-link), F_t , causes a moment at the torsional spring given by:

$$T = F_t \left(L + \frac{l}{2} \right) \quad (2.16)$$

Since this component of the force contributes to the deflection of the pseudo-rigid-body link, it is called the active force. The force normal to the path, F_n , is called a passive force because it does not contribute to the bending moment. If the angle of the applied force remains the same throughout the beam deflection, the active and passive components change when the deflection changes. The active force is:

$$F_t = F \sin(\phi - \theta) \quad (2.17)$$

Thus the relationship between the applied force F and the bending angle of the beam θ can be derived by combining equations 3, 6 and 7.

$$T = F \sin(\phi - \theta) \times \left(L + \frac{l}{2} \right)$$

$$F = \frac{K\theta}{\left(L + \frac{l}{2} \right) \sin(\phi - \theta)} \quad (2.18)$$

2.4 MICRO-ACTUATORS

2.4.1 Introduction to piezoelectric ceramics

Piezoelectric materials can be used to convert electrical energy into mechanical energy and vice versa. As a result, piezo-actuators are used in many applications such as measuring technology, disk drive, microelectronics and precision mechanics. Polycrystalline ferroelectric ceramic materials such as barium titanate and lead zirconate titanate (PZT) have been developed with improved properties. PZT ceramics are available in many variations and are still widely used materials for actuator and sensor applications [39].

The ceramic consists of tiny piezoelectric crystallites at random orientation, quite different from that of a single crystal. The random orientation causes the overall ceramic to be inactive with no piezoelectricity, until some methods are found to polarize the ceramic as a whole entity. One method is poling which requires special compositions that are not only piezoelectric but also ferroelectric. This method involves polarizing the ceramic under the influence of an electric field. Electrodes are applied to the ceramic and a sufficiently high electric field is applied to the material such that the dipoles within the individual crystallites are reoriented and aligned in the direction of the electric field. During this process, there is a small expansion of the material along the poling axis and a contraction in both directions perpendicular to it. An illustration of this process is shown in Figure 2.27.

Figure 2.27: Dimensional change of a ceramic as a result of poling [39].

There are two main types of piezo-actuators. They are the low voltage devices that require about 100 volts for full motion, and the high voltage devices that require about 1000 volts for full extension. The maximum electrical field the PZT ceramic can withstand is 1 - 2 kV/mm.

2.4.2 Piezoelectric coefficients and directions

The piezoelectric effects are dependent on direction because of the anisotropic nature of PZT ceramics. Translational axes 1, 2 and 3 and rotational axes 4, 5 and 6 are introduced in Figure 2.28. The direction of polarization is established during the poling process by a strong electrical field applied between two electrodes. For actuator applications, the piezo properties along the poling axes are the most essential with the largest deflection. Piezoelectric materials are characterized by several coefficients. The important coefficient is the strain coefficient, d_{ij} . Double subscripts are introduced to link electrical and mechanical quantities. The first subscript gives the direction of the excitation while the second describes the direction of the system

response. For example, d_{33} applies when the electric field is along the polarization axis number 3 and the strain is along the same axis. While d_{31} applies if the electric field is in the direction of axis 3, but the strain is in the axis 1 which is orthogonal to the polarization [40].

Figure 2.28: Orthogonal system describing the properties of a poled piezoelectric ceramic [40].

2.4.3 Advantages of piezoelectric actuator

The piezo effect directly converts electrical energy into mechanical work only by consuming the electrical energy during movement. Static operation like holding loads does not consume power. The piezo actuator has neither gears nor rotating shafts. Its displacement is based on solid state dynamics and shows little wear and tear. General PZT has excellent endurance properties even after several billion cycles. In addition piezo actuators offer the fastest response time available in microseconds time constants. This characteristic makes it very suitable to act as an actuator [40].

2.4.4 Axial and transversal piezoelectric actuator [41]

Piezoelectric ceramic materials like lead zirconate titanate (PZT) have been used as solid-state actuators and sensors for many applications such as precision positioning, noise and vibration sensing and cancellation, linear motor etc. Piezoelectric actuators are used in very precise control positioning in applications as small efficient motors. Piezoelectric actuators have short reaction times and fast acceleration. Another advantage of using piezoelectric actuators is that there is no electromagnetic field or heat generated. Thus piezoelectric motors offer significant advantages relative to electromagnetic devices. The significant disadvantage of piezoelectric actuators is that they are limited by very small displacements which are around 0.15 – 0.2% of their overall dimensions.

When a piezoelectric actuator is stimulated by an electrical signal, it responds with a precisely controlled mechanical displacement. If the displacement is prevented, a force will develop. There is an approximately linear relationship between the input signal and the actuator response as a stroke or force. In positioning applications, an infinitely small change in the operating voltage produces an infinitely small change in position. Piezoelectric actuators are usually categorized according to the type of mechanical displacements which are axial, transverse or flexional. An axial actuator receives a signal applied parallel to the direction in which the piezoelectric ceramic element is polarized and responds with an extension in the same direction, known as the d_{33} -mode. The ceramic element progressively extends in height as voltage is increased. A transversal actuator receives a signal applied parallel to the direction in which the ceramic element is polarized and responds with a contraction perpendicular

to the direction of polarization, known as the d_{31} -mode. The ceramic element progressively shortens in length as voltage is increased. Figure 2.29 shows the directions that are used to define axes of the piezoelectric ceramic. The axial and transversal actuators are illustrated in Figure 2.30.

Figure 2.29: Directions of forces affecting a piezoelectric element [41].

Figure 2.30: (a) Axial actuator, (b) Transversal actuator [41].

The axial and transversal actuators can be constructed as a single ceramic element or a compound actuator that consists of multiple ceramic elements stacked together connected in parallel. The actuators exhibit high stiffness and produce small movements or large forces. However the actuators are not independent of the environment as temperature, humidity and other factors can affect the performance of

the actuators. Therefore the piezoelectric actuators may not perform as anticipated when used in a real application.

The important governing equations for a simple axial actuator consisting of a single ceramic element are shown below [41].

Stroke with force held constant: $\Delta h = d_{33}V$ (2.19)

Force with height of ceramic element held constant:

$$F = \frac{d_{33}LwV}{s_{33}^E h} \quad (2.20)$$

Stiffness: $K_E = \frac{Lw}{s_{33}^E h}$ (2.21)

Minimum impedance frequency (resonance frequency) of ceramic element with one surface of element secured to a base:

$$f_m = \frac{N_L}{2h} \quad (2.22)$$

where h is the height or thickness of the element, V is the voltage, w is the width of the ceramic element, s_{33}^E is the elastic compliance and N_L is the longitudinal frequency constant for ceramic material.

For a transversal actuator, its corresponding equations are given as:

Stroke with force held constant: $\Delta L = \frac{d_{31}LV}{h}$ (2.23)

Force with length of ceramic element held constant:

$$F = \frac{d_{31} w V}{s_{11}^E} \quad (2.24)$$

Stiffness:
$$K_E = \frac{h w}{s_{11}^E L} \quad (2.25)$$

Minimum impedance frequency (resonance frequency) of ceramic element with one surface of element secured to a base:

$$f_m = \frac{N_p}{2L} \quad (2.26)$$

where h is the height or thickness of the element, V is the voltage, w is the width of the ceramic element, s_{11}^E is the elastic compliance and N_p is the longitudinal frequency constant for ceramic material.

The piezoelectric charge constants d_{33} and d_{31} typically increase when the electric field strength is increased. Thus at high field strength, the stroke of the actuator and the force can exceed the anticipated values. As the charge constants are also temperature dependent, the stroke of the actuator may increase with increasing temperature. The method in which the actuator is mounted can further complicate the theoretical calculations and thus measurements made under operating conditions are always the reliable indications for actual actuator performance. As the maximum stroke of a simple axial actuator can approach only up to 0.2% of its dimension of the elements, the stroke may be too small for practical use. Thus a compliant mechanism is required to amplify the stroke so as to achieve the desired actuation.

2.4.5 Shear mode piezoelectric actuator

The mechanical motion generated by piezoelectric ceramic actuators is usually related to either d_{33} longitudinal mode or d_{31} transverse mode. Although the shear mode d_{15} is the highest piezoelectric charge coefficient in soft PZT ceramic materials, the piezoelectric shear vibration mode is seldom used in actuator and transducer applications [42]. For shearing motion configuration, the soft PZT ceramic plate is poled along its length and electrically driven across its thickness. An example of the application of shear mode piezoelectric micro-actuator is in the use of dual stage actuator system for magnetic disk drives as shown in Figure 2.31 [43].

Figure 2.31: Schematic view of a shear mode piezoelectric actuator [43].

The actuator that uses the shear mode of piezoelectric elements is shown in Figure 2.32. The piezoelectric element is polarized horizontally and when the voltage is applied, the element shears as indicated by the solid lines. The displacement per volt of the piezoelectric element can be estimated by

$$\frac{L}{V} = nd_{15} \quad (2.27)$$

where L is the element displacement, n is the number of layers, V is the applied voltage and d_{15} is the shear mode piezoelectric charge constant. This actuator can be easily manufactured at low cost because of its simple stacked structure. Two layers and multi-layers shear mode actuator are also investigated to reduce the driving voltage required.

Figure 2.32: Schematic view of the shear mode of piezoelectric element and the arrow indicating the poling direction [43].

2.5 CONTACT MECHANICS THEORY

2.5.1 Introduction [44]

In general, when two solid bodies are brought into contact, they will either be in contact at a point, along a line or over a surface. In complex loading and geometries, a combination of the above contacts is possible. The initial contact region will change

when a load is applied on the bodies in contact. The change in the contact regions depends on the magnitude of the load, the elastic properties of the bodies and the frictional properties of the contact surfaces. Thus the contact problem becomes highly nonlinear and requires the use of advanced numerical methods to analyze the problem.

The finite element method (FEM) is widely used to perform the analysis of contact problems. The three approaches of the finite element method to solve contact problems are the Lagrange Multiplier method, the Penalty method and the Hybrid method.

Contact mechanics is the study of load transfer in mechanical assemblages whereby the nature of contact interaction between the two contacting surfaces is complicated. The difficulty of the contact problem is also due to the fact that the contact behaviour is affected by the material properties of the two contacting surfaces, their texture, surface finish, local topology at the contact surface, rate of loading, magnitude of loading, direction of loading and the supports of the bodies. While considerable success has been achieved in solving frictionless contact problems using the numerical approach, the complicated nonlinear nature of friction still presents a major challenge in the development of accurate modeling of contact problems.

2.5.2 Classification of contacts [44]

Contact problems can be classified as conforming and non-conforming problems. A brief classification is given for both conforming and non-conforming contact in a frictionless or frictional situation.

Figure 2.33: (a) Conforming and (b) Non-conforming contact problems.

Conforming contact problems involve contact surfaces of similar profile. They are considered to be load independent as the character of their final contact solution is not affected by the magnitude of the external load. Since the size of the contact area is independent of load, the load history is therefore not important in this case. An example of a conforming contact problem is a flat-ended punch resting on a flat surface as shown in Figure 2.33 (a). In the absence of friction and if the contact area is known, the conforming contact problems are completely linear and thus can be solved directly with one single load step without iterations. With the presence of friction, a load incremental procedure is not required but iterations have to be done to determine the contact status along the contact interface. Since lateral displacement is restrained due to friction, the problem is solved with the final load applied in one step.

Non-conforming contact problems involve contact bodies with different profiles. When two unloaded solid bodies are brought into contact, they touch only at a point like contact of 2 spheres or along a line like contact of a cylinder on a flat surface. The size of the initial contact area will change when the bodies are subjected to an external load. This change is affected by many factors such as initial profiles, material properties, rate of applied load, direction of load, contact characteristics and methods of support. An example of the non-conforming problem is a sphere resting on a flat surface as shown in Figure 2.33 (b). In the absence of friction, lateral displacement is not restrained and the frictionless contact problem is a linear one. Step by step load increment is not necessary but iteration has to be performed in order to obtain the correct contact area of the deformed bodies under the applied load. The presence of friction makes the solution of the contact state an iterative procedure. Since the size of the contact area is not known, an incremental method is employed to find the load step corresponding to each elemental increase or decrease in the area of contact until the total external load or final area of contact is reached.

2.5.3 Hertz contact theory [44]

The contact pressure distribution, contact area and deformation between two smooth circular bodies in contact were studied and presented by Hertz in 1881 [44]. Hertz contact theory is restricted to non-conforming linear elastic solids. The Hertz contact model is also based on the assumptions that the dimensions of the contact area must be smaller than that of the radii of the two bodies, and the surface contact is frictionless so that only a normal force or pressure is transferred. Two non-conforming surfaces with radius R_1 and R_2 brought into contact over an infinitesimal

area are considered. Under the application of a normal load F , the bodies will undergo deformation which gives rise to a contact area as shown in Figure 2.34.

Figure 2.34: The Hertz contact model of two circular bodies in contact.

Assuming that the deformations are fully elastic, the Hertz contact formula can be used. The contact radius, a , and the normal deformation, δ , are given by [44]:

$$a = \left(\frac{3FR}{4E_e} \right)^{\frac{1}{3}} \quad (2.28)$$

$$\delta = \left(\frac{9F^2}{16RE_e^2} \right)^{\frac{1}{3}} = \frac{a^2}{R} \quad (2.29)$$

The composite Young's modulus, E_e , is defined such that:

$$\frac{1}{E_e} = \frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \quad (2.30)$$

where ν_1 , E_1 and ν_2 , E_2 are the Poisson's ratio and elastic modulus of materials 1 and 2 respectively. R is the composite radius of curvature defined by the radii of the contact bodies 1 and 2 such that:

$$\frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2} \quad (2.31)$$

However Hertz's equations are valid under the assumption that $a \ll R_1$ or R_2 . The values of the contact area and the deformation are to the power of $\frac{2}{3}$ to that of the applied force. Equation (2.29) can be rewritten as:

$$F^2 = \frac{16}{9} RE^2 \delta^3$$

$$F = \left(\frac{16}{9} R \right)^{\frac{1}{2}} E \delta^{\frac{3}{2}} = \frac{4}{3} R^{\frac{1}{2}} E \delta^{\frac{3}{2}}$$

Differentiating F with respect to δ ,

$$\begin{aligned} \frac{\partial F}{\partial \delta} &= \frac{3}{2} \times \frac{4}{3} R^{\frac{1}{2}} E \delta^{\frac{1}{2}} = 2 R^{\frac{1}{2}} E \delta^{\frac{1}{2}} \\ &= 2 R^{\frac{1}{2}} E \left(\frac{9 F^2}{16 R E^2} \right)^{\frac{1}{6}} = 2 R^{\frac{1}{2}} E \left(\frac{3 F}{4 R^{\frac{1}{2}} E} \right)^{\frac{1}{3}} \\ &= \left[\left(2 R^{\frac{1}{2}} E \right)^3 \right]^{\frac{1}{3}} \left(\frac{3 F}{4 R^{\frac{1}{2}} E} \right)^{\frac{1}{3}} = \left(8 R^{\frac{3}{2}} E^3 \right)^{\frac{1}{3}} \left(\frac{3 F}{4 R^{\frac{1}{2}} E} \right)^{\frac{1}{3}} \\ &= (6 E^2 F R)^{\frac{1}{3}} \end{aligned}$$

Thus the vertical stiffness, K_v , is defined as:

$$K_v = \frac{\partial F}{\partial \delta} = (6 E^2 F R)^{\frac{1}{3}} \quad (2.32)$$

Figure 2.35: Schematic drawing of the pressure distribution at the contact area between the 2 bodies.

The contact pressure distribution is semi-elliptical with radius r and has a maximum value at the origin and zero value at the ends as shown in Figure 2.35. The pressure distribution function at the contact region can be written as [45]:

$$P(r) = \frac{3F}{2a^2\pi} \sqrt{1 - \frac{r^2}{a^2}} \quad (2.33)$$

Thus the maximum contact pressure at $r = 0$ can be given by:

$$P_{\max} = \frac{3F}{2a^2\pi} \quad (2.34)$$

or when equation (2.28) is substituted into equation (2.34),

$$P_{\max} = \frac{1}{\pi} \left(\frac{6FE_e^2}{R^2} \right)^{\frac{1}{3}} \quad (2.35)$$

For the situation when two cylindrical bodies are in contact, the contact half-width and the maximum contact pressure are given by [44]:

$$a = \left(\frac{4F'R}{\pi E_e} \right)^{\frac{1}{2}} \quad (2.36)$$

and

$$P_{\max} = \left(\frac{F'E_e}{\pi R} \right)^{\frac{1}{2}} \quad (2.37)$$

where F' is the applied force per unit length in the axial direction of the cylinder.

2.5.4 Contact with adhesion [45]

With the use of simple models, the effects of contact in the sliding and rolling of elastic bodies can be investigated. The simplest type of contact is that of a smooth, frictionless sphere or cylinder of one material touching or pressed against a flat plane surface of another material. The contact is maintained with a constant sustained force on the flat surface. The contact and deformation between the two bodies are illustrated in Figure 2.36.

Figure 2.36: Model of a nonlinear elastic circular body in contact with a plane surface at a normal force F .

In the scale of micrometers, the effects of the surface interaction forces in the immediate vicinity of the contact region become significant. Thus in some cases, the observed contact radius is higher than that predicted by Hertz contact theory as shown

in Figure 2.37. The increased contact area can be explained by the attractive interaction between the two bodies.

Figure 2.37: Larger contact radius than predicted value due to adhesive forces.

Two theories take into account the adhesion effect at the surfaces and modified the Hertz contact theory by introducing an adhesive interaction parameter called work of adhesion, defined by the symbol w . The first theory is by Johnson, Kendall and Roberts (JKR) in which the contact radius is defined as:

$$a^3 = \frac{3R}{4E_e} \left[F + 3\pi w R + \sqrt{6\pi w R F + (3\pi w R)^2} \right] \quad (2.38)$$

which is also known as the JKR model. In this model, a strong adhesive force is supposed to act at the equilibrium distance z_0 . For all distances larger than z_0 , the interaction force is disregarded. Thus attractive forces act only within the contact region. The JKR model predicts the force required to separate the two contacting bodies is given by the pull-off force, F_c [44].

$$F_c = \frac{3}{2} \pi R w \quad (2.39)$$

The second theory is the Derjaguin, Muller and Toporov (DMT) model which includes the adhesion outside the contact area by considering the long-range attractive Van der Waals forces. The pull-off force in the DMT model is given by [45]:

$$F_c = 2\pi R w \quad (2.40)$$

The interaction forces for the Hertz, JKR, and DMT models are compared to a realistic interaction in Figure 2.38. There is no attractive force in the Hertz model but only hard wall repulsion at the contact region. The JKR model includes a short-range adhesion force which act only within the contact zone. The DMT curve shown represents a long-range surface attractive force. For an actual interaction force, the integral of the force– distance attractive well corresponds to the work of adhesion, w .

Figure 2.38: The interaction forces used in the different models [45].

CHAPTER 3

THEORY AND DESIGN

3.1 COMPLIANT MECHANISM

3.1.1 Microgripper mechanism design approach

The primary design objective is to design a microgripper which can produce the desired output force and displacement in the desired location and direction with the specified input from a microactuator. The objective function can be shown as a flow chart below.

Figure 3.1: The initial design domain for the microgripper.

The initial design problem specification is shown in Figure 3.1. The forces and displacements of the actuation input and the end-effector output are stated with the desired directions. The design domain (or workspace) is to be within a given size limit of 10 mm by 10 mm. The boundaries of the design would be fixed to a support base. An initial trial and error design is introduced and the mechanical advantage and geometrical advantage are calculated using rigid free body diagram method. The design is then modeled and analyzed with finite element analysis simulation.

3.1.2 Parallel mechanism

The gripper design includes a parallel guiding mechanism in which two opposing links remain parallel throughout the mechanism's motion. The mechanism consists of a simple four-bar in which the opposing links have the same length, thus forming a parallelogram. With compliant parallel guiding mechanism, the joint friction, backlash and need for lubrication can be eliminated. The design proposed is shown in Figure 3.2. Figure 3.3 illustrates the motion of the rigid link parallel guiding mechanism.

Figure 3.2: Compliant parallel guiding mechanism design.

Figure 3.3: Rigid link parallel guiding mechanism.

The link BCP is the coupler link between links AB and CD and it moves parallel to itself during motion. Defining Δx and Δy as the horizontal and vertical displacements of point B when it moves to point B' and r as the length of link AB and CD, the following simple equations can be shown.

$$\Delta x = r \cos \beta \quad (3.1)$$

$$\Delta y = r (1 - \sin \beta) \quad (3.2)$$

Figure 3.4 shows a closer look at the links DEF and DE'F' and it can be seen that they are equilateral triangles. The link is at position E'F'G' after a vertical displacement d is imposed downwards on the link FG.

Figure 3.4: Schematic diagram of links DEF and DE'F'.

Letting length ED be equal to m ,

$$DF = m \tan \alpha = DF'$$

Thus getting,

$$DF' \cos \beta = d$$

$$m \tan \alpha \cos \beta = d$$

$$\cos \beta = \frac{d}{m \tan \alpha}$$

From Equation (19),

$$\Delta x = r \cos \beta = \frac{rd}{m \tan \alpha} \quad (3.3)$$

The geometrical advantage for this mechanism design can be written as

$$GA = \frac{\text{output_displacement}}{\text{input_displacement}} = \frac{2\Delta x}{d} = \frac{2r}{m \tan \alpha} \quad (3.4)$$

3.1.3 Displacement and force calculations

In order to achieve an output displacement that is perpendicular to the direction of the input displacement, the design is chosen as shown in Figure 3.5. The parallel linkages act as transmission links between the input and output displacements.

Figure 3.5: Schematic model of flexure arm bending movement for force and displacement calculations.

By defining F_t as the force that is applied on the flexure arm with respect to time as shown in Figure 3.5, it can be calculated from Equation (2.18) as:

$$F_t = \frac{K_T \theta}{(L + \frac{l}{2}) \sin(\phi - \theta)} \quad (3.5)$$

K_T is described as the total stiffness provide by the four flexural hinges.

Thus $K_T = 4K = \frac{4EI}{l}$, where E is the Young Modulus and I is the area moment of inertia.

$$F_t = \frac{\frac{4EI}{l}\theta}{(L + \frac{l}{2})\sin(\phi - \theta)} \quad (3.6)$$

Since $\phi = (180^\circ - \alpha) + \theta$, $m = L + \frac{l}{2}$ and $\cos \theta = \frac{d}{m \tan \alpha}$ from equation 3,

$$F_t = \frac{4EI \cos^{-1}(\frac{d}{m \tan \alpha})}{(lm) \sin(180^\circ - \alpha)} \quad (3.7)$$

In order to show that the above equation is correct in terms of SI units, the units on the left hand side (LHS) of the equation must be equivalent to the right hand side (RHS).

Defining $F_t = [N]$, $E = [\text{Pa}] = [\text{Nm}^{-2}]$, $I = [\text{m}^4]$, $d = [\text{m}]$, $m = [\text{m}]$, $l = [\text{m}]$

$$RHS = \frac{[Nm^{-2}][m^4] \frac{[m]}{[m]}}{[m][m]} = [N] = LHS$$

However if the *Paros and Weibord* model for flexural stiffness is used from Equation (2.6):

$$K = \frac{2Ebt^{\frac{5}{2}}}{9\pi R^{\frac{1}{2}}}$$

Then

$$F_i = \frac{4(2Ebt^{\frac{5}{2}})\cos^{-1}(\frac{d_i}{m \tan \alpha})}{9\pi R^{\frac{1}{2}}m \sin(180^\circ - \alpha)}$$

$$= \frac{8Ebt^{\frac{5}{2}}\cos^{-1}(\frac{d_i}{m \tan \alpha})}{9\pi R^{\frac{1}{2}}m \sin(180^\circ - \alpha)} \quad (3.8)$$

The SI units on both sides of the equation are verified by:

$$RHS = \frac{[Nm^{-2}][m][m^{\frac{5}{2}}]}{[m^{\frac{1}{2}}][m]} = \frac{[Nm^{\frac{3}{2}}]}{[m^{\frac{3}{2}}]} = N = LHS$$

Figure 3.6 shows the forces and displacements applied to the flexure mechanism. d_o and d_i are defined as the output and input displacements respectively.

Figure 3.6: Diagram of the model to show the relationship between the input and output displacement.

Deriving from Equation (3.3),

$$d_o = 2 \times \frac{rd_i}{m \tan \alpha} \quad (3.9)$$

The actuating force, F_t can be written as:

$$F_T = 2F_t \cos(180^\circ - \phi) = 2F_t \cos(\theta - \alpha)$$

$$F_T = 2F_t \cos[\cos^{-1}(\frac{d_i}{m \tan \alpha}) - \alpha] \quad (3.10)$$

An initial dimension model was drawn out as shown in Figure 3.7. The units of dimensions are all in micrometers (μm).

Figure 3.7: The initial design model showing the dimensions in μm .

In order to establish a relationship between d_i (input) and d_o (output), the required parameters of the design model are interpreted as shown in Table 2.

Table 1: Parameters for the displacement graph.

r (μm)	m (μm)	α (deg)	d_i (input) (μm)
1 st variable	2 nd variable	3 rd variable	4 th variable

Equation (3.9) is used to plot the graph of d_o (output) against $\alpha(^{\circ})$. The values of r and m are computed as $100\text{ }\mu\text{m}$ and $50\text{ }\mu\text{m}$ respectively. The value of d_i (input) is $1\text{ }\mu\text{m}$ and α is in the range of 1° to 90° . The result was then plotted as shown in Figure 3.8.

Figure 3.8: Graph of d_o (output) against $\alpha(^{\circ})$.

Equation (3.9) is used again to obtain the graph of d_o (output) against d_i (input). In the graph the values of α are varied as 35° , 45° and 55° , and the value of d_i (input) is varied in the range of 1 to $4\text{ }\mu\text{m}$. The results are shown in Figure 3.9.

Figure 3.9: Graph of d_o (output) against d_i (input).

It is also desirable to calculate the value of the force required to produce the input displacement d_i . The input force to be calculated will be F_T as illustrated in the previous Figure 3.6. Equations (3.8) and (3.10) in the earlier pages are used to relate the force F_T to the displacement d_i . The values of the parameters used in both equations are given in Table 3.

Table 2: Parameters for calculating the input force.

E	b	t	m	α	R	d_i
0.078 N/ μm^2	100 μm	5 μm	50 μm	45°	5 μm	variable

Figure 3.10 shows the result of how the input force varies with the value of the input displacement.

Figure 3.10: Graph of input force F_T against d_i (input).

3.2 PIEZOELECTRIC ACTUATION METHOD

The objective is to design and develop a simple structured, piezoelectric based actuation method for the microgripper. The actuator utilizes the shear deformation of piezoelectric elements to drive the compliant mechanism of the microgripper. Single - layer piezoelectric elements are used in order to simplify the integration and assembly process. The shearing motion of the piezoelectric ceramic will allow relative movement of the top and bottom electrode surfaces. The entire microgripper consists of basically three parts which are the compliant gripper mechanism, shear mode piezoelectric actuator and the base. An exploded assembly view of the microgripper

system is shown in Figure 3.11. The integration of the shear piezoelectric actuator will enable the sliding motion between the movable part of the microgripper mechanism and the support base.

Figure 3.11: Assembly view of the microgripper system.

Figure 3.12: (a) Shear piezoelectric actuator drives the arm forward when voltage signal is given, (b) Actuator pulls the arm back when the polarity is reversed.

The shear piezoelectric actuator will drive the movable arm of the compliant gripper mechanism back and fro, thus providing the desired actuation to grasp and release the target micro-parts or biological cells. A cross-section schematic diagram is shown in Figure 3.12 to illustrate the driving principle of the shear mode piezoelectric actuator.

For the assembly process, the piezoelectric actuator is secured to a solid base surface. Of the three possible methods of gluing, soldering and clamping, the overall best method would be gluing. An epoxy or acrylate glue will provide a strong, flexible and non-conducting bond. The flexible nature of a glue bond also eliminates the possibility of vibration-induced fatigue during operation. Soldering is a less reliable method of securing the actuator because vibrations over time in the piezoelectric ceramic element can cause the bond to fail [47]. Figure 3.13 shows the 3-D assembled view of the microgripper system.

Figure 3.13: 3-D assembled view of the microgripper system.

3.3 CONTACT MECHANICS ANALYSIS

3.3.1 Contact between microgripper and target object

The objective is to investigate the contact mechanics in which two contacting bodies are the end-effector of the microgripper and the target object, which is either an industrial micro-part or a biological cell in the micrometer range. Due to the complexity of contact problems, it is assumed that the contacting bodies have frictionless surfaces and the emphasis is to determine the relationship between the applied forces, contact area and contact stress. Examples of micro-objects that require grasping are shown in Figure 3.14.

Figure 3.14: Diagrams of an embryo cell and polystyrene particles in the microns range [46].

The proposed microgripper design uses two parallel end plates in motion to grasp and release the target object as shown in Figure 3.15.

Figure 3.15: Model of an elastic circular body in contact with gripper surfaces.

However, the flat surface of the microgripper is modified to a curved surface. The centre of curvature of the radius R_2 is on the same side as the centre of curvature of radius R_1 . Thus the model to investigate the contact mechanics relationship between the contacting microgripper and the object is illustrated in Figure 3.16.

Figure 3.16: Model of an elastic circular body in contact with curved gripper surfaces.

The assumptions for this model are:

- 1) The two bodies are made of elastic material.
- 2) The dimensions of the contact area must be smaller than that of the radii of the two bodies.
- 3) The surface contact is frictionless so that only a normal force is present and no tangential force is regarded.
- 4) The deformation of the target object is very small so that there is no plastic deformation and the elasticity law is still applicable.
- 5) The adhesive forces at the contact region are neglected.

Hertz contact theory is commonly used for simple contact problems consisting of a cylindrical geometry and a flat surface. Thus Hertz contact theory is used to obtain values of the contact area, deformation and contact stress of the target object. The calculations of the contact mechanics include the target object of a spherical geometry made up of materials which are commonly used in research and life science experiments. Spherical glass beads are used as models for analytical, sedimentation, and separation applications. The other uses for glass beads are as spacers, insulators and reflectors. Spherical pollens and spores provide model specimens for analyzing and controlling pollens, a common industrial contaminant. Pollens can also be used as controls for test systems, or as models of biological cells when stained with cytochemical dyes. Metal and mineral particles and spheres meet the need for an assortment of materials for chemical, metallurgical and engineering studies. In addition, suspensions of large copolymer microspheres are used as model systems for fluid mechanics experiments, and as challenge particles for large pore filtration

systems. The microspheres are composed of cross-linked polystyrene polymer and are also useful as experimental particles for acoustical and optical analytical systems.

Polystyrene microspheres and borosilicate glass beads are selected for usage in the contact problem as their technical material properties can be easily obtained. For the microgripper material, it is proposed to be either Foturan glass or silicon. The essential properties of the materials used in the contact problem are shown in Table 4.

Table 3: Properties of the various materials

	Microgripper		Target object	
Material	Foturan glass	Silicon	Borosilicate glass	Polystyrene
Elastic modulus (N/mm ²)	78 x 10 ³	150 x 10 ³	63 x 10 ³	3 x 10 ³
Poisson ratio	0.22	0.17	0.20	0.33

3.3.2 Contact interaction results

For this simplified contact problem between the microgripper and the target object, the Hertz equations are utilized. The parameters, R_1 and R_2 , which are the radius of the target sphere and the radius of the curvature of the microgripper end, are fixed at values of 0.01 mm and 0.03 mm respectively. The variables are the material properties which are namely ν_1 , ν_2 , E_1 and E_2 , and the applied force F . Four cases are considered for the calculations and the graph plots. The four cases are summarized in Table 5.

Table 4: The materials of the microgripper and object.

Case 1	Foturan glass microgripper grasping polystyrene object
Case 2	Foturan glass microgripper grasping borosilicate glass object
Case 3	Silicon microgripper grasping polystyrene object
Case 4	Silicon microgripper grasping borosilicate glass object

The graphs show the relationships of the contact radius, deformation and maximum contact pressure with the applied normal force. The range of the applied force is from 0.01 to 0.05 N. Any higher value than this range will result in undesirable large deformation of the target object and most probably plastic deformation as well.

Figure 3.17: Graph of deformation against the applied force.

Figure 3.17 shows the deformation against applied force graph for the different cases. Since the material polystyrene has smaller elastic modulus compared to borosilicate glass, it is expected that polystyrene will undergo a larger deformation compared to borosilicate glass when the same force is applied. The selection of Foturan glass or silicon as material for the microgripper does not have much significant effect on the deformation of the target object as seen from the graph.

Figure 3.18: Graph of contact radius against the applied force.

As the deformation increases, the corresponding contact radius increases as well. This is shown in Figure 3.18. In addition, the graph shows that the polystyrene object will have a larger contact radius with the microgripper as compared to the borosilicate glass object.

Although the deformation of the object is relatively small, the maximum contact pressure on the object itself is significantly large as illustrated in Figure 3.19. The graph shows that the borosilicate glass material has higher maximum contact pressure than polystyrene. This is due to the smaller deformation experienced by the borosilicate glass and since pressure is inversely proportional to the contact area, the resulting pressure becomes very high for the borosilicate glass.

Figure 3.19: Graph of maximum pressure against the applied force.

The results obtained in this chapter will be compared with the results from the finite element analysis and the experimental results found in the later chapters.

CHAPTER 4

FINITE ELEMENT ANALYSIS OF MICROGRIPPER MECHANISM

4.1 Introduction

Finite element analysis (FEA) software ANSYS is used as a modeling and calculation tool. The program can provide solutions to structural and contact problems. The simulation work involves the use of structural elements to model the compliant mechanism and also the use of contact elements to model the interacting bodies in the contact problem. Visual animations as well as numerical calculation results are obtained from the program.

4.2 Simulation of compliant mechanism

The basic objective of the proposed compliant mechanism design is to obtain desirable output displacements of the tip of the microgripper with a series of known input actuating displacements. The compliant mechanism is an elastic mechanism that performs its kinetic function through its intended flexibility. In this case the flexibility is enabled by the notch hinges on the parallel arms linkage. Thus the performance of the compliant will be largely dependent on the material properties and the geometrical design of the mechanism.

The 3-dimensional model of the proposed microgripper created in a CAD environment is given in Figure 4.1. The design is a planar, monolithic mechanism consisting of parallel links, flexural notch hinges and actuating links.

Figure 4.1: CAD model of the microgripper mechanism.

The diagram of the final design of the monolithic compliant mechanism for analysis in the simulation is shown in Figure 4.2.

Figure 4.2: Compliant mechanism design of the microgripper.

The shaded areas represent the area designed to be bonded to the support base and the piezoelectric actuator. The mechanism inside the dotted box is more clearly shown in Figure 4.3. The dimensions in Figures 4.2 and 4.3 are in mm. The dimensions of the components shown in Figure 4.3 are optimized in the CAD and simulation program, so as to enable an output displacement of up to 200 μm at the microgripper tips. The

notch hinges are given a radius of 2.5 μm so that the flexural hinges can allow deformation and yet can avoid fatigue stress failure.

Figure 4.3: A close-up of the gripping mechanism.

For the structural simulation, the material of the microgripper compliant mechanism is defined to be Foturan glass. The finite element mesh used is triangular meshing and the element type is PLANE 82. The structural area is subdivided into 2166 elements and represented with 3638 nodes. The essential parameters to be defined for the simulation are summarized in Table 6. The parameters r and l define the different heights of the parallel links as shown in Figure 4.3. The values of r and l are varied in

the simulation and the corresponding performance of the mechanism is obtained from the analysis.

Table 5: Parameters for the structural simulation

Elastic modulus of material, E	78000 N/mm ²
Poisson ration of material, ν	0.22
Density of material	2.37 g/cm ³
Element type	PLANE 82
Mesh size	4

PLANE 82 is a high order version of the 2-D, 8-noded element. The 8-node element is defined by eight nodes having two degrees of freedom at each node with translations in the x and y directions. The element has plasticity, creep, swelling, stress stiffening, large deflection, and large strain capabilities [47]. Free meshing of triangle elements is applied on the design area. Structural constraints of zero displacement in both x and y directions are applied on lines 2 to 6. This will hold the bulk of the body in place. A displacement load is then applied onto line 1 in the negative y direction. The load is described as the input displacement, d_i , and is varied in the range of 2 to 10 μm . Figure 4.4 shows the meshing and the locations of the load constraints. The plane thickness is set to be 50 μm .

Figure 4.4: The finite element mesh and load constraints of the mechanism.

4.3 Compliant mechanism simulation results

When an input displacement, d_i , of $10\ \mu\text{m}$ is applied on the simulation model, the simulation results show the structural deflection of the microgripper mechanism as illustrated in Figure 4.5. The dotted lines show the outlines of the shape of the original position of the mechanism.

Figure 4.5: FEA diagram of the structural deformation of the mechanism.

Figure 4.6 shows the diagram of the close-up of the microgripper tip ends after deflection. It can be seen that the tip ends have moved towards each other to perform a grasping action. In addition, the center tip withdraws backwards to allow more space for the target object to be gripped. The output displacement of each of the microgripper end tip is defined as d_o as shown in Figure 4.6. Thus the overall output displacement of the pair of microgripper end tips, D_o , can be described as $2d_o$.

Figure 4.6: Movement of the microgripper tip ends.

The von Mises stresses on the compliant mechanism design upon deformation are illustrated in Figure 4.6. The maximum stress and the bulk of the stresses are concentrated at the flexural hinges of the compliant mechanism shown in Figure 4.7. The stress result is expected as the flexural hinges experience the most bending deformations within the whole mechanism. The FEA model used in Figure 4.7 has a value of 1.5 mm for its parallel arm length and the input displacement is 10 μm . The value of the maximum stress obtained at the flexural hinges is 828 N/mm^2 , which is within the elastic region of the mechanism material.

Figure 4.7: FEA stress analysis of the compliant mechanism.

For the simulation, the variables are the length of the parallel arms, l , and the input displacement values, d_i . The FEA results of the total gripping distance, D_o , are tabulated in Table 7.

Table 6: Results of the total gripping displacement, D_o .

	D_o (μm)				
d_i (μm)	$l = 1 \text{ mm}$	$l = 1.25 \text{ mm}$	$l = 1.5 \text{ mm}$	$l = 1.75 \text{ mm}$	$l = 2 \text{ mm}$
2	7.06	8	8.88	9.72	10.54
4	14.16	16.06	17.8	19.48	21.12
6	21.28	24.14	26.78	29.3	31.78
8	28.44	32.24	35.78	39.18	42.48
10	35.64	40.4	44.84	49.1	53.26

The ratio of the output displacement and the input displacement is characterized as the geometrical advantage. Figure 4.8 shows that the longer the parallel arms, the greater the geometrical advantage is.

Figure 4.8: Graph of total gripping distance against input displacement.

From Equation (27) in chapter 3, the theoretical gripping distance can be described as:

$$D_o = \frac{2rd_i}{m \tan \alpha}$$

m is defined as 0.375 mm and α is 45° for the model in the simulation.

The results of the theoretical D_o are plotted along with the results obtained from the FEA simulation for varying values of l . The combined graph presented in Figure 4.9 shows that the theoretical results have higher geometrical advantage compared to the simulation results. The reason is that the theoretical results are based on a simplified 2-D joints and linkage model which does not take into account the deformation of the mechanism.

Figure 4.9: Graph of theoretical results of gripping distance compared to the FEA results.

Another simulation is performed in which the applied input displacement is substituted by an input force instead. This will provide the results of the amount of force required to create the desired input displacement.

Figure 4.10: Input force applied on the compliant mechanism.

The force is applied on a node as shown in Figure 4.10. The length of the parallel arms is fixed at 1.5 mm. The input and output displacements of the compliant mechanism are plotted against the input force in Figure 4.11. The input and output displacements increase linearly as the input force is increased.

Figure 4.11: Graph of input and output displacement against input force.

4.4 Simulation of contact between microgripper and micro-part

The estimation of grasping forces and displacements requires the knowledge of contact characteristics including the relationship between the normal force and contact area, and the pressure distribution profile at the gripper-object interface. The objective here is to study the contact between the microgripper and the target micro-part with large and nonlinear deformations based on nonlinear elastic finite element method (FEM) modeling. FEM allows the investigation of parameters such as contact stress within the contact area of the gripper and object, and the nonlinear force-displacement relationship. The end-effector of the microgripper and the target object are modeled as

homogeneous, linear elastic and isotropic materials. Hyperelastic contacts involving the gripper and the object resulting in large hyperelastic deformations are also investigated in the later portion of the chapter.

Contact problems fall into two general classes which are rigid-to-flexible and flexible-to-flexible. In rigid-to-flexible contact problems, one or more of the contacting surfaces are treated as rigid which means that it has a much higher stiffness relative to the deformable body it contacts. In general, any time a soft material comes in contact with a hard material, the problem may be assumed to be rigid-to-flexible [47].

4.5 Contact simulation of microgripper

The contact relationship between a circular object and the end-effectors of the microgripper is shown in 2-dimensions in Figure 4.12. A circular area is located at the origin and the two ends of the microgripper are placed at a distance apart. In the simulation, the microgripper ends are displaced towards the target circle. The dimensions in Figure 4.12 are in mm.

Figure 4.12: Illustration of the 2-D contact simulation.

In this case, the circular object is flexible while the microgripper is rigid. Thus upon contact, the circular object will be deformed whereas the microgripper will still retain its shape. The dimensions shown in Figure 4.12 are all in mm. For the first simulation, material 1, which is the material of the microgripper, is defined as Foturan glass. Material 2, which is the material of the target object, is defined as polystyrene. The finite element mesh used is quadrilateral meshing and the element type is PLANE 82. The circular object is subdivided into 99 elements and represented with 274 nodes. The microgripper is subdivided into 116 elements and represented with 297 nodes.

If there is no friction on the surface of contact, the circular object will tend to slip away from the jaws of the microgripper. In order to avoid slipping, a coefficient of friction value of 0.2 is included inside the simulation. Two additional contact element types are also defined in the simulation to solve the contact relationship. The essential parameters to be defined for the simulation are summarized in Table 8.

Table 7: Parameters for the contact simulation

Elastic modulus of material 1, E_1	78000 N/mm ²
Poisson ration of material 1, ν_1	0.22
Elastic modulus of material 2, E_2	3000 N/mm ²
Poisson ration of material 2, ν_2	0.33
Element type 1	PLANE 82
Element type 2	TARGE 169
Element type 3	CONTA 172
Mesh size	4
Coefficient of friction	0.2

TARGE 169 is used to represent 2-D target surfaces for the associated contact elements. Translational or rotational displacement can be imposed on the target segment element. CONTA 172 is used to represent contact and sliding between a 2-D target surface like TARGE 169 and a deformable surface, defined by this element. The element is applicable to 2-D structural and coupled field contact analyses. This element is located on the surfaces of 2-D solid elements with mid-side nodes. Contact occurs when the element surface penetrates one of the target segment elements on a specified target surface [47].

4.6 Contact simulation results

The values of the nominal deformation, δ , and the maximum pressure, $P_{\max}$, can be obtained from the simulation results. The two nodes at the circumference of the circle are the initial points of contact and their locations are where the maximum pressure will occur as shown in Figure 4.13.

Figure 4.13: Maximum pressure at the initial point of contact.

When a deformation of 1 μm is applied on both sides of the circular object, the contact pressure is obtained as shown in Figure 4.14.

Figure 4.14: Contact pressure results from the FEA.

The contact pressure distribution in Figure 4.14 shows that the maximum pressure occurs at the center of the contact region and decreases exponentially to zero value towards the outer end of the contact region. This distribution is similar to the Hertz's contact theory. The corresponding stress experienced by the object is represented in Figure 4.15. The negative stress values represent the compressive nature of the stress while the positive values represent the tensile stresses.

Figure 4.15: Stress results on the object from the FEA.

Figure 4.16: Graph of maximum contact pressure against deformation of target object.

The relationship between the nominal deformation and the maximum pressure on the surface of the target object can be obtained from the simulation results. The values of the deformation are varied by changing the deflection of the microgripper towards the target object. The values of the maximum contact pressure from the theoretical results and from the FEA results are plotted in the graph in Figure 4.16. The values of the maximum pressure obtained from the FEA are significantly lower than the theoretical results. This may be because the theoretical calculations do not take into account the increase of the contact surface area between the target object and the gripper when the target object begins to deform under pressure.

CHAPTER 5

MICROGRIPPER PROTOTYPE BASED ON MEMS FABRICATION

In this research, silicon micro-machining method is used to fabricate the MEMS prototype of the microgripper model for investigative and evaluation purposes in order to achieve a working prototype for future tests. The micro-machining processes in making this MEMS prototype will be described in this chapter.

5.1 Mask preparation

The stencil being applied for the generation of a desired pattern or design in resist-coated wafers is called a photomask. The mask data containing the design of the microgripper is streamed out using Tanner L-Edit software for the procurement of the photomask. In practice, a photomask, a nearly optically flat glass (transparent to UV) or quartz plate (transparent to UV) with a metal (chromium layer) absorber pattern, is placed into direct contact with the photoresist coated surface with the wafer exposed to the ultraviolet radiation. The absorber pattern on the photomask is opaque to ultraviolet light while the glass or quartz is transparent. The mask being used in this project is the plastic mask which is low in cost.

Figure 5.1: Chrome photomask with pattern design layout.

Tanner L-Edit Pro is the program used for the design of the mask layout. It integrates layout editing, verification, and place and route in one program tool. After the mask has been laid out, the layout is given to a mask vendor to procure the chrome mask. The chrome photomask containing the microgripper design patterns is shown in Figure 5.1. The light field or dark field image on the mask is then transferred onto the wafer surface. The silicon microgripper is to be fabricated with conventional micromachining technology using a single mask level in order to simplify the whole process. The smallest dimension on the mask pattern is 25 μm . A P-type $\langle 100 \rangle$ silicon wafer was used as the substrate. The next step is to prepare the silicon wafer with pre-process cleaning.

5.2 Pre-process wafer preparation

The preparation process begins with initial cleaning of the boron-doped, P-type silicon wafers for the purpose of removing contaminants prior to photoresist coating.

The silicon wafers have diameter of 100 mm, thickness of 450 μm , <100> orientation and resistivity of 1 to 10 $\Omega\text{-cm}$. The wafer surfaces were cleaned using hydro-fluoric (HF) acid to free the wafer from contamination of foreign particles. Wafer cleaning is an important process to remove the contaminants before wafer priming. The contaminants include solvent strains such as acetone, isopropyl alcohol and xylene, smoke particles as well as dust from the environment.

In this wafer preparation, wet cleaning is used and the chemical solvent used to clean the wafer was Piranha ($\text{H}_2\text{SO}_4 + \text{H}_2\text{O}_2$) at a temperature of 120°C with the wafer being soaked in the solvent for 20 minutes. De-ionized water (DI H_2O) was then used as the rinsing agent. The rinsed wafer was placed in a cassette and is spin dry for a total of about 4 minutes. The 4 minutes include spinning dry at a speed of 800 rpm while being rinsed with de-ionized water and after which, it was spun dry at a speed of 2300 rpm followed by 2500 rpm. The wafer cleaning tank and spin-dry machine used for this wet cleaning process are shown in Figure 5.2.

(a)

(b)

Figure 5.2: (a) Wafer cleaning tank, (b) spin-dry machine.

Next, photolithography was used to transfer the photo mask pattern of the microgripper onto the silicon wafer surface. In this process, the mask material for subsequent dry etching process was photoresist.

5.3 Photolithography process

In this case, photolithography is used to transfer the pattern of the microgripper design onto the silicon wafer. The lithography process flow is briefly depicted in the steps shown in Figure 5.3. The pattern drawn with L-Edit program on a computer (Figure 5.3a) is transferred onto a photomask (Figure 5.3b) as mentioned earlier. The mask is an optical plate that has on its surface a photo definable opaque chrome material in the desired pattern prepared by a mask vendor.

After mask making, the pattern transfer begins when the wafer substrate (Figure 5.3c) is spin-coated with a photoresist which is a photosensitive organic polymer (Figure 5.3d). The adhesion promoter, which in this case is hexamethyldisilazane (HMDS), is used to assist the subsequent resist coating. The photoresist that is spin coated as the masking layer has a uniformed thickness of approximately 10 μm . Hard bake process on a hotplate at 100~110 $^{\circ}\text{C}$ is performed to evaporate the coating solvent and to densify the resist after spin coating.

The substrate and mask are brought into contact and UV light is beamed through the mask and onto the photoresist (Figure 5.3e). Thus the photoresist is exposed with the patterned mask and the exposed photoresist is developed to create the desired etch mask for subsequent dry etching in order to create the finished silicon mold wafer.

The photoresist under the transparent portions of the mask will be exposed, causing it to become soluble in a developing solution. This is known as a positive photoresist while a negative photoresist will give the inverse pattern. The wafer and mask are then separated, and the exposed photoresist is removed in the developing solution (Figure 5.3f). The photoresist is now used as a protective mask to transfer the pattern into the underlying material via etching. When the etch patterning is completed, the photoresist is removed by a solvent.

Figure 5.3: Photolithography process flow depicted in steps a to f.

The more detailed steps and equipments involved in the photolithography for the micro-fabrication of the microgripper are as described below.

- i) The hexamethyldisilazane (HMDS) priming under vacuum using the Karl Suss Delta VPO Primer started off with baking and nitrogen purging at 100 °C for 60 seconds for the removal of surface moisture. The HDMS priming then continued

for 20 seconds and is followed by baking and nitrogen purging for another 60 seconds, as shown in Figure 5.4.

Figure 5.4: HMDS priming equipment.

- ii) The wafer underwent cooling at a temperature of 24 °C for 60 seconds.
- iii) Spin coating was performed using the Karl Suss Delta 80 Coater with the covered lid closed at a speed of 1500 rpm to achieve a photoresist (AZ – 9260) thickness of 10 μm as shown in Figure 5.5.

Figure 5.5: Spin coating to achieve photoresist.

- iv) The resist AZ 9260 was pre-baked at 110 °C for about 4 minutes as shown in Figure 5.6.

- v) After baking, the process was then followed by cooling the resist for 60 seconds at a temperature of 24 °C.
- vi) The wafer was subsequently exposed to a radiation source with wavelength of 365 nm and ultraviolet intensity at 9.3 mW/cm² for 55 seconds as shown in Figure 5.7.

The type of contact is hard, which means contact printing. The AZ 9260 was then developed by soaking the wafer in acetone with a developing time of 90 seconds.

vii) The wafer was rinsed with de-ionized water and placed under the microscope for inspection of the transfer of the pattern.

viii) Finally, resist hard baking was carried out at a temperature of 110 °C for 3 minutes to finish the whole photolithography process.

5.4 Etching process

The post treatment of photolithography process is the etching process. It is used to remove the unwanted material of the substrate. After the pre-treatment processes were completed, deep silicon etching was performed using Deep Reactive-Ionized Etching (DRIE) to achieve the desired microgripper silicon prototype. The STS deep RIE machine is used to perform etching on the silicon wafer. The deep RIE process is controlled by a computer that is linked to the deep RIE machine as shown in Figure 5.8.

Figure 5.8: STS deep RIE machine for the etching process.

After the desired trench depth, approximately 450 μm , was reached, the etching process was completed. Then the photoresist was stripped and removed by soaking in acetone solvent for 30 min as shown in Figure 5.9.

Figure 5.9: Removal of photoresist by acetone.

The series of the micro-fabrication techniques (photolithography and deep reactive ionized etching) discussed here are being practiced to machine a working prototype of the microgripper. The purpose of the micro-fabricated prototype is to explore one of the several different fabrication methods that can be used to produce a working prototype of the actual size as the proposed microgripper design, so that subsequent testing or experiments can be performed on the prototype for further studies. During the course of fabricating the desired working prototype, many wafers were used with several attempts on micro-fabricating the prototype being performed. The fabrication results were obtained and observed using the Scanning Electron Microscope (SEM) and will be discussed later in this chapter.

5.5 Silicon prototype of microgripper

Using the photolithography and etching methods mentioned in the previous sections, the micro-size silicon prototypes of the microgripper were obtained. Figure 5.10 shows the SEM view of the fabricated silicon microgripper device. The microgripper size is 7 mm x 10 mm in area and 0.45 mm in thickness. The parallel arm length is 100 μm and its width is 10 μm . The gap distance between the two end tips is 100 μm .

Figure 5.11 shows the SEM photo of another modified design of the microgripper in which the area of the center tip is increased. The larger area of the center tip layout means that the etching time is shortened as the volume of material to be removed is reduced.

Figure 5.11: SEM photo of the modified design.

The 3-D nature of the silicon microgripper can be seen in Figure 5.12. The vertical sidewalls of the silicon microstructure are fully etched through by the DRIE process. The structure designs of the flexure notch hinges at the ends of the linkage arms are formed as depicted in Figure 5.13. These fine features on this silicon microstructure were obtained only after numerous attempts on the photolithography and etching processes.

Figure 5.12: Tilted view of the microgripper

Figure 5.13: Closer view of the etched features on the silicon prototype

5.6 Advantages and disadvantages of MEMS micro-fabrication

MEMS micro-fabrication can be performed as easily on one device as on a thousand. Batch processing can make thousands of identical devices not subject to the variations present in individually constructed objects. Geometrical control is very important for microstructures such as the microgripper mechanism. Photolithography allows largely varying geometries to be patterned in the same space with micrometer dimensional accuracy. In addition, dimensions of the same feature can be varied on a mask to create many different but similar structures. Micro-fabrication have many different types of advantages such as higher surface-area-to-volume ratio, small size, small sample volumes, geometrical control, constrained geometries, single-cell analysis, batch processing, high throughput, or integration. With the advantages of mass production of similar microstructures and good geometrical control, micro-fabrication using MEMS method for the microgripper mechanism is most suitable and efficient.

However the MEMS fabrication process has its disadvantages as well. MEMS fabrication usually has multiple complex process steps and long development times, depending on the complexity of the structure. The challenging issues faced in this research during the MEMS micro-fabrication include samples failure due to causes such as contaminations, poor quality substrate, over etching, and basically design flaws in the initial stage. These factors result in the failed samples that have distortion in the design shape, as well as fine features which are not completely etched through. An example of a failed silicon sample is shown in Figure 5.14, in which fracture occurs at the critical areas such as the notch hinges due to poor etching.

The diagram showing another failed microgripper sample is shown in Figure 5.15. The failure occurs at the hinges where the features have the finest resolution. Once taken out of the cleanroom environment, the weak critical points on the features of the microgripper will break easily when subjected to external forces from human hands or from mechanical tools. This is because silicon microstructures fabricated by MEMS have relatively poor structural integrity and strength.

Figure 5.15: Another failed silicon sample.

5.7 Piezoelectric ceramic actuator

Two types of piezoelectric (PZT) actuators have been considered for integration into the microgripper mechanism. One is a low voltage piezoelectric translator which consists of many piezoceramic disks inside a steel cylinder under pre-load, and the other is a shear mode piezoelectric ceramic which is of dimension $2 \times 2 \times 0.5 \text{ mm}^3$. The former is utilized to drive the WEDM microgripper prototype which will be described in the next chapter, while the latter will be used to integrate into the micro-fabricated silicon microgripper.

The shear mode piezoelectric actuator is made up PZT ceramic material which is referred to as American Piezo Ceramic (APC) material number 850. Both sides of the ceramic are coated with nickel electrodes with wire leads attached to the surfaces to provide electrical connections. Figures 5.16 to 5.18 show the PZT ceramic in more

details. The piezoelectric properties of the ceramic are obtained from the vendor. The shear piezoelectric charge constant, d_{15} , of the ceramic is 0.59 nm/V.

Figure 5.16: PZT shear ceramics (a) with electrical leads and (b) without leads.

Figure 5.17: Close-up view of the PZT ceramic and the Ni electrodes.

Figure 5.18: Side view of the PZT ceramic with its electrical leads.

The PZT ceramic is made up of lead zirconate titanate material and the SEM micrograph is used to observe its microstructure. The SEM pictures are shown in Figure 5.19.

5.8 Piezoelectric simulation

Piezoelectric is the coupling of structural and electric fields, which is a natural property of materials such as quartz and ceramics. Applying a voltage to a piezoelectric material creates a displacement, and vibrating a piezoelectric material generates a voltage. Possible piezoelectric analysis types are static, modal, pre-stressed modal, harmonic, pre-stressed harmonic, and transient. A piezoelectric model in ANSYS requires permittivity (or dielectric constants), the piezoelectric matrix, and the elastic coefficient matrix to be specified as material properties.

Figure 5.20: Piezoelectric model for simulation.

A piezoelectric transducer consists of a rectangular block of PZT material with its polarization direction aligned along the Z axis as shown in Figure 5.20. Electrodes are placed on the two surfaces orthogonal to the polarization axis. The objective is to determine the shear mode and linear mode displacements of the piezoelectric ceramic when a potential difference is applied across its electrodes. The $2 \times 2 \times 0.5 \text{ mm}^3$ design of a PZT APC 850 material is modeled inside ANSYS. The parameters of the ceramic material properties are computed into the ANSYS data. An electric potential is applied across the electrode terminals of the ceramic and the results of the shear and linear mode displacement are investigated. The simulation model is shown in Figure 5.21.

Figure 5.21: Model of the piezoelectric ceramic being investigated.

The dielectric matrix defines the electrical permittivity in units of Farad/meter and is denoted by $[\epsilon]$. The matrix is a 3×3 diagonal matrix for a 3-D model and the data is entered as an orthotropic material property as PERX, PERY and PERZ. The dielectric matrix is thus defined as:

$$[\epsilon] = \begin{bmatrix} \epsilon_{11} & 0 & 0 \\ 0 & \epsilon_{22} & 0 \\ 0 & 0 & \epsilon_{33} \end{bmatrix} \text{ F/m}$$

$$= \begin{bmatrix} 1750 & 0 & 0 \\ 0 & 1750 & 0 \\ 0 & 0 & 1750 \end{bmatrix} \text{ F/m}$$

The piezoelectric matrix relates the electric field to stress and has units of Coulombs/meter². The matrix is a 6×3 matrix for a 3-D model and the data is input as a material data table of type PIEZ. The matrix is defined as:

$$[e] = \begin{bmatrix} e_{11} & e_{12} & e_{13} \\ e_{21} & e_{22} & e_{23} \\ e_{31} & e_{32} & e_{33} \\ e_{41} & e_{42} & e_{43} \\ e_{51} & e_{52} & e_{53} \\ e_{61} & e_{62} & e_{63} \end{bmatrix} \text{ C/m}^2$$

$$= \begin{bmatrix} 0 & 0 & -80.65 \\ 0 & 0 & 0 \\ 0 & 0 & 38.46 \\ 0 & 0 & 0 \\ e_{51} & 0 & e_{53} \\ 0 & 0 & 0 \end{bmatrix} \text{ C/m}^2$$

For piezoelectric coupled-field analysis, the stiffness matrix $[c]$ is used instead of the elastic modulus and Poisson's ratio.

$$[c] = \begin{bmatrix} 6.3 & 0 & 0 & 0 & 0 & 0 \\ & 6.3 & 0 & 0 & 0 & 0 \\ & & 5.4 & 0 & 0 & 0 \\ & & & 2.0 & 0 & 0 \\ & & & & 1.0 & 0 \\ & & & & & 1.0 \end{bmatrix} \times 10^{10} \text{ N/m}^2$$

For linear mode, the value of e_{51} in the piezoelectric matrix is input as 27.78. For the shear mode, the value of e_{53} in the piezoelectric matrix is input as 27.78.

The deformation results of the piezoelectric ceramic in the linear mode and shear mode when a voltage of 100 volts is applied on the electrodes are shown in Figures 5.22 (a) and 5.22 (b) respectively. For this simulation, no displacement constraint is pre-applied on the model.

(a) Linear mode

(a) Shear mode

Figure 5.22: The deformed shapes of the piezoelectric ceramic with 100 V applied voltage.

Figure 5.23: The piezoelectric ceramic with the bottom electrode constrained.

Figure 5.23 shows the deformation of the piezoelectric ceramic when the ground electrode is constrained at zero displacement. This is to simulate the ceramic as being attached to a non-movable support. The voltages applied across the electrodes are varied from 0 to 500 volts and the corresponding maximum displacements in the actuating direction, which is x-direction, are obtained.

Figure 5.24: Graph of maximum actuation displacement against applied voltage.

The results are plotted in Figure 5.24. The graph shows that the piezoelectric ceramic in shear motion can provide slightly higher displacements than that in linear motion. The simulation of the shear mode with boundary constraints on one side of the PZT ceramic shows a more linear plot compared to the unbounded models because its motion of displacement is more relative to the ground.

5.9 Measurement and testing of PZT ceramic

An experimental setup is developed to analyze the performance of the shear-mode piezoelectric actuator. The schematic diagram of the setup is shown in Figure 5.25. The purpose is to evaluate the relationship between the displacement and voltage applied to the actuator so that calibration can be done. The $2 \times 2 \times 0.5$ mm piezoelectric ceramic actuator is driven by a circuitry which includes a function generator and a piezoelectric power amplifier.

Figure 5.25: Schematic diagram of the experimental setup for testing of the PZT actuator.

The voltage signal generated by the function generator is amplified by the power amplifier to the required driving voltage value. The magnitude of the amplified voltage can reach up to 250 V. The power amplifier drives the high voltages to the PZT actuator through the electrical leads soldered onto the nickel electrodes on the surface of the piezoelectric ceramic.

For the experiment, a square wave voltage signal is generated. The amplitude and frequency of the square wave is varied using the function generator. The oscilloscope monitors and displays the high voltage output that is supplied by the power amplifier.

Figures 5.26 to 5.28 show the experimental setup and equipment used for the tests.

Figure 5.26: Complete experimental setup for testing.