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On-Chip Actuation of Compliant Bistable Micro-Mechanisms

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ON-CHIP ACTUATION OF COMPLIANT BISTABLE MICRO-MECHANISMS

by

Michael S. Baker

A thesis submitted to the faculty of
Brigham Young University
in partial fulfillment of the requirements for the degree of
Master of Science

Department of Mechanical Engineering
Brigham Young University
April 2002
of a thesis submitted by

Michael S. Baker

This thesis has been read by each member of the following graduate committee and by majority vote has been found to be satisfactory.

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Date John N. Harb
As chair of the candidate’s graduate committee, I have read the thesis of Michael S. Baker in its final form and have found that (1) its format, citations, and bibliographical style are consistent and acceptable and fulfill university and department style requirements; (2) its illustrative materials including figures, tables, and charts are in place; and (3) the final manuscript is satisfactory to the graduate committee and is ready for submission to the university library.

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ABSTRACT

ON-CHIP ACTUATION OF COMPLIANT BISTABLE MICRO-MECHANISMS

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Department of Mechanical Engineering
Masters of Science

Two compliant bistable micro-mechanisms have been developed which can be switched in either direction using on-chip thermal actuation. The energy storage and bistable behavior of the mechanisms are achieved through the elastic deflection of compliant segments. The pseudo-rigid-body model was used for the compliant mechanism design, and for analysis of the large-deflection flexible segments. To achieve on-chip actuation, the mechanism designs were optimized to reduce their required rotation, allow them to be switched using linear-motion thermal actuators. The modeling theory and analysis are presented for several design iterations. Each iteration was successfully fabricated and tested using either the MUMPs or SUMMiT surface micromachining technology.
ACKNOWLEDGMENTS

There are several people that deserve thanks for their help and their contribution to this work. First, I’d like to thank my advisor, Dr. Larry Howell, for his help and encouragement. It is largely because of him that I decided to continue for my Masters degree. His excitement and enthusiasm for doing cool stuff is inspiring and he has been an excellent example and a mentor.

Also, thanks goes out to everyone in the graduate lab. It’s been a great environment to work in and it will be missed. Specifically, I would like to thank Daniel Wilcox and Craig Lusk for their assistance in taking scanning electron microscope images and Scott Lyon for his help with everything.

Finally, and most importantly, I’d like to thank my wife Tami for all her support and encouragement. Our late night talks about my work and its associated challenges have helped me to organize my thoughts and ideas and have been invaluable to me throughout this entire process. Thanks...

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CHAPTER 1 INTRODUCTION

1.1 Background and Thesis Objective

The purpose of this research was to develop a compliant bistable microelectromechanical device that could be switched in either direction using a method of on-chip actuation. The device modeling, design and optimization will be presented as well as a discussion of the actuation methods and a presentation of test results.

Microelectromechanical systems (MEMS) are mechanical systems made with feature sizes measured in microns. They are fabricated using the same techniques and materials as integrated circuits, and as such are typically planar devices. Current commercial examples of MEMS devices are limited, and include an airbag sensor by Analog Devices and the Texas Instruments Digital Micromirror Device. However, because of the advantages of MEMS devices, including small size, low cost, and the potential for integration with on-chip electronics, several applications are currently under active research including accelerometers, gyroscopes, micro-relay applications and fiber-optic switching.
A bistable mechanism is defined as a mechanism that has two stable positions within its range of motion. As the mechanism is moved from its first stable position to its second stable position, it passes through an unstable equilibrium position. A primary advantage of a bistable mechanism is that it only requires energy input to switch the mechanism from one state to the other, but input energy is not required to maintain the mechanism in either of the stable equilibrium positions. This behavior of the bistable mechanism is of particular importance at the micro-level where power consumption and heat dissipation are often a major concern. The advantages of increased stability, positional accuracy and low power consumption have been important driving factors in the research and development of micro-bistable devices.

1.2 Importance of Research

A bistable switch has numerous potential applications, including micro-valves, micro-switches, relays and micro-positioners. Because of its bistable nature, it would also be useful in fully autonomous micro-systems where the power supply is limited. Because a bistable device only requires energy to switch from one stable position to the other, but requires no energy input to maintain either stable state, the power consumption and heat generated by the device could be lower than a continuously powered switch or solid-state relay. Such a device could find use in mechanical memory arrays, where high data security is necessary, because they would maintain their states in the event of a loss in electrical power. In addition, a successful bistable relay could have a significant impact in the area
of telecommunications or any other application which requires a high isolation, low insertion loss relay.

Because MEMS devices are batch fabricated, they can be mass produced at very low cost. This low cost fabrication, combined with their small size and low power requirements has the potential to reduce significantly the cost of the products in which they are used.

1.3 Thesis Contributions

The primary contribution of this thesis is the demonstration of two bistable mechanism configurations that can be successfully paired with thermal actuators for on-chip actuation. Specifically, the design of the bistable mechanism itself was optimized to reduce its required rotation between stable positions such that it could be switched using existing thermal actuators. A model will be presented which allows for application specific design of a bistable mechanism through the use of an optimization algorithm. Some work was done to minimize the actuator power requirements, however, extensive optimization of the actuators is beyond the scope of this thesis. This work will focus on the bistable mechanism design itself, with attention to potential applications. The development of a commercially viable product will not be addressed and is left for future consideration.

1.4 Outline

To provide a more complete background and understanding of compliant bistable MEMS devices, the next chapter will review MEMS fabrication, compliant mechanism
design using the pseudo-rigid-body model (PRBM), and bistable mechanism theory.

Chapter 3 then reviews prior work in the field of MEMS relays and bistable devices.

Methods of actuation, including an overview of the thermal actuators used in this work, as well as a review of alternative actuation methods is also presented. The “Young” mechanism design is discussed in Chapter 4, with details of the design optimization performed to achieve on-chip actuation of this device. Chapter 5 presents details on the linear displacement bistable mechanism. Conclusions and Results are summarized in Chapter 6.
Before continuing with a discussion of this work, a review of MEMS technologies, bistable mechanism theory, and compliant mechanism design is necessary. This chapter will present an overview of MEMS technologies as well as a discussion of some of the challenges and difficulties associated with MEMS product development. It will also review compliant mechanism theory and the pseudo-rigid-body model.

2.1 MEMS Technologies

Microelectromechanical Systems (MEMS) have become a very active area of research in recent years. This interest is driven by the potential for MEMS to reduce the cost and size of many existing devices such as switches, relays and sensors. Also, in some applications, MEMS technologies present a way to accomplish tasks that have not otherwise been feasible, such as optical switching and high isolation, low insertion-loss RF switching.
MEMS are fabricated using processes and technologies developed for the integrated circuit industry. There are three primary fabrication techniques that are classified as MEMS technologies, including surface micromachining, bulk micromachining and LIGA. The work in this thesis has been done exclusively with surface micromachining. However, an understanding of all three techniques is important so a brief overview of each technology will be presented.

2.1.1 Surface Micromachining

In surface micromachining, devices are fabricated in planar levels that are deposited on a silicon wafer. Two different materials are deposited in alternating levels, with one material forming the structural levels, and the other forming the sacrificial levels. After deposition, each level is independently patterned using photolithography and then etched to create the structures and shapes of interest. After the deposition, pattern and etch process is complete, the sacrificial material is completely removed using a selective chemical etch that removes the sacrificial level but does not attack the structural material. The most common materials used in surface micromachining are polysilicon (poly) for the structural levels and silicon dioxide (SiO₂) or phosphosilicate glass (PSG) for the sacrificial levels. The release etch is then performed using hydrofluoric acid.

To illustrate the surface micromachining process, the steps for fabrication of a simple cantilevered beam will be presented. The fabrication begins with a silicon wafer substrate on top of which is deposited a uniform insulating layer, typically a low-stress nitride. The first sacrificial material is the deposited uniformly over the wafer and is pat-
terned and etched to form the cantilevered beam anchor as shown in Figure 2.1. Next, the polysilicon layer is deposited using low-pressure chemical vapour deposition (LPCVD), and is patterned and etched to form the cantilevered beam shape. The LPCVD process is conformal such that the polysilicon fills in the anchor holes in the sacrificial material as shown in Figure 2.2. For more advanced processes, additional layers of sacrificial and
structural materials can be deposited, however for this simplified example a single structural layer is sufficient. The final step is to remove the sacrificial material, leaving only the polysilicon cantilevered beam as illustrated in Figure 2.3.

Two commercial processes exist at the time of this work to fabricate surface micro-machined parts, the Multi-User MEMS Process (MUMPs) at Cronos Integrated Microsystems (Koester et al., 2001), and the Sandia Ultra-planar Multi-level MEMS Technology (SUMMiT V) at Sandia National Laboratories (Sniegowski and de Boer, 2000). The MUMPs process consists of a ground level (Poly0) that is attached to the insulated substrate for electrical paths, and two releasable polysilicon levels (Poly1 and Poly2), with PSG used for the sacrificial levels. A final gold layer is deposited and patterned at the end of the process for improved electrical path conduction and bond pads. The minimum feature size in MUMPs, which is determined by the photolithography step, is 3 microns. The SUMMiT V process also has a ground level (Poly0), but has four structural polysilicon levels (Poly1, Poly2, Poly3 and Poly4), with silicon dioxide used as the sacrificial mate-

Figure 2.3 - Final step of cantilevered beam fabrication showing fully released beam
rial. There is not currently a metallization layer in the SUMMiT V process; however, it does have the advantage of allowing a smaller minimum feature size of only 1.0 micron.

The SUMMiT V process is considered more advanced than the MUMPs process for several reasons. First, it contains more layers, with a smaller minimum feature size, allowing more complex mechanical structures to be realized. It also has a specialized pin-joint cut process which significantly improves the fabrication of both fixed and floating pin-joints. Finally, the last two oxide levels are polished flat using a chemical-mechanical polishing (CMP) process before depositing the Poly3 and Poly4 levels. This is important because as each layer is deposited, it conforms to the shape of the levels beneath it. After several layers have been deposited and etched, there can be significant topography which limits the motion of the upper levels and creates problems in the focus and resolution of the photolithography process during the patterning and etching steps. The CMP step smooths out the top surface of the sacrificial oxide levels deposited before the Poly3 and Poly4 levels, allowing the polysilicon in each of these two levels to be deposited flat. This significantly opens the design space for devices fabricated in the SUMMiT V process because the upper levels are then able to pass over the lower levels without the interference that would normally be present due to the level conformity.

While significant advancements have been made in the processing for MEMS fabrication, as well as in the area of device design, there are still several challenges that have hindered the rapid development of commercially viable products. For example, packaging and long-term reliability are still difficult issues to resolve for many potential products, especially in areas where the MEMS devices interact with the environment, such as sen-
sors, optical switches and bio-fluidic devices. The resolution of some of these problems will be discussed, including drying processes and stiction issues.

2.1.1.1 Stiction and Long-Term Reliability

When compared with more traditional macro fabrication techniques, surface micromachined devices have a very high surface to volume ratio. MEMS devices are fabricated at the micron length scale, with gaps between levels typically in the range of 2 to 6 microns, layer thicknesses on the order of 2 to 3 microns, and in-plane dimensions typically measured in the hundreds of microns. Because of this, surface forces dominate their performance and behavior, and are much more significant than the effects of gravity or inertial forces.

Because of the dominance of surface effects in MEMS devices, a common difficulty arises in which two surfaces will stick together whenever they come into contact with each other. This phenomenon is typically referred to as stiction or adhesion, and is the result of the high surface forces present, from van der Waals forces or solid bridging, relative to the elastic restoring force of the MEMS structure. This adhesion between two polysilicon surfaces is the cause for many of the reliability problems associated with MEMS designs (de Boer et al., 1999). Because of the reliability issues introduced by stiction, many of the commercially available MEMS products have been designed without contacting surfaces (accelerometers and gyroscopes for example). In working towards a solution to these stiction related problems, several surface treatments have been developed which have been shown to reduce the effects of stiction in polysilicon. These coatings will be discussed in more detail in the next section.
The long term reliability of a MEMS device is also affected by other factors including relative humidity, and ambient conditions. It has been demonstrated that wear caused by rubbing polysilicon surfaces increases with decreasing relative humidity (Tanner et al., 1999), but adhesion increases exponentially with increasing relative humidity (de Boer et al., 1998). While their small size does introduce several reliability problems, it proves to be a benefit in some instances. Because of their small mass, some MEMS devices do not seem to be sensitive to shock and vibration (Tanner et al., 2000a, Tanner et al., 2000b).

2.1.1.2 Release Processes and Surface Treatments

After fabrication, the sacrificial material must be completely removed to allow the MEMS devices to move and interact. This release process is typically performed in an aqueous solution of hydrofluoric acid, followed by rinses in de-ionized water. The entire release process is performed with the MEMS part completely submerged, and after completion the released MEMS parts must be dried. If they are removed from the rinse water and allowed to air dry, significant capillary forces develop in the gap between the substrate and the free-standing MEMS parts. This capillary force is large enough to pull the polysilicon structure down in contact with the substrate, causing it to stick. This stiction is typically referred to as release stiction and is primarily due to the hydrophillic nature of polysilicon which causes the contact angle of the meniscus, which forms in the water trapped between the structure and the substrate, to be less than 90 degrees resulting in an attractive force. Several methods have been developed to prevent this capillary force from pulling down and adhering the MEMS device to the substrate. The two most common
solutions to the problem of release stiction are the application self-assembled monolayer (SAM) coatings on the polysilicon surface and the use of supercritical carbon dioxide drying.

In the application of a SAM coating a single molecule thick film is deposited onto the polysilicon surface, changing the surface from hydrophillic to hydrophobic. When air dried, the hydrophobic surface has a contact angle greater than 90 degrees and repels the water at its surface, eliminating the capillary force and allowing the structures to dry fully suspended and released from the substrate. There are two different SAM coatings in common use, octadecyltrichlorosilane (OTS) (Houston et al., 1996) and perfluorodecyltrichlorosilane (FDTS) (Srinivasan et al., 1997), each resulting in a water contact angle of greater than 100 degrees. Cantilevered beams as long as 1 mm that have been treated with a SAM coating have been reported fully released, whereas beams as short as 100 μm were adhered when air dried without any coating applied. In addition to the benefit of reduced release stiction, SAM coatings have also been shown to significantly reduce the in-use adhesion energy and coefficient of friction of coated polysilicon surfaces (Srinivasan et al., 1998a; Srinivasan et al., 1998b).

The other method in common use for drying released MEMS parts is supercritical carbon dioxide drying. This technique relies on the transition of a liquid to a supercritical fluid. Release stiction is caused by the surface tension forces in the evaporating liquid between the MEMS device and the substrate. By transitioning the liquid first to a supercritical fluid and then to a vapor, the liquid/vapor interface, and the associated surface tension forces, are eliminated. Because of the relatively low temperature and pressure of the
critical point for carbon dioxide, it is typically used as the drying liquid. After release and rinse, the rinse water is displaced by methanol, which is easily soluble in liquid carbon dioxide. The methanol is then displaced by liquid carbon dioxide which is then heated to above the critical point. Once in the supercritical region, the carbon dioxide is vented leaving a fully released and dried MEMS structure (Mulhern et al., 1993). Supercritical drying of MEMS structures is an excellent method for eliminating release stiction. However, it does not have the benefit of reducing in-use stiction, whereas SAM coatings provide both benefits.

2.1.2 Bulk Micromachining

Bulk micromachining is a term used to characterize processes where the MEMS device is etched directly in the single crystal silicon substrate. This can be done with either a wet or dry etch. Wet etch processes are commonly used for accelerometers and pressure sensors and typically use potassium hydroxide (KOH), ethylene diamine pyrocatechol (EDP) or tetramethyl ammonium hydroxide (TMAH) as the etching liquid with Boron doping used as the etch stop and SiO₂ as the etch mask. The wet chemistry etches are selective to the crystal planes in silicon and can be used to create angled sidewalls.

Another method of bulk micromachining that is gaining popularity is silicon-on-insulator (SOI) MEMS. This process uses a dry deep reactive-ion etch (DRIE), which is an anisotropic etch and can be used to create high aspect-ratio vertical sidewalls. Silicon-on-insulator wafers must be used which consist of two silicon wafers bonded together using silicon fusion bonding which creates an oxide layer in between the two fused silicon wafers. This oxide layer is then used as the etch stop for the deep reactive-ion etch. The
high aspect-ratio that can be achieved with SOI bulk micromachining can be advanta-
geous in that it allows for a much greater mass and spring stiffness than can be attained
with the thin surface micromachined levels. However, as with wet chemistry bulk micro-
machining, this process is limited to only a single layer, whereas surface micromachining
allows for multiple stacked layers.

2.1.3 LIGA

LIGA is a German acronym which stands for Lithographie Galvanoformung
Abformung (X-ray lithography, electroforming, injection molding). With LIGA, very high
aspect-ratio devices can be fabricated out of electroplated metals. The process uses an X-
ray source, called a synchrotron, to pattern a mold in polymethylmethacrylate (PMMA).
The patterned PMMA is then used as a mold for electroplating metal, such as nickel or
copper, in the patterned shape. LIGA parts are typically measured in the hundreds of
microns and are large when compared to surface or bulk micromachined devices.

Advantages of LIGA processing include the ability to fabricate parts with very
high aspect ratios (greater than 1000:1) and with large mass and spring stiffness. Also, the
final parts are made of electroplated metals, as opposed to polysilicon. However, the
released parts are no longer attached to any substrate and as such require assembly.
Another significant disadvantage is the high cost and limited availability of a synchrotron
for creating the PMMA molds.
2.2 Compliant Mechanisms

A compliant mechanism is defined as any mechanism that gains some or all of its motion from the deflection of flexible segments, rather than from traditional rigid-body pin-joints (Howell and Midha, 1993). Compliant mechanism examples are common throughout history, and include the crossbow and the catapult. Nature also makes use of compliance, especially as the size of the object decreases, such as in insect wings and plants. The use of compliance in mechanism design offers several advantages over traditional rigid-body kinematics including improved tolerances, reduced backlash, reduced part count and weight, and a lower total cost of manufacturing. Reliability is improved because the rubbing surfaces of a traditional pin-joint, as well as the associated need for lubrication, are eliminated. Also, in some cases the ability to integrate energy storage into a mechanism without the need for separate springs can be a significant advantage (as is the case in this work).

However, compliant mechanism design also brings with it several disadvantages, the most notable of which is the increased analysis complexity. To form useful mechanisms, the deflection of a compliant segment is normally well outside of the small-deflection regime, where simplified beam mechanics models have been developed and proven over the years for the force and deflection characteristics of a member. Analysis of large deflections requires the solution of non-linear beam equations. Closed-form solutions are possible for only simple geometries and loading cases, and require the use of complex elliptic-integral solutions or finite-element methods. Also, while it is listed as an advantage, the energy storage of the deflected flexible segments can be a disadvantage in some
situations. Additionally, because the compliant segments can often experience high stress states, failure due to fatigue is a significant concern.

For MEMS designs, compliant mechanisms offer additional benefits. Because MEMS fabrication techniques are planar in nature, the use of compliance allows complex mechanism motion in a device fabricated in a single level. Due to fabrication constraints, pin-joints can be difficult to manufacture in a MEMS process, and when used, they can present significant long-term reliability concerns due to stiction and wear of the rubbing surfaces (Miller et al., 1997). This becomes even more significant when it is understood that traditional lubrication techniques cannot be used at the MEMS scale.

2.2.1 Non-Linear Large Deflection Analysis

The deflection of a beam is defined by the well known Bernoulli-Euler equation, which states that the change in slope of the beam along its length is proportional to the moment in the beam, or

$$\frac{d\theta}{ds} = \frac{-M}{EI}$$

(2.1)

where $M$ is the moment at a distance $s$ along the curved beam, $E$ is Young’s Modulus of the material, $I$ is the beam area moment of inertia and $\theta$ is the angle of the beam. In an $x$-$y$ coordinate system, where $x$ is the coordinate parallel to the beam and $y$ is positive down, this equation can be written as (Gere and Timoshenko, 1990)

$$\frac{d^2y}{dx^2} = \frac{-M}{EI} \left[ 1 + \left(\frac{dy}{dx}\right)^2 \right]^{3/2}$$

(2.2)
For small deflections, the angle \( \theta \) is assumed to be equal to \( dy/dx \) and is assumed to be very small. As such, \( (dy/dx)^2 \) can be assumed to be zero. This simplifying assumption reduces Eq. (2.2) to the commonly used linear equation for beam deflections,

\[
\frac{d^2y}{dx^2} = \frac{-M}{EI}
\]  

(2.3)

However, in compliant mechanisms the deflection is large and the simplified solution of Eq. (2.3) cannot be used. For simple end-loading cases, Eq. (2.2) has been solved by Bisshopp and Drucker (Bishop and Drucker, 1945; Bisshop, 1973) using elliptic-integrals (Byrd and Friedman, 1954). Frisch-Fay also treated non-linear beam deflections (Frisch-Fay, 1962). However, while these solutions are useful for understanding the nature of large deflections, and have been crucial to the development of simplified techniques such as the pseudo-rigid-body model, they are still cumbersome and difficult to use in a design synthesis environment.

2.2.2 The Pseudo-Rigid-Body Model

While the elliptic-integral solutions and finite-element analysis are useful for a final detailed analysis of a compliant mechanism, a more simplified approach is necessary in the initial design stages. The pseudo-rigid-body model (PRBM) is a method of analysis that allows the large deflections to be modeled using rigid-body kinematics, greatly simplifying the design of compliant mechanisms (Howell, 2001). By converting all compliant segments to rigid-body counterparts, rigid-body kinematics can be used in mechanism design, leveraging the extensive mechanism synthesis techniques that have been devel-
oped in traditional kinematics analysis. A different model is required for each type of flexible segment in a mechanism.

2.2.2.1 Small-Length Flexural Pivot

A small-length flexural pivot is a flexible segment that is significantly shorter and more flexible than the other rigid links in the mechanism, as shown in Figure 2.4. As such, it can be treated like a single pin-joint with a torsional spring added to model the stiffness of the flexible member. Its deflection is easily calculated by placing a pin joint, called the characteristic pivot, at the center of the flexible segment. The stiffness of the added torsional spring is calculated as

\[ K = \frac{EI}{l} \]  \hspace{1cm} (2.4)

where \( l \) is the length, \( E \) is Young’s Modulus, and \( I \) is the cross-sectional area moment of inertia of the small-length flexural pivot. The output torque of the pivot at any point in its rotation is then defined as

\[ T = K\Theta \]  \hspace{1cm} (2.5)
where $\Theta$ is the angle of the pseudo-rigid body link, as shown in Figure 2.4 (Howell and Midha, 1994).

2.2.2.2 End-Loaded Cantilevered Beam

Using the pseudo-rigid-body model, a constant cross-section cantilevered beam with a force applied at the free end can be modeled as a rigid link with a pin joint and a torsional spring (a fixed-pinned segment). This model was developed by observing that the path of the tip of a flexible beam with an end-force applied is nearly a circular arc, as predicted by the elliptic-integral solution. The center of this arc lies on the flexible beam at a distance of $\gamma l$ (the characteristic radius) from the end of the beam, where $\gamma$ is the characteristic radius factor and $l$ is the length of the flexible segment as shown in Figure 2.5. The flexible beam can then be replaced with a rigid link and pin-joint, with the joint located at the characteristic radius. A torsional spring must be added at the characteristic pivot to
account for the stiffness of the beam in bending. This PRBM will be discussed in more detail in Section 4.1.

2.2.2.3 Initially-Curved End-Loaded Beam

As an extension to the end-loaded cantilevered beam model, a PRBM has also been developed for the case where the cantilevered beam has some initial radius of curvature $R_i$ as shown in Figure 2.6 (Howell and Midha, 1996). Again, a torsional spring must be added at the characteristic pivot to account for the beam stiffness. This PRBM will be described in more detail in Section 5.1.

2.2.2.4 Additional Pseudo-Rigid-Body Models

By utilizing symmetry in a compliant segment, the models presented in this section can be built upon in the development of additional pseudo-rigid-body models. For example, the end-loaded cantilevered beam model can be extended through symmetry to model a fixed-guided segment (Howell, 2001), and the initially-curved end-loaded cantilevered
beam model can be extended to model a circular-arc functionally-binary pinned-pinned segment (Edwards et al., 1999). The circular-arc model will be described in detail in Section 5.1 with direct application to a pinned-pinned segment. Several additional pseudo-rigid-body models have been developed for use in compliant mechanism design, however they are beyond the scope of this work. An end-loaded cantilevered beam with both an end force and moment has been modeled (Lyon et al., 2000), extending the work done previously for only force-loaded or moment-loaded beams. A PRBM has also been developed for a tapered cantilevered beam (Parkinson et al., 2000b).

2.3 Bistable Mechanism Theory

The stability of a system can be observed based on the response of the system to a small disturbance. If the system returns to its original state after being disturbed, then it is considered to be in a stable state. If, however, the system diverges and does not return it is considered unstable. If a system is disturbed and simply remains in its new disturbed state it is considered neutrally stable (Timoshenko, 1961).

A common analogy used to illustrate the concept of stability is that of a ball on a hill (Figure 2.7). If either ball ‘A’ or ball ‘C’ is lightly bumped, or given a small disturbance, the ball will oscillate and return to its original position. These two balls are each in stable equilibrium positions. Because there are two stable positions in this system it is considered bistable. Ball ‘B’ can be balanced on the top of the hill so that it is in a state of equilibrium. However, any small disturbance will cause it to roll down the hill to the nearest stable position, and as such it is in an unstable equilibrium position. It should be noted
that an unstable equilibrium position will always be located between any two stable positions. Finally, ball ‘D’ is in a neutral state because if it is tapped, it will roll slightly and remain in its disturbed position.

Similar to the ball-on-a-hill analogy, the stability of any mechanism can be determined by examining the total potential energy stored in the system. The stable mechanism positions will be located at the minimums of the potential energy curve (Opdahl et al., 1998; Simitses, 1976). This same method can be used to analyze the ball-on-a-hill analogy. The maximum points in the potential energy curve identify the unstable equilibrium positions of the mechanism, and will be located between the stable positions.

For a compliant mechanism, energy is stored in the deflection of the compliant segments. If the compliance in the segment is modeled using the pseudo-rigid-body-model, as a torsional spring, the potential energy, \( V \), is defined by

\[
V = \frac{1}{2}K(\Theta - \Theta_i)^2
\]  

where \( K \) is the pseudo-rigid body model spring constant and \( \Theta_i \) is the initial pseudo-rigid body link angle. The total potential energy in a mechanism can be plotted by summing the

Figure 2.7 - Ball-on-a-hill analogy for stability. Balls ‘A’ and ‘C’ are stable, ball ‘B’ is unstable and ball ‘D’ is neutrally stable
potential energy stored in each of the pseudo-rigid-body model springs through a mechanisms range of motion. Potential energy can also be stored in the system as height differences in a gravity field; however, this term can be ignored if the device is operating in a horizontal plane, or if the forces of gravity are negligible when compared to the elastic stiffness of the flexible members. The force required to move the mechanism can also be found by differentiating the potential energy equation with respect to the desired input variable.
CHAPTER 3  SURVEY OF PRIOR WORK

Because a primary application of bistable mechanisms is in the area of relays and switches, this chapter will present a brief overview of current MEMS relay/switch solutions as well as an overview of the state of the art in bistable devices, including both compliant and non-compliant designs. Various methods of MEMS actuation will also be discussed.

3.1 Non-Latching Relays and Switches

A more straightforward relay design, often easier to implement than a bistable mechanism, is the non-latching relay. In this design the switch is fabricated in either a normally-open or normally-closed state, and the switch position is changed by applying power to the actuator. This type of relay has the clear disadvantage of requiring constant power application to hold the switch in its second position. This disadvantage is one of the primary motivations for using bistable or latching switch designs.
Several different non-latching designs have been proposed for relays and switches, both for electrical and optical switching. The most well known optical device is the Texas Instruments Digital Micromirror Device™ (DMD) (Meier, 1998). This device is fabricated in a proprietary surface micromachining technology and has seen commercial success when used in digital projectors and displays. It uses a complex electrostatic drive signal to tilt a mirror 10 degrees in either direction from center, reflecting light through a color wheel towards the projector screen to create an image. Other electrostatically actuated optical switches used for fiber optic switching applications include a bulk micromachined switching mirror (Marxer et al., 1997) and a silicon micromachined switch attenuator (Giles et al., 1999).

Electrical switching applications have also been developed which typically consist of a mechanism for opening or closing a physical gap in a conducting line. Surface micromachining seems to be the most common fabrication technique used for these switches and numerous examples can be found in the literature using both electrostatic (Zavracky et al., 1998) and thermal actuation methods (Wood et al., 1998; Kruglick and Pister, 1999). Bulk micromachined switches have also been demonstrated (Schiele and Hillerich, 1999).

### 3.2 Bistable MEMS Devices

There are two different ways to implement a bistable device. The most direct method of achieving bistability is with a latching design. In this situation, the switch or mechanism is moved to a second position and then held in that position by a latch or some other type of stop. This configuration does not require continuous power application.
However, it often suffers from slower switching speeds. The other type of bistable mechanism is one which achieves its two stable positions through the stored energy in the elastic deflection of some of its members (see Section 2.3). The analysis of an elastically bistable mechanism is more complex than a latching switch, as compliant mechanism theory must be used in the design and analysis; however, they are more simple in their implementation.

### 3.2.1 Latching Relay Examples

A latching positioner for aligning and switching fiber optic cables has been demonstrated using bulk micromachining technologies (Hoffman et al., 1999). It uses a bimetallic actuator paired with thermal expansion in the mechanism itself, which moves and positions an input fiber relative to an output fiber. The device is then clamped in position so that the actuator power can be disconnected. A second device, which opens and closes an electrical switch by operating out of plane, has been demonstrated using a similar bimetallic thermal actuator approach (Sun et al., 1998).

### 3.2.2 Compliant Bistable Mechanism Examples

Several different elastically bistable devices and configurations have been demonstrated in the literature. Both in-plane and out-of-plane switches have been shown that utilize the snap-through effect of a fixed-fixed beam buckled under compressive residual stress (Halg, 1990; Vangbo and Backlund, 1998). A thermally actuated bistable device has been demonstrated which uses a film deposited in a state of residual tension to achieve bistable motion (Matoba et al., 1994). Each of these devices requires control of the residual stress in the deposited films to achieve the bistable effect. Only the residual-tension
mechanism by Matoba has been demonstrated with successful on-chip actuation. However, it requires three different timed drive signals for proper actuation.

Beam buckling has also been used to create fully-compliant bistable devices that do not rely on the residual stress in the structural material (Parkinson et al., 2000a; Qiu et al., 2001). Because of the high stresses developed in the buckled beams, these designs tend to be very large when compared to other devices reviewed. These mechanisms have also not been demonstrated with on-chip actuation.

The last category of surface micromachined bistable devices found in the literature include partially-compliant mechanisms where energy storage is achieved through compliance in flexible members, but pin-joints are also required for some of the mechanism motion. A relay with electrical contacts has been demonstrated, and the contact resistance extensively tested, for use as a switching relay (Kruglick and Pister, 1998). While it was not demonstrated with actuation, its force and displacement requirements fall within the output specifications of thermal stepper motors. The two mechanisms described in this thesis have also been demonstrated without actuation in previous works (Jensen et al., 1999; Baker et al., 2000).

3.3 Methods of Actuation

For a switch or bistable mechanism to be of use, it must be fabricated with some means of on-chip actuation. For research purposes, a mechanical probe tip can be used to test the functionality of a new bistable device. However, for practical use it must be actu-
ated with some form of an electric signal. The two primary methods of actuation in MEMS are electrostatic and thermal, and each will be discussed briefly.

### 3.3.1 Electrostatic Actuation

Electrostatic actuators rely on the electrostatic attractive force that exists between two members at different voltage potentials. For two parallel plates, this force is given by

\[
F = \frac{\varepsilon_0 V^2 A}{2g^2}
\]  

(3.1)

where \(\varepsilon_0\) is the dielectric constant, \(V\) is the voltage difference between the two plates, \(A\) is the overlap area of the plates and \(g\) is the gap between them (Figure 3.1). However, because the force increases with \(1/g^2\), a parallel plate electrostatic drive would be unstable and would have a very small possible displacement. Because of this, electrostatic drives are typically configured as comb finger arrays (Legtenberg et al., 1996), with the force given by

\[
F = \frac{n\varepsilon_0 tV^2}{g}
\]  

(3.2)
where \( t \) is the thickness of each finger and \( n \) is the total number of fingers in the array as shown in Figure 3.1. In a comb finger configuration, the force is proportional to the voltage squared, but is not a function of the actuation distance, allowing for a much more direct and stable control.

While electrostatic comb drives have seen widespread use, they suffer from the disadvantage of requiring very high operating voltages. Also, because the electrostatic force generated is small, the comb drives must have a large number of fingers to generate a usable force. Because of this they tend to be quite large and are only able to generate forces in the tens of microneutons range. However, because the force generation is based on electrostatics, there is virtually no current flow, and as such, the power requirements for electrostatic drives are very low.

There have been several different actuator designs demonstrated in the literature for electrostatic drives. An advantage of the comb drive is that it is equally suited for fabrication using both surface and bulk micromachining. Two orthogonal comb drives have been combined to create a rotary microengine that can be operated at hundreds of thousands of revolutions per minute using the SUMMiT surface micromachining technology (Rodgers et al., 1998). A higher force, lower voltage actuator has also been developed at Sandia National Laboratories using the SUMMiT process which has a more limited output displacement when compared to other comb drives, but which operates at significantly lower voltages (Rodgers et al., 2000). Finally, a rotary electrostatic actuator has also been demonstrated (Barnes et al., 2000).
3.3.2 Thermal Actuation

Thermal actuators are devices that utilize constrained thermal expansion of thin members to achieve an amplified motion. Typically, the thin members are heated by passing a current through them. Several different design configurations have been proposed for thermal actuation, including the heat-drive actuator illustrated in Figure 3.2 (Comtois et al., 1998). This design is similar in concept to a bimetallic expansion. One leg is fabricated thinner than the other so that as a current is applied, it will heat up and expand more than the thicker leg. The uneven expansion of the two legs causes the device to rotate.

A second thermal actuator design is the thermomechanical in-plane microactuator (TIM), shown in Figure 3.3 (Cragun and Howell, 1999; Que et al., 1999). The thin legs in this design are constrained from freely expanding, and are fabricated at some small initial angle such that when heated they expand and displace the movable shuttle. This actuator design only requires that there be 2 legs per side for stable operation. However, the number of legs can be increased to increase the output force of the actuator. The leg lengths can also be modified to control the maximum output displacement.
Actuators based on thermal expansion offer advantages over electrostatically actuated devices because they can produce a much greater force output per unit area at significantly lower voltages, while also being capable of relatively large displacements. The disadvantage of thermal actuators when compared to other methods is that they require a greater amount of power in their operation. The higher energy requirements are due to the energy loss from the heated legs to the surrounding air and into the substrate. However, this disadvantage can be minimized through proper actuator design. Operating in a vacuum has been shown to reduce the power requirements of the actuators used in this study by over an order of magnitude due to the absence of a medium for heat conduction from the heated legs. It has also been found that applying a rapid current pulse to the actuators, where a higher current is passed through the actuator legs for a short time interval, increases the deflection amplitude and improves the operating efficiency when compared to a steady-state operating mode (Lott, 2001).
3.3.2.1 Modified Thermal In-plane Microactuators

Multiple-leg thermomechanical in-plane actuators were used in this work because of their relatively large force and displacement output (Figure 3.4). However, they were modified slightly from the published work in an attempt to decrease the power requirements, and match the actuator output force and displacement to that required by the bistable mechanisms being actuated. Specifically, the actuator legs were grouped closer together, as seen in Figure 3.4, to reduce the heat loss due to conduction in the surrounding air. In the SUMMiT process the legs were also elevated higher off the substrate to increase the thermal insulation between the legs and the substrate. Both of these changes were found to improve the actuator efficiency.

A second modification was also made to the thermomechanical in-plane microactuator to increase its maximum displacement. While increasing the length of the heated legs does increase the output displacement, there is a practical limit to how much the legs
can be lengthened. As the length is increased, the critical buckling force is decreased and eventually the legs will simply buckle when heated instead of expanding and moving the shuttle. The cross-sectional area of the heated legs cannot be increased to overcome this limitation because the current requirements to heat a larger leg would be prohibitive. To increase the maximum displacement the TIM was staged as shown in Figure 3.5. In this configuration, two small-displacement, high-force TIMs each push on a constrained amplifier similar in function to the TIM legs themselves. However, because the amplifier legs are not heated, they can be made wider to overcome the critical buckling limitation. An SEM image of an amplified TIM (ATIM) is shown in Figure 3.6.

Figure 3.5 - Amplified TIM (ATIM) design with two small, high-force actuators pushing on a wide leg amplifier for larger maximum displacement without buckling
Figure 3.6 - SEM image of an ATIM fabricated in the MUMPs process
This chapter discusses the design of an in-plane compliant bistable micro-mechanism, called the Young Mechanism, which has a rotary motion between its two stable equilibrium positions. The energy storage and bistable behavior of the mechanism is achieved through the elastic deflection of compliant segments. The pseudo-rigid-body model (PRBM) was used to model and analyze the large-deflection members. After modeling each of the flexible segments in the device with the PRBM, the mechanism motion and energy storage were analyzed using traditional rigid-body mechanism analysis methods.

To achieve on-chip actuation, the mechanism design was optimized to allow it to be switched using linear motion thermal actuators. Because of the potential for use in autonomous systems, the power consumption of the actuators has also been reduced. Three optimized designs were fabricated and tested and the results are presented using paired thermal actuators for single pulse on-chip actuation.
4.1 Pseudo-Rigid-Body Model of a Fixed-Pinned Segment

As described in Section 2.2.2.2, a PRBM has been developed for a cantilevered beam with a force applied at the free end. This model is illustrated in Figure 4.1. By replacing the flexible segment with a rigid link and a pin joint, the force and deflection characteristics of the member can be analyzed using rigid-body kinematics. The location of the pin joint is determined by the characteristic radius factor, $\gamma$. The value for $\gamma$ is a function of the angle of the applied load; however, it is reasonable to assume a value of 0.85 for a wide range of loading angles (Howell and Midha, 1995). The location of the beam end can then be determined by
\[
\frac{a}{l} = 1 = \gamma(1 - \cos \Theta) \tag{4.1}
\]

and

\[
\frac{b}{l} = \gamma \sin \Theta \tag{4.2}
\]

where \( \Theta \) is the pseudo-rigid-body link angle, \( a \) is the horizontal distance to the beam end, and \( b \) is the vertical distance to the beam end. A torsional spring must be added to model the resistance of the beam to bending, with the torque defined as

\[
K = \gamma K_\theta \left( \frac{EI}{l} \right) \tag{4.3}
\]

where \( \gamma \) is the characteristic radius factor, \( l \) is the length of the flexible beam and \( K_\theta \) is called the stiffness coefficient. The value of the stiffness coefficient is also a function of the load angle, but it can be approximated as 2.65 for most loading conditions (Howell et al., 1996).

### 4.2 Young Mechanism Design and Analysis

The basic form of the mechanism developed in this work consists of two thin flexible segments pinned to ground and connected to each other by a single rigid section. The mechanism consists of only a single link, but it is able to achieve its motion because of the compliance in the flexible segments. This type of mechanism has been named the *Young Mechanism* in reference to Brigham Young University, where it was originally developed (Jensen et al., 1999). To analyze this compliant mechanism, the two fixed-pinned segments are replaced with their pseudo-rigid-body models, allowing the motion to be ana-
analyzed as a rigid-link four-bar mechanism with torsional springs located on each of the two pin joints that are not connected to ground (Figure 4.2). Using the following closed-form kinematics solution for a rigid-body four-bar mechanism, the device motion can be calculated at any point in its rotation. For $\theta_2$ between 0 and $\pi$,

$$\begin{align*}
\theta_3 &= \psi - \beta \\
\theta_4 &= \pi - \lambda - \beta
\end{align*}$$

and for $\theta_2$ between $\pi$ and $2\pi$,

$$\begin{align*}
\theta_3 &= \psi + \beta \\
\theta_4 &= \pi - \lambda + \beta
\end{align*}$$

where

$$\begin{align*}
\beta &= \cos^{-1}\left(\frac{r_1^2 + \delta^2 - r_2^2}{2r_1\delta}\right) \\
\psi &= \cos^{-1}\left(\frac{r_3^2 + \delta^2 - r_4^2}{2r_3\delta}\right)
\end{align*}$$
The stability of the mechanism can be determined by examining the potential energy stored in the system (Section 2.3). The compliance in each flexible segment is modeled, using the pseudo-rigid-body model, as a torsional spring with the potential energy, \( V \), defined as

\[
V = \frac{1}{2}K(\theta - \theta_i)^2
\]  

(4.10)

where \( K \) is the torsional spring constant, \( \theta \) is the link angle and \( \theta_i \) is the initial fabricated link angle. The total energy for the mechanism is found by summing the energy stored in each of the compliant segments. For this mechanism, there are two torsional springs, and the total potential energy of the mechanism is defined as

\[
V = \frac{1}{2}K_2[(\theta_2 - \theta_{2i}) - (\theta_3 - \theta_{3i})]^2 + \frac{1}{2}K_3[(\theta_4 - \theta_{4i}) - (\theta_3 - \theta_{3i})]^2
\]  

(4.11)

where \( K_2 \) and \( K_3 \) are the PRBM spring constants as defined by Eq. (4.3), and \( \theta_2, \theta_3, \) and \( \theta_4 \) are the PRBM link angles. The stable and unstable equilibrium positions are then determined by plotting the total mechanism potential energy as a function of the input angle. Stable positions are located at the minimums of the potential energy curve (Opdahl et al., 1998; Simitses, 1976). The maximum points in the potential energy curve identify the unstable equilibrium positions of the mechanism, and will be located between the stable

\[
\lambda = \cos \left( \frac{r_1^2 + \delta^2 - r_2^2}{2r_4\delta} \right) \quad (4.8)
\]

and

\[
\delta = \sqrt{r_1^2 + r_2^2 - 2r_1r_2\cos\theta_2} \quad (4.9)
\]
positions. Taking the derivative of the potential energy curve with respect to the input crank angle gives the input torque required to rotate the mechanism, and is defined by

\[
T = K_2[(\theta_2 - \theta_2^{i}) - (\theta_3 - \theta_3^{i})]\left(1 - \frac{d\theta_3}{d\theta_2}\right) + K_3[(\theta_4 - \theta_4^{i}) - (\theta_3 - \theta_3^{i})]\left[\frac{d\theta_4}{d\theta_2} - \frac{d\theta_3}{d\theta_2}\right]
\]

(4.12)

where

\[
\frac{d\theta_3}{d\theta_2} = \frac{r_2 \sin(\theta_4 - \theta_2)}{r_3 \sin(\theta_3 - \theta_4)}
\]

(4.13)

and

\[
\frac{d\theta_4}{d\theta_2} = \frac{r_2 \sin(\theta_3 - \theta_2)}{r_4 \sin(\theta_3 - \theta_4)}
\]

(4.14)

The maximum stress in each of the two flexible segments can be calculated at any position in the mechanism motion using the following equations:

\[
\sigma_2 = \frac{M_2 c}{I} = \frac{\{K_2[(\theta_2 - \theta_2^{i}) - (\theta_3 - \theta_3^{i})]\}}{I}\left(\frac{w_2^2}{2}\right)
\]

(4.15)

and

\[
\sigma_4 = \frac{M_4 c}{I} = \frac{\{K_3[(\theta_4 - \theta_4^{i}) - (\theta_3 - \theta_3^{i})]\}}{I}\left(\frac{w_4^2}{2}\right)
\]

(4.16)

where \(w\) is the width of the segment and \(I\) is the cross-sectional moment of inertia.

Both the linear thermal actuator and the rotary bistable mechanism have been successfully demonstrated independently in previous works (Cragun and Howell, 1999; Jensen et al., 1999). However, because of the large rotations required to switch the bistable mechanism, it was not possible to actuate the device on-chip using the existing actuators.
The thermal actuators are limited to a maximum steady-state linear displacement of approximately 12 \( \mu m \). In order to toggle the mechanism directly with a single 12 \( \mu m \) actuator pulse using the MUMPs surface-micromachining technology, the maximum rotation between the stable and unstable positions is limited to approximately 25 degrees. This limit is based on the configuration shown in Figure 4.2, where the actuator linear motion is used to flip the rotary mechanism by pushing on an extension of the mechanism input link. The minimum length of this input link extension is specified by the design rules for the fabrication process used. In addition to the displacement constraints, the input torque requirements of the mechanism are also limited by the upper bound of the actuator output force. The first design of the bistable mechanism had a required rotation of 60 degrees between the stable and unstable positions, with a total rotation of 95 degrees between the two stable positions, and as such could not be switched using a single-pulse of the actuator (Jensen et al., 1999). These devices were tested by manually displacing them with a mechanical probe. For successful on-chip actuation the bistable mechanism design must be optimized to reduce its required rotation to within the allowable output of the thermal actuators.

4.3 Design Optimization for On-Chip Actuation

These restrictions required the development of a new mechanism with small rotations that would also fall within the allowable torque restrictions. To better examine the full design space of the mechanism, the pseudo-rigid-body model and energy equations for the Young Mechanism (Eqs. 4.4 - 4.16) were programmed and linked to an optimiza-
tion algorithm (Parkinson et al., 1992) with the objective of minimizing the angle of sweep between the two stable positions. This was done by changing

- The lengths of the pseudo-rigid-body links, \( r_1, r_2, \) and \( r_4 \).
- The pseudo-rigid-body link initial angles, \( \theta_{2i} \) and \( \theta_{4i} \).

subject to the following constraints:

- A size limitation to reduce the effects of stiction and adhesion to the substrate.
- A geometric check to ensure that the mechanism could be fabricated from a single layer and rotate through its motion without interference.
- A constraint on the magnitude of the stress in the flexible segments to prevent failure.
- A required minimum potential energy difference between unstable and stable positions to ensure that the device has enough energy to overcome stiction.

The width of each of the two flexible links was fixed at the minimum allowed by the fabrication process. Because of the numerous local minimums in the design space, gradient-based optimization routines such as Sequential Quadratic Programming (SQP) and Generalized Reduced Gradient (GRG) failed to give good results; however, a simulated annealing routine provided satisfactory results and was used extensively in determining the optimum design (Kirkpatrick et al., 1983). The simulated annealing algorithm randomly perturbs the design variables in a search for the best design. If a better design is found, it is accepted; however, if a worse design is found, there is a probability that it will also be accepted as the new best design. This probability is based on the Boltzmann probability factor and decreases as more evaluation cycles are performed. By allowing a
worse design to be accepted, the algorithm is able to jump out of local minimums in the
design space in the search for a global optimum. It is this characteristic of the simulated
annealing algorithm that allows it to perform well in this design problem because the
design space for the young mechanism is very discontinuous with numerous local mini-
mums.

4.3.1 Model Validation

To validate the design selected from the optimization routine, the final mechanism
configuration was modeled and analyzed using a commercial finite element analysis code
(ANSYS). Because of the large elastic deformations, a nonlinear analysis was performed.
From the finite-element analysis, the position of the unstable equilibrium point and the
second stable point were found as well as the value of the maximum stress in the mecha-
nism at its second stable equilibrium position, and the mechanism’s final deformed shape
(Figure 4.3). The values obtained from finite element analysis were within 2.8% of the
results predicted using the PRBM.
4.3.2 Initial MUMPs Design from Optimization Routine

The final mechanism design selected from the optimization results has a total input crank sweep of only 23 degrees between stable positions. The potential energy and input torque curves are included in Figure 4.4, and the dimensions of the mechanism are listed in Table 4.1. The mechanism was paired with thermal actuators in the configuration shown in Figure 4.5 and was fabricated using the Multi-Layer MEMS Processes (MUMPs). Two thermal actuators were used, with one actuator used to switch in each direction. The device proved to be bistable as predicted and was able to be actuated in either direction by a single pulse of a thermal actuator. Figure 4.5 is an SEM of the device and actuators, with the mechanism shown in both its fabricated position and its second stable position. In testing done in air, the actuators required less than 8 volts and under 55 milliamps, or 440 milliwatts, to transition the device between stable positions. When tested under vacuum, as discussed in Section 3.3.2, the power requirements were reduced by almost an order of magnitude.
magnitude, requiring less than 4.0 volts and under 12.0 milliamps, or a total of 48 milli- 

watts.

4.3.3 MUMPs Design Refinement And Power Reduction

A noted disadvantage of the mechanism just described is the large difference between the required rotations to toggle the mechanism in each direction. As can be seen from the potential energy curve shown in Figure 4.4, the mechanism must rotate 19 degrees from its fabricated stable position to the unstable equilibrium position, but only needs to rotate back 4 degrees to go back from the second stable position to the unstable

<table>
<thead>
<tr>
<th>Table 4.1 - Summary of Young mechanism designs</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Parameter</strong></td>
</tr>
<tr>
<td>$r_1$ (microns)</td>
</tr>
<tr>
<td>$r_2$ (microns)</td>
</tr>
<tr>
<td>$r_4$ (microns)</td>
</tr>
<tr>
<td>$\theta_{20}$ (degrees)</td>
</tr>
<tr>
<td>$\theta_{40}$ (degrees)</td>
</tr>
<tr>
<td>Width of $r_2$ (microns)</td>
</tr>
<tr>
<td>Width of $r_4$ (microns)</td>
</tr>
<tr>
<td>Rotation from fabricated position to unstable equilibrium (degrees)</td>
</tr>
<tr>
<td>Rotation from second stable position to unstable equilibrium (degrees)</td>
</tr>
<tr>
<td>Max Stress (GPa)</td>
</tr>
<tr>
<td>Minimum potential energy difference (pJ)</td>
</tr>
<tr>
<td>Actuation power in air (milliwatts)</td>
</tr>
</tbody>
</table>
equilibrium position. The size of the thermal actuator is determined by its required force and displacement, so a much larger actuator is needed to switch the mechanism from its fabricated position. To reduce the power requirements, and achieve a more symmetric behavior, it is desirable to have the unstable equilibrium position be more centered between the two stable positions.

To achieve this, a second objective was added to the optimization setup to drive the unstable position more towards the midpoint between the two stable positions. The widths of each of the two flexible segments were also added as design variables to increase the available design space. It was determined from the previous design that the clearance in the pin-joints introduced a significant variation in the mechanism performance. This clearance is on the order of 0.75 \( \mu \text{m} \) and is a result of the MUMPs surface micromachining process. In an attempt to account for this variation, a tolerance on the length of the ground link of plus or minus 0.75 \( \mu \text{m} \) was included in the optimization. This causes the resulting design to be more robust to the errors introduced by the clearances.
The final mechanism design has a total required rotation of 23 degrees between stable positions, the same as the previous iteration. The potential energy and torque curves are included in Figure 4.6, and an SEM image of the mechanism in both the fabricated position and the second stable position is shown in Figure 4.7. While similar to the previous design in total rotation, it is more symmetric about the unstable equilibrium point, with 9 degrees of rotation from the fabricated position to the unstable position, and 14
degrees from the unstable position to the second stable position. This gives it a maximum required rotation of 14 degrees, compared to 19 degrees for the previous mechanism. The peak torque values for the second iteration are similar in magnitude to the previous design, but the input torque curve has a smaller gradient so better performance is expected. This mechanism configuration was also fabricated using the MUMPks process and was successfully actuated in each direction using thermal actuators. Because of its lower maximum required angular displacement, smaller thermal actuators could be used, resulting in lower power requirements; however, for the initial fabrication the same actuators were used. Because of this, testing of this mechanism resulted in similar current and voltage requirements as for the previous iteration. The design is also summarized in Table 4.1.

4.3.4 Sandia SUMMiT V Design Optimization

Both of the previous two designs were optimized for and fabricated in the MUMPks process, which has a minimum line width of 3 µm, and a pin-joint clearance of plus or minus 0.75 µm. Because these two process constraints significantly effect the performance and size of the Young mechanism, it would be advantageous to reduce these processing parameters. The Sandia SUMMiT V process allows for 0.8 µm minimum line widths, and has a specialized pin-joint cut that creates pin-joints with less than 0.3 µm clearance. These two advantages alone make the SUMMiT process much more favorable for Young mechanism design. The optimization routine used in Section 4.3.3 was modified slightly to allow for the smaller minimum line width and pin-joint clearance, and the device was re-optimized for the SUMMiT process. The resulting potential energy and
torque curves are shown in Figure 4.8, with the resulting geometry listed in Table 4.1 along with the previous two MUMPs designs for comparison. SEM images of the mechanism in both positions are included in Figure 4.9. Because of the availability of additional planarized layers in the SUMMiT process, a bar was added across the mechanism to prevent any motion out of the plane of the device. The addition of this feature is not possible using the MUMPs process. However, the additional layers in the SUMMiT process are not
required for the Young mechanism. This design was successfully tested with the lowest power requirements of any of the three designs, requiring only 3.0 volts and 11.0 milliamps in air, or a total of 33 milliwatts. This design resulted in a lower power requirement in air than the previous two iterations required in vacuum. Much of this improvement is due to design changes in the thermal actuators that were only possible in the SUMMiT process, such as elevating the heated legs higher off the substrate using the Poly3 and Poly4 layers. This increased gap decreases the heat loss to the substrate, increasing the efficiency of the actuator. In addition, significant savings were realized because the actuators could be smaller due to the reduced force and displacement requirements of the SUMMiT Young mechanism design.

4.4 Summary

By combining an accurate large-deflection model of a bistable mechanism configuration with an optimization algorithm, a device has been developed which can be actuated on-chip using linear thermal actuators. Three successful design iterations have been demonstrated, two having been fabricated and tested using the MUMPs surface micromachining technology, and one in the SUMMiT process. Each configuration can be switched in either direction with a single pulse of a thermal actuator. By minimizing the required displacement of the actuators their size has been reduced, lowering the power requirements from previous design iterations. The smaller line-widths and geometries that are possible in the SUMMiT process have proven to be a significant advantage in the design
of both the thermal actuator and the Young mechanism, allowing the smallest mechanism
with the lowest actuator power requirements.
This chapter discusses the design of a bistable micromechanism with linear displacement between its stable equilibrium positions. The equations used to develop the mechanism, as well as test results from a device fabricated in the MUMPs process will be presented.

5.1 Pseudo-Rigid Body Model of Functionally Binary Pinned-Pinned Segments

A functionally binary pinned-pinned segment is a compliant segment which is pinned on both ends, and is loaded only at the pin joints to achieve equilibrium (Figure 5.1). Because of the pin joints, the segment can only support loads in line through the pins, and cannot support any moments or other loads. As such, the segment behaves in the same way as a linear spring, pinned at both ends. However, the general force-deflection relationship is not linear, but is a function of the geometry of the segment. If the compliant segment is constrained to be a constant radius circular arc (Figure 5.2), a pseudo-rigid body
model can be used to completely characterize the force and deflection relationship of the segment (Howell and Midha, 1996). Due to symmetry, the arc can be divided at the midpoint and analyzed using the pseudo-rigid-body model for an initially-curved cantilevered beam with a horizontally applied load (Figure 5.3). The symmetry of the model is maintained by requiring that the link lengths, spring constants ($K_{\text{left}}$ and $K_{\text{right}}$) and PRBM link angles ($\Theta_{\text{left}}$ and $\Theta_{\text{right}}$) for each half model are the same. With the link lengths and torsional spring constants determined for a half-model, the deflection characteristics of the complete segment are easily determined.

Figure 5.1 - The general form of a functionally binary pinned-pinned segment (FBPPS). Equilibrium requires that the vertical forces be zero, so it only experiences loading in-line with the pin joints.

Figure 5.2 - (a) Functionally binary pinned-pinned segment (FBPPS) in shape of circular arc, and (b) the half model used for analysis.
For this analysis, the important parameters in the half-model that need to be determined are the pseudo-rigid-body link lengths and the torsional spring constant. The length of the fixed segment is determined by the non-dimensional parameter $\gamma$, the characteristic radius factor, and is given by $L(1-\gamma)$, with $L$ being the length of the half-segment. Optimum values for $\gamma$ have been determined by Edwards et al., 1999, and are given for different values of $\kappa_0$ as

1. $\gamma = 0.8063 - 0.0265\kappa_0$ for $0.500 \leq \kappa_0 \leq 0.595$ (5.1)
2. $\gamma = 0.8005 - 0.0173\kappa_0$ for $0.595 \leq \kappa_0 \leq 1.500$ (5.2)

with $\kappa_0$ being the beam’s initial curvature, as defined by

$$\kappa_0 = \frac{L}{R_0}$$ (5.3)

where $R_0$ is the radius of curvature of the segment. For segments without any initial curvature, the length of the rotating link is determined by $\gamma L$. However, because of the initial
curvature of the beam, another non-dimensional factor $\rho$ must be used. The length of the rotating link is then given as $\rho L$, with $\rho$ defined from the geometry as

$$\rho = \sqrt{\left(\frac{a_i}{L} - (1 - \gamma)\right)^2 + \left(\frac{b_i}{L}\right)^2} \quad (5.4)$$

where $a_i$ and $b_i$ are the initial horizontal and vertical coordinates of the beam end, and are given by

$$\frac{a_i}{L} = \frac{1}{\kappa_0} \sin \kappa_0 \quad (5.5)$$

and

$$\frac{b_i}{L} = \frac{1}{\kappa_0} (1 - \cos \kappa_0) \quad (5.6)$$

The initial pseudo-rigid-body link angle $\Theta_i$, is defined as

$$\Theta_i = \tan^{-1}\left(\frac{b_i}{a_i - L(1 - \gamma)}\right) \quad (5.7)$$

and the new link angle after a load has been applied is given as

$$\Theta = \tan^{-1}\left(\frac{b_p}{a_p - L(1 - \gamma)}\right) \quad (5.8)$$

with $a_p$ and $b_p$ defining the new vertical and horizontal position of the end point of the beam with the load applied (see Fig. 4). They are defined as

$$\frac{a_p}{L} = 1 - \gamma + \rho \cos \Theta \quad (5.9)$$

and
The value of the torsional spring constant in the pseudo-rigid-body model can then be determined by

\[ K = \frac{\rho K_\Theta EI}{L} \]  

(5.11)

The non-dimensional factor \( K_\Theta \) has been given by Edwards et al., 1999, to be

\[ K_\Theta = 2.568 - 0.028\kappa_0 + 0.137\kappa_0^2 \]  

(5.12)

The force deflection relationship can now be defined where the force of the pinned-pinned segment, \( F_s \), is

\[ F_s = \frac{K(\Theta - \Theta_i)}{b_p} \]  

(5.13)

The force and deflection characteristics of any constant radius functionally binary pinned-pinned segment can now be fully characterized. The maximum stress in the segment occurs in the center, where the moment is greatest, and has a magnitude of

\[ \sigma_{max} = \pm \frac{F_s b_p c}{I} - \frac{F_s}{A} \]  

(5.14)

where \( c \) is the distance from the neutral axis to the outer surface of the beam, or half the beam height, \( A \) is the cross sectional area of the beam, and \( I \) is the moment of inertia of the cross section.
5.2 Linear Displacement Bistable Mechanism

Attaching several FBPP segments at an angle to a center slider, with the segments symmetric about the center line of the slider, results in a bistable mechanism with linear motion as illustrated in Figure 5.4. As the slider moves down, the FBPP segments are compressed, resulting in a tendency for the slider to snap back to its first stable position. The mechanism passes through an unstable equilibrium point when the segment pin-joints line up perpendicular to the slider, after which the force exerted by the segments causes the slider to snap down to its second stable position.

Because the pin-joint attached to the slider is constrained to move only in the vertical direction, the distance through which the pinned-pinned segments are compressed is
known throughout the motion of the slider. As such, $a_p$ can be determined directly from the geometry as

$$a_p = \frac{\sqrt{w^2 + (h-y)^2}}{2}$$  \hspace{1cm} (5.15)

where $w$ and $h$ define the initial orientation of the pinned-pinned segment and $y$ is any linear displacement of the slider (Figure 5.4). The maximum displacement between the two stable equilibrium positions is $y = 2h$.

Once $a_p$ has been determined, it can be used to solve for $\Theta$ and $b_p$ by using Eq. (5.9) and Eq. (5.10) respectively. Using Eq. (5.13), the compression force, $F_s$, on each of the pinned-pinned segments can then be calculated. The total force, $F_t$, required to displace the slider a distance $y$ can then be found by summing the vertical component of the force applied by each of the pinned-pinned segments, and is given by

$$F_t = NF_s \sin(\theta)$$  \hspace{1cm} (5.16)

where $F_s$ is defined by Eq. (5.13), $N$ is the total number of pinned-pinned segments in the mechanism, and $\theta$ is the angle of the pinned-pinned segment. For any slider displacement $y$, the angle $\theta$ is given by

$$\theta = \arcsin\left(\frac{h - y}{\sqrt{w^2 + h^2}}\right)$$  \hspace{1cm} (5.17)

For equilibrium, the pinned-pinned segments must be symmetric about the center line of the slider so that the component of the force normal to the slider caused by the left side segments cancels with the force components normal to the slider from the right side segments.
5.3 Application to a Micro-Bistable Mechanism

A linear displacement bistable mechanism has been fabricated and tested at the micro level using the MUMPs fabrication process. The mechanism was fabricated with an electrical contact switch on the slider to illustrate a potential application for the device (Figure 5.5). When the mechanism is switched to its second stable position it closes the electrical contact between the two contact pads. The pinned-pinned segment radius of curvature, $R_o$, is 144 $\mu$m, with a total length, $L$, of 85.5 $\mu$m. The curvature $\kappa_o$, can be calculated using Eq. (5.3) as

$$\kappa_o = 0.5934$$  \hspace{1cm} (5.18)

and the fundamental radius factor $\gamma$ can be calculated by Eq. (5.1) to be

$$\gamma = 0.7906$$  \hspace{1cm} (5.19)
The initial coordinates of the beam end \(a_i\) and \(b_i\) can be found by Eq. (5.5) and Eq. (5.6) to be

\[
a_i = 80.5 \mu m \\
b_i = 24.6 \mu m
\]  

(5.20) (5.21)

With the initial coordinates known, the characteristic radius factor \(\rho\) can be calculated using Eq. (5.4) to be

\[\rho = 0.7875\]  

(5.22)

The initial pseudo-rigid body angle, \(\Theta_i\) is defined by Eq. (5.7) as

\[\Theta_i = 0.3745 \text{ rads}\]  

(5.23)

To calculate the torsional spring constant, the non-dimensional parameter \(K_\Theta\) can be calculated using Eq. (5.12) as

\[K_\Theta = 2.60\]  

(5.24)

and the spring constant can then be found to be

\[K = 43,188 \ \mu \text{N} \ \mu \text{m}\]  

(5.25)

by using Eq. (5.11). The initial geometry of the pinned-pinned segments as shown in Figure 5.4 is given by

\[w = 210 \ \mu \text{m}\]  

(5.26)

and

\[h = 55 \ \mu \text{m}\]  

(5.27)

Because of the limitations of the MUMPS fabrication process, and of the floating pin-joints used (Clements et al., 1999), the flexible segment does not extend the complete
length, from pin-joint to pin-joint. However, because the deflection of the beam is a function of the moment, and the moment is very small near the pin-joints, the error introduced is negligible.

With $K$ known, and the geometry of the mechanism defined, its force-deflection properties can now be fully defined with the pseudo-rigid body model by plotting the total force $F_t$, as defined by Eq. (5.16), through the entire range of motion, as show in Figure 5.6. Notice that at a displacement of 55 $\mu$m the force transitions from positive to negative. This corresponds to the unstable equilibrium point where the pinned-pinned segments are at their maximum compression, and are normal to the slider. As would be expected, the force on the slider is negative after this point as the slider snaps to its second stable equilibrium point.

The energy in the mechanism can be found by either integrating the force-deflection curve, or by calculating the sum of the potential energy stored in the pseudo-rigid-body model torsional springs as discussed in Section 2.3 (Figure 5.6). When calculating
the potential energy in the torsional springs it is important to remember that each pinned-pinned segment contains two identical torsional springs (Figure 5.3). At 55 µm the potential energy curve is at a local maximum, indicating that this position is an unstable equilibrium position. At both 0 µm and 110 µm the curve is at a local minimum which represents the two stable equilibrium positions.

The mechanisms did exhibit bistable behavior as expected, showing an unstable equilibrium position about the point where the pinned-pinned segments were perpendicular to the slider. The mechanism shown in Figure 5.5 was not allowed to snap completely to its second stable position because of the electrical contact on the end of the slider. By placing the contact at a position before the stable point, the mechanism will apply a force on the contact to secure it firmly in the closed position. Using the bistable mechanism in this configuration is advantageous because when closed, it applies a continuous force on the contact without requiring any power to maintain its output force or position.

By combining the linear-displacement bistable mechanism with an amplified thermomechanical in-plane microactuator (ATIM) as described in Section 3.3.2.1, the bistable mechanism can be actuated on-chip (Figure 5.7). The actuator required 85 milliamps and 11 volts, or 935 milliwatts to toggle the mechanism between its two stable positions. The mechanism configuration is summarized in Table 5.1.

5.3.1 **Design Refinement for Size Reduction**

While it does work as designed, the linear displacement bistable mechanism just described is quite large and requires a large force and large displacement actuator to
Figure 5.7 - ATIM configured with a linear-displacement bistable mechanism for on-chip actuation

Table 5.1 - Comparison of both linear-displacement bistable mechanism designs

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Initial Design</th>
<th>Reduced-Size Design</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R_o$ (µm)</td>
<td>144.0</td>
<td>43.0</td>
</tr>
<tr>
<td>$L$ (µm)</td>
<td>85.5</td>
<td>57.0</td>
</tr>
<tr>
<td>$w$ (µm)</td>
<td>210.0</td>
<td>100.0</td>
</tr>
<tr>
<td>$h$ (µm)</td>
<td>55.0</td>
<td>35.0</td>
</tr>
<tr>
<td>FBPPS cross-section width (µm)</td>
<td>4.0</td>
<td>3.0</td>
</tr>
<tr>
<td>FBPPS thickness (µm)</td>
<td>2.0</td>
<td>2.0</td>
</tr>
<tr>
<td>Total Displacement (µm)</td>
<td>110.0</td>
<td>70.0</td>
</tr>
<tr>
<td>Maximum Required Force (µN)</td>
<td>150.0</td>
<td>76.8</td>
</tr>
<tr>
<td>Actuation power in air (milliwatts)</td>
<td>935</td>
<td>-</td>
</tr>
</tbody>
</table>
enable on-chip actuation. To allow for smaller actuators, and lower the actuator power requirements, the bistable mechanism was redesigned with the objective of reducing the force and displacement needed to transition the mechanism from one stable position to the other. The final design configuration is shown in Figure 5.8 with the actuation force and potential energy plots included in Figure 5.9. It was fabricated with the same actuator as the previous design iteration. However, because of its reduced displacement requirement,
it was able to be toggled between stable positions with a lower power applied to the actuator. A comparison of the two different mechanism designs is included in Table 5.1.

5.4 Summary

The equations which govern the force-deflection relationship for a circular-arc functionally binary pinned-pinned segment have been developed. It has been shown that these pinned-pinned segments can be combined to create a linear displacement bistable mechanism. The force-deflection relationship for this mechanism has been characterized, and a specific example has been provided. The model which has been developed could be used to design a mechanism by varying the geometric parameters to achieve the force-displacement characteristics needed for the specific application.
CHAPTER 6  CONCLUSIONS AND RECOMMENDATIONS

This chapter will present the conclusions found in this work as well as recommendations for future development.

6.1 Conclusions

Two different bistable compliant micro-mechanisms have been developed which can be paired with thermomechanical in-plane microactuators for on-chip actuation. Each of these devices has been successfully fabricated and tested using the MUMPs surface micromachining process, with the Young mechanism also fabricated in the SUMMiT process. The design improvements from each iteration of the Young mechanism are summarized in Table 6.1, and improvements in the linear-displacement bistable mechanism are summarized in Table 6.2.

The pseudo-rigid-body model has been found to be a useful tool in the design and analysis of nonlinear large-deflection compliant mechanisms. By using this tool, a model
of both the Young mechanism and the linear-displacement bistable mechanism has been developed which allowed for the realization of important design improvements.

The Young mechanism model has been paired with an optimization routine to significantly improve the mechanism performance relative to the initial design. Specifically, the total required displacement to toggle the mechanism between its two stable positions has been reduced. This improvement was necessary to achieve the goal of on-chip actuation using the existing thermal actuators, and will be important in the development of low-power switching applications.

Table 6.1 - Summary of Young Mechanism Designs

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Initial MUMPs</th>
<th>Refined MUMPs</th>
<th>SUMMiT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum Link Size (microns)</td>
<td>245.4</td>
<td>228.2</td>
<td>81.4</td>
</tr>
<tr>
<td>Total rotation between stable positions (degrees)</td>
<td>22.9</td>
<td>23.4</td>
<td>15.9</td>
</tr>
<tr>
<td>Maximum rotation required to switch mechanism (degrees)</td>
<td>18.7</td>
<td>14.1</td>
<td>9.9</td>
</tr>
<tr>
<td>Max Stress (GPa)</td>
<td>1.22</td>
<td>0.92</td>
<td>0.70</td>
</tr>
<tr>
<td>Actuation power in air (milliwatts)</td>
<td>440</td>
<td>440</td>
<td>33</td>
</tr>
</tbody>
</table>

Table 6.2 - Summary of design improvements for the Linear-Displacement Bistable Mechanism

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Initial Design</th>
<th>Reduced-Size Design</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total mechanism width (µm)</td>
<td>210.0</td>
<td>100.0</td>
</tr>
<tr>
<td>Total Displacement (µm)</td>
<td>110.0</td>
<td>70.0</td>
</tr>
<tr>
<td>Maximum Required Force (µN)</td>
<td>150.0</td>
<td>76.8</td>
</tr>
</tbody>
</table>
While the total required displacement of the linear-displacement bistable mechanism has also been reduced, its reduction was not large enough to fit within the current output displacement of the actuators. As such, an amplified actuator design was implemented which increased the output displacement of the actuator enough to allow for on-chip actuation of both linear-displacement bistable device iterations.

This actuator improvement is one of several that were designed to improve the output displacement and increase the actuator efficiency. Specifically, it was found that moving the heated actuator legs closer together, and elevating them higher off the substrate improved the efficiency of the actuator by reducing the heat losses to the surrounding air. The most dramatic improvement was found by operating the actuator in a vacuum to completely remove the medium for heat loss to the surrounding air.

6.2 Recommendations for Future Work

For each of the bistable devices examined in this work, some consideration was taken to include the effects of the pin-joint clearances in the mechanism design optimization. However, a detailed clearance analysis was not performed. It has been seen during the testing of these devices that this clearance can have a significant impact on the bistable characteristics of the device. By including these effects in the model, the safety factor in the design could be reduced and the model would be able to more accurately predict the device performance.

While each design iteration presented in this work was tested successfully in air, and the actuator power requirements reported, it would be beneficial to conduct additional
testing of these devices in a vacuum to take advantage of the significant improvement in thermal actuator efficiency seen when operating under vacuum. While this testing would not affect the performance of the mechanisms themselves, it would illustrate the low-power potential of the actuation system.

Several design iterations were presented for both of the bistable mechanisms discussed in this work. Each design iteration represented a reduction in the required force and displacement output of the actuator, which would allow for smaller, lower power actuators to be used. However, because an adequate model of the thermal actuator output force and displacement was not available at the time of the design work, actuators were used which were believed to be conservative in their design to ensure that they would have the force and displacement required to test the mechanism. As low-power actuation is an important characteristic for certain applications, additional modeling of the actuators should be performed to enable them to be more closely matched with their output requirements. This actuator optimization would allow the benefits of the reduced power mechanisms to be fully realized.

Finally, the most significant area of future consideration is in the development of commercial applications for this technology. It is clear that bistable mechanisms could provide significant benefit in several potential applications including relays, positioners and valves. Their use in emerging fields such as micro-fluidics could be significant. This work did not explore these potential applications, but focused instead on improving the modeling and design of each mechanism. However, for the work to be of true benefit, it
must be applied in an area where it can make a significant contribution. One area that is currently being explored for application of this work is in autonomous micro-systems.
LIST OF REFERENCES


#include "supportC.h"
#include <math.h>
#define RAD 1.74532925199e-2
#define DEG 57.2957795132
#define PI 3.14159265359
#define ZERO 0.000001

/*=============================================================================  
Function anapreC  
Preprocessing Function  */
#else
#include <stdio.h>  
#endif  
void anapreC( modelName )
    char *modelName;
#endif
void anapreC( modelName )
    char *modelName;
#endif
#define __STDC__
void anapreC( char *modelName )
#endif
    void anapreC( modelName )
    char *modelName;
#endif
    {  
        /* set model name (16 chars max) */
        strcpy( modelName, "Model Name" );
    }

/*=============================================================================  
Function anafunC  
Analysis Function  */
#else
#include <stdio.h>  
#endif
void anafunC( void )
#endif
void anafunC( )
#endif
    {  
        /* don't forget to declare your variables and functions to be double precision */
        int i, inRange, stable;
double gamma, kTheta, youngs, yieldStress, r1, r2, r4, theta1i, theta2i, theta4i, typeK[4], refLen[4], height, width[4], pinR, extend, scanInc, calcInc,
    r3, r3x, r3y, theta3i, theta2prime, delta, beta, lambda, k[4],
    inertia, len, offset, theta2, theta3, theta4, diff[4], angle,
    PE, d4d2, d3d2, firstStable, secondStable, firstPE, secondPE,
    maxPE, maxTorque, minTorque, bistable, stress, torquePrev,
    maxStress, safetyFactor, sweep, PE1, PE2, minTorqueAbsolute,
    minPE, torque, before, after, centerAngle, fit, minAng, firstT4,
    secondT4, maxFlipAngle, minFlipAngle;

    /* get AV values from OptdesX (Variable names 16 chars max) */
    /* be sure to use the ADDRESS of the variables in the function calls */
    avdscaC( &gamma, "Gamma" );
    avdscaC( &kTheta, "Ktheta" );
    avdscaC( &youngs, "Young_Mod" );
    avdscaC( &yieldStress, "Yield_Stress" );
    avdscaC( &r1, "PRBM_r1" );
    avdscaC( &r2, "PRBM_r2" );
    avdscaC( &r4, "PRBM_r4" );
    avdscaC( &theta1i, "Theta1" );
    avdscaC( &theta2i, "Theta2" );
    avdscaC( &theta4i, "Theta4" );
    avdscaC( &typeK[0], "K1_Type" );
    avdscaC( &typeK[1], "K2_Type" );
    avdscaC( &typeK[2], "K3_Type" );
    avdscaC( &typeK[3], "K4_Type" );
    avdscaC( &refLen[0], "Ref_Len 1" );
    avdscaC( &refLen[1], "Ref_Len 2" );
    avdscaC( &refLen[2], "Ref_Len 3" );
    avdscaC( &refLen[3], "Ref_Len 4" );
    avdscaC( &height, "Height" );
    avdscaC( &width[0], "Width_1" );
    avdscaC( &width[1], "Width_2" );
    avdscaC( &width[2], "Width_3" );
    avdscaC( &width[3], "Width_4" );
    avdscaC( &pinR, "Pin_Radius" );
    avdscaC( &extend, "Extension" );
    avdscaC( &scanInc, "Scan_Inc" );
    avdscaC( &calcInc, "Calc_Inc" );

    /* Round values that should be integers */
    for(i=0;i<4;i++) {
        typeK[i]=floor(typeK[i]);
        refLen[i]=floor(refLen[i]);
    }

    /* Calculate theta3 and r3 */
    theta1i*=RAD;
    theta2i*=RAD;
    theta4i*=RAD;
    theta2prime=theta2i-theta1i;
    offset=0;
    if(theta2prime<0) offset=-2*PI;
    if(theta2prime>2*PI) offset=-2*PI;
    theta2prime=offset;
    r3x=r1*cos(theta1i)+r4*cos(theta4i)-r2*cos(theta2i);  
    r3y=r1*sin(theta1i)+r4*sin(theta4i)-r2*sin(theta2i);  

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\[
\begin{align*}
r_3 &= \sqrt{r_3x^*r_3x+r_3y^*r_3y}; \\
delta &= \sqrt{r_1*r_1+r_2*r_2-2*r_1*r_2*\cos(\theta_2\prime)}; \\
\beta &= \arccos((r_1*r_1+\delta*\delta-r_2*r_2)/(2*r_1*\delta)); \\
\psi &= \arccos((r_3*r_3+\delta*\delta-r_4*r_4)/(2*r_3*\delta)); \\
\lambda &= \arccos((r_4*r_4+\delta*\delta-r_3*r_3)/(2*r_4*\delta)); \\
\text{if} (\theta_2\prime \geq 0 \text{ and } \theta_2\prime < \pi) \theta_3i &= (\psi-(\beta-\theta_1i)); \\
\text{else} \theta_3i &= (\psi+(\beta+\theta_1i)); \\
\end{align*}
\]

/* Find the spring constants */

for(i=0;i<4;i++)
{
  if(typeK[i]==1){
    k[i]=0;
    continue;
  }
  inertia=width[i]*width[i]*width[i]*height/12;
  if(typeK[i]==2){
    k[i]=youngs*inertia/refLen[i];
    continue;
  }
  if(typeK[i]==3 || typeK[i]==4){
    if(refLen[i]==1) len=(r1-pinR)/gamma;
    else if(refLen[i]==2) len=(r2-pinR)/gamma;
    else if(refLen[i]==3) len=(r3-pinR)/gamma;
    else len=(r4-pinR)/gamma;
    k[i]=gamma*kTheta*youngs*inertia/len;
    continue;
  }
  if(typeK[i]==4) k[i]*=2;
}

/* Find start and end angles */

after=0;
do{
  after+=(scanInc*RAD);
  theta2prime=(theta2i+after)-theta1i;
  if(theta2prime<0) offset=2*PI;
  if(theta2prime>=2*PI) offset=-2*PI;
  theta2prime+=offset;
  delta=sqrt(r1*r1+r2*r2-2*r1*r2*\cos(\theta_2\prime));
} while((r3*r3+\delta*\delta-r4*r4)/(2*r3*\delta)<1 && after<2*PI);

before=0;
do{
  before+=(scanInc*RAD);
  theta2prime=(theta2i-before)-theta1i;
  if(theta2prime<0) offset=2*PI;
  if(theta2prime>=2*PI) offset=-2*PI;
  theta2prime+=offset;
  delta=sqrt(r1*r1+r2*r2-2*r1*r2*\cos(\theta_2\prime));
} while((r3*r3+\delta*\delta-r4*r4)/(2*r3*\delta)<1 && before<2*PI);

if((after+before)>=2*PI){
  before=PI;
  after=PI;
}

/* Step through range from theta2i */

inRange=0;
maxPE=0;
maxTorque=0;
minTorque=0;
torquePrev=0;
bistable=0;
maxStress=0;
firstStable=0;
secondStable=0;
firstPE=0;
secondPE=0;
safetyFactor=0;

for(angle=-before;angle<=after;angle+=(calcInc*RAD)){
    theta2=theta2i+angle;
    theta2prime=theta2-theta1i;
    offset=0;
    if(theta2prime<0) offset=2*PI;
    if(theta2prime>=2*PI) offset=-2*PI;
    theta2prime+=offset;
    delta=sqrt(r1*r1+r2*r2-2*r1*r2*cos(theta2prime));
    beta=acos((r1*r1+delta*delta-r2*r2)/(2*r1*delta));
    psi=acos((r3*r3+delta*delta-r4*r4)/(2*r3*delta));
    lambda=acos((r4*r4+delta*delta-r3*r3)/(2*r4*delta));
    if(theta2prime>=0 && theta2prime<PI){
        theta3=(psi-(beta-theta1i));
        theta4=(PI-lambda-(beta-theta1i));
    }
    else{
        theta3=(psi+(beta+theta1i));
        theta4=(PI-lambda+(beta+theta1i));
    }
    theta3-=offset;
    theta4-=offset;
    diff[0]=theta2-theta2i;
    diff[1]=theta2-theta2i-(theta3-theta3i);
    diff[2]=theta4-theta4i-(theta3-theta3i);
    diff[3]=theta4-theta4i;
    PE=.5*(k[0]*diff[0]*diff[0]+k[1]*diff[1]*diff[1]+k[2]*diff[2]*diff[2]+k[3]*diff[3]*diff[3]);
    d4d2=(r2*sin(theta3-theta2))/(r4*sin(theta3-theta4));
    d3d2=(r2*sin(theta4-theta2))/(r3*sin(theta3-theta4));
    torque=k[0]*diff[0]+k[1]*diff[1]*(1-d3d2)+k[2]*diff[2]*(d4d2-d3d2)+k[3]*diff[3]*d4d2;
    stable=0;
    if(angle<(-before+0.5*calcInc*RAD) && torque>0 && before!=PI) stable=1;
    else if(angle>(after-0.5*calcInc*RAD) && torque<0 && after!=PI) stable=1;
    else if(torquePrev<0 && torque>0) stable=1;
    if(stable==1){
        bistable++;
        if(inRange==0){
            inRange=1;
            firstStable=theta2;
            firstPE=PE;
            maxTorque=torque;
            minTorque=torque;
            maxPE=PE;
            firstT4=theta4;
        } else{
            inRange=0;
            secondStable=theta2;
            secondPE=PE;
            secondT4=theta4;
        }
    }
}
if(inRange==1){
    if(torquePrev>0 && torque<0) centerAngle=angle+theta2i;
    if(torque>maxTorque) maxTorque=torque;
    if(torque<minTorque) minTorque=torque;
    if(PE>maxPE) maxPE=PE;
    for(i=0;i<4;i++){
        stress=(6*k[i]*diff[i])/(width[i]*width[i]*height);
        if(fabs(stress)>maxStress) maxStress=fabs(stress);
    }
    torquePrev=torque;
}

/* Calculate if it fits or not... */
fit=0;
if(r2+extend+pinR<r1) fit++;
else{
    minAng=theta2i;
    if(secondStable<minAng) minAng=secondStable;
    if(firstStable<minAng) minAng=firstStable;
    if(minAng<(80*RAD) && minAng>0){
        if(minAng>asin((pinR+(extend/2))/r1)) fit++;
    }
    else if(minAng>(70*RAD)) fit++;
}
if(r4+extend+pinR<r1) fit++;
else{
    minAng=theta4i;
    if(firstT4<minAng) minAng=firstT4;
    if(secondT4<minAng) minAng=secondT4;
    if(minAng>(100*RAD) && minAng<(180*RAD)){
        if((PI-minAng)>asin((pinR+(extend/2))/r1)) fit++;
    }
    else if(minAng<(110*RAD)) fit++;
}

/* Check stress and convert angles to degrees */
if(maxStress*maxStress<ZERO) safetyFactor=0;
else safetyFactor=yieldStress/maxStress;
sweep=fabs((secondStable-firstStable)*DEG);
if(fabs(secondStable-centerAngle)>fabs(firstStable-centerAngle)){
    maxFlipAngle=fabs(secondStable-centerAngle)*DEG;
    minFlipAngle=fabs(firstStable-centerAngle)*DEG;
} else {
    maxFlipAngle=fabs(firstStable-centerAngle)*DEG;
    minFlipAngle=fabs(secondStable-centerAngle)*DEG;
}
PE1=maxPE-firstPE;
PE2=maxPE-secondPE;
if(PEx<PE2) minPE=PE1; else minPE=PE2;
if(fabs(maxTorque)<fabs(minTorque)) minTorqueAbsolute=fabs(maxTorque);
else minTorqueAbsolute=fabs(minTorque);
centerAngle*=DEG;
firstStable*=DEG;
secondStable*=DEG;
before*=DEG;
after*=DEG;
firstT4*=DEG;
secondT4*=DEG;

/* send functions to OptdesX (Function names 16 chars max) */
afdscaC( minTorque, "MinTorque" );
afdscaC( maxTorque, "MaxTorque" );
afdscaC( minTorqueAbsolute, "Min_Flip_Torque" );
afdscaC( PE1, "MinPE" );
afdscaC( PE2, "MaxPE" );
afdscaC( minPE, "Min_Flip_PE" );
afdscaC( maxStress, "Max_Stress" );
afdscaC( safetyFactor, "Safety_Factor" );
afdscaC( firstStable, "First_theta2" );
afdscaC( secondStable, "Second_theta2" );
afdscaC( centerAngle, "Center_theta2" );
afdscaC( sweep, "Theta2_sweep" );
afdscaC( maxFlipAngle, "Max_flip_ang" );
afdscaC( minFlipAngle, "Min_flip_ang" );
afdscaC( firstT4, "First_theta4" );
afdscaC( secondT4, "Second_theta4" );
afdscaC( bistable, "Bistable?" );
afdscaC( before, "Range_Before" );
afdscaC( after, "Range_After" );
afdscaC( r1, "r1" );
afdscaC( r2, "r2" );
afdscaC( r3, "r3" );
afdscaC( r4, "r4" );
afdscaC( theta2i*DEG, "theta2i" );
afdscaC( theta3i*DEG, "theta3i" );
afdscaC( theta4i*DEG, "theta4i" );
afdscaC( fit, "Fit?" );

/*=============================================================================*/
Function anaposC
Postprocessing Function
/*=============================================================================*/
#ifdef __STDC__
void anaposC( void )
#else
void anaposC( )
#endif
{
}