Thermal optimization of a polyimide V-groove actuator for a walking micro-robot

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THERMAL OPTIMIZATION OF A POLYIMIDE V-GROOVE ACTUATOR FOR A WALKING MICRO-ROBOT

by

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Abstract

The objective of this thesis was to develop a Finite Element Model for the Polyimide V-groove actuator (fabricated by T. Ebefors, Sweden). Extensive FEM simulations for this MEMS actuator were performed using ANSYS 5.6. An optimization module was used to improve the performance of the existing design. A substantial improvement in the performance was observed for the proposed design. In short, this research established a methodology that can be extended for modeling and simulation of other MEMS devices.

A computer simulated FEM model for heat and deflection analysis was validated for two configurations of the Polyimide V-groove Actuator (i.e. a Serpentine Heater Configuration and a Polysilicon Heater Configuration). Some differences between the simulated and experimental results (reported by T. Ebefors) were noted in the low frequency domain. The role of various parameters including thermal conductivity and wall temperature has been investigated to eliminate these discrepancies.

To improve the performance of the actuator, different design geometries were proposed and each design was simulated for various frequencies. Significant performance improvement was observed for the case of "uniform diaphragm thickness at the V-groove bottom". The optimization module of ANSYS was used for optimizing the thickness of the silicon diaphragm (referred to as "single variable optimal design"). Steady state analysis showed that there is an improvement in the deflection and the force developed for the single variable optimal design over T. Ebefors' design. Transient analysis showed improvement in the cooling characteristics of the single variable optimal design over T. Ebefors' design.

In the second optimization exercise (referred to as "overall optimization"), all the dimensions of the V-grooves were used as design variables. A three times increase in the deflection was observed in the overall optimal design as compared to the single variable optimal design. Also, there is a three times reduction in the maximum force developed by the overall optimal design.
Transient analysis revealed that the overall optimal design has better cooling characteristics compared to the single variable optimal design.

Hence, for an application where the applied force is not a critical factor, the “overall optimal design” would be suitable, e.g. if a lightweight mirror is mounted on the end of the actuator, the mirror can be moved through a larger distance. For micro robotics applications, the “optimal design with a single variable” could be useful, where the load carrying capacity of this design is superior.
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**Nomenclature**

\( V \)  
Length of the top edge of the V-groove (\( \mu m \))

\( W \)  
Length of the bottom edge of the V-groove (\( \mu m \))

\( H \)  
Thickness of the Polyimide V-groove actuator (\( \mu m \))

\( D \)  
Thickness of the thin silicon diaphragm at the bottom of the V-groove (\( \mu m \))

\( \theta_{KOH} \)  
KOH etching angle for single crystal silicon wafer (54.74 ° for <100> orientation wafers)

\( l_{uncured} \)  
Dimension of the polyimide before curing (\( \mu m \))

\( l_{cured} \)  
Dimension of the polyimide after curing (\( \mu m \))

\( \varepsilon \)  
Shrinkage coefficient of the polyimide, expressed in terms of (%)

\( N \)  
Number of V-grooves connected in series

\( \alpha_s \)  
Static out-of-plane bending angle of the Polyimide V-groove structure

\( \alpha_d \)  
Dynamic out-of-plane bending angle of the Polyimide V-grooves, Stucture

\( \alpha_T \)  
Thermal expansion coefficient of polyimide (ppm / °C),

\( \Delta T \)  
Temperature increase caused by heating (°C)

\( \Delta x \)  
Horizontal displacement of the actuator at a particular frequency (\( \mu m \))

\( \Delta x_0 \)  
Reference displacement or deflection at cut-off frequency (\( \mu m \))

\( \delta_x \)  
Deflection of the actuator in X direction (\( \mu m \))

\( \delta_y \)  
Deflection of the actuator in Y direction (\( \mu m \))

\( T_{x=0} (t) \)  
Temperature of the left wall at time = t

\( h \)  
Natural Convective coefficient (W / m\(^2\) °K)

\( Q_g \)  
Hear generation rate of the heater (W / m\(^3\))

\( \Delta t \)  
Time sub step for ANSYS simulations.

\( \rho \)  
Density of the material (Kg / m\(^3\))

\( C \)  
Specific Heat of the material (J / Kg °K)

\( K \)  
Thermal Conductivity of the material (W / m °K)

\( T \)  
Time period for an input cycle (sec)
\( f \) Frequency of the input signal (Hz)

\( L \) Total length of the actuator (\( \mu m \))

\( \delta_{Ty} \) Deflection at the tip of the actuator in Y direction (\( \mu m \))

\( \delta_{Cy} \) Deflection at the center of the actuator in Y direction (\( \mu m \))

\( \delta_t \) The depth of the silicon fin into the V-groove (\( \mu m \))

\( L_1 \) The distance of the first fin from the edge of the V-groove (\( \mu m \))

\( L_2 \) The center to center-to-center distance between the two fins (\( \mu m \))

\( V_{heater} \) Volume of each heater (m\(^3\))

\( \Omega \) Sheet resistance of the film (ohms / square)

\( \rho \) Resistivity of the material (\( \Omega \) m)

\( t_h \) Thickness of the resistor (\( \mu m \)).

\( w_h \) Width of the resistor (\( \mu m \)).

\( I_h \) Length of the resistor (\( \mu m \)).

\( \sigma_{ut} \) The ultimate tensile strength (Mpa)

\( \sigma_{max} \) The maximum induced Van Misses stress (Mpa)

F.O.S. Factor of safety

\( P_R \) Radiation Heat loss (watts)

\( \varepsilon \) Emissivity of the material

\( \sigma \) Steffen’s-Boltzmann constant (5.6703x\( 10^{-8} \) W / m\(^2\) K\(^4\))

\( T_b \) Body Temperature (\(^oK\))

\( T_0 \) Ambient Temperature (\(^oK\))

\( A_s \) Surface area (m\(^2\))

**Special notations used for the ANSYS optimization module (Chapter 5).**

\( S \) The maximum induced Van Misses stress (Mpa)

\( T_{MAX} \) The maximum temperature developed in the actuator (\(^oK\))

\( Y \) Deflection at the tip of the actuator in Y direction (\( \mu m \))
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Chapter 1: Introduction.

1.1 Micro Electro Mechanical Systems (MEMS):

Micro-Electro-Mechanical Systems (MEMS) represents the integration of mechanical elements, sensors, actuators, and electronics on a common silicon substrate using IC (integrated circuit) micro fabrication technology. Nobel award winner Dr. Richard Feynman is considered as one of the principle investigators of the field of MEMS. In his famous paper "There is Plenty of Room at the Bottom", which was presented at the annual meeting of American Physics Society in 1959, [1], Dr Feynman proposed a new and exciting field of "miniaturization". He proposed to build micro machines using the fabrication techniques used for IC manufacturing (evaporation of thin films along with the photolithography and etching). Silicon wafers are used for IC fabrication. Along with well-established electronic properties, silicon also exhibits excellent mechanical properties. In his paper "Silicon as a Mechanical Material" [2], Kurt Petersen described the advantages of employing silicon as a mechanical material because of its excellent mechanical characteristics. Hence, silicon is suitable for micro mechanical structures as well.

MEMS refer to devices (sensors and/or actuators), with one of its dimensions less than 1 mm. T. Ebefors has defined MEMS as "a set of micro fabrication techniques for producing two- or three-dimensional structures featuring micrometer to millimeter dimensions" [3]. Today MEMS devices are used in many industrial and research applications. A MEMS based accelerometer was the first successful industrial MEMS device (used in automobiles). Some current MEMS-based devices include pressure sensors, accelerometers for airbags, optical switches and heads for inkjet printers, etc. There are three primary components of any complete system. These are, a sensor to record the change in its environment, a processing unit that will process the sensor data and send action signals, and an actuator to perform the required action depending upon the signals given by processing unit. Today, there is no successful industrial MEMS device that incorporates all three
components together. However, there are many everyday life products that contain micro-functional elements (either a micro sensor or micro actuator). Laser heads in CD and DVD units and read write heads in hard disks in computers are some of the everyday product containing micro functional elements [4].

1.2 Advantages of MEMS:

- **Batch Fabrication**: MEMS devices are fabricated using “Batch Fabrication” technique. Because of their small size, hundreds of these devices can fit on a silicon wafer. When this wafer completes a set of fabrication processes, hundreds of these tiny devices are available simultaneously. This technique is well suited for high volume applications, causing considerable reduction in the manufacturing cost. Although, mechanical capabilities of one MEMS device may not be sufficient to perform a useful task, combining many devices together could solve this problem. This is possible with MEMS devices because of batch fabrication capabilities.

- **Size**: The small size of MEMS devices, allows them to be used in applications where macroscopic devices are not feasible. The reduction in size also leads to a significant reduction in the weight of the device. Because of their smaller size and lower weight, micro devices make entirely new applications possible.

- **Speed of operation**: Because of the low mass of MEMS devices, they can act faster than conventional mechanical devices.

- **Power Consumption**: The small size of MEMS devices typically leads to lower power consumption. Although much of a research would be required in the field of micro power generation. Today, there is no micro-power generation device available. MEMS devices are operated through macro power sources.
• **Reliability**: MEMS structures are usually fabricated as a single monolithic structure. E.g. cantilever beam in an accelerometer also acts as an electrical conductor. This eliminates the requirement of separate wires. Also, because of their size, MEMS devices can be easily encapsulated thereby preventing corrosion and dust, and do not require lubricants thus improving reliability.

(http://www.memgen.com/resource_center/MEMS_benefits.htm)

1.4 **Challenges ahead of MEMS researchers**:

- **Fabrication Technique**: IC fabrication processes were primarily developed for two-dimensional structures like transistors. Some of these processes are not useful to realize complex three-dimensional MEMS structures. Hence, new processes will have to be developed to fabricate MEMS devices.

- **Materials**: IC fabrication technology can incorporate a limited number of materials. Some of these materials include costly metals like gold; platinum etc. Research must be focused to include all the materials in the fabrication process to get inexpensive MEMS devices.

- **Modeling of MEMS**: Current MEMS research is oriented towards fabrication of the devices. Lesser attention is provided towards modeling and simulation of the MEMS. There are two main reasons for this trend. First, the material properties at the micro scale differ from that of the macro scale properties. These properties are more dependent upon the fabrication technique and very limited data is available about the physical properties of microfilms. Secondly, the governing equations of the macro scale do not hold true at the micro scale. For example, at the micro scale, lagging behavior is observed in case of heat conduction [5].

  Use of differential equations for designing a system is limited to fairly simple systems. There are a few good books available on modeling of a micro system [6,7]. For complex micro systems (like the Polyimide V-groove
actuator, which is studied in this thesis), it is very difficult to come up with a governing equation. There is great need to develop simulation software packages that can address micro scale issues.

1.4 Microrobotics:

Insects can be considered as the inspiration behind the development of microrobotics. Using high-speed video photography and computer simulations, traits of insects such as the cockroach have been analyzed [8, 9]. One of the important locomotion requirements of a micro robot is a large out-of-plane deflection of the actuators forming its legs. As stated in the previous section, limitations of current lithographic techniques generally limit a MEMS device to planer geometries (this is referred as "Quasi 3 D techniques by T. Ebefors [3]). Using surface micro machining techniques, researchers have fabricated some freestanding structures, e.g. cantilever beams [10]. Although very long structures are possible using surface micro machining techniques, the thickness of such structures is limited to few microns due to limitations of material deposition techniques [3]. There are certain techniques available to obtain out-of-plane rotation of the actuator (the actuator is oriented in a plane perpendicular to the plane of the wafer). K. Pister at. al. developed a micro fabricated hinge structure to address out-of-plane rotation [11]. It involves a number of polysilicon layers deposited on one another and separated by sacrificial layers. Sacrificial layers are finally etched to release the structure. This structure could not be used as an actuator by itself. An external actuating mechanism using a comb-drive was required for controlled actuation [11]. G. Lin at. al. used the principle of bimorph structures to obtain a three dimensional actuator [12]. Bimorph structures consist of two metals with different thermal expansion coefficients that are sandwiched together. G. Lin at. al. used two layers of polyimides with different expansion coefficients. When heated, due to differential thermal expansion, out-of-plane actuation takes place. The structure developed by G. Lin at. al had a large bending radii, and the actuation required high driving voltage [3].
T. Ebefors, at the Royal Institute of Technology, Sweden, proposed a better solution using polyimide inside a V-groove [3] to obtain out-of-plane bending. The principle of actuation is based on the thermal expansion of polyimide inside a V-groove using an electrical heater. The initial out-of-plane orientation of the leg is due to shrinkage of the polyimide (because of chemical cross-linking between the molecules) when cured. Due to the larger absolute contraction at the top edge of the V-groove as compared to the bottom edge, the actuator (cantilever beam) attached to the V-grooves curls out of the plane, forming a freestanding leg [3]. When heated, the cured leg will straighten, due to differential expansion between the V-groove top and bottom. Details about this actuator are given in the next chapter.

This thesis deals with developing a finite element model for the Polyimide V-groove actuator. The model was validated using the experimental data published by T. Ebefors [3]. A brief outline of the thesis is presented in the next section.

1.5 Outline of this thesis:

The next chapter (#2) provides details about polyimide and its material properties. This is followed by the principle of polyimide V-groove actuation. Different configurations of the actuator fabricated by T. Ebefors along with their test results are discussed at the end.

Chapter # 3 presents details about the methodology used by the author to develop an FEM model of the actuator using ANSYS 5.6 followed by boundary conditions used in the simulations. The results of the simulations are compared with the experimental data. A parametric analysis was performed to resolve discrepancies in the FEM model at lower frequencies. Possible areas of improvements in the model are identified and the model is simulated with corrected boundary conditions.

In Chapter # 4, numbers of different design configurations are proposed to improve the performance of the actuator. FEM models of all these designs were
simulated for different frequencies and the results were compared with T. Ebefors’ design. The best alternative was selected for further analysis.

Chapter # 5 presents the results of the optimization exercise carried out on the best alternative identified in chapter # 4. Two optimized geometries were obtained each by evaluating different dimensions of the actuator. Maximum stress and temperature induced in each of these geometries was estimated at the time of the optimization exercise. The results of force versus deflection analysis conducted on these two optimized geometries are presented at the end of the chapter.

Conclusions and comments about future work are presented in Chapter #6.
Chapter 2: Polyimide V-groove actuator

2.1 Introduction:

The first Polyimide V-groove actuator was developed, fabricated, and tested by Ebefors et al. [3] at the Royal Institute of Technology, Sweden. This is a unique actuator with three dimensional actuation characteristics in the plane perpendicular to the plane of the wafer (out-of-plane). The project started with the objective to design and fabricate a gas flow sensor in 1986 at the Royal Institute of technology, Sweden [3]. Some of the specifications of this project at Royal Institute of Technology, Sweden included [3]:

- The actuator design must be robust enough to withstand high forces generated by the fluid,
- The structure should have small bending radius in order to lower the chip area consumption,
- The microfabrication process must be compatible with standard IC process (i.e. materials used in fabrication must be compatible with IC processes),
- The out-of-plane erected structures must be fixed at a bending angle close to 90° without the need of a manual assembly mechanism,
- The entire fabrication process of the actuator should be cost-effective for production and render good accuracy for the position of the out-of-plane standing structure,

This flow sensor project lead to the development of a new technology of using V-grooves filled with polyimide for three dimensional static structures. In 1997, T. Ebefors applied this technology for fabricating a MEMS-based walking microrobot platform. In his research, T. Ebefors successfully integrated an electrical micro-heater within the polyimide V-groove structure. Thus, a controlled actuation was made possible by supplying electrical pulses at different frequencies. [3] The out-of-the-plane (three dimensional) orientation and compatibility with the existing IC processes makes this actuator very unique for
microrobotics applications. The application of this actuator was further extended for developing a microconveyor [3].

2.2 Polyimide:

Polyimide is a commercially used, high temperature polymer [13]. Due to excellent physical properties among the polymer family, polyimides are frequently used in IC fabrication as an insulator. Some of the physical properties listed in [13] are as follows:

- Polyimide films have excellent thermal stability (up to 450 °C).
- Good dielectric properties (dielectric constants ~ 3.3, resistivity ~1016 Ω-cm)
- Superior chemical resistance, toughness and wear resistance.

Polyimides find their applications in IC packaging as a stress relief layer (also known as a buffer coat) [3]. Polyimide is already a compatible material with the existing IC processes. Because of high flexibility (Young’s Modulus = 2 – 3 GPa), polyimide was used in the fabrication of micro hinges [14]. Polyimide also finds its application as a sensing element in micro-sensors [15].

Most of the polyimides are inert in common organic solvents (e.g. amines, esters, alcohols, aldehydes). The first step in the formation of a solid polyimide film is spin coating a “polyimide precursor” on the silicon wafer. Polyimide precursor is a very viscous liquid. This precursor is transformed into solid polyimide by a prolonged heating process called “curing”. Curing is performed in an inert gas (usually in N₂ at SMFL, RIT) or under vacuum. Solidification of the precursor takes place because of the cross-linking between the molecules in hydrocarbon chain, a phenomenon called as “imidization” [3]. The curing process also results in weight and thickness loss due to out-gassing of the solvents at high temperature. This thickness loss is expressed in terms of the “shrinkage coefficient” (ε). The shrinkage coefficient “ε” is temperature dependent property, which increases with the curing temperature [3]. (For more details about the behavior of polymers, refer to [16].) This shrinkage also induces mechanical
stresses in the substrate. To reduce these mechanical stresses, polyimides with lower "shrinkage coefficient" are used in IC fabrication process [3]. In order to obtain a smaller bending radius, polyimide with higher shrinkage coefficient was used in the Polyimide V-groove actuator [3].

A wide variety of polyimides are commercially available ranging from low coefficient of thermal expansion (CTE) values (~5 ppm/°C) to high CTE values (>50 ppm/°C). Polyimide with a higher coefficient of thermal expansion was used by T. Ebefors to obtain a large actuation from the V-grooves, [3].

Further classification of the polyimides is based on its photosensitivity. Non-photosensitive polyimides find their applications in stress relief coats in IC packaging, as explained earlier in this section. Photosensitive polyimides have found their application in IC fabrication as well as in MEMS structures. Photosensitive polyimide is a negative resist. When exposed to UV light, the cross linking between the molecules takes place and during the development process the unexposed polyimide is removed [13]. A photosensitive polyimide

<table>
<thead>
<tr>
<th>Property</th>
<th>Unit</th>
<th>Typical Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tensile Strength at Break</td>
<td>MPa</td>
<td>260</td>
</tr>
<tr>
<td>Young’s Modulus</td>
<td>GPa</td>
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</tr>
<tr>
<td>Tensile Elongation at Break</td>
<td>%</td>
<td>80</td>
</tr>
<tr>
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<tr>
<td>Thermal Decomposition Temperature</td>
<td>°C</td>
<td>597</td>
</tr>
<tr>
<td>Coefficient of Thermal Expansion</td>
<td>ppm/°C</td>
<td>32</td>
</tr>
<tr>
<td>Coating Stress (100 silicon)</td>
<td>MPa</td>
<td>20</td>
</tr>
<tr>
<td>Dielectric Constant 1MHz: 0%-50% RH</td>
<td>3.1-3.4</td>
<td></td>
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<tr>
<td>Dissipation Factor 1MHz: 0%-50% RH</td>
<td>0.003-0.009</td>
<td></td>
</tr>
<tr>
<td>Dielectric Strength; room temp. - 50%RH</td>
<td>V/μm</td>
<td>342</td>
</tr>
</tbody>
</table>

Table 2.1: Physical properties of the cured "Durimide HTR-3” film [3].
was used in the fabrication of this actuator. T. Ebefors used “Durimide HTR-3” (polyimide precursor manufactured by “Arch Chemicals”, Connecticut, NY) in the fabrication of the Polyimide V-groove actuator [3]. Table 2.1 shows the physical properties of the “Durimide HTR-3”. This table was reproduced from the Arch Chemicals website (http://www.archmicro.com/products/4prob100.PDF).

2.3 Principle of actuation of the Polyimide V-groove actuator:

Initial out-of-plane bending (curling) of the Polyimide V-groove actuator is based on the shrinkage characteristic of the polyimide. As stated in the previous section, when cured at high temperatures, the polyimide film shrinks (due to degassing of solvents) causing reduction in the weight and dimensions. This shrinkage is quantified in the terms of Shrinkage coefficient “ε”. Figure 2.1 illustrates the shrinkage behavior of a polyimide film [3].

![Figure 2.1: Illustration of the polyimide film shrinkage after curing.](image)

Thickness of the polyimide film reduces, while the lateral dimension remains constant.

From Figure 2.1, when an uncured polyimide film of thickness “a” is cured the thickness reduces to “ a’ ”, where a > a’. The shrinkage coefficient is calculated using following formula [3],

\[
\varepsilon = \frac{\Delta l}{l} = \frac{l_{\text{uncured}} - l_{\text{cured}}}{l_{\text{uncured}}} = \frac{a - a'}{a}
\] (2.1)
When a rectangular opening on the surface of a single crystal silicon wafer is etched in KOH solution, a V-shaped (tapered) groove is formed at an angle of 54.74° [13]. If this V-groove is filled with polyimide and cured, the polyimide shrinks causing an out-of-plane bending (curling) of the beam attached at the end of the V-groove. Figure 2.2 illustrates the curing of the polyimide inside a V-groove.

Figure 2.2: Principle of the Polyimide V-groove joint. The curing causes the polyimide in the V-groove to shrink. The absolute lateral contraction length of the polyimide is larger at the top of the V-groove than at the bottom (ε a > ε b) resulting in a rotation which bends the free standing structure out of the wafer plane.

When the polyimide inside the V-groove is cured, the dimensions of the V-grooves are reduced due to shrinkage of the polyimide. The reduction (contraction) of the top edge of the polyimide is ‘ε V’ and that of the bottom edge will be ‘ε W’. Since V > W, the contraction of the top edge will be more than that of the bottom edge (i.e. ε V > ε W). This behavior leads to a static out-of-plane curling of the beam attached to the end of the V-groove. A larger static bending angle can be obtained by connecting a number of V-grooves in series. Assuming that the shrinkage of the polyimide in the horizontal plane is uniform throughout the V-groove (i.e. constant ‘ε’), and neglecting the shrinkage of the polyimide in
the vertical plane, the formula for the static bending angle of the structure reported in [3]

$$\alpha_s = 2.0.90 - \theta_{KOH} - \arcsin(\cos(\theta_{KOH}) \times (1 - \epsilon)) \] \quad (2.2)$$

where

$$\alpha_s = \text{Static out-of-plane bending angle of the structure},$$

$$N = \text{Number of V-grooves connected in series},$$

$$\epsilon = \text{Shrinkage coefficient of the polyimide},$$

$$\theta_{KOH} = \text{KOH etching angle (54.74° for <100> silicon wafer.)}$$

According to Equation 2.2, the static bending angle “$$\alpha_s$$” is independent of the thickness of the V-groove. Hence, the thickness of the V-groove can be chosen to fit the purpose of the device [3]. The top edge dimension of the V-groove (represented by “a” in Figure 2.2) will increase with the thickness of the actuator, thereby increasing the bending radius of the structure. The bending angle of the structure is referred as “static” because it is a result of an irreversible curing of the polyimide. Hence, this bending angle is fixed or static, unless the structure is heated by an external source. Figure 2.3 [17] illustrates the comparison of the bending angle from experimental data (represented by solid line in Figure 2.3) with the results of Equation 2.2 (represented by the dotted line in Figure 2.3) for structures with different number of V-grooves. The curing temperature was kept uniform (400 °C for 3 hours) for all structures. A strong correlation exists between the experimental and theoretical results. Figure 2.3 gives a static bending angle of 31.5° per V-groove from experimental data, versus 35.6 ° per V-groove from Equation 2.2. The value of the shrinkage coefficient was found experimentally. A 20 μm thick polyimide film was coated on number of silicon wafers. These wafers were cured at different temperatures for 3 hours. The thickness after curing was measured using a “profilometer” [17], and the shrinkage coefficient was calculated using Equation 2.1. (For more details, refer [17]). Appendix D shows the shrinkage coefficient vs. temperature characteristics of a sample tested at RIT by the author.
Figure 2.3: Bending angle $\alpha_d$ for joints with different numbers of V-grooves cured at 400 °C / 3 hours ($\varepsilon = 48\%$) [17].

Figure 2.4: Polyimide shrinkage coefficient versus the curing temperature [17].
Figure 2.4 [17] illustrates the relation between the shrinkage coefficient of polyimide and the curing temperature. Unlike curing, the thermal expansion of the cured polyimide is a reversible process (up to 300 °C). Hence, the dynamic bending angle could be achieved by integrating micro-heaters within the V-groove structure. Two different configurations of the heater have been used by T. Ebeffors [3] : i) serpentine heater configuration and ii) polysilicon heaters configuration. These resistive micro-heaters when excited with electrical pulses, produce local heat which is conducted into the V-grooves. The resulting temperature rise causes expansion of the polyimide and dynamic (reversible) change in the bending angle of the V-grooves. This dynamic bending is suitable for actuator applications. The dynamic bending angle can be estimated from Equation 2.3 [3].

\[
\alpha_d = 2. N \left[ 90^\circ - \theta_{KOH} - \arcsin(\cos(\theta_{KOH}) \times (1 - \varepsilon + \alpha_T \cdot \Delta T)) \right] \quad (2.3)
\]

where

- \( \alpha_d \) = Dynamic bending angle of V-grooves,
- \( \alpha_T \) = Thermal expansion coefficient of polyimide,
- \( \Delta T \) = Temperature increase caused by heating.

Other notations (i.e. N and \( \varepsilon \)) in Equation 2.3 represent the same parameters as in Equation 2.2. Figure 2.5 show the comparison of experimental data with the theoretical results given by Equation 2.3 [17]. A Polyimide V-groove (PVG) joint structure with five V-grooves was cured at 400 °C for 3 hours. The structure was then placed on a hotplate, and the bending angles were measured. The experimental setup for measuring the deflection is explained in the next section (Section 2.4). The solid line represents the experimental results and the dotted line represents the bending angle obtained from Equation 2.3.
2.4 Test set-up to measure the characteristics of a Polyimide V-groove (PVG) actuator:

As stated in the previous section, a Polyimide V-groove actuator could be operated under static or dynamic modes. In the static mode mode, the bending angle is a result of the curing of the polyimide (and the shrinkage associated with it), which is an irreversible process. In the dynamic mode, a reversible change in the bending angle could be achieved through thermal expansion of the polyimide (below 400 °C). T. Ebefors fabricated two different configurations of microheater to dissipate heat into the polyimide. Details about the design and fabrication of these two configurations are presented in the following sections. This section describes the test setup built by T. Ebefors (Details are available in [3]). Figure 2.6 [3] illustrates the schematic representation of the test setup used by T. Ebefors for measuring static and dynamic response of the actuators.
Test structures (or actuators) consisted of 600x500x30 \( \mu \text{m}^3 \) silicon plate bent (curled) out of the wafer plane using Polyimide V-groove joints [3]. Each actuator had a different number of V-grooves, anywhere between one and seven. Each V-groove was 30 \( \mu \text{m} \) thick and 70 \( \mu \text{m} \) wide at the top edge of the V-groove. A visible laser beam was aligned parallel to the test structure (alignment method is not described in the literature). The number of photo diodes recorded the position of the reflected beam (Figure 2.6). Using simple trigonometry, the angle between the incoming and the reflected laser beam was calculated (formulae are not provided in the literature published by T. Ebefors). T. Ebefors reported the accuracy of the test set-up to be better than 0.1° [3]. A static bending angle variation of +/- 1.5 ° per V-groove was observed over a whole wafer [3]. In other
words, for a wafer containing only 3 V-groove structures, the variation in the static bending angle would be \((1.5 \times 3 =) \pm 4.5^\circ\) over the whole wafer.

To measure the dynamic response, the actuator was excited using the square wave, at different frequencies [18]. Diodes indicated the change in the bending angle of the actuator. The stroke length (deflection at the tip of the actuator) was estimated using trigonometric relationship (no formulas provided in the literature). To measure the mechanical response, the actuator was actuated with the square wave of very low frequency to obtain the steady state amplitude. Two photodiodes were place at 10\% and 90\% level of this maximum amplitude. Time needed for the laser spot to move between the two photodiodes was used as the mechanical response time [18]. No differences in the response time between the serpentine type heater and the polysilicon type heater configuration were observed [3]. The cut-off frequency (frequency at which actuator attains steady state) for the actuators was observed to remain between 1 – 10 Hz, which corresponds to the measured response time of 100 – 200 ms [3].

2.5 Serpentine Heater Configuration:

Figure 2.7 [3] illustrates the schematic representation of the Serpentine Heater Configuration. The dimensions of the V-groove reported in Figure 2.7 were taken before the polyimide was cured. In this configuration, the aluminum was acting as an electrical conductor as well as an electrical heater. The width of the aluminum forming the electrical conductor (referred as an “interconnects” in the literature [3]) was more than the width of the aluminum forming the electrical heater. Multiple stripes of heater elements were placed inside each V-groove to ensure uniform heating across the length of the V-groove. It is important to note that Figure 2.7 is just a schematic representation of the system. For more details about the device, the reader is encouraged to refer to the PhD thesis of T. Ebefors [3].
Figure 2.7: Schematic of the Serpentine Heater Configuration fabricated and tested by T. Ebefors (*all dimensions are in µm*) [3].

Figure 2.8: Bending angle versus supply frequency for Serpentine Heater Configuration (90 mW/actuator) [3]
Figure 2.8 [3] illustrates the dynamic response characteristics of 3 V-groove and 4 V-groove actuators. The bending angle is measured using the test setup described in the previous section. Input power for both actuators (3 V-grooves and 4 V-grooves) was kept constant (90 mW/actuator). Dotted lines in Figure 2.8 represent the theoretical bending angle calculated using Equation 2.3. Cut-off frequencies (frequencies below which the steady state of the actuator is attained) were 4 Hz and 3 Hz for the three and four V-groove actuator respectively. The fabrication steps for this actuator are explained in the Section 2.7.

One of the major drawbacks of the Serpentine Heater Configuration is the fact that aluminum has been used as a heater material. Aluminum has a low resistivity and is generally used as an electrical conductor [13]. By reducing the width of the aluminum, electrical heaters can be formed, however the width of a layer inside a V-groove is limited by the existing lithographic technology. Metal stripes of very small width cannot be patterned inside a 30 μm deep V-groove. Hence, the resistance of the heater is fairly limited. Inducing artificial stresses inside the film the resistivity of the aluminum can be changed. However the induced stresses may affect the performance of the actuator and decrease its fatigue life. Another alternative is to reduce the film thickness. Since, the contacts are also formed with the aluminum, a thinner film would increase the resistance of the contacts, causing heat loss.

2.6 Polysilicon Heater Configuration:

The second configuration is referred as the "Polysilicon Heater Configuration". In this configuration, polysilicon heaters replaced the aluminum electrical heaters. Heaters are placed on top of the silicon substrate, between the two V-grooves. By changing the doping concentration of the boron (or phosphorus) the resistivity of the polysilicon can be changed [3]. Aluminum is used as an electrical conductor. Figure 2.9 [3] illustrates the schematic of the Polysilicon Heater Configuration. The dimensions of the V-groove reported in Figure 2.9 were measured before the polyimide was cured.
Figure 2.9: Schematic of the Polysilicon type heater configuration fabricated and tested by T. Ebefors (*all dimensions are in \( \mu m \)). [3]

Figure 2.10: Frequency dependence of the horizontal displacement (stroke length) of the four V-groove Polysilicon Heater Configuration. (90 mW / actuator) [3]
It is noteworthy that the polysilicon heaters do not go all the way across the width of the actuator. This modification was introduced to conduct more heat out of the polyimide V-groove joint (faster cooling) during the cooling period. At the same time two adjacent polysilicon heaters are in parallel. This provides a redundancy for failure [3]. If a heater on one of the sides fails due to fabrication defects, the heater on the other side would still be functional.

Figure 2.10 [3] illustrates the plot of horizontal displacement versus supply frequency for an input power of 90-mW/actuator, for the Polysilicon Heater Type Configuration. The test setup described in Section 2.4 was used for testing this actuator. The horizontal displacement in Figure 2.10 is reported in the terms of decibels. The decibel, or dB, is used to express the gain or the loss with respect to a reference. It is frequently used in the electronic industry for expressing huge quantities like gain of an amplifier (which is in millions). To obtain the linear displacement of the actuator, the following formula was used:

\[ dB = 20 \times \log_{10} \left( \frac{\Delta x}{\Delta x_0} \right) \]  \hspace{1cm} (2.4)

Where \( \Delta x \) = Horizontal displacement at a particular frequency
\( \Delta x_0 \) = Reference displacement (deflection at cut-off frequency).

The value of \( \Delta x_0 \) was not directly available in the literature. In his thesis report [3], T. Ebefors reported the maximum stroke length of 340 \( \mu \)m for an input power of 180 mW/actuator for the actuator length of 1000 \( \mu \)m. Data presented in Figure 2.10 is for 500 \( \mu \)m long actuator with 90 mW/actuator power. Linear approximation could be made to estimate the horizontal displacement. However the relation between the power and the deflection is not linear. In a private communication, T. Ebefors mentioned that the reference displacement (\( \Delta x_0 \)) was in the range of 50 \( \mu \)m to 60 \( \mu \)m for Figure 2.10. From the validation exercise, it was observed that for \( \Delta x_0 = 50 \mu \)m in Equation 2.4, the model developed in this
thesis (Section 3.9) predicts closer values to the experimental data. Hence, in this thesis, the value of \( \Delta x_0 \) was assumed to be 50 \( \mu \text{m} \).

2.7 Fabrication of Polyimide V-groove actuator [3]:

This section briefly describes the micro-fabrication sequence used for fabricating Polyimide V-groove actuators by T. Ebefors. For a detailed fabrication process refer to [3]. The fabrication process for both the configurations (i.e. Serpentine Type and Polysilicon Type) was similar. The only difference between the two processes was the method of depositing the micro heaters. The fabrication sequence for the Polysilicon Heater Configuration is described here. The differences in the fabrication sequence for Serpentine Heater configuration will be mentioned in \{ \}. Figure 2.11 [3], shows the schematic of the fabrication steps.

- 100 mm diameter, 500 \( \mu \text{m} \) thick SOI wafer (Silicon On Insulated) was used for the fabrication. SOI wafers consist of a thick silicon substrate known as "handle" (500
μm), and thin silicon substrate known as “device” (30 μm). These two silicon substrates are separated by a thin insulator layer (1 μm of oxide in this case), and hence the name.

- A polysilicon layer (1.5 μm thick) was deposited using a LPCVD (Low pressure chemical vapor deposition) technique and was doped with boron to form heaters. The heaters were patterned using lithographic techniques. {This process was not performed for the Serpentine Heater configuration}. A low stress silicon nitride layer was deposited (LPCVD) to encapsulate the heaters. 30 μm deep and 70 μm wide V-grooves were etched in KOH solution. The insulator (oxide) between the two silicon substrates acted as an etch stop (Figure 2.11 (a))

- Silicon oxide (LOCOS) was grown inside the V-grooves to insulate the metal contacts from silicon. 1.5 μm of aluminum was sputtered and patterned (using lithography) to form metal contacts. {Instead of patterning the contacts inside the V-groove, electrical heaters were patterned to form the Serpentine Heater configuration}. (Figure 2.11 (b))

- Photosensitive polyimide was spun coated on the wafer and patterned using lithographic techniques. The polyimide was soft baked at 150 °C. To protect the front side features (Heaters, contacts, polyimide etc), while performing processes on the backside of the wafer; the front side was covered with black wax. A 500 μm thick <111> silicon wafer was stuck to the black wax. Using a double side mask aligner, the backside release area was patterned. Backside etching of 500 μm thick wafer was done in KOH solution. Buried oxide layer acted as etch stop. (Figure 2.11 (C))

- The buried oxide layer was etched using BOF (Buffered Oxide Etch). Black wax was dissolved in acetone - propanol solution to release the actuator structure. Polyimide was then cured in N₂ atmosphere at 400 °C, to bend the structure out of plane. The structure was then tested using the probe station with the test setup described in Section 3.4. (Figure 2.11 (d)).
2.8 Applications of Polyimide V-groove actuator:

Apart from its application in the flow sensors, the Polyimide V-groove actuator had been used in variety of other applications [3]. Some of the MEMS devices realized using the Polyimide V-groove actuator, have been described in this section.

- **Micro-robotics application:** In previous attempts at fabricating a leg for microrobot, R. Yeh et al. used surface micro-machined micro-hinges as legs for micro robot [19]. In another attempt, Bright et al. used thermal actuation of thin manually erected silicon legs [20]. However, the main drawback of these designs was the difficulty to integrate the actuators in an array configuration [3]. At the same time, these were not true three-dimensional structures in terms of controlled out-of-plane actuation. The unique out-of-plane bending feature of the Polyimide V-groove actuator makes it the most suitable actuator for the locomotion of a micro-robot. An array of Polyimide V-groove actuators was fabricated with the static bending angle of 135 ° for each actuator [3]. A larger static bending angle required more V-grooves in series, which in turn gave larger

![Figure 2.12: Operating principle of walking micro-robot. [3]](image)

![Figure 2.13: Walking speed as a function of frequency for different loads and power conditions for a walking micro-robot. [3]](image)
dynamic bending angles. Figure 2.12 [3] illustrates the working principle of a walking micro-robot. Displacement equal to $2 \cdot \Delta x$ is obtained in one cycle because of the fixed phase difference of $90^\circ$ between the two sets of legs. Figure 2.13 shows the walking speed of the micro-robot as a function of frequency for different load and power conditions. Experimental results presented in Figure 2.13 were obtained for a micro robot with eight legs (PVG actuators) and each leg was 1 mm long.

- **Micro-conveyor:** Micro-conveyor is basically micro-robot flipped upside down. To calibrate the conveyor for the speed versus the load on the conveyor, a Whetstone Bridge strain gage was fabricated on the conveyor itself. The polysilicon heater layer (Figure 2.11 (a)) was utilized for fabricating the piezoresistors of the Whetstone Bridge [3]. Based on the information from these piezo resistive tactile sensors, advanced control of each actuator was possible [3]. (Details about the "advanced control" are not provided in [3].)

### 2.9 Objective of this thesis:

The objective of the current research, presented in this thesis, is to optimize the design of the Polyimide V-groove (PVG) actuator to maximize the deflection, using FEM software (ANSYS).

### 2.10 Methodology used in this thesis:

Earlier research on the Polyimide V-groove actuator was entirely focused towards developing a micro-fabrication process [3]. Two different heater configurations (Serpentine heater and Polysilicon heater) were fabricated and tested by T. Ebefors [3]. A brief summary about the published test data available has been presented earlier in this section. T. Ebefors did not perform FEM simulations on the actuator design. The research in this thesis presents the results of FEM simulations performed on the Polyimide V-groove actuator design.
The first step in this simulation exercise was focused towards model validation. A two-dimensional model of the actuator fabricated by T. Ebefors was generated using ANSYS 5.6. Simulation results were compared with the published experimental data [3]. Some differences between the simulated data and experimental data were observed in the low frequency region (1 Hz to 10 Hz). Parametric analysis was performed to analyze the effect of various parameters on the output of the actuator. With new boundary conditions, discrepancies at low frequencies were resolved.

To improve the performance of the actuator, various designs modifications were simulated. The best alternative was found on the basis of improvement in the maximum deflection and cooling of the actuator. Using a trial and error method the range of optimized dimensions was obtained. The effect of heater placement on the performance of the actuator was studied for the optimized dimension.

To estimate the exact optimized dimensions, a multi variable, non-linear, sub-problem optimization method available in ANSYS was used. The optimized design was further simulated for transient conditions and the output was compared with T. Ebefors' design. Stress analysis was carried out to estimate the factor of safety of the design. Finally, the force versus deflection characteristics was obtained for the optimized design.
Chapter 3: Model Development and Validation

3.1 Introduction:

The Finite Element Method (FEM) is a numerical procedure that can be applied to obtain solutions of variety of problems. ANSYS is a general-purpose finite element computer program, which is capable of performing static, dynamic, heat transfer, fluid flow and electromagnetic analysis [21]. The first step in any simulation exercise (using finite element software) is the validation of the model. There are two most commonly used validation techniques available. Either the simulation output is compared with an analytical solution or it is compared with experimental data. The first approach is suitable for less complex systems. For complex systems, like the PVG actuator, it will be very difficult to find a set of governing equations predicting the behavior of the actuator. Hence, for the validation of the Polyimide V-groove actuator model, the second approach will be followed. The model will be validated using the published experimental data available from the T. Ebefors’ thesis [3].

3.2 Outline of the chapter:

The development and validation of the Polyimide V-groove actuator model consisted of following steps:

- Develop an FEM model of Serpentine Heater Configuration (3 V-grooves and 4 V-grooves) for transient conditions.
- Validate the model by comparing the simulation results with the published experimental data.
- Identify the role of conduction, convection and wall temperature in the heat transfer process, thereby eliminating any discrepancies in the simulation.
- Develop and validate the model for the Polysilicon Heater Configuration, based on the concepts developed in Serpentine Heater Configuration.
- Understand the transient phase of the heat transfer for further optimization of the actuator.
3.3 Model development:

A finite element model of the Serpentine Heater Configuration was generated using ANSYS 5.6. The following assumptions were made in order to simplify and speed up the simulation:

- An equivalent two-dimensional model was created representing the three dimensional geometry of the Serpentine Heater Configuration.
- All the material properties were assumed to remain constant over the operating temperature range. (Refer Appendix D for material properties).
- The out-of-plane curling of the actuator was neglected (thus the actuator was assumed to remain in the plane of the wafer as a straight beam).
- The top and bottom edge dimensions of the V-grooves were reduced as per the shrinkage coefficient (~ 45% for curing temperature 350°C [3]).
- To investigate the role of various parameters (convection, wall temperature, etc) the entire length of the actuator was modeled.
- The thickness of the aluminum layer was assumed to be uniform (1.5 μm) throughout the actuator [3].
- The two dimensional model would assume the width of the aluminum layer to be equal to the width of the structure. This would change the resistance to the heat flow through the contacts. To correct for this, the thermal conductivity of the aluminum forming the contacts, was scaled down by multiplying it by the ratio of the actual width of the contact to the entire width of the structure. Similar treatment was applied to the aluminum heaters inside the V-grooves.

Since the modeling was done in two dimensions, the supplied electric power was converted into an equivalent heat generation rate by dividing the total input power per leg by the product of the volume of the aluminum heater times the number of V-grooves.
• Further simplification was done by neglecting the oxide layer deposited inside the trenches to act as an electrical insulator, and on top of the wafer to act as a masking layer for etching the V-grooves. Since the power was converted into an equivalent heat generation rate for this simulation, it was reasonable to neglect it.
• The V-grooves were assumed to be entirely filled with polyimide after curing.
• Heat transfer in the Z-direction (perpendicular to the plane of the paper) was neglected since the electrical heaters spread throughout the width of the V-groove [3].
• Internal stresses generated due to curing of the polyimide were neglected.
• Radiation heat loss was neglected. (justification provided in chapter 5)
• After the parametric studies, it was observed that convection has a very little effect on the simulation output. Hence, the convection boundary condition was neglected.

Figure 3.1 shows the dimensional view of the model generated using ANSYS 5.6. The published experimental data is available in the form of deflection angle versus the supply frequency of the input electrical pulses. This requires transient analysis of the model. The mesh was generated with “coupled field”, “PLANE 13” elements since this is the only element available in ANSYS, which allows transient analysis (against the indirect analysis which involves breaking down problem into separate heat transfer and structural analysis and solve them separately). According to the ANSYS reference manual [22] “the PLANE13 element has a 2-D magnetic, thermal, electrical, piezoelectric, and structural field capability with limited coupling between the fields. PLANE13 is defined by four nodes with up to four degrees of freedom per node. PLANE13 has large deflection and stress stiffening capabilities. When used in purely structural analyses, PLANE13 also has large strain capabilities”.

PLANE 13 is the only two-dimensional element available in ANSYS, which can couple thermal and elastic behavior. As stated earlier, PLANE 13 is a
Figure 3.1: (a) Enlarged view of the meshed model for Serpentine Heater Configuration (3 V-grooves) generated using ANSYS 5.6. (b) Complete model of the 3 V-grooves actuator. (All dimensions are in µm).

quadratic element (four nodes per element). Displacements UX and UY, in the x and y directions (respectively), and temperature were selected as degrees of freedom per node. The element behavior was set as "plane strain" which refers to no heat transfer in the z direction. A bottom-up technique was used to generate the geometry of the model. The key-points defining the corners of the first V-groove were first plotted. Joining these key points in an appropriate, order the areas defining the polyimide and the aluminum contacts were created. Then all the areas were "Glued" to each other. According to the ANSYS manual, "Glue is similar to overlap, except that it applies to cases in which the intersection between entities occurs only at the boundary (edges). Two glued entities (in this case, two glued areas) maintain their individuality (they are not "added"), but they
become connected at their intersection. Next the aluminum and the polyimide areas were copied (reproduced) at equal distances to generate multiple V-grooves. Connecting the key points of the V-groove edges silicon substrates between the V-grooves were generated. All the areas were “Glued” again thus generating the complete geometry of the actuator. Using the “Mesh Attribute” function available in ANSYS, appropriate material properties were allocated to the areas. “Mesh Attribute” identifies the material number to be assigned to subsequently defined elements.

There are three meshing options are available in ANSYS, i.e. free meshing, mapped meshing, and smart meshing. According to ANSYS manual, free mesh has no restrictions in terms of element shapes, and has no specified pattern applied to it. Compared to a free mesh, a mapped mesh is restricted in terms of the element shape it contains and the pattern of the mesh. A mapped area mesh contains either only quadrilateral or only triangular elements. In addition, a mapped mesh typically has a regular pattern, with obvious rows of elements. Smart element sizing (smart sizing) is a meshing feature that creates initial element sizes for free meshing operations. Smart Sizing gives a better chance of creating reasonably shaped elements during automatic mesh generation.

When “free meshing” and “smart sizing” options were used, very fine mesh was generated. There was no difference between the simulation results of “smart” or “free” meshing versus “mapped” meshing. Since stresses developed in the model were not taken into account during this validation exercise, the mesh density will not be a critical factor. However, the simulation time increases drastically for finer mesh. Hence “mapped meshing” was used. Each line forming the geometry of the model was divided into an appropriate number of parts. For the model shown in Figure 3.1, all the edges of the V-groove were divided into ten equal divisions. This number-of-divisions guides the number of elements and their size while meshing. A denser mesh was generated near the V-grooves. The silicon substrate was divided into two regions. The region near to the V-grooves had a denser mesh as compared to rest of silicon substrate.
3.4 Transient Analysis:

Transient analysis was performed for one heat cycle. The cycle was divided into three load steps. Figure 3.2 shows the graphical representation of the three load steps. The first load step (represented by “Step 1” in Figure 3.2) is applied for a very small amount of time (~1 x 10^{-8} sec). This step defines the initial boundary conditions of the system. These initial boundary conditions include:

- The left wall (edge) of the actuator was restrained from having displacements in the x and y directions (\( \delta_x = \delta_y = 0 \)), (Figure 3.3)
- A heat sink was created at the same edge by enforcing a constant temperature boundary condition (\( T_{x=0}(t) = 20^\circ \))

![Figure 3.2: Load steps applied during the transient simulation of the actuator for one cycle.](image)

**Figure 3.2:** Load steps applied during the transient simulation of the actuator for one cycle.

![Figure 3.3: Schematic diagram representing applied boundary conditions for the serpentine heater type actuator model. (Figure not to the scale)](image)

**Figure 3.3:** Schematic diagram representing applied boundary conditions for the serpentine heater type actuator model. (Figure not to the scale)
Convection was applied on the top and bottom edges of the actuator \((h = 20 \text{ W/m}^2 \text{oK at 20 oC})\).

- Zero heat generation rate was applied on the aluminum heaters inside the trench.

In the second step ("Step 2" in Figure 3.2), the heat generation rate was applied to the aluminum heaters inside the trench. This represents the heating period and is applied for a 50% duty cycle. In order to accommodate various losses, the heat generation rate was truncated by 10%. All other boundary conditions applied in the first step were kept unchanged. The time sub-step was calculated using the Fourier number given in the following Equation 3.1 [22]

\[
\Delta t = \frac{\rho C (\Delta x)^2}{K} \quad (3.1)
\]

For micro systems, the average element length is of the order \(1 \times 10^{-6} \text{ 1e-6 meters, which leads to an extremely small time sub-step (~1 x 10^{-8} sec.})\). When a simple system of only one V-groove was simulated for this time sub step, the student version of ANSYS 5.6 terminated the simulation after 1e-4 seconds (10000 sub steps) and it took almost 48 hrs to simulate this much. The same behavior was observed for a time step equal to 1e-7 and 1e -6 sec. Figure 3.4 compares the simulation output for different sizes of time sub steps.

The simulation stopped at point ‘A’ (for time step values from \(1 x 10^{-8}\) to \(1 x 10^{-6}\) sec, Figure 3.4). According to Figure 3.4 results nearly match for a time step equal to \(1 \times 10^{-4}\) sec and \(1 \times 10^{-5}\) sec. For achieving faster simulation, a time sub-step value was selected as \(1e-4\) sec for all the transient simulations here onwards. The highest simulated frequency used in this thesis is 100 Hz, which corresponds to the cycle time period \((T)\) of

\[
T = \frac{1}{f} = \frac{1}{100} = 0.01 \text{sec} \quad (3.2)
\]
With the time sub step = $1 \times 10^{-4}$ sec, 100 data points ($0.01/1 \times 10^{-4} = 100$) will be calculated for 100 Hz. For all the frequencies lower than 100 Hz, more than 100 data points will be calculated by ANSYS. To obtain a better understanding of transient behavior, all the output data points were stored in the result file.

In the third step ("Step 3" in Figure 3.2), the heat generation rate was again equated to zero, which represents the cooling period. The cooling period is also applied for the 50% of the duty cycle.

![Graph showing temperature over time with different time substeps](image)

**Figure 3.4: Comparison of the simulated temperature values at the top of the polyimide for single V-groove actuator with serpentine heater configuration (90 mW/leg, 10 Hz)**

### 3.5 Effect of Convection:

The 3 V-grooves configuration was first simulated with convection applied at the top and bottom edges of the model ($h=20$ W/m² °C). This represents the
Figure 3.5: Effect of convection boundary convection on the output of a 3 V-Groove Serpentine Heater Type Configuration (90 mW/leg).

natural cooling of the actuator. The same configuration was again simulated with no convection applied at the top and bottom edges of the model (h=0 W/m²°C). All the other boundary conditions and simulation parameters explained in Section 3.3 Transient Analysis' were kept unchanged. Both the boundary conditions were simulated for different frequencies. Figure 3.5 shows the comparison of the results obtained from the two simulations. It is apparent from the deflection patterns of these two simulations that convection has a very small effect on the output (deflection of the actuator). There is hardly any change in deflection values even if convection is neglected. Complete simulation without convection boundary condition took lesser time. Accordingly, for all further simulations, convection was neglected. Analysis of this behavior is done in more detail in the Parametric Analysis section.
3.6 Results:

The simulations were carried out for actuator geometries with 3 V-grooves and 4 V-grooves. Each model was simulated for various frequencies. Throughout this validation exercise the input power to each actuator was kept constant (90 mW/leg). The deflection in the vertical direction (along Y-axis) was recorded for the tip of the actuator ($\delta_1$) and the center of the actuator ($\delta_2$) over the period of time. The bending angle was calculated using following formula 3.3.

$$\theta = \sin^{-1}\left(\frac{\delta_1 - \delta_2}{L/2}\right)$$ (3.3)

Where L is the total length of the actuator. Figure 3.6 shows the comparison of experimental results vs the simulated values for the 3 V-grooves.

![Figure 3.6: Comparison of Simulated values with experimental data for 3 V-grooves with serpentine heater type actuator (90 mW/leg)](image)
configuration. It can be observed from Figure 3.6 that the simulated values agree with the experimental data for the high frequency domain. For analysis purposes, the Figure 3.6 can be divided into three regions. The first region contains high frequencies from 100 Hz to about 60 Hz. It is clear from the Figure 3.6 that FEM model closely predicts the behavior of the 3 V-grooves actuator at high frequency region.

The second region contains the moderate frequencies from 50 Hz to 20 Hz. The simulation trend rises rapidly as compared to the experimental data for moderate frequency region. In the Model Development section, the material properties were assumed to remain constant throughout the operating temperature range. At the lower frequencies the heating period increases, causing a higher temperature rise within the actuator. The material properties of the actuator are likely to change at moderate frequency region. Hence the model shows the deviation as compared to the experimental data. It is important to note from Figure 3.6 that the deviation of simulated output increases inversely with the frequency value. i.e. the deviation is increasing with the lowering of the simulation frequency. This trend supports the reasoning about the deviation made above.

For the lower frequency region from 15 Hz to 3 Hz, the model fails to predict the behavior of the actuator. Issues in the low frequency domain will be addressed in the Parametric Analysis and Discussion sections, later in this chapter.

A similar analysis was done for the four V-grooves configuration. Figure 3.7 shows the comparison of results for four V-groove configuration. For analysis purposes Figure 3.7 will also be divided into the same three regions as Figure 3.6. For the high frequency region (100 Hz to 60 Hz), the model closely predicts the behavior of the actuator with 4 V-grooves. This confirms the validation of all the assumptions made while developing the model (at high frequencies).

In the moderate frequency region (60 Hz to 20 Hz), the model for 4 V-grooves agrees better than the model for 3 V-grooves. From Figure 3.7, the model for 4 V-grooves closely predicts the actual behavior from 60 to 40 Hz. The
simulation output deviates from the experimental data below 40 Hz. The amount of deviation in 4 V-grooves model is less as compared to the 3 V-grooves model. Since the same power is being dissipated in both the cases (90 mW/leg), the power dissipated per V-groove is less in 4 V-grooves actuator. This observation leads to the fact that the temperature rise in the case of the 4 V-grooves actuator would be lower as compared to the 3 V-grooves actuator. Hence, the model with constant material properties is better in predicting the experimental output for 4 V-grooves as compared to 3 V-grooves. For the further optimization of the actuator, a model with four V-grooves will be used. Issues in the low frequency region will be addressed in the Parametric Analysis and Discussion sections.

When tests were conducted on the actuator, it was observed that the outermost V-groove was getting heated more as compared to the innermost V-groove [3]. A similar behavior was observed in the simulation. Figure 3.8 illustrates the simulated temperature evolution at the center of the top edge of the V-grooves. The innermost V-groove (or the leftmost V-groove) is referred as “1st
Figure 3.8: Temperature evolution at the center of the top edge of the V-grooves in the 4 V-groove serpentine heater type configuration. (90 mW/leg, 20 Hz)

V-groove" in the Figure 3.8. All the other V-grooves are number in the sequential order. It can be seen that the temperature in the 4th V-groove (outermost V-groove) is maximum among the four. This behavior is a result of the innermost V-groove acts as a thermal insulator because the thermal conductivity of the polyimide is very small (0.16 W/m °K). Thus most of the heat is trapped between the outer V-grooves, which heats up the outer V-grooves. The outermost V-groove gets heated the most. Similar behavior was observed in the case of the 3 V-grooves configuration.

From a performance standpoint; higher temperature rise in the innermost V-groove is desirable since this would cause higher deflection at the tip of the actuator. Also, as the innermost V-groove acts as an insulator, there is no cooling path available. These two issues will have to be dealt while optimizing the design of the actuator.
3.7 Parametric analysis:

In order to understand the effect of various parameters on the transient response, a parametric analysis was conducted. This analysis was aimed at two objectives. The first objective was to address the model discrepancies at the lower frequency region (15 Hz to 3 Hz). From Figure 3.6 and 3.7, it was observed that the FEM model reaches steady state earlier as compared to the actual system (steady state in Figure 3.6 and 3.7 is represented by no change in the angular deflection with reduction in frequency in frequency (or increasing heating time)). For the three V-groove configuration, the maximum deflection for the model is constant for all the frequencies lower than 10 Hz (Figure 11). Hence the model reaches steady state after \((1/f = 0.1 \text{ Sec})\). From the experimental data (Figure 11), this behavior is observed for the frequencies lower than 3 Hz (steady state = \(1/f = 0.33 \text{ Sec}\)). The second objective is to understand the effect of various parameters on the transient response, which will help in the optimization exercise. The following trends were obtained:

- **Effect of convection:** From Figure 3.5, convection plays a very limited role in the heat transfer process of this actuator. Similar behavior was observed when the heat transfer coefficient was increased to \(h=100 \text{ W/m}^2 \text{ °C}\). This observation helped in reducing the simulation time by not applying a convection boundary condition. However it brings forth the essential requirement of an effective cooling mechanism for the actuator. From Figure 3.8, the innermost V-groove is acting as an insulator. Thus, most of the supplied heat is being trapped in the actuator itself. This behavior will be very helpful for achieving higher deflections, but it further complicates the cooling of this actuator. These observations lead to the conclusion that an active cooling mechanism is required for the actuator. However, complexities involved in fabrication will be an important issue while designing an active cooling mechanism for this actuator.
Effect of conductivity of silicon: The simulation was run for three different thermal conductivity values of silicon. Figure 3.9 shows the comparison of simulation results for 3 V grooves. As the conductivity of silicon is reduced to 70 W/m °K, more heat is being conducted into the polyimide, thus increasing the deflection of the actuator. It can be noted from Figure 3.9 that the variation in conductivity value causes a small change in deflection at higher frequencies, but at lower frequencies, the magnitude of change is significant. Increasing the thermal conductivity of silicon to 140 W/m °K, the deflection of the actuator reduces as higher amount of heat is being conducted into the silicon as compared to polyimide. As far as the validation of the model is concerned, this data is of little use. Since a single crystal silicon wafer was used for the fabrication of device, there would be a small variation in its thermal conductivity. It is noteworthy that the performance of

![Figure 3.9: Effect of the change of the thermal conductivity of silicon substrate on the maximum deflection of 3 V-grooves serpentine heater type actuator (90 mW/leg)](image)
the actuator can be improved by using the base layer with lower thermal conductivity. By introducing some impurities, the thermal conductivity of the silicon can be changed [13]. However, due to the process limitations, this change is limited for a very small thickness of the silicon. Use of Single crystal quartz wafer could be an alternative since the thermal conductivity of the quartz is less than that of silicon [13]. However, new set of processes will have to be developed to etch V-grooves in the quartz wafer.

- **Effect of conductivity of polyimide:** The effect of thermal conductivity variation of polyimide is shown in Figure 3.10. When the conductivity of polyimide is increased, the deflection value increases. Due to increased thermal conductivity of polyimide, more heat is conducted into the V-groove. This causes a higher temperature rise and higher thermal expansion which

![Graph](image)

**Figure 3.10:** Effect of the change of the thermal conductivity of polyimide on the maximum deflection of 3 V-grooves serpentine heater type actuator (90 mW/leg)
results in a higher deflection. This increase in deflection is nearly constant over the frequency range. Since, variety of polyimides are available in the market, selecting a polyimide with higher thermal conductivity is relatively simple as compared to changing the thermal conductivity of silicon.

- **Effect of wall temperature:** For all the simulations performed so far (Figures 3.6 and 3.7), a constant temperature boundary condition (20°C) was enforced at the left wall. However from the steady state analysis of the 3 V-grooves Polysilicon Heater Configuration in the T. Ebefors’ thesis, the wall temperature has rose to 33.75°C [3]. Figure 3.11 shows the effect of linearly increasing wall temperature from 20°C to 35°C over the period of 150 ms (settling time obtained experimentally [18]). Figure 3.11 shows better agreement with experimental results compared to Figure 3.6. The trend of

![Graph showing effect of wall temperature on deflection](image)

**Figure 3.11:** Effect of the linear increase of the wall temperature boundary condition, on the maximum deflection of 3 V-grooves serpentine heater type actuator (90 mW/leg)
the simulation results is more consistent with the experimental data at lower frequency region. A similar trend was observed for the 4 V-groove configuration also.

3.9 Discussion of the Results:

The plots of deflection vs. frequency (Figures 3.6, 3.7 and 3.11) are consistent with the experimental data for frequencies of 40 Hz and above. The difference is more significant for the 3 V-groove configuration as compared to the 4 V-groove configuration for frequencies between 20 and 40 Hz (see Figures 3.6 and 3.7). Since the same power is distributed in each case, the power dissipation per groove is more in the 3 V-grooves configuration as compared to the 4 V-grooves configuration. Higher power dissipation results in the higher temperature rise in the case of 3 V-grooves actuator. The material properties have been assumed to remain constant with temperature. Hence the deviation between the experimental and simulation data is higher in the case of the 3 V-grooves configuration. From Figure 3.5 it was observed that the convection plays a very little role in heat dissipation. This leads to the conclusion that an active cooling mechanism will be required for dissipating the entire heat during the cooling cycle. Failure to remove all the heat may result built up of residual deflection over time. Further simulation will be required for optimizing the heat transfer.

The simulated values differ from the experimental values at lower frequencies. This deviation may be explained due to the following reasons:

- Although the shape of the polyimide is assumed to be a perfect V, in actual practice the shape of the experimental V may differ dimensionally. One of the reasons for this shape difference in the experimental V is due to the presence of the aluminum contact, which expands at the high curing temperature of polyimide (as opposed to shrinkage in polyimide). As a result, the actual shrinkage pattern will be different than assumed.
- Model neglects the polyimide shrinkage in the vertical direction. However, curing of the polyimide causes shrinkage in all the direction. Hence, the V-grooves were not filled completely after curing [3].

- Constant material properties were assumed over the temperature range. At lower frequencies, the temperature increases inside the system due to longer heating time. Therefore, at lower frequencies the assumption of constant material properties rather than temperature dependent properties will induce more error.

- The innermost V-groove (nearest to the left wall) acts as a heat insulator, thereby causing higher temperatures in the outer V-grooves, see Figure 3.8. This higher temperature increases the resistance of the heating element of the outer V-groove, causing higher power generation inside the outer V-groove. This results in the outermost V-groove (farthest from the left wall) contributing more to the total deflection. For this analysis, equal power generation in all V-grooves was assumed. Therefore, at lower frequencies, the simulated heat supplied to the outermost V-groove is less than the actual value. The power generation inside each V-groove needs to be changed over time, to correct this error.

- Increasing the wall temperature over time gives better results than the constant temperature assumption (Figure 3.11). Instead of using a linear temperature rise (as done here) perhaps a exponential rise may prove to be more effective.

- Due to the thermal expansion coefficient mismatch between polyimide and aluminum, residual stresses would get developed in the structure after curing, which have been neglected in the modeling.

- As reported by T. Ebefors, the data acquisition system used had significant uncertainties [3]. Deviations as high as +/- 0.25° can be involved in the measurement system.
3.9 Model Validation for Polysilicon Heater Configuration:

The primary aim of this exercise is to validate the FEM model for Polysilicon Heater Configuration. The model for the Polysilicon Heater Configuration was developed using the techniques developed while validating the model for Serpentine Heater Configuration. If the model for the Polysilicon Heater Configuration could be validated using the same technique developed before, it would build confidence with the entire simulation exercise. As compared to Serpentine Configuration, the Polysilicon Heater configuration appears more promising. Polysilicon has higher resistivity as compared to aluminum and by changing the doping concentration the resistivity of the polysilicon can be changed. Also, the polysilicon heaters are on the top of the V-grooves (Figure 2.9). Hence, the fabrication of these heaters is much easier. For the Serpentine Heater Configuration, it is very difficult to change the resistivity of the aluminum. The Serpentine Heaters have a smaller width of the aluminum inside the deep trenches. A complicated two-layer photoresist process is required in order to fabricate small features inside the trench [3]. Considering above factors, the Polysilicon Heater Configuration has been used for further optimization in this

![Figure 3.12: Enlarged view of the meshed model for Polysilicon Heater type Configuration generated using ANSYS 5.6.](image)

*(All dimensions are in \( \mu \text{m} \)).*
thesis. Figure 3.12 shows the close up of the meshed model generated for Polysilicon Heater Configuration. The model was simulated using the same boundary conditions that are described in the Section 3.4. Figure 3.13 shows the comparison of simulated results with the experimental data.

![Graph showing comparison](image)

**Figure 3.13: Effect of the linear increase of the wall temperature boundary condition, on the maximum deflection of 4 V-grooves Polysilicon Heater Type Configuration (90 mW/leg)**

From Figure 3.13, the model is capable of closely predicting the behavior of Polysilicon Heater Configuration for high frequency region. When the fixed wall temperature boundary condition was replaced with a linearly changing wall temperature, the model closely predicts the experimental results. Since the same set of boundary conditions are used from Serpentine Heater Configuration, the validity of the entire simulation exercise has been confirmed. The model for Polysilicon Heater Configuration will be used for further optimization.
Chapter 4: Alternative Designs

4.1 Introduction:

Engineering is about minimizing the losses and improving the efficiency of the system. Hence, optimization plays a very important role in the design process. Influence of all the parameters have to be evaluated when performing an optimization exercise. These parameters may include material properties, fabrication sequence, device packaging, cost etc. The optimization study identifies the influence of each parameter on the performance of the system. The influence of individual parameter on the performance of the system could be evaluated using experimental techniques as well. However experimental methods are costly and time consuming. Finite Element Analysis software is a big aid in performing this exercise. A number of different systems can be simulated in short time. Optimization is especially important in the field of MEMS, where it is often necessary to reduce power consumption, weight, and cost. Coupled field Finite Element Analysis (FEA) software packages are widely used for simulations to investigate feasibility [23]. This chapter presents the results of the thermal management study conducted on the Polyimide V-groove actuator using a trial and error method.

4.3 Outline of the chapter:

This chapter deals describes the alternative designs proposed to improve the performance of the Polyimide V-groove actuator. In the preliminary simulations, an emphasis was placed on achieving Maximum Deflection and Recovery (where the leg returns to its initial configuration). The same model has been used for this thermal management exercise and further optimization. Following four alternate designs were proposed and simulated at various frequencies. The performance of each alternative design was compared with that of T. Ebefors’ design (Figure 3.13)

1. Fin Type V-groove Configuration
2. V-grooves with Varying Diaphragm Thickness
3. V-grooves with Uniform Diaphragm Thickness
4. Placement of the electrical heater

4.3 Definitions

Various new parameters have been used in this chapter. This section deals with defining those parameters. All the definitions have been explained with the context of Figure 4.1, which is the plot of deflection at the tip of the actuator in the lateral direction over time for one cycle.

- **Heating Period**: It is the period of the (power) input cycle, for which the heat is supplied (by applying voltage across the heater). Heating Period is normally

![Diagram](image)

**Figure 4.1**: Plot of Deflection versus Time comparing the response of existing design with the proposed design for one heat cycle. This plot will be used to explain various parameters used in the study.
expressed in terms of “Duty Cycle”. Duty Cycle is the fraction of the total cycle time for which required voltage is applied (or zero voltage is applied, for cooling period). Hence, heating period of 50% Duty Cycle means the voltage has been applied for half the total cycle time. In Figure 4.1, the Heating Period is represented by “OH”

- **Cooling Period**: It is the period of the input cycle, for which the no heat is supplied. This could be achieved by applying zero voltage across the electrical heaters. Cooling Period is also expressed in terms of “Duty Cycle”. In Figure 4.1, the Cooling Period is represented by “HK”

- **Maximum Deflection (MD)**: The maximum value of deflection at the tip of the actuator in lateral direction during one input cycle. The Maximum Deflection usually occurs at the end of the Heating Period. From Figure 4.1, for the Existing design plot (solid line), the Maximum Deflection is shown by point “B”. The length of “OB” gives the numerical value of Maximum Deflection for the Existing design. For the Proposed design, the same is shown by point “A”.

- **Recovery (R)**: There are different definitions available for recovery in the literature. For this study the Recovery is defined as, the value of the deflection at the tip of the actuator (in the lateral direction) at the end of the Cooling Period. In other words, Recovery is the amount of residual deflection at the end of the Cooling Periods. For the efficient performance of the actuator, the lower Recovery values will be preferred. Point “D” on Figure 4.1 represents the recovery for the Existing design. Similarly, E₁ and E₂ are the Recovery values for proposed design.

- **Deflection Gain**: It is the increase (or decrease) in the Maximum Deflection value of the Proposed design over the Existing design. The following formula will be used for calculating the Deflection Gain:
Deflection Gain = (MD)\text{Proposed} - (MD)\text{Existing} \quad (4.1)

For this optimization study, the Polysilicon Heater Configuration has been used as an existing design (Figure 3.13). For calculating the Deflection Gain values for different actuator designs, the simulated Maximum Deflection value from Figure 3.13 will be used. From Figure 4.1, the Deflection Gain for the Proposed design will be given by (A - B) and is shown by letter “C”. A positive value of the Deflection Gain represents performance improvement of the actuator.

Recovery Gain: It is the increase (or decrease) in the Recovery value of the Proposed design over the Existing design. It will be calculated by subtracting the Recovery value of the existing design from that of a proposed device. The following formula will be used for calculating the Recovery Gain:

\[ \text{Recovery Gain} = (R)\text{Proposed} - (R)\text{Existing} \quad (4.2) \]

Since one of the objectives of this study is to improve the cooling of the actuator, lower Recovery values will be desirable. A closer look at the Equation 4.2 will show that a performance improvement will be represented by a negative value of Recovery Gain.

For example, from Figure 4.1, points E_1 and E_2 represent the Recovery of the Proposed design. If the recovery of the proposed design is E_1, then the Recovery Gain will be given by F_1 (= E_1 - D), which will be a positive value. Here, the residual deflection of the proposed design (E_1) is greater than the existing design (D). Hence, from performance improvement perspective, positive values of the Recovery Gains are not desirable.

On the other hand, if the recovery of the proposed design is E_2, then the Recovery Gain will F_2 (= E_2 - D), which will be a negative value. Here, the residual deflection of the proposed design (E_2) is smaller than the existing design (D). Hence, from performance improvement perspective, negative
values of the Recovery Gains are highly desirable. The intention of calculating the Recovery Gain in this form is that Deflection Gain and Recovery Gain for the same configuration will be plotted on the same graph (see Figures 4.10 and 4.12).

- **Net Gain**: Overall improvement in performance of the proposed design over the existing design (Polysilicon Heater Configuration) will be referred as Net Gain. Improvement in the Maximum Deflection will be represented by a positive value of the Deflection Gain and an improvement in the Recovery will be represented by a negative value of the Recovery Gain. Hence, the Net Gain of the proposed design over the Plane Wall V-groove configuration will be calculated using following formula:

\[
\text{Net Gain} = \text{Deflection Gain} - \text{Recovery Gain.}
\]  

From Figure 4.1, the Net Gain will be given by \((C - F_1)\) or \((C + F_2)\) depending on the response of the proposed design. Graphically, this will be the length of the line joining the Maximum Deflection and Recovery points for a particular configuration parameter. This has been explained in more detail in later sections.

- **Relative expansion of a V-groove**: The difference between the lengths of the top and bottom edges of a thermally expanded V-groove, shown in Figure 4.2. Here, the configuration of the V-groove before expansion is shown with dotted lines, and after expansion, with solid lines. Due to the thermal expansion, the bottom edge has increased from \(A\) to \(A'\).
and the top edge has increased from B to B'. The relative expansion of the V-groove will then be given by the following formula:

\[
Relative \ Expansion = B' - A'
\] (4.4)

4.4 Fin type Configuration:

Thermal conductivity of the polyimide is very low (0.16 W/m °K)[18] compared to the thermal conductivity of the silicon (140 W/m °K) [13]. Because of such a high difference in the thermal conductivity values of these two materials, most of the heat is conducted into the silicon substrate. Heat dissipation into the polyimide V-grooves takes place through the two inclined sidewalls of V-grooves. Increasing the area of contact between the polyimide and the silicon the heat dissipation could be increase. Figure 4.3 shows the

![Figure 4.3: Schematic sketch showing the proposed arrangement of Fin type V-groove configuration](image-url)
schematic of a Fin Type Configuration proposed to perform this action. The triangular projections of silicon will act as fins, dissipating more heat in the polyimide, thus causing a higher temperature rise. While proposing this design, the basic dimensions of the V-groove are kept the same (Figure 4.3). According to Figure 4.3, \( \delta_r \) is the depth of the silicon fin into the V-groove, and \( L_{r2} \) is the center to center-to-center distance between the two fins. \( L_{r1} \) is the distance of the first fin from the edge of the V-groove and will be governed by the width of the aluminum contacts. To obtain precise information about the working of this configuration, a 3D model would be required. In order to perform a feasibility study, a 2D version of this configuration was generated.

Some modifications were necessary for generating a two dimensional model. Figure 4.4 shows the two-dimensional model of this configuration generated in ANSYS 5.6. Note that the triangular projections were modeled inside the V-groove in a perpendicular plane of the paper. In actual design these fins protrude into the plane of V-groove. The depth of the projections was taken as 5 \( \mu \text{m} \). This value was selected arbitrarily for the feasibility exercise. Fins were equally spaced. Equation would be derived to backtrack the actual values of \( \delta \).
L₁, L₂ if the Fin Type Configuration shows improvement over the Polysilicon Heater Configuration. This geometric modification in the model will accommodate two effects of the Fin Type Configuration. First, additional heat will be supplied into the polyimide due to more contact area. Secondly, the mass of the polyimide is reduced in the proper proportion. These modifications simulate the actual configuration in Figure 4.3. The model was simulated for a range of frequencies, and the simulation results were compared with T. Ebefores’ design (Polysilicon Heater Configuration). Figure 4.5 displays the simulation results for the Fin Type and Polysilicon Heater Type configuration (referred in this section as Plane Wall Type Configuration) over the frequency range chosen.

It is noteworthy that the Maximum Deflection for the Fin Type Configuration is reduced with respect to the Plane Wall Configuration. Hence the Deflection Gain is negative, and so there is no performance improvement.

Figure 4.5: Comparison of Maximum Deflection of Fin type configuration with Plane Wall V-groove configuration for different supply frequencies. Supply power is kept same at 90 mW.
In order to calculate the Net Gain, it is important to calculate the Recovery Gain value. For the input frequency of 40Hz, the Deflection Gain will be

\[ \text{Deflection Gain} = MD_{\text{proposed}} - MD_{\text{Existing}} = 11.2 - 12.3 = -1.1 \mu m \] (4.4)

Recovery is an important issue for thermal actuators. Although at lower frequencies (< 10 Hz), the Recovery is lower than 3% of the Maximum Deflection, at higher frequencies, the Recovery of Plane wall V-groove configuration is substantially higher [18]. Hence, one of the important objectives of this study is to reduce the Recovery (residual deflection) at the end of the Cooling Period for higher frequencies. Figure 4.6 shows a comparison of the Deflection versus time relationship for the two configurations over one cycle. The curve with circular dots represents the simulated results for the Plane wall V-groove Configuration and the curve with squares represents the fin type V-groove configuration.

![Figure 4.6: Deflection Vs Time plot comparing the performance of Fin type V-groove configuration with Plane wall V-groove configuration for a single cycle. (Supplied power = 90 mW, supply frequency = 40 Hz)](image-url)

- 56 -
configuration. Note that the Recovery in the Fin Type Configuration is better than the Plane Wall Configuration. Hence the Recovery Gain is

\[ \text{Recovery Gain} = (R)_{\text{Proposed}} - (R)_{\text{Existing}} = 4.4 - 5.1 = -0.7 \, \mu m \quad (4.5) \]

Now the net gain for fin type configuration at 40 Hz is

\[ \text{Net Gain} = \text{Deflection Gain} - \text{Recovery Gain} = -1.1 - (-0.7) = -0.4 \quad (4.6) \]

According to Equation 4.6, there is no improvement in the performance with the Fin Type Configuration over Plane Wall Configuration (T. Ebefors’ design).

Fabrication of the Fin Type Configuration will be very complicated. Due to the non-uniform cross section of the V-groove, complex stresses are likely to get generated during the curing process. The corners of the fins will act as stress concentration points, adding the severity to the stresses. Considering the complexity involved in fabrication of the Fin Type Configuration, this case will not be developed further. For succeeding simulations discussed in this study, the supplied power was kept constant (90 mW) and the simulations were preformed at 40 Hz frequency.

### 4.5 V-grooves with varying silicon diaphragm at the bottom of the V-grooves:

The lateral deflection at the tip of the actuator is a function of relative lateral expansion between the top and bottom edge of the Polyimide V-groove. From a closer look at formula of the Relative Expansion (Section 4.3), it can be observed that the expansion of the bottom edge of the V-groove reduces this Relative Expansion. Therefore, by fixing the bottom of the V-Groove, the Relative Expansion could be improved. This can be easily accomplished by leaving a thin silicon diaphragm at the bottom of V-groove, as shown in Figure 4.7. Here, a thin diaphragm was left at the bottom of the center two V-grooves. This is possible by
Figure 4.7: FEM model showing schematic representation of V-grooves with varying Silicon diaphragm at the bottom of the device. (All dimensions are in \( \mu m \))

limiting the etching process inside these two center V-grooves. This modification will not change the overall dimensions of the actuator (for other dimensions, refer Figure 3.12). By not providing diaphragms at the two extreme V-grooves, the intent was to trap the heat in the center portion of the actuator. This arrangement should develop more temperature change in the middle two V-grooves. Due to the thin silicon diaphragm at the bottom of the central V-groove, there would be an increase in the Relative Expansion. Figure 4.8 shows the simulated results for Maximum Deflection versus varying diaphragm thickness values. The straight-line represents the Maximum Deflection value for Plane Wall V-grooves Configuration (or Polysilicon Heater Configuration or T. Ebefors’ Design, Figure 3.13) with no diaphragms, as shown in Figure 3.12 (It is important to note that the curve for plane wall V-groove configuration is shown for comparison purposes only, and has no actual dependence on diaphragm thickness values).
Figure 4.8: Comparison of Maximum Deflection for different diaphragm thickness values with the Plane Wall V-groove configuration (for 90 mW/leg at 40 Hz)

Figure 4.9: Comparison of Recovery for different diaphragm thickness values with the Plane Wall V-groove configuration (for 90 mW/leg at 40 Hz)
Note from Figure 4.8 that the Maximum Deflection initially increases with diaphragm thickness. Deflection Gain peaks at 5 µm diaphragm thickness, and then it decreases continuously. For the actuator with a thin diaphragm thickness of 3 µm, the Maximum Deflection at the end of heating cycle is about 11.9 µm. For the Plane Wall V-groove Configuration the simulated value of Maximum Deflection at the end of heating cycle is about 12.4 µm. Hence, for the diaphragm thickness of 3 µm, there is loss in the Maximum Deflection value. In other words, the Deflection Gain is -0.5 µm (from Equation 4.1). Note that the largest Maximum Deflection is obtained for a diaphragm thickness of 5 µm, and the Maximum Deflection for this thickness is about 12.9 µm. Hence, the Deflection Gain for 5 µm thick diaphragm is +0.5 µm over the Plane Wall V-groove Configuration in Figure 3.12.

Figure 4.9 shows a similar comparison of Recovery for the actuator at different diaphragm thickness. Again, the straight line represents the Recovery Value for a Plane Wall V-groove Configuration. Note that for a diaphragm thickness of 3 µm, the Recovery of the proposed device is about 3.5 µm as compared to the Recovery of 4 µm for the Plane Wall V-grooves Configuration. Hence, the Recovery Gain is -0.5 µm for this case (from Equation 4.2). For a diaphragm thickness of 5 µm, the final Recovery of the device is 3.7µm. Hence, the Recovery Gain is about -0.3 µm. Note that the Recovery Gain is reducing as the diaphragm thickness is increasing in the range of 3 µm to 5 µm. However, after attaining a peak at 5 µm, the curve falls continuously, thereby increasing the Recovery Gain. In the next step the Gain values (Deflection Gain and Recovery Gain) were plotted on the same graph. Figure 4.10 shows the plot of Net Gain vs. diaphragm thickness. Recall that Net Gain is computed by subtracting Recovery Gain from Deflection Gain (Equation 4.3). Hence, for a diaphragm thickness of 5 µm, the Net Gain is (0.5- (-0.3) =) 0.8 µm. According to the discussion in the definitions Section (4.3), Net Gain will be the length of the line joining the Deflection Gain and Recovery Gain points for a given diaphragm thickness. Hence, for a 5 µm thick diaphragm, the length of the line ‘A’ in Figure 4.10 represents the Net Gain.
Note also from Figure 4.10 that the two curves intersect each other at point B and C. In the region between these two points, the numerical value of the Deflection Gain exceeds that of Recovery Gain. Hence, in this region, there will be improvement in the performance of the proposed actuator. In other words, for diaphragm thickness between 3 μm to 7 μm, the performance of the proposed actuator will be better than the Plane wall V-groove configuration (shown in Figure 3.12). Beyond point C, the numerical value of the Deflection Gain is less than that of the Recovery Gain. Hence, the performance of the proposed design will be poor as compared to the Plane Wall V-groove configuration. For a 10 μm diaphragm thickness (represented by D in Figure 4.10), the Deflection Gain is –2.5 μm, and Recovery Gain is about –1.4 μm. Hence, the Net Gain is (-2.5-(-1.4)=) –0.9 μm.
4.6 V-grooves with uniform silicon diaphragm at the bottom:

Different diaphragm thickness at the bottom of the V-grooves would require a two-step etching process, which adds complexity. By incorporating uniform diaphragm thickness for all V-grooves, the etching process can be simplified. Figure 4.11 shows the schematic representation of the proposed design with uniform diaphragm at the bottom of the V-grooves. This case was also simulated for varying diaphragm thickness.

![Image of V-grooves with uniform silicon diaphragm](image)

Figure 4.11: FEM model showing schematic representation of V-grooves with uniform Silicon diaphragm at the bottom of the device.

Because of the heat conduction path provided at the two end V-Grooves, more heat would dissipate during the Cooling Period, providing more Recovery. However, less heat is likely to be trapped during the Heating Period. This should not be a problem during the Heating Period, as the Relative Expansion has increased since, the expansion at the bottom of all the V-grooves being curtailed. Figure 4.12 shows curves for Deflection Gain and Recovery Gain versus diaphragm thickness, plotted in a manner similar to Figure 4.10. Compared to the Different Silicon diaphragm design (Section 4.5) the Deflection Gain shows a substantial increase (1.1 μm) for a 1 μm diaphragm thickness. But for this diaphragm thickness, there is no Recovery Gain over the Plane wall V-groove configuration. Hence, the Net Gain will be 1.1 μm. However, for a diaphragm thickness range between 2 μm and 3 μm, the device shows a Recovery Gain of about -0.45 μm, along with a Deflection Gain of about +1.0 μm. Hence, for the V-grooves with uniform silicon diaphragm thickness, the maximum Net Gain is obtained for a diaphragm thickness range of 2-3 μm. In order to determine
optimize value of the silicon diaphragm thickness, an optimization study was conducted on this design, which is presented in the next chapter.

The V-groove configuration with uniform diaphragm thickness (Figure 4.11) shows better performance as compared to the varying diaphragm thickness configuration (Figure 4.7). First, note the improvement in the maximum Deflection Gain in Figure 4.12 is 1.1 μm, compared to 0.5 μm in Figure 4.10. This improvement can be attributed to the increased relative expansion for all V-grooves as a result of the bottom expansion being curtailed. Secondly, the Recovery cycle in Figure 4.12 has improved substantially. For a diaphragm thickness beyond 3 μm, Figure 4.11 shows higher Recovery over Figure 4.10. For a thickness of 4 μm, the Recovery Gain in Figure 4.12 is 0.8 μm compared to 0.4 μm in Figure 4.10. This improvement can be attributed to the improved heat conduction path provided by the thin diaphragm at the bottom of the device.
4.7 Effect of the heater placement:

In all the previous simulations, heat generation rate at all the heaters was assumed to be uniform. The heat generation rate was calculated using following formula.

\[
q_g = \frac{InputPower}{V_{heater} \times N}
\]

Where \( V_{heater} \) = Volume of each heater  
\( N \) = Number of V-grooves.

Polyimide V-groove actuators use polysilicon as a heater material. In the T. Ebefors’ design, the dimensions of all heaters were kept constant [3]. This ensured that, for a given supply voltage, all heaters would generate the same amount of heat. In the micro-fabrication process, it is possible to deposit heaters with different dimensions without any difficulty. Resistance of a heater is given by

\[
R = \rho \frac{l}{w \times t} = \Omega \frac{l}{w}
\]

Where \( \Omega = \rho / t \) = Sheet resistance of the film (ohms/square)  
\( \rho \) = Resistivity of the material  
\( t_h \times w_h \times l_h \) = Thickness x Width x Length of the resistor.

For a given film thickness (here, LPCVD deposited polysilicon) the sheet resistance is constant. By changing the length (l) and/or width (w) the resistance of the heater can be changed. For the present design of the Polyimide V-groove actuator actuator, the change of the length of the heater would cause non-uniform heating across V-grooves. Hence, the option of changing the length of the heater is not feasible. However, by changing the width of the heater the resistance of the heater can changed. Simulations were performed on a model of V-groove with uniform diaphragm configuration for different heater locations. The
Figure 4.13: Schematic diagram of the actuator design used for understanding the effect of heater placement. For rest of the dimensions, refer Figure 3.12 and 4.10.

thickness of the diaphragm for these simulations was taken as 3 μm. Figure 4.13 shows the nomenclature used to identify heater positions (Refer Figure 3.12 and 4.11 to obtain the overall dimensions of the actuator). The heater position to the left wall is # 1, and other heaters are numbered sequentially from left to right. To identify the role of each heater on the output of the actuator, total power was supplied through a single heater location. In actual device, this would be accomplished by providing a heater element at only one of the positions shown in Figure 4.13 (either 1 or 2 or 3). The transient output in each case was compared with the transient output of the actuator (shown in Figure 4.13), with equal power dissipated through all heaters.

Figure 4.14 demonstrates the effect of the different heater location on the output of the actuator. In Figure 4.13, (0 0 1) represents that the total power has been dissipated through heater # 3, and (1 0 0) represents the total power being dissipated through heater # 1 and so on. Comparing the effect of heater placement with that of uniform heating, the following observations can be made.

- By supplying total power through heater # 3 (shown by points D_1 and R_1 on Figure 4.14), a higher Maximum Deflection can be obtained as compared to uniform heating (shown by points D_3 and R_3 on Figure 4.14), (i.e. D_1 > D_3). Hence, Deflection gain is positive (From Eq 4.1). However, there is loss in the
Figure 4.14: Effect of placement of heater on the performance of the actuator.

Recovery of the actuator (i.e. $R_1 > R_3$). Hence, the Recovery gain is also positive (Eq 4.2). It is clear from Figure 4.13 that the magnitude of Deflection Gain and the Recovery Gain is almost same. Hence, this configuration will have zero Net Gain (Eq 4.3). This behavior could be explained with the help of following reasoning. Since the heater is away from the left wall (which is acting as heat sink), more heat would be trapped into the system. This results in higher the Maximum Deflection. V-grooves on the left side, act as an insulator. During cooling period, heat could either escape through the thin diaphragm, which has higher heat resistance. Instead, heat flows into the beam on the left of the V-grooves. Since convection plays a very little role in the heat transfer process (Section 3.5), because of the improper cooling mechanism, there is increase in the recovery value.

- When entire heat is supplied through heater # 1 (shown by points D$_1$ and R$_4$ on Figure 4.14), there is loss of Maximum Deflection (i.e. $D_4 < D_3$). However, there is improvement in the Recovery of the actuator (i.e. $R_4 < R_3$). From
Figure 4.13, the values of Deflection Gain and the Recovery Gain are almost same. Hence, there will be zero Net Gain for <0 1 0> heater placement. Since the heater is near to the left wall (heat sink), more heat is being escaped through the thin diaphragm into the left wall. (There is only one thin diaphragm on the left of the heat as compared to two on the right). This explains the decrease in Maximum Deflection and improvement in the Recovery of the actuator.

- When entire heat is supplied through heater # 2 (shown by points D_2 and R_2 on Figure 4.14), there is very small increase in the Maximum Deflection (D_2 > D_3). However, as compared to the Deflection Gain in case of <0 0 1>, the Deflection gain for <0 1 0> is negligible. From Figure 4.13, there is no change in the Recovery (R_2 = R_3). Due to the symmetric heating in the case of <0 1 0>, the final output pattern matches with that of the uniform heating case.

None of the above cases shows actual improvement in the performance of the actuator. Also, none of the above cases are practical, since they contain only one heater per actuator. Hence, a non-working heater would mean a non-working actuator (leg). It can be concluded from the above analysis that the performance of the Polyimide V-groove actuator does not depend on the placement of the heater.
Chapter 5: Optimization and Force analysis.

5.1 Introduction:

Optimization of a system using trial and error method works effectively only if single parameter is involved in the study. For the systems where two or more parameters are involved, the optimization is practiced through simulation software programs. Many special purpose software packages are available today to optimize a multi disciplinary problem. An optimization module available in ANSYS will be used in this chapter to confirm the results from previous chapter. In the previous chapter, out of the three proposed designs, "V-grooves with uniform diaphragm at the bottom of the V-grooves" appears to have the best performance so far. When the simulations were performed, it was observed that the Net Gain was highest for the diaphragm thickness ranging from 2 \( \mu \text{m} \) to 3 \( \mu \text{m} \). In this chapter, the optimization module available in ANSYS has been used to estimate the optimal value of the diaphragm thickness. During all the previous simulations for determining optimal thickness range, the induced stresses have completely been ignored. While using the optimization module available in ANSYS the maximum stress will be one of the constraints. One of the important parameters for any actuator is the amount of force developed by that actuator. After determining the optimal design, a steady state analysis will be performed to obtain the relation between the force developed and the displacement of the actuator.

5.2 Outline of the chapter:

The optimization of the Polyimide V-groove actuator with uniform diaphragm at the bottom configuration consisted of following steps:

- Find the optimal value of diaphragm thickness to "maximize the deflection" of the actuator using ANSYS optimization module. To save time computational time, this exercise will be performed with a steady state model.
• Two optimization exercises were conducted. The first exercise was aimed towards optimizing the thickness of silicon diaphragm, and confirming the validity of the Section 4.6. In the second optimization exercise, all dimensions of the V-groove were optimized to maximize the deflection.

• Confirm the maximum stresses developed in the actuator for the optimal diaphragm thickness. These stresses have to be less than the ultimate strength of the weakest material of the actuator, i.e. polyimide.

• Compare the transient response of the optimal design with that of T. Ebefors’ design.

• Perform steady state analysis to estimate the maximum force developed by the optimized design and compare it with that of T. Ebefors’ design.

• A second optimization exercise was conducted to optimize the overall dimensions of the V-groove to maximize the deflection.

5.3 Design optimization:

Design optimization is the process of creation or modification of the design to make it as effective as possible. Hence, the optimum design is the best design, which meets all the specified requirements of the designer and minimize (or maximize) the key objective function. (e.g. the key objective function in this optimization exercise is the maximum deflection of the actuator) an optimization process demands large computational resources to handle the nonlinear problems associated with it. In the early years of digital computers, the optimization process was very costly and time consuming. As a result of these constraints, most of the products were designed without being optimized. Tremendous developments in inexpensive computational resources in recent years have helped in the utilization of optimization techniques for product development [24]. Optimization is especially important in the field of MEMS, where it is often necessary to reduce power consumption, weight, and cost.

ANSYS 5.6 contains an optimization module (referred as “/OPT”) that can be employed to determine the optimum design for coupled field elements. Most of the theory presented in this chapter is picked up from [22]. The ANSYS
program offers two optimization methods to address the optimization problem, “Subproblem Approximation” and “First Order” method. Details about the methods are explained in detail in Section 5.6. For both the “Subproblem Approximation” and “First Order” method, the program performs a series of analysis-evaluation-modification cycles (optimization loops). That is, an analysis of the initial design is performed, the results are evaluated against specified design criteria, and the design is modified as necessary. This process is repeated until all specified criteria are met [22]. For the optimization exercise presented this chapter, “Subproblem Approximation” has been used. In the next section, terminologies used in defining optimization problem will be explained.

5.4 Optimization terminology:

This section will introduce the formal terminology being used to define an optimization problem.

- **Design Variables**: These are the independent parameters that identify a particular design (e.g. length, width). During the optimization process, these parameters will be change over a prescribed range. Upper and lower limits have to be specified by the designer, which serve as "Design Variable constraints". These limits define the range of variation for the Design Variable. In present problem. For Polyimide V-groove actuator, the thickness of the diaphragm and the dimensions of the V-groove (top and bottom edges, depth, etc) were used as Design Variables. Lower constraints were no defined on the Design Variables used in this chapter. However, the dimensions of the T. Ebefors’ Design were used as upper constraints on all the Design Variables. This insured that the optimum design is always smaller than the T. Ebefors’ design.

- **State Variable**: These are the parameters that set the design constraints. These parameters are dependent on the Design Variables and are also defined in terms of upper and lower limits, known as "State Variable
constraints”. State Variable have no direct role to play in determining the optimum design. However, they are evaluated using Design variables and satisfaction of the constraints establishes the validity of the design. In the case of Polyimide V-groove actuator, maximum stress and maximum temperature were used as State Variables.

- **Objective Function**: This is a function (quantity), which is explicitly or implicitly depends on Design Variables. The value of the Objective Function should change by changing the values of the Design Variables. The aim of the optimization exercise is to minimize (or maximize) the Objective Function by changing the values of Design Variables, under the constraints imposed by State Variables. The optimization module of the ANSYS allows defining only one Objective Function. For the Polyimide V-groove actuator, the Maximum Deflection was used as an Objective Function.

- **Design Set**: A set of parameters including Design Variables and State variables that represent a particular model configuration (or a particular design). These parameters are changed with the iterations performed during optimization process (also called as “optimization loop”) to obtain optimized design. Depending on the State Variable constraints, design sets are divided into three categories as explained bellow.

- **Feasible Design**: This is the design (or design set) that satisfies all specified constraints on State Variables as well as constraints on the Design Variables. While performing optimization iterations, ANSYS changes the values of the Design Variable (Design Sets). Depending on the effect of this change on the value of Objective Function, the next Design set is chosen. While performing these iterations, there may be number of Design Sets generated by ANSYS that satisfy all the constraints. These Design Sets are called as Feasible Design. Although only one of them is an optimum design, these feasible sets
could be important in understanding the performance of the system under different conditions.

- **Infeasible Design**: This is the design (or design set) that violates one or more constraints set by either State Variables constraints or Design Variables constraints. When the result of optimization iterations is displayed in ANSYS, the violated parameters from the Infeasible Design sets are shown with “>” sign before the parameter.

- **Best Design**: This is the Design set that satisfies all constraints and produces the minimum objective function value. If all Design Sets are infeasible, the best design set is the one closest to being feasible, irrespective of its objective function value. When the result of optimization iterations is displayed, a “ * ” symbol is used to indicate the Best Design set.

### 5.7 Method of Optimization using ANSYS:

The following process sequence was followed for optimizing the Polyimide V-groove actuator (with uniform diaphragm at the bottom) for maximum deflection.

- A model of “Polyimide V-groove actuator with uniform diaphragm at the bottom” was build using the dimensions shown in Figure 3.12 and 4.11. Diaphragm thickness was selected as 3 μm for the first iteration. Design Variables were defined in terms of parameters (E.g. parameter “D” was assigned for diaphragm thickness, etc)

- The model was simulated to obtain the steady state solution. Since optimization performs many iterations, steady state analysis was chosen (instead of transient solution) to save the time. The input power was selected as 90 mW per actuator.

- Parameters to be used as State Variable and Objective Functions were assigned with appropriate quantities. (E.g. parameter “S” was assigned to
the maximum Von Misses stress and "TMAX" was assigned to the maximum temperature

- A "log file" was created for the above model. A "log file" typically contains a log (record) of all the commands used in constructing and simulating the model. The "log file" was edited, i.e. unnecessary command s were deleted (E.g. command for zooming on a particular area of model is an unnecessary command for ANSYS, etc). Editing of the log file is required in order to make it suitable for optimization looping.

- The optimization module was started and the "log file" created in the previous step was assigned as the "analysis file" for optimization looping. The "analysis file" will be analyzed by ANSYS to obtain optimized solution.

- Optimization variables were assigned. For the first optimization exercise, thickness of the diaphragm was assigned as the only Design Variable. In the second exercise, all the dimensions of the V-groove were assigned as Design Variables (details in Section 5.11). For both the attempts, maximum Von Misses stress and maximum temperature were assigned as State Variables, and the deflection at the tip of the actuator was assigned as an Objective Function.

- Sub-problem Approximation Method was used for performing optimization loop. Details about the selection of optimization method are given in next section.

- Optimization loop was executed and the resulting Design Sets were reviewed. Further force analysis was performed on the optimized design.

5.7 Selection of Method of optimization:

Two methods are available in ANSYS to solve optimization problem. These two methods have been briefly discussed in this section [22].

- **Subproblem Approximation Method**: This is an advanced zero-order method, which uses approximations (curve fitting) for optimization. The term zero-order means it requires only the values of the dependent variables, and not
their derivatives. It is a commonly used method that can be applied efficiently to wide range of engineering problems. By calculating the Objective Function for several sets of Design Variables, a relationship is established between the Objective Function and Design Variables (by a least squares fit between the data points). The resulting curve is called an approximation and hence the name. With every optimization loop a new data point is generated thus updating the Objective Function approximation. Now this approximation is minimized to obtain the optimum design. This method is fast and can be applied to variety of engineering problems.

Since the approximations are used for determining the objective function, the optimum design will be only as good as the approximations itself. One technique to overcome this shortcoming is by running the Subproblem Approximation method with different initial designs (Design Sets) and verifying the optimal design.

- **First Order Method**: This method uses derivative information, i.e. gradients or rate of change of the dependent variables with respect to the Design Variables. For every iteration, gradient calculations are performed in order to determine a search direction, and a line search strategy is adopted to minimize the unconstrained problem. It is highly accurate and works well for problems having dependent variables that vary widely over a large range of design space. However, this method can be computationally intense.

For the optimization of the Polyimide V-groove actuator, the Subproblem Approximation Method was used, since it is faster. To avoid the errors mentioned above, each optimization loop was run twice (i.e. for two different initial design parameters) and the results were crosschecked.
5.7 Optimization for Maximum Deflection with thickness of the silicon diaphragm as the only Design Variable.

The objective of the first optimization exercise was to estimate the optimal thickness of the silicon diaphragm at the bottom of the V-groove (Figure 4.11) in order to maximize the deflection at the tip of the actuator. The optimization module of the ANSYS is capable of handling multiple Design Variables at the same time. However, in this exercise, only one Design Variable has been used. For optimization problems with single Design Variable, the method used in Section 4.5 and 4.6 (trial and error) would prove equally effective. However, ANSYS was used for this exercise for following reasons,

- Optimization using ANSYS would give an exact value of optimal diaphragm thickness. This will be useful for targeting etch time and other process parameters during fabrication of the actuator.
- In an earlier simulation (Section 4.5 and 4.6), the maximum stress induced in the actuator was not taken into consideration. For very thin silicon diaphragm thickness, the maximum stress may exceed the safe stress limit. As the maximum induced stress is one of the State Variables, the optimal design (from ANSYS optimization module) will be safe.
- By performing optimization with a single variable, the author got chance to get himself familiar with the optimization module of ANSYS.

Since the optimization module of ANSYS performs a number of iterations, steady state analysis was chosen (instead of transient solution) to save the simulation time. Two optimization loops were performed. In the first loop, the initial diaphragm thickness was set at 4 \( \mu \text{m} \) and for the second loop, it was set at 6 \( \mu \text{m} \). the following parameters were defined with the appropriate constraints:
Design Variable:

Diaphragm Thickness (D): The diaphragm thickness was the only Design Variable for the first optimization exercise. From Figure 4.11, Deflection Gain (for transient analysis) was highest in the diaphragm thickness range of 2 \( \mu \text{m} \) to 3 \( \mu \text{m} \). Beyond this range, there was a reduction in the Deflection gain values with the increase in diaphragm thickness. To observe the effect on the deflection of actuator over a wider range of diaphragm thickness values, the upper constraint on the Design Variable was set at 10 \( \mu \text{m} \).

State Variable:

Stress (S): When transient simulations were performed on the Polyimide V-groove actuator (in the previous chapter), the induced stresses were totally ignored. Due to difference in the thermal expansion coefficient values of polyimide (30 ppm/ °C) and silicon (2.33 ppm/ °C), heavy stresses are likely to get developed at their junction. To limit the maximum stress, the maximum Von Misses stress induced in the actuator was assigned as a State Variable. Polyimide is the weakest material in this actuator. The tensile strength at the break (ultimate strength) of the polyimide is 260 MPa (Table 2.1: Physical properties of Durimide HTR-3). Hence, the upper constraint on the stress was set at 250 MPa.

Temperature (TMAX): Although, Polyimide belongs to the family of high temperature polymers; it burns beyond 400 °C. This temperature is critical in the design of a Polyimide V-groove actuator. With the change in the dimensions of the actuator, the maximum temperature is likely to change. Hence, the maximum temperature was set as State Variable. The upper constraint on the temperature was set at 400 °C.
**Objective Function:**

Deflection (Y): The optimization of the actuator is always related to reducing the input power (or improving the output power). A direct optimization for reducing the input power is not possible in ANSYS. The output power of the actuator is measured in terms of two parameters, the deflection or force developed. The output force depends on the application of the actuator. In the previous chapter (# 4), the optimization was obtained in terms of the Maximum Deflection of the actuator. Keeping consistency with the previous analysis, the deflection at the tip of the actuator in the lateral direction was chosen as an Objective Function. For the Polyimide V-groove actuator, the deflection occurs with lowering of the tip of the actuator. Hence, the deflections are obtained in terms of negative values. This is helpful for optimization module in ANSYS, because this module can perform only minimization. Minimization of the negative quantity would mean maximization of the deflection of the actuator.

### 5.8 Result of optimization for thickness of the silicon diaphragm

The optimization loop was executed two times, first with an initial diaphragm thickness of 4 μm and second with an initial diaphragm thickness of 6 μm. ANSYS performed 7 iterations the first time and 8 iterations the second time during optimization loop. Table 5.1 (a) and (b) shows gives the list of iterations performed with initial thickness of 6 μm and 4 μm respectively. At the end of the first loop (initial thickness of 6 μm), the optimum design consists of the diaphragm thickness of approximately 2.2 μm (Table 5.1 (a)). Similar results were obtained for the second iteration, Table 5.1 (b) (initial thickness of 4 μm). The optimized Design Sets have been shown with yellow color in Table 5.1 (a) and (b). Figure 5.1 shows the plots of diaphragm thickness (D) versus maximum deflection (Y) for the design sets given by two loops. (i.e. Figure 5.1 (a) is a plot of “Y" versus "D" from Table 5.1 (a). Similarly Figure 5.1 (b) is a plot of “Y" versus “D" from Table 5.1 (b).
Table 5.1 (a): Details of iterations performed on Polyimide V-groove actuator with initial diaphragm thickness of 6 \( \mu \text{m} \)

Table 5.1 (b): Details of iterations performed on Polyimide V-groove actuator with initial diaphragm thickness of 4 \( \mu \text{m} \).

From the discussion in Section 5.6 one way to minimize the errors incorporated with the Subproblem Approximation method is by running the optimization loop with different initial designs (Design Sets) and verifying the optimal design. A small error associated with the Subproblem Approximation method can be observed from Table 5.1 (a) and (b). For the initial diaphragm thickness of 6 \( \mu \text{m} \), the optimized value of the diaphragm was given as 2.1956 \( \mu \text{m} \) (Table 5.1 (a)). When the loop was simulated with initial diaphragm thickness of 4 \( \mu \text{m} \), the optimum value of diaphragm was 2.2187 \( \mu \text{m} \) (Table 5.1 (b). There is a difference of 0.0231 \( \mu \text{m} \) between two simulations, which is extremely small and can be
neglected. Hence, it can be said that Figure 5.1 (a) and (b) confirm with each other and the optimization has been validated for minimal error.

Comparing figure 5.1 (a) and (b) with Figure 4.12, it can be observed that the Maximum Deflection versus diaphragm thickness characteristics shows the same trend for steady state as well as for transient solution. This observation further confirms the validity of the optimization exercise. For the optimized design, with the diaphragm thickness of 2.2 μm, the maximum induced Von Misses stress ($\sigma_{\text{max}}$) is 187 MPa. From Table 2.1, the ultimate tensile strength ($\sigma_{\text{ut}}$) of the polyimide is 260 MPa. Hence, the factor of safely (F.O.S.) for the actuator is,

$$F.O.S. = \frac{\sigma_{\text{ut}}}{\sigma_{\text{max}}} = \frac{260}{187} = 1.39$$

When the contour plot of Von Misses stress was analyzed, it was observed that the maximum stress gets developed in the silicon diaphragm underneath the outermost V-groove.

Figure 5.1: Comparison of deflection results given by two optimization loops. [(a): Initial thickness = 6 μm and (b): initial thickness = 4 μm].
5.9 Force developed by the actuator

One of the important parameters in choosing a proper actuator for certain application is the amount of force generated by the actuator. In this section the force developed by the optimized actuator due to thermal actuation, has been estimated using the steady state simulations. The force versus deflection data has been for the optimized actuator has been compared with the T Ebefors' design (Figure 3.12). The amount of force developed depends on the stiffness of the actuator. With the addition of the thin silicon diaphragm the stiffness of the beam has been increased. Hence, the steady state force should also increase, causing overall improvement in the efficiency of the actuator.

The amount of force developed is inversely proportional to the deflection. Force versus deflection characteristics will be constant for a particular design of the actuator. There are two methods available to estimate the amount of force developed. In the first method, the deflection of the actuator is specified. Now the reaction force at the tip of the actuator will be the amount of force developed by the actuator for the specified deflection. This method is difficult to simulate.

In the second method, a force of known magnitude is applied at the tip of the actuator, in the direction opposite to the deflection. Now, the resulting steady state deflection of the actuator (due to the electrical heating) will correspond to this applied force. The system can be simulated for forces of increasing magnitude, until the resulting deflection reduces to zero.

A better alternative would be to obtain the maximum force developed by the actuator first. As mentioned earlier, the amount of force developed is inversely proportional to the deflection. Hence, the actuator will generate maximum force when the deflection at the tip of the actuator is zero. This condition can be simulated in ANSYS by enforcing a boundary condition of “y = 0” at the tip of the actuator. Figure 5.2 shows the schematic of these boundary conditions. In Figure 5.2 (a), the tip of the actuator is fixed in Y direction at point “A”. The actuator was then simulated for the input power of 90 mW using steady state analysis. The reaction in vertical direction at point “A” was the maximum force developed by the actuator. In Figure 5.2 (b) the fixed boundary condition
Figure 5.2: (a) The tip of the actuator is fixed in Y direction. Now the reaction force will be the maximum force developed by the actuator. (b) To obtain force versus displacement characteristics, forces of different quantities were applied at the tip of the actuator.

was removed and a force of known quantity (shown as 1 μN in the Figure) was applied at point “A”. The actuator was again simulated for steady state, and the deflection at the tip of the actuator corresponds to the applied force. The magnitude of force was increased in steps, and same simulation was executed again. Deflection of the actuator under load is given by following formula:

\[ d = d_0(a) + C \times F \]  
(5.2)

Where
- \( d_0 \) = No load deflection of the actuator. This deflection is a function of input parameter “a”.
- \( a \) = Input parameter, in this case, the input power.
- \( C \) = Compliance of the actuator (length per force)
- \( F \) = Applied force.
Figure 5.3: Comparison of T. Ebefors' design with the optimized design for maximum deflection of the actuator under the action of externally applied force.

Compliance is reciprocal of stiffness. Force analysis was conducted on T. Ebefors' design (as explained above) and on the optimized design (with the diaphragm thickness of 2.2 μm) of the Polyimide V-groove actuator. Figure 5.3 shows the comparison of the force analysis for the two designs.

From Figure 5.3, the relation between the deflection of the actuator and the applied force for T. Ebefors' design and for the optimized design is given by the following Equations respectively:

\[ Y_T = -3.483X + 29.085 \]  \hspace{1cm} (5.3)

\[ Y_O = -3.1742X + 30.654 \]  \hspace{1cm} (5.4)
Throughout this thesis report, the deflection of the actuator tip is presented as a positive quantity for the sake of simplicity. Whereas, the actual deflection of the actuator would be a negative quantity, as the tip of the actuator goes down after supplying the power. Figure 5.4 shows the schematic diagram representing deflection of the actuator.

Figure 5.4: Schematic representation of the heating of the actuator. “d” represents the deflection of the actuator.

From Figure 5.4, the dotted line shows the position of the actuator after heating. Distance “d” represents the deflection of the actuator. Hence, the deflection at the tip of the actuator is really a negative quantity. Although throughout this thesis, the deflection is represented in terms of positive quantities (for simplicity), it is important to substitute the deflection values in Equation 5.3 and 5.4 in their actual form. Replacing “X” in Equation 5.3 with the notation “F” (for force”), “Y” with the notation “d” (for deflection) and making appropriate sign changes as explained above, the relation between deflection ($d_T$) and the applied force ($F$) for T. Ebefors' design will be:

$$d_T = C \times F + d_o (a) = 3.484 \times F - 29.085$$  \hspace{1cm} (5.5)

Where, the compliance ($C$) of the T. Ebefors' design is 3.484 $\mu$m/$\mu$N, and the no load deflection ($d_o$) is $-29.085 \ \mu$m. Applying similar treatment to Equation 5.4,
the deflection \((d_0)\) versus applied force \((F)\) relation for the optimized design will be:

\[
d_0s = C \times F + d_o(a) = 3.1742 \times F - 30.654
\]  

(5.6)

Where, the compliance \((C)\) of the optimized design is 3.1742 \(\mu \text{m/\mu N}\), and the no load deflection \((d_{os})\) is \(-30.654 \mu \text{m}\).

The compliance of the optimized actuator has reduced as compared to the T. Ebefors' Design. As stated earlier, compliance is inversely proportional to the stiffness of the actuator. Thus, the reduction in the compliance value indicates that the stiffness of the actuator has increased. In the optimized design, a part of polyimide inside the V-groove has been replaced with silicon. Young's modulus of silicon (190 GPa) is much greater than that of polyimide (3.3 GPa) [13]. Hence, the inclusion of silicon diaphragm has increased the stiffness of the actuator, the result of which can be seen in the increase in the force developed by the optimized actuator. For any cantilever beam, the deflection produced by the application of external force will reduce if the stiffness of the beam is increased. During simulation, the amount of input heat was kept same for T. Ebefors' design and for optimal design. If the inclusion of silicon diaphragm increases the stiffness of the actuator, the maximum deflection of the optimal design should have reduced. From Figure 5.3, the maximum steady state deflection has increased from 29.08 \(\mu \text{m}\) to 30.654 \(\mu \text{m}\) (i.e. deflection corresponds to zero applied force) and the maximum force has increased from 8.35 \(\mu \text{N}\) to 9.56 \(\mu \text{N}\). In order to understand this behavior, the actuating mechanism (expansion of V-grooves) in two cases will have to be understood.

In the case of T. Ebefors' design (Figure 3.12), the deflection at the tip of the actuator is a function of relative expansion of the V-grooves (refer Section 2.3 for more details), which are completely filled with polyimide. Expansion of the bottom edge of the polyimide plays a major role in reducing the relative expansion, and the maximum deflection. With the addition of a thin silicon diaphragm at the bottom of the V-groove, the expansion of the bottom edge of
the V-groove has been curtailed. Hence, it would be appropriate to state that, by adding a thin silicon diaphragm, the actuation mechanism itself has been modified. The relation between the stiffness (or compliance) and the maximum deflection would hold if this relationship is applied to the actuators with the same actuating mechanism only.

5.10 Transient analysis of the optimized design with a single variable:

The optimization exercise was performed on a steady state model of the actuator. Maximum Deflection at the end of steady state simulation was the Objective Function. A similar exercise could be attempted for the transient solution, but the time for completing one optimization loop would be about 8 to 9 hrs. Hence, transient analysis was not recommended for this study. In this section, the optimized model (with the diaphragm thickness of 2.2 µm) was simulated and the results were compared with the simulation results of the T. Ebefors' design. The simulation parameters were be kept same as Section 4.5 and 4.6, i.e. input power = 90 mW per actuator and supply frequency = 40 Hz. From Figure 3.6 and 3.7, model of the actuator is capable of predicting the experimental results fairly close without the wall temperature effect. In addition to that, for the Polysilicon Heater type configuration model was did not show any change in the simulation output with or without wall temperature effects (Figure 3.13). Hence, to speed up the simulations, no wall temperature effects were included. In other words, temperature of the left wall was assumed to remain constant throughout simulation. Figure 5.5 shows the comparison of transient simulations.

From Figure 5.5, the Maximum Deflection of the T. Ebefors' design is represented by $D_T$ (= 12.3 µm), and that of the optimized design is represented by $D_O$ (= 13.28 µm). Similarly $R_T$ (= 4.00 µm) and $R_O$ (= 3.56 µm) represent Recoveries for T. Ebefors' design and optimized design respectively. Hence the Deflection Gain for the optimized design will be (from Equation 4.1):

$$\text{Deflection Gain (DG)} = D_O - D_T = 13.28 - 12.3 = 0.98 \mu m$$ (5.7)
Figure 5.5: Comparison of transient simulation results of T. Ebefors’ Design with the Optimized design. (90-mW/actuator, 40 Hz). Wall temperature effects are neglected.

And the Recovery gain will be calculated from Equation 4.2

\[
Recovery\ Gain\ (RG) = R_O - R_T = 3.56 - 4.00 = -0.46\ \mu m \quad (5.8)
\]

Now the Net Gain of the actuator at 40 Hz will be calculated from Equation 4.3

\[
Net\ Gain = DG - RG = 0.98 - (-0.46) = 1.44\ \mu m \quad (5.9)
\]

Thus, addition of the diaphragm shows an overall improvement in the performance of the actuator. Since there is performance improvement in the Recovery of the actuator (i.e. negative Recovery Gain), thin silicon diaphragm also addresses cooling issues as well. However, a diaphragm alone is not
sufficient to eliminate the need of active cooling mechanism for achieving complete recovery at the end of cooling period.

5.11 Optimization for Maximum Deflection with multiple Design Variables:

The second optimization exercise was aimed towards optimizing the overall dimensions of the V-grooves. Three design variables were defined, i.e. depth of the V-groove, thickness of the diaphragm and width of the diaphragm.

![Diagram of V-groove with labeled dimensions](image)

**Figure 5.6: Schematic of design variables used in optimization for Maximum Deflection with multiple Design Variables**

The aim of this exercise was to obtain the maximum deflection, by reducing the size of the actuator. Hence, the dimensions of the T. Ebefors' design (Figure 3.12) were used, as upper limit constraints on Design Variables. The optimization loop was simulated twice, and the optimal dimensions were obtained.

Thickness of the actuator (H), diaphragm thickness (D) and width of the diaphragm (W) were defined as scalar parameters in ANSYS. The geometry of the actuator was constructed with using these scalar parameters. Figure 5.6 shows the schematic of the V-groove representing the scalar parameters (H, D...
and W) and the key-points (a to f). Key-point “a” was placed at the origin (0,0). The X-coordinate of the key-point “b” was defined by the angle of KOH etching (54.7° [c]) and shrinkage coefficient “ε” (=48 % [A]). Equation for calculating the X-coordinate of is given by

\[ x_b = (H - D) \times \left( \frac{1}{\tan \theta_{KOH}} \right) \times (1 - \varepsilon) = 0.388(H - D) \]  

\[ (5.10) \]

<table>
<thead>
<tr>
<th>KEY POINTS</th>
<th>X COORDINATE</th>
<th>Y COORDINATE</th>
</tr>
</thead>
<tbody>
<tr>
<td>a</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>b</td>
<td>0.388 x (H - D)</td>
<td>- (H - D)</td>
</tr>
<tr>
<td>c</td>
<td>0.388 x (H - D) + W</td>
<td>- (H - D)</td>
</tr>
<tr>
<td>d</td>
<td>2 x 0.388 x (H - D) + W</td>
<td>0</td>
</tr>
<tr>
<td>e</td>
<td>0.388 x (H - D)</td>
<td>- H</td>
</tr>
<tr>
<td>f</td>
<td>0.388 x (H - D) + W</td>
<td>- H</td>
</tr>
</tbody>
</table>

Table 5.2: Coordinates of Key points defining the geometry of V-groove in terms of scalar parameters H, D and W.

Coordinates for the rest of the key-points in terms of H, D and W are given in Table 5.2.

- **Design Variables:**

  Diaphragm Thickness (D): From the optimization results of single Design Variable analysis, the optimal diaphragm thickness was 2.2 μm for the thickness of actuator 30 μm. From this data, it can be said that for optimal performance, the diaphragm thickness should be about 10 % of the thickness of the actuator. Since, the aim of this exercise is to reduce the overall dimensions of the V-
groove, the maximum thickness of the actuator will be 30 μm. The upper constraint on the diaphragm thickness was set at 5 μm.

Thickness of the actuator (H): If other dimensions of the V-grooves are kept the same, and only the thickness of the actuator is changed, then the stiffness of the actuator will be reduced with reduction in thickness. The deflection of the beam is inversely proportional to its stiffness. Hence, the deflection of the actuator will increase by reducing the thickness of the beam. Since the coordinate of the key-point “b” depends on factor “H − D” (Table 5.2) and the thickness of the aluminum contact is 1.5 μm, the lower constraint on the thickness was set at 7 μm. From the discussion made at the start of Section 5.11, the dimensions of the T. Ebefors’ design set the upper constraint. Hence, the upper constraint was set at 30 μm.

Width of the diaphragm (W): If other dimensions of the actuator are kept the same, and only the width of the silicon diaphragm is varied, then the stiffness of the actuator will reduce by increasing the width of the diaphragm. As the width of the diaphragm increases, it adds additional amount of polyimide, which has lower Young’s Modulus as compared to silicon. Hence, to increase the deflection of the actuator, the width of the diaphragm should increase, to the point that the stresses induced in the actuator do not cross ultimate strength limit (250 MPa). Since, the dimensions of the T. Ebefors’ design set the upper constraint, the upper constraint was set at 15.2 μm.

- State Variables:

  Stress (S): Due to difference in the thermal expansion coefficient values of polyimide (30 ppm/ °C) and silicon (2.33 ppm/ °C), heavy stresses are likely to get developed at the junction. To limit this maximum stress, the maximum Von misses stress induced in the actuator was chosen as a State Variable. Polyimide is the weakest material in this actuator. The tensile strength at the break (ultimate
strength) of the polyimide is 260 MPa (Table 2.1: Physical properties of Durimide HTR-3). Hence, the upper constrain on the stress was set at 250 MPa.

Temperature (TMAX): Although, Polyimide belongs to the family of high temperature polymers it burns beyond 400 °C. This is a critical factor in the design on Polyimide V-groove actuator. With the change in the dimensions of the actuator, the maximum temperature is likely to change too. Hence, the maximum temperature was set as State Variable. The upper constraint on the temperature was set at 600 °K.

- **Objective Function:**

  Deflection (Y): Objective function was kept same, i.e. maximizing the maximum deflection. For the Polyimide V-groove actuator, the deflection occurs with lowering of the tip of the actuator. Hence, the deflections are obtained in terms of negative values. This is helpful for optimization module in ANSYS, because this module performs only minimization. Minimization of the negative quantity would mean maximization of the deflection.

### 5.12 Optimal geometry of the V-grooves for maximum deflection:

<table>
<thead>
<tr>
<th></th>
<th>OPTIMAL SET 1</th>
<th>OPTIMAL SET 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>S (MPa)</td>
<td>206.60</td>
<td>240</td>
</tr>
<tr>
<td>TMAX (°K)</td>
<td>365.47</td>
<td>366.82</td>
</tr>
<tr>
<td>D (μm)</td>
<td>0.83788</td>
<td>0.97984</td>
</tr>
<tr>
<td>H (μm)</td>
<td>10.049</td>
<td>9.4759</td>
</tr>
<tr>
<td>W (μm)</td>
<td>15.171</td>
<td>14.835</td>
</tr>
<tr>
<td>Y (μm)</td>
<td>-93.087</td>
<td>-96.637</td>
</tr>
</tbody>
</table>

Table 5.3: Overall optimal geometry of the V-groove for maximum deflection
Since three Design variables were involved, optimization loop made as many as nineteen iterations before giving optimal design. Table 5.3 shows the results (optimal geometries) of these two optimization loops.

5.13 Analysis of the optimal geometry of V-groove for maximum deflection:

The results of two optimization loops are very close and hence the error involved in use of Subproblem Approximation method has been minimized. Values of H, D and W are given up to three decimal places. During micro fabrication of geometry, it would be extremely difficult to maintain these dimensions. Hence, for further analysis of optimal geometry of the actuator, the following values will be used:

\[ H = 10 \, \mu m, \quad D = 1 \, \mu m, \quad W = 15 \, \mu m \]

Comparing these values with the overall optimal design obtained using a single Design Variable \( H = 30, \quad D = 2.2 \, \mu m, \quad W = 15.2 \, \mu m, \quad \text{maximum deflection} = 30.65 \, \mu m \), obtained in the previous section, the following observations can be made.

- For the optimal design, diaphragm thickness is about 10% of the thickness of the actuator.
- The thickness of the actuator has been reduced to a third of the original value. At the same time, the maximum deflection has also increased to three times that of the original value. With the reduction in the thickness of the actuator, the volume of the actuator has also been reduced by same proportion. Since, the input power was kept same, the power density of the actuator has increased three times. The direct impact of this can be seen in the maximum temperature value. The maximum temperature has increased from 338°C to 366 °K, with the increase of 28 °K. The maximum operating temperature of the polyimide is about 700 °K. Hence, it can be said that, the overall optimal design will not be able to operate for as high powers as the optimal design with a single variable.
Although the thickness (H) and the diaphragm thickness (D), has reduced, the value of the width of the diaphragm (W) is almost the same. Considering the proportionality of the diaphragm in the overall optimal case, the length of the diaphragm is large compared to the optimal design with single variable. Figure 5.7 illustrates this point.

![Diagram of actuator geometries](image)

**Figure 5.7: Comparison of the optimal geometries (a) optimal geometry with single variable (b) overall optimal geometry.**

Comparing Figure 5.7 (a) with 5.7 (b), although the thickness of the actuator (H to H1) and the diaphragm thickness (D to D1) have changed, the width of the diaphragm is same in both cases. Considering the proportionality, the overall optimal design (Figure 5.7 (b)) has a larger width of the diaphragm. This would mean that the stiffness of the overall optimal geometry is very small. This has helped in increasing the maximum deflection by threefold. At the same time, the force generated by the actuator will reduce considerably.

### 5.14 Force developed by the overall optimal actuator:

The Maximum deflection versus externally applied force characteristic of the overall optimal actuator was estimated using the technique explained in Section 5.9. Figure 5.8 shows the plot of deflection versus applied force. The relation between developed force and the maximum deflection is

\[
Y_{ov} = -50.452X + 90.692
\]  

(5.11)
Applying the method used for deriving Equations 5.5 and 5.6, the relation between the deflection and the applied force will be:

\[ d_{ov} = C \times F + d_0(a) = 50.452 \times F - 90.692 \]  

(5.12)

Where, the compliance (C) of the optimized design is 50.452 \( \mu \text{m}/\mu \text{N} \), and the no load deflection \( (d_0) \) is \(-90.692 \mu \text{m}\).

From Figure 5.8, there is a substantial increase in the compliance value of the actuator. Compliance is inversely proportional to the stiffness of the actuator. Hence, the stiffness of the overall optimum design is very less as compared to the other designs. This has resulted in the increase in the maximum deflection
and reduction in the force developed by the actuator. From the discussion made at the end of Section 5.9, a direct comparison of these two actuators (Overall Optimal Design and Optimal Design for Single Variable), follows the relationship between the stiffness, force and maximum deflection.

From Figure 5.8, the overall optimal design gives higher deflections at the expense of the developed force. Hence, for the application where the applied force is not a critical factor, the overall optimal design would be suitable. One of such application could employ optical mirrors. If the mirror is mounted on the tip of the actuator (and assuming that micro mirror will not be heavy), the mirror can be moved through a larger distance.

For applications such as micro robotics, the optimal design with a single variable will be useful, as the load carrying capacity of this design is much higher.

5.15 Transient analysis of the overall optimized design:

Figure 5.9 shows the comparison of transient simulation results of the overall optimal design with the optimal design with single design variable.

The overall optimal design appears to have better cooling as compared to the optimal design with single design variable (Figure 5.9). The Maximum Deflection for the optimal design with single Design Variable ($D_{ov}$) is 13.28 μm, and the Recovery ($R_{ov}$) is 3.567 μm. The ratio of Recovery to Maximum Deflection, also known as yield is

$$Y_{ov} = \frac{R_{ov}}{D_{ov}} = \frac{3.567}{13.28} = 0.2685 = 26.85\%$$  \hspace{1cm} (5.13)

Similarly, the Maximum Deflection for the overall optimal design ($D_{os}$) is 60.02 μm, and the Recovery ($R_{os}$) is 7.17 μm. The yield is

$$Y_{os} = \frac{R_{os}}{D_{os}} = \frac{7.17}{60.02} = 0.1195 = 11.95\%$$  \hspace{1cm} (5.14)
Comparing Equation 5.11 and 5.12, there is a large improvement in the performance of the actuator, from heat transfer point of view. The overall optimal design is capable of recovering to the lower percentage recovery. e.g. assuming that the Maximum Deflection of both the actuators is 100 μm. For the optimal design with single Design Variable, the recovery of the actuator will be ~ 27 μm (Equation 5.13), and for the overall optimal design, the recovery will be ~12 μm (Equation 5.14). Since, one of the objectives of the optimization study is to reduce the recovery (improve cooling), hence the overall optimal design is better for cooling.

Comparing the overall optimal design with T. Ebefors’ design, the Net Gain of the overall optimized design is 44.02 μm. (The value of Net Gain for the overall optimized design is calculated using the method and formulae described in Section 5.10).
5.16 Justification for neglecting the radiation boundary condition:

For a body at temperature $T_b$ (°K), the radiation heat loss will be given by the following formula:

$$P_R = \varepsilon \times \sigma \times A_s \times (T_b^4 - T_o^4)$$  \hspace{1cm} (5.15)

where

$P_R = \text{Radiation Heat loss (watts)}$,

$\varepsilon = \text{Emissivity of the material}$,

$\sigma = \text{Steffen's-Boltzmann constant (5.6703 \times 10^{-8} \text{ W} / \text{m}^2 \text{K}^4)}$,

$A_s = \text{Surface area (m}^2)$,

$T_b = \text{Body Temperature (°K)}$,

$T_o = \text{Ambient Temperature (°K)}$.

The radiation heat loss will be highest for the blackbody (A hypothetic body that completely absorbs all of the thermal radiation incident on it). The Polyimide V-groove actuator will be assumed as a blackbody (this is a conservative assumption, since the actual radiation loss will always be less than the calculated value), hence $\varepsilon = 1$. The magnitude of this loss will depend on the local temperature. However, analysis of the local radiation loss for the entire length of the actuator will be very complicated. Hence, it will be assumed that the entire V-groove actuator is heater to the steady state temperature. According to Table 5.1 (a) and 5.1 (b), the maximum temperature at steady state ($T_b$) is 338 °K. According to Figure 2.7 and 3.1 (b), the length of the actuator is 1200 μm, and the width of the actuator is 600 μm. Both the surfaces (front and back surface) of the actuator will contribute in the radiation loss. The radiation loss will be given by:

$$P_R = 1 \times 5.6703 \times 10^{-8} \times (2 \times 1200 \times 10^{-6} \times 600 \times 10^{-6} \times (338^4 - 293^4)) = 4.7 \times 10^{-4} W$$  \hspace{1cm} (5.16)
Hence, the heat loss due to radiation loss is 0.47 mW. The supplied power is 90 mW. Hence the percentage heat loss due to radiation is

\[
%P_r = \frac{0.47}{90} \times 100 = 0.5\% \quad (5.17)
\]

In spite of all the conservative assumptions, the radiation heat loss is 0.5 % of the input heat. The actual heat loss will be much lower than 0.47. This justifies the assumption of neglecting the radiation heat loss for modeling the Polyimide V-groove actuator (Section 3.3).
Chapter 6: Concluding remarks.

The objective of this thesis was to develop a finite element model for a Polyimide V-groove actuator developed by T. Ebefors, which has never been attempted previously. Entire FEM simulations for a MEMS actuator were performed using ANSYS 5.6, which is a generic finite element package. The optimization module was used to improve the performance of the existing design. A substantial improvement in the performance was observed for the proposed design. In short, this research has established a methodology that can be extended for modeling and simulation of other MEMS devices.

6.1 Conclusions derived from this research:

- A computer simulated FEM model for heat and deflection analysis was validated for the Serpentine Heater case in the high frequency domain for three and four V-groove cases. Some differences between the simulated and experimental results (reported by T. Ebefors) were noted in the low frequency domain. Results of the parametric analysis proposed a need for an active cooling mechanism, since convection plays a small role in the heat transfer (cooling) process. It was also observed that all the V grooves do not contribute equally to the actuation process. Thus, the effect of the heater location on the performance of the actuator has been analyzed. The role of various parameters (thermal conductivity and wall temperature) has been investigated. Using a linear increase in the wall temperature, the discrepancies in the model in the lower frequency region were addressed to a certain extend. The modeling concepts and the boundary conditions developed in the validation of Serpentine Heater configuration were applied to the Polysilicon Heater configuration. The model of the Polysilicon Heater Configuration was successful in predicting simulation results close to the experimental data. Since the same set of boundary conditions were used for the Serpentine Heater and Polysilicon Heater Configuration, these results confirm the validity of entire FEM exercise.
To improve the performance of the actuator, different design geometries were proposed and each was simulated for various frequencies. "Fin type V-groove" geometry did not show any significant net performance improvement over the T. Ebefors' design. Although an improvement was noted in the cooling cycle here, the Maximum Deflection during the heating cycle was reduced. Due to complexities of fabrication, this alternative did not appear promising. Actuator geometry involving "varying silicon diaphragm thickness at the bottom of the device" showed performance improvement in Net Gain. More significant Net Gain was observed for the case of "uniform diaphragm thickness at the V-groove bottom". Considering the simplicity of fabrication, the "uniform diaphragm V-groove geometry" was selected for further investigation.

The thickness of the silicon diaphragm (referred as single Design Variable optimal design) at the bottom of the V-groove was optimized in order to maximize the deflection at the tip of the actuator. The optimization module of ANSYS was used for this optimization exercise. It was found that for the deflection of the actuator was maximum for the diaphragm thickness of 2.2 μm. This diaphragm thickness result confirms with the result of the "trial and error" method used previously. Force versus deflection analysis showed that there is improvement in the deflection as well as the force generated by the optimized design with single Design Variable as compared to T. Ebefors' design. Transient analysis (at 40 Hz) of the optimal design with single Design Variable showed a Net Gain of 1.44 μm over T. Ebefors' design.

In the second optimization exercise, all the dimensions of the V-grooves were used as Design Variables, and is referred to as overall optimization. A three times increase in the deflection was observed in the overall optimal design as compared to single variable optimal design. At the same time, there is a three times reduction in the maximum force developed by the overall optimal
design. It was concluded that three times increases in the deflection (accompanied by reduction in developed force) is a result of reduction in the stiffness of the overall optimized design. Transient analysis (at 40 Hz) revealed that the overall optimal design has better cooling characteristics compared to optimal design with single Design Variable. A Net Gain for the overall optimal design was 44 μm over T. Ebefors’ design.

6.2 Recommendations for future Work:

Completion of this thesis is not the end of this project; in fact it is just the start. The modeling and simulation of the actuator model presented in this thesis, helped in identifying certain areas for further research. They include:

- Inclusion of the temperature dependant material properties in modeling.
- It is necessary to couple the electrical domain into the current thermal-elastic modeling.
- Out-of-the plane bending of the actuator was neglected in this thesis. It is important to carry out further simulation for modeling the curing process of the polyimide inside the V-groove. This simulation will quantify the amount of stresses developed during curing process
- These curing stresses will then have to be included in to the modeling of the actuator.
- Modeling discrepancies for the lower frequency domain were addressed by applying linearly increasing wall temperature boundary conditions. For accurate modeling of wall temperature increases an exponential function will need to be used instead of a linear approximation.
- Proposed optimal designs include diaphragm thickness in the range of 1 μm to 3 μm. Addition of these diaphragms in the design may incorporate a lagging behavior in transient heat transfer as described for micro scale heat transfer. The model will have to be modified to accommodate microscale issues.
- Constant temperature boundary condition applied to the left wall should be eliminated and a body of large mass has to be attached at the left wall. Now by enforcing constant temperature boundary condition on this body should remove some of the low frequency range discrepancies in the model.

- Evaluate the effect of V-groove placement with respect to the left wall of the actuator needs to be simulated.

- Finish the fabrication and test of a leg actuator prototype, which is currently being undertaken at Semiconductor and Microsystems Fabrication Laboratory (SMFL), at RIT by the author. The outline of the fabrication process is presented in Appendix A.
References


[22]. Training Manual “Introduction to ANSYS- part 1 for Release 5.6”, Nov 1999. ANSYS® is a registered trademark of SAS IP Inc.


Appendix A: Block diagram of proposed fabrication process

1. Nitride layer deposition for masking
2. Lithography and Etching Nitride for trench opening
3. KOH Etching of deep trench (450μm) from backside
4. Boron Diffusion for the thin diaphragm below V-grooves
5. Lithography and etching the Nitride for V-grooves
6. KOH etching of the V-grooves
7. Deposition, Lithography and etching of heater material
8. Deposition, Lithography and etching of contact material
9. Spin coating Polyimide inside V-grooves
10. View of the structure so far
11. Lithography and Etching Nitride for releasing the leg
12. DRI etching of the leg to release the structure and curing
Appendix B: Sample “log file” (generated by ANSYS) used for the Overall Optimization of the V-groove.

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*SET,W , 15.2
*SET,D , 4
!* /NOPR
/PMETH, OFF, 0
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KEYW, PR_STRUCT, 1
KEYW, PR_THERM, 1
KEYW, PR_FLUID, 0
KEYW, PR_ELMAG, 0
KEYW, MAGNOD, 0
KEYW, MAGEDG, 0
KEYW, MAGHFE, 0
KEYW, MAGELC, 0
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KEYW, PR_CFD, 0
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KEYOPT, 1, 4, 0
KEYOPT, 1, 5, 0
!* !*
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K, 0.388*H+W,-H,,
K, 0.776*H+W,0,,
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ADELE, 3, , 1
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LGEN, 2, P51X, , 1.5, , , 0
FLST, 3, 1, 4, ORDE, 1
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LGEN, 2, P51X, , -1.5, , , 0
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CMDELE, _Y2
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CMDELE, _Y2
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CMDELE,_Y2
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NSORT,S,EQV,1,1, ,0
*GET,S,SORT,,MAX
NSORT,TEMP, ,1,1, ,0
*GET,Tmax,SORT,,MAX
NSORT,S,X,1,1, ,0
*GET,sxmax,SORT,,MAX
NSORT,S,Y,1,1, ,0
*GET,symax,SORT,,MAX
Appendix C: Material Properties

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<th>SILICON</th>
<th>POLYIMIDE</th>
<th>ALUMINUM</th>
<th>POLYSILICON</th>
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<tr>
<td>Young’s Modulus (N / m²)</td>
<td>190 x 10⁹</td>
<td>25 x 10⁹*</td>
<td>70 x 10⁹</td>
<td>190 x 10⁹</td>
</tr>
<tr>
<td>Poisson’s Ratio</td>
<td>0.28</td>
<td>0.34</td>
<td>0.33</td>
<td>0.28</td>
</tr>
<tr>
<td>Density (Kg / m³)</td>
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<td>2000*</td>
<td>1410*</td>
<td>2330</td>
</tr>
<tr>
<td>Thermal Conductivity (W / m °K)</td>
<td>140*</td>
<td>0.16*</td>
<td>0.33*</td>
<td>29</td>
</tr>
<tr>
<td>Specific Heat (J / Kg °K)</td>
<td>707*</td>
<td>2000*</td>
<td>900*</td>
<td>753</td>
</tr>
<tr>
<td>Thermal Expansion Coefficient (ppm / °K)</td>
<td>2.33 x 10⁻⁶</td>
<td>40 x 10⁻⁶</td>
<td>23.1 x 10⁻⁶</td>
<td>2.33 x 10⁻⁶</td>
</tr>
</tbody>
</table>

Table C.1: Material properties used for simulations. (Properties marked with “*” were taken from [18], and the other from [13])
Appendix D: Shrinkage coefficient Vs Curing temperature

The author is currently fabricating the optimized design of the polyimide V-groove actuator at SMFL (Semiconductor and Microsystem Fabrication Laboratory), in RIT. The “Durimide 7520” (polyimide) of the “Arch Chemicals, NY” is used in the fabrication of the actuator. Preliminary tests were performed on the “Durimide 7520” sample by the author to investigate the shrinkage coefficient ($\varepsilon$) characteristics. Figure D1 illustrates the shrinkage coefficient as a function of curing temperature. The method described in the Section 2.12 has been used to obtain shrinkage characteristics of “Durimide 7520”. The shrinkage coefficient shows a significant rise over a temperature range from 250$^\circ$ C to 300$^\circ$ C. This behavior is a result of the degassing of the solvent and the cross-linking taking place between the Polyimide molecules at this temperature. Beyond this temperature range, the shrinkage coefficient value remains fairly constant. In order to maximize the out-of-plane curling of the leg actuator, curing will be done above 300$^\circ$ C.

![Figure D.1: Characteristic of the “Durimide 7520”: - shrinkage coefficient for a 50 $\mu$m thick polyimide film spun coated over a 4” bare silicon wafer vs. curing temperature.](image-url)