Rotary Micro-Ball Bearing Designs for MEMS Applications

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Abstract

Micro-electro-mechanical systems (MEMS) technology allows the fabrication of small mechanical systems in silicon using standard micro-fabrication processes. MEMS techniques have found wide acceptance in such devices as accelerometers, micro-mirrors, resonators, probes, and micro-tweezers to name a few. Though small linear motions are common in MEMS applications, few devices exhibit reliable rotary motion. This work explores several methods of fabricating rotary bearings using micro-balls as the support mechanism. Micro-ball bearings have several advantages over other MEMS bearing technologies in that they provide robust mechanical support, require no external control systems, and basic designs require very few fabrication steps.

Ball cages or retainers are common in macro-scale bearings, providing uniform spacing between the balls. Several cage designs are proposed and explored in this work: a radial ball bearing with an integrated ball cage, a dual-row style cage, and five unique cage geometries integrated into silicon micro-turbines (SMTs.) Also, an example of a curved or angular contact raceway is presented as an example of this type of raceway geometry in MEMS devices. Each is presented with a discussion of the design considerations and fabrication process. This is followed by a characterization of the performance of each design.

These studies found that the integrated cage in the radial ball bearing performs well at speeds ranging up to 20 000 RPM. Minimal wear was observed after 6 hours of continuous testing. However, the solder bond in the cage was a common failure point in these devices, limiting the reliability and longevity. The dual groove style cage was designed to eliminate the solder bond. However, the higher frictional forces between the ball and the cage in
this design resulted in higher losses during operation. Taking into account the higher losses and the added complexity of the design, it seems unlikely that this approach would be appropriate for further study. However, the design does represent a novel approach for releasing multi-wafer rotary structures and is presented here as example of this technique. Testing of the cage designs for the SMTs indicated that a full ring design (a full annulus with holes for the balls) performed the best of the 5 cage geometries. However, these devices do not perform as well as cage-less designs for high speed applications due to higher fictional forces and increased raceway wear at the interface between the ball and the raceway edge. Finally, the curved raceway has shown excellent performance up to 2500 RPM with normal loads up to 40 mN in tribometer testing. SMTs with this raceway design were also tested for over 10 million revolutions and at speeds over 70 000 RPM. The test results for all of the bearings designs presented here show that the devices exhibit stable operation at low to moderately high speeds.
Contents

Acknowledgments 3

Abstract 5

Table of Contents 7

List of Figures 8

List of Tables 9

Nomenclature 10

1 Introduction 14
   1.1 Background ................................................................. 14
   1.2 Motivation and Goals ....................................................... 16
   1.3 Research Objectives ......................................................... 16
   1.4 Description of Thesis ....................................................... 17

2 Bearings in MEMS Devices 19
   2.1 Competing Bearing Technologies ........................................... 19
   2.2 Pin Joint Bearing ............................................................... 20
   2.3 Magnetic or Electrostatic Levitation ..................................... 22
   2.4 Jewel Bearing ................................................................. 22
   2.5 Air Bearing ................................................................. 23
   2.6 Liquid Bearing ............................................................... 26
   2.7 Previous Work in Micro-Ball Bearings .................................... 27
       2.7.1 Linear Micro-Ball Bearing ........................................... 27
       2.7.2 Rotary Micro-Ball Bearings .......................................... 29
       2.7.3 Conventional Micro-Ball Bearings ................................... 30
   2.8 Discussion and Conclusions ................................................. 32
3 Micro-Fabrication Technologies 33
  3.1 Patterning and Masking ........................................... 33
    3.1.1 Photo-Masks ................................................. 34
    3.1.2 Photolithography ............................................. 34
    3.1.3 Masking Materials .......................................... 35
      3.1.3.1 Photoresist ........................................... 35
      3.1.3.2 Silicon Dioxide ....................................... 36
  3.2 Deep Reaction Ion Etching (DRIE) ................................ 37
    3.2.1 Basic Principles ............................................. 37
    3.2.2 Through-Wafer Etching ....................................... 38
    3.2.3 Halo Mask for Through Wafer Etching ......................... 39
    3.2.4 Etch Lag ..................................................... 40
  3.3 Electroplating .................................................... 41
  3.4 Eutectic and Solder Wafer Bonding ............................... 43
  3.5 Conclusion ........................................................ 44

4 Ball Bearing Design Considerations 45
  4.1 Ball Cage .......................................................... 46
  4.2 Raceway and Roller Element Materials ............................ 49
  4.3 Raceway Geometry ................................................. 50
  4.4 Raceway Tolerance ................................................ 54
  4.5 Raceway Fill ........................................................ 55
  4.6 Wear ............................................................... 56
  4.7 Bearing Loss ....................................................... 57
  4.8 Discussion and Conclusions ....................................... 61

5 Testing Methodologies 63
  5.1 Tribometer .......................................................... 63
    5.1.1 Tribometer Measurement Technique ................................ 64
    5.1.2 Test Platform .................................................. 66
      5.1.2.1 Torque Measurement Platform (Inner Platform) ............ 66
      5.1.2.2 Normal Load Measurement Platform (Outer Platform) ....... 69
    5.1.3 Tribometer Setup and Operation ................................ 71
  5.2 Silicon Micro-Turbine (SMT) ....................................... 76
    5.2.1 Turbine Design and Operation .................................. 77
    5.2.2 Testing Setup .................................................. 79
  5.3 Discussion and Conclusions ....................................... 86
6 Proof of Concept of a Radial Ball Bearing with Integrated Ball Cage 88

6.1 Design ......................................................... 89
6.2 Device Fabrication ........................................... 91
   6.2.1 Step A - Plate Solder Pads ......................... 91
   6.2.2 Step B - Etch Cage Arms and Bearing Raceway ... 92
   6.2.3 Step C - Pattern Cage Release Channels ......... 92
   6.2.4 Step D - Device Assembly ......................... 92
   6.2.5 Step E - Release Etch ............................. 92
   6.2.6 Fabricated Devices ................................. 93
6.3 Testing ........................................................ 94
   6.3.1 Cage Bond Failure .................................. 95
   6.3.2 Tribometer Results ................................ 96
   6.3.3 Wear .................................................. 99
6.4 Discussion and Conclusions ............................. 101

7 In-Situ Fabrication of a Monolithic Silicon Ball Cage 103

7.1 Design ........................................................ 105
7.2 Device Fabrication .......................................... 110
   7.2.1 Top and Bottom Die Fabrication .................. 110
      7.2.1.1 Step A (Front Side)- Plate Solder Pads .... 111
      7.2.1.2 Step B (Front Side) - Pattern Raceway Stand- off Trench ........................................... 112
      7.2.1.3 Step C (Front Side) - Pattern the Raceway and Etch the Raceway Stand-Off ......................... 112
      7.2.1.4 Step D (Front Side) - Etch Bearing Raceway . 112
      7.2.1.5 Step E (Back Side) - Define Cage Release Window and Identification Marks .......................... 112
      7.2.1.6 Step F (Back Side) - Define Testing Adaptor Feature and Bearing Release Channels; Etch Cage Release Window Through the Wafer ...................................... 113
      7.2.1.7 Step G (Back Side) - Pre-Etch Testing Adaptor Feature and Bearing Release Channels; Etch Cage Release Window Through the Wafer ...................................... 113
      7.2.1.8 Images of Fabricated Top and Bottom Die ...... 113
   7.2.2 Center Die Fabrication ............................... 115
      7.2.2.1 Step A (Top Side) - Plate Solder Pads ...... 116
      7.2.2.2 Step B (Top Side) - Define Cage with Support Beams ...................................................... 117
      7.2.2.3 Step C (Back Side) - Plate Solder Pads ...... 117
7.2.2.4 Step D (Back Side) - Define Cage with Support Beams
7.2.2.5 Step E (Back Side) - Pattern Cage Without Support Beams and Pre-Etch Cage With Support Beams
7.2.2.6 Step F (Back Side) - Etch Cage to Final Depth and Reduce Beam Thickness
7.2.2.7 Step G (Top Side) - Pattern Cage Without Support Beams and Pre-Etch Cage With Support Beams
7.2.2.8 Step H (Top Side) - Etch Area Around Cage Through the Wafer and Reduce the Beam Thickness
7.2.2.9 Images of Fabricated Center Die
7.2.3 Device Assembly and Release Etching
7.2.3.1 Step A - Device Assembly
7.2.3.2 Step B - Release Etching
7.2.3.3 Images of Assembled and Released Devices
7.3 Testing
7.4 Test Results
7.5 Wear
7.6 Discussion and Conclusions
8 Micro-Turbines with Integrated Silicon Ball Cage
8.1 Retainer Ring Design Considerations
8.2 Retainer Ring Design Variations
8.3 SMT Design
8.4 Device Fabrication
8.4.1 Retainer Fabrication
8.4.2 Turbine Wafer Fabrication
8.4.2.1 Step A (Top Side) - Electroplate Solder Pads
8.4.2.2 Step B (Top Side) - Pattern Turbine Release Channel
8.4.2.3 Step C (Top Side) - Pattern Cage Raceway Standoff
8.4.2.4 Step D (Top Side) - Pattern Cage Raceway and Etch Bearing Release Channel
8.4.2.5 Step E (Top Side) - Etch Cage Raceway Stand Off
8.4.2.6 Step F (Top Side) - Etch Cage Raceway
8.4.2.7 Step G (Back Side) - Pattern Turbine Release Channel ........................................ 143
8.4.2.8 Step H (Back Side) - Pattern Turbine Blade and Etch Bearing Release Channel .......... 143
8.4.2.9 Step I (Back Side) - Pre-Etch Turbine Blades ......................................................... 143
8.4.3 Thrust Wafer Fabrication ......................................................................................... 143
8.4.3.1 Step A (Top Side) - Electroplate Solder Pads .......................................................... 144
8.4.3.2 Step B (Top Side) - Etch Bearing Raceway ............................................................. 144
8.4.3.3 Step C (Back Side) - Pattern and Pre-Etch Thrust Release Channel ......................... 145
8.4.4 Assembly and Release Etch .......................................................... 145
8.4.4.1 Step A - Place Cage, Solder Balls, and Steel Balls in Thrust Die ............................... 146
8.4.4.2 Step B - Align Turbine Die and Bond the Device ................................................... 146
8.4.4.3 Step C - Turbine Release Etch .................................................................................. 147
8.4.4.4 Images of Fabricated SMT Devices with Integrated Cages ...................................... 147
8.5 Testing ......................................................... 148
8.5.1 Comparison of Cage Design Performance ............................................................... 149
8.5.2 Performance Repeatability ....................................................................................... 152
8.6 Longevity and Wear ................................................................. 153
8.7 Discussion and Conclusions .......................................................... 155
9 Curved Raceway .................................................................................. 157
9.1 Device Design ................................................................................ 157
9.2 Device Fabrication ......................................................................... 159
9.2.1 Fabrication of SMTs with Angular Contact Raceways ............................... 159
9.2.1.1 Bearing Raceways ......................................................................................... 159
9.2.1.2 Turbine Side ................................................................................................. 161
9.2.1.3 Thrust Side .................................................................................................... 162
9.2.1.4 Device Assembly ........................................................................................... 163
9.2.2 Fabrication of Devices for Tribometer Testing ........................................ 164
9.2.2.1 Bearing Raceway ........................................................................................... 164
9.2.2.2 Backside ....................................................................................................... 166
9.2.2.3 Adaptor Side .................................................................................................. 167
9.2.2.4 Device Assembly ........................................................................................... 168
9.2.3 Images of Fabricated Devices ................................................................. 169
9.3 Performance .................................................................................... 171
9.3.1 Performance of SMT with Angular Contact Raceway ............................... 171
9.3.2 Tribometer Measurements ...................................................................... 173
9.3.3 Wear ................................................................................................. 175
9.4 Discussion and Conclusions .................................. 176

10 Conclusions and Further Work .......................... 178
  10.1 Contributions of this Thesis ............................. 178
  10.2 Objectives Revisited ..................................... 179
  10.3 Conclusions ................................................. 182
  10.4 Further Work ............................................... 182

11 List of Published Works ..................................... 185

List of Published Works ...................................... 185

Bibliography ....................................................... 201

A AZ9260 Photoresist Coating Process ....................... 202
## List of Figures

<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.1</td>
<td>A. Image of the gear arrangement in the polysilicon pin joint studies; B. Excessive wear found on drive gear pin hole; C. Undamaged pin hole on side gear to compare with above image; D. Cross section of the pin joint bearing interface showing unworn structure (top) and worn structures (bottom). Images reproduced from [Tanner 98].</td>
</tr>
<tr>
<td>2.2</td>
<td>Conceptual drawing of a common jewel bearing design, with jewels forming a cup on the stator to hold a pin attached to the rotor.</td>
</tr>
<tr>
<td>2.3</td>
<td>Optical photograph (top) and schematic (bottom) of the cross section of a silicon micro-turbine supported on an air bearing. Diagram reproduced from [Frechette 05].</td>
</tr>
<tr>
<td>2.4</td>
<td>Air bearing operating modes presented in [Hara 03]. Figure reproduced from [Hara 03].</td>
</tr>
<tr>
<td>2.5</td>
<td>(a) Schematic of the linear micro-motor on micro-ball bearings (b) Square wave motor drive signal. Figure reproduced from [Modafe 06].</td>
</tr>
<tr>
<td>2.6</td>
<td>A. Optical image of ball jamming in the bearing raceway; B. SEM image of ball without wear (top) and ball after 39 min. operation (bottom); C. SEM image of the unworn bonded silicon raceway interface (top) and the worn bonded silicon raceway interface. Images reproduced from [Waits 07a].</td>
</tr>
<tr>
<td>2.7</td>
<td>Picture of the Timken S100 micro-ball bearing, photo by Timken Co.</td>
</tr>
<tr>
<td>3.1</td>
<td>Halo mask concept (left) with the etch trenches in white, the device structure in pink and the sacrificial substrate in green. SEM (right) of a device fabricated by through wafer etching using a halo mask.</td>
</tr>
</tbody>
</table>
3.2 SEM image of high aspect ratio silicon trenches, with feature sizes ranging from 2.2\( \mu m \) on the left to 5.5\( \mu m \) on the right. Larger trench sizes have a higher etch rate and lower aspect ratio. Image from [Chung 04].

4.1 Examples of macro-scale cage designs. A. Nylon snap cage for ball bearing, B. Nylon cage for cylindrical bearing, C. Nylon cage of high-angular-contact bearing, D. Phenolic cage for precision ball bearing, E. Angular contact bearing with annular cage, F. deep-groove Conrad-assembly bearing with riveted cage, G. Dual row ball bearing with snap cage, H. Thrust ball bearing with annular cage design. Images reproduced from [Harris 06b].

4.2 Examples of the 2 raceway geometries explored in this thesis. The contact region for each race is indicated by the red regions in the diagram, the rectangular raceway (left) provides smaller elliptical contacts, while the curved (angular) raceway (right) provides much larger elongated elliptical contact regions. A thrust load (W) is applied to the rotor as indicated by the blue arrow.

4.3 The contact angle (\( \alpha \)) is shown for a ball bearing under thrust load. The contact angle is the angle at which the ball contacts the raceway under load or during operation. The contact angle can be different between the rotor and the stator.

4.4 Parameters for Calculating \( P_d \).

4.5 Examples of wear and denting on macro-scale bearings. A. Smearing on the raceway of a tapered bearing, B. Extreme micropitting on ball bearing inner raceway, C. Advanced surface initiated fatigue on a thrust ball bearing, and D. Rolling element denting of a ball. Images from [Harris 06a].

4.6 Diagram showing the net force on the stator from clockwise motion of the rotor during sliding (clockwise) and rolling (anticlockwise.)

5.1 Top view of the center test platform under applied torque.
5.2 Side view of the outer platform under applied normal load.
5.3 Assembled and labeled test platform.
5.4 SEM of the torque of a torque platform with all 50 beam (left) and a CAD rendering of the entire platform (right).
5.5 Top view of torque platform, \( w \) - beam width, \( L \) - beam Length.
5.6 Top view of normal load platform, \( w \) - beam width, \( L \) - half the folded beam length.

5.7 CAD drawing of the fully assembled test platform showing top, side and bottom views.

5.8 Drawing of the underside of the test platform, depicting the torque measurement laser path from the source, reflected from the prism, and redirected to the sensor by the piezo actuated reflector and fixed mirrors.

5.9 A. CAD Drawing of the top (left) and bottom (right) of the device adaptor, B. SEM of the bottom of the adaptor, this is the part that is inserted into the device, C. picture of the adaptor inserted into the device, the device is inserted into the sample holder, and the motor coupler can be seen above the adaptor.

5.10 Screen capture of the LabVIEW screen used for testing.

5.11 Diagram showing the net force on the stator from clockwise motion of the rotor during sliding (clockwise) and rolling (anticlockwise).

5.12 SMTs with 5 mm rotor diameter (left) and 10 mm (right).

5.13 A cutaway showing the turbine test setup. Pressurized gas is used to power the turbine. The input power is controlled by an electronically controlled proportional valve and input power is monitored by a flow sensor and inlet pressure sensor. The top of the turbine is vented to atmospheric pressure. Some of the inlet gas leaks through the bearing to the backside of the turbine. This gas applies a net upward force, monitored by the thrust pressure sensor, on the turbine allowing the bearing to function in the proper mode. In order to reduce this pressure a bleed valve is attached to the bottom side of the test enclosure. The bleed rate is controlled by an electronically controlled proportional solenoid.

5.14 CAD drawing of the turbine with half of the plumbing wafer cutaway with a conceptualization of the gas flow through the turbine with a stalled rotor.

5.15 Conceptual diagram showing the parts and connections in the SMT test setup.

5.16 Optical Signal Digitizer comparator circuit (top) and a digital oscilloscope capture (bottom) of the conversion from the analog signal to digital.
5.17 Picture of the test enclosures for both 10 mm and 5 mm diameter devices (top) and the assembled test enclosure set up for testing (bottom) .............................................. 85

6.1 CAD rendering of the assembled bearing (left) and an exploded diagram showing all of the individual part (right) ........ 89

6.2 Depiction of the design parameters and how they relate to the bearing design ................................................................. 90

6.3 Process flow for a radial ball bearing with an integrated ball cage. A) electroplate solder pads, B) etch cage and raceway features, C) define DRIE mask in backside oxide, D) insert steel balls and bond die, and E) release the rotor and cage using oxide mask from Step C. ............................................. 91

6.4 SEM of the parts of the bearing: the ball cage (upper left), the rotor (upper right) the stator (lower left), and fully assembled bearing (lower right) ................................................. 93

6.5 SEM of the stator after the release etch. The boxes and arrows indicate damage caused to the raceway by over-etching during fabrication ................................................................. 94

6.6 SEM of a design with 8 balls with the top of the stator removed to show the inside of the bearing ........................................... 94

6.7 SEM image of a cage that quickly during testing. The ...... 96

6.8 The measured torque of the bearing at low speeds and the predicted torque from Equation 6.1. The error bars indicate the range of the measurement for each speed and the connected points are the average of the measurements ..................... 97

6.9 The average of the measured torque for the bearing over 3 test runs at speeds of 1000 RPM to 20 000 RPM with the predicted torque from Equation 6.1 ............................................. 98

6.10 The graph shows the measured power loss in the bearing from 1000 RPM to 20 000 RPM. At 1000 RPM the bearing loss is approximately 0.5 mW. As the speed increases and the centrifugal force plays a more dominate role and increases the bearing loss ................................................................. 99

6.11 SEM image of the cage after testing. The circles indicate the locations of the wear on the ball pockets. The damage to the cage arms is due to wear as similar damage is not seen in Figure 6.7 on a cage that failed after very little testing ........ 100
6.12 SEM image of the top of the silicon rotor after testing. Fabrication damage is indicated in the boxes and wear damage is contained in the ellipse. The wear damage appears as a rounding at the edge of the rotor where the rotor and ball contact each other. This damage is not seen in untested devices.

6.13 SEM Image showing the wear on the bottom half of the stator. Wear damage is contained in the ellipse. The wear is a rounding of the stator edge where the stator and the ball contact each other. This rounding is not present in untested device.

7.1 Conceptual drawing of the multi-wafer release etch technique (left), DRIE is used to etch a sacrificial beam through windows that have been etched in the upper and lower wafers of the 3 wafer stack. A CAD drawing of the bearing design with red arrows to indicate the location of the sacrificial beams and the rotor release channel before release etching (right A) and after release etching (right B.).

7.2 Conceptual drawing of the dual groove style bearing design. The cutaway (lower right) shows the orientation of the parts of the device under a thrust normal load.

7.3 Conceptual drawing of the cage support release etching. This is done in two DRIE etch steps, one from the top and one from the bottom.

7.4 Drawing of the cage release window showing the cage support beam, the top and bottom balls, the release channel and measurement labels for the cage features.

7.5 Layouts on both sides of the top/bottom wafers, and on the center wafer of the dual row style design with labels indicating important features.

7.6 Device fabrication process flow for the top and bottom die used in the dual row style cage device.

7.7 SEM image of the raceway of the bottom die for the dual row style cage device.

7.8 SEM of the raceway side of the bottom die for the dual row style cage device, the arrows indicate features that have been etched completely through the wafer.

7.9 SEM image of the top of the top die for the dual row style cage device. The device identification marks, test adaptor features, bearing release channels and the cage release windows are clearly visible.
7.10 Device fabrication process flow for the center die used in the
dual row style cage device. . . . . . . . . . . . . . . . . . . . . 116

7.11 SEM image of the center die used for the dual row style cage
device. The cage can be seen at the center of the device sup-
ported by 4 beams. The black regions in the image have been
etched through the wafer. . . . . . . . . . . . . . . . . . . . . 119

7.12 SEM image of the center die used in the dual row style cage
device. The angle of the device in the image shows the dif-
ference in the height between the top of the wafer, the ball
pockets and the tops of the beams. The black regions have
been etched through the wafer. . . . . . . . . . . . . . . . . . 120

7.13 Device fabrication process flow for assembly and release etch-
ing of the dual row style cage device. The red arrows show
the location of the sacrificial beams and the bearing release
channel before (A) and after (B) the release etches. . . . . . . . 121

7.14 SEM image of the top of an assembled and released dual row
style cage device. The test adaptor, bearing release channel,
cage release windows and identification marks are visible. . . . 123

7.15 SEM image of the bottom of an assembled and released dual
row style cage device. The bearing release channel, cage re-
lease windows and the identification marks are visible. . . . . . 124

7.16 Torque measurements of 4 tests of the dual row ball bearing
design. . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . 125

7.17 Measured bearing loss of the dual row ball bearing design for
all 4 tests plotted with the power loss for the radial bearing
over the same range. Though the power loss varies with $\omega$
rather than $\omega^2$, as seen in the radial design, the loss of the
dual row bearing is almost an order of magnitude greater than
the radial design over the same range. . . . . . . . . . . . . . . . 126

7.18 SEM image of the cage after testing. Wear at the edge of the
ball pocket is indicated in the ellipse on the left. The location
of the support beam is indicated in the ellipse on the right;
there is very little indication of the location of the support
beam after the release etch. . . . . . . . . . . . . . . . . . . . . . 127

7.19 SEM image of the rotor after testing. The arrow on the left
points to rotor wear which appears as a discolored region on
raceway. The ellipse in the figure indicates where the support
beam was located before etching. . . . . . . . . . . . . . . . . 127
7.20 SEM image the stator after testing. The arrow indicates the location of the wear, which appears as a discolored region on the raceway. Pillar defects are present at the edge of the raceway and are an artifact of the multi-step etching processes.

8.1 Photographs of the both the 5 mm and 10 mm devices with a British Pound coin for scale (top), and a cutaway view of the device showing the retainer ring (bottom).

8.2 Cut-away schematic view of a conventional thrust bearing with a ball-riding retainer.

8.3 (a) SEM images showing the 5 mm retainer ring designs: Full Ring (top left); Full Skeleton (top right); Half Skeleton (bottom left); Outer Open (bottom right); Inner Open (centre).
(b) Schematic cross-sections of bearings with the different retainer types.

8.4 SEM image of a SMT with the rotor removed to expose the retainer and balls, release etch damage to the retainer is indicated by the arrow.

8.5 Process flow for the turbine wafer of the SMT with a silicon ball cage.

8.6 Photograph showing the different mask layers on the turbine raceway before the bearing release channel is etched. The solder pads are also covered with photoresist.

8.7 Process flow for the thrust wafer of the SMT with a silicon ball cage.

8.8 Process flow for the assembly and release etch of the SMT with a silicon ball cage.

8.9 SEM of the 5 mm diameter SMT after the all of the parts have been assembled on the thrust die but before the turbine die has been placed on top.

8.10 SEM of the 5 mm diameter device after release etching.

8.11 SEM image showing the Full Ring cage and steel balls in the 5 mm SMT stator after rotor has been removed.

8.12 SEM image of a rotor and cage that have been removed for the SMT stator. The turbine blades have been severely damaged.

8.13 Performance curves for the 5 mm devices plotted in RPM versus Input Power W.

8.14 Performance curves for the 10 mm devices plotted in RPM versus Input Power W.

8.15 Measured repeatability of performance over 12 ramp tests for a 5 mm Full Ring Device operating at difference power levels.
8.16 SEM images of the raceway wear on the rotor (top image) and the stator (bottom image) of the 10 mm diameter device after longevity testing. 154

8.17 SEM image showing the wear (indicated by the arrows) on the top of the retainer after longevity testing. 155

8.18 SEM image showing the wear on the bottom of the retainer after longevity testing. 155

9.1 SEM of micro-lens mold with a diameter of 116.7 µm created using HNA etching, picture from [Albero 09]. 158

9.2 SEMs of the curved profile created by the ICP method, picture from [Larson 05]. 158

9.3 Device fabrication process flow for SMTs with angular contact raceways. 160

9.4 Device fabrication process flow for devices with angular contact raceways for tribometer testing. 165

9.5 SEM image of a rotor used for tribometer testing. A steel ball has been placed next to the rotor to show the curvature of the raceway in relation to the ball used in the device. 169

9.6 SEM image of the SMT rotor with a ball next to the raceway to show the curvature of the raceway in relation to the ball. 170

9.7 SEM image of a SMT stator raceway. The large depression on the raceway are damage for over-etching during the release etch step. 170

9.8 SEM image of a stator raceway with the ideal geometry and no etch damage. 171

9.9 Graph of the speed and input power of the SMT after each 2 million revolution longevity test. After an initial improvement in performance after 2 million revolution, the turbine performance begins to decline. 172

9.10 Comparison of the performance of the 5 mm SMT with the Full Ring and the curved raceway SMT. 173

9.11 Graph of the tribometer results for the angular contact bearings. The line represents the expected torque value for the devices and the measured data points for 3 tests are scattered plotted on the graph. 174

9.12 The power loss predicted by using Equation 9.1 to calculate the frictional torque and the power loss calculated using the average of the measured frictional torque. 175
9.13 SEM image of the tested SMT stator raceway. The wear pattern is enclosed in the box and the over-etch damage to the raceway is indicated by the arrows. The wear can be identified as the lighter regions on the raceway region enclosed by the box.176
## List of Tables

<table>
<thead>
<tr>
<th>Table</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.1</td>
<td>Comparison of micro-bearing technologies</td>
<td>19</td>
</tr>
<tr>
<td>3.1</td>
<td>DRIE process parameters for anisotropic etching for recipes named Dark3S and Dark4S</td>
<td>38</td>
</tr>
<tr>
<td>3.2</td>
<td>Electroplating parameters for the metals used in the fabrication processes in this research</td>
<td>42</td>
</tr>
<tr>
<td>3.3</td>
<td>Etchants used to remove the seed layer after electroplating</td>
<td>43</td>
</tr>
<tr>
<td>4.1</td>
<td>ISO 3290-1:2008 standard values for Grade 5 balls. The values represent the maximum deviation allowed for each parameter</td>
<td>49</td>
</tr>
<tr>
<td>4.2</td>
<td>Values used to calculate the contact areas of the rectangular and curved raceways. Steel values are from [Bhushan 01] and silicon values are from [Hull 99]</td>
<td>52</td>
</tr>
<tr>
<td>5.1</td>
<td>Typical design parameters for the torque measurement platform</td>
<td>68</td>
</tr>
<tr>
<td>5.2</td>
<td>Design parameters for the normal load measurement platform</td>
<td>71</td>
</tr>
<tr>
<td>6.1</td>
<td>Values for the design parameters of the tested devices</td>
<td>90</td>
</tr>
<tr>
<td>7.1</td>
<td>Design parameters for the cage support beam and etch windows for the first and second design iterations</td>
<td>108</td>
</tr>
<tr>
<td>8.1</td>
<td>Key design parameters for large (10 mm) and small (5 mm) devices. All dimensions are in µm.</td>
<td>135</td>
</tr>
<tr>
<td>8.2</td>
<td>Comparison of retainer parameters by geometry</td>
<td>136</td>
</tr>
<tr>
<td>9.1</td>
<td>DRIE process parameters for etching the angular contact raceway</td>
<td>162</td>
</tr>
</tbody>
</table>
Nomenclature

Symbols Used - In Order of Appearance

\( D \) Ball diameter
\( \nu_{Si} \) Poisson Ratio Silicon
\( \nu_{St} \) Poisson Ratio 440C Steel
\( E_{Si} \) Young’s Modulus Silicon
\( E_{St} \) Young’s Modulus 440C Steel
\( r \) Raceway curvature radius
\( W \) Force on the rolling element
\( d_m \) Bearing Pitch Diameter
\( E \) Contact Modulus
\( R \) Ball Radius
\( a \) Elliptical Contact along Major Axis, or radius for point contact
\( b \) Elliptical Contact along Minor Axis
\( p_0 \) Maximum Pressure at Center of a Contact Ellipse
\( \sigma_{\rho} \) Geometric Parameter for Calculating Elliptical contacts
\( P_d \) Diametral Clearance
\( d_o \) Outer Diameter of Ball Raceway
\( d_i \) Inner Diameter of Ball Raceway
\( F\% \) Raceway Fill
\( N_b \) Number of Balls in Bearing Design
Centrifugal Force from Rotating Balls
Orbital Speed of the Balls in RPM
Rotational Speed of Balls in rad s\(^{-1}\)
Bearing Rotational Speed
Material Density
Bearing Orbital Speed in RPM
Frictional Torque
Frictional Torque for Applied Load and Centrifugal Forces
Toque from Lubrication Viscous Friction
Total Bearing Power Loss Due to Frictional Torque
Rotational Displacement
Displacement in the z-axis
Beam Length
Beam Width
Beam Depth (Thickness)
Number of Beams
Transverse Shear Force
Couple for Calculation Torsional Stiffness
Torque
Torsional Stiffness
Z Axis Displacement
Force Applied in the Z Direction
Separation Between Beam Center Lines (Chapter 5)
Cage Thickness
Raceway Inner Diameter (Same as \(d_i\))
Raceway Outer Diameter (Same as \(d_o\))
Cage Release Gap (Chapter 6)
t  Raceway Tolerance Equal to 0.5\(P_d\)
\(W_b\)  Sacrificial Beam Width
\(W_w\)  Etch Window Width
\(H_w\)  Etch Window Height
\(R_{BP}\)  Radius of Ball Path
\(R_I\)  Retainer Inner Radius
\(R_O\)  Retainer Outer Radius
\(g\)  Ball Pocket Opening (Chapter 8)
\(R_B\)  Ball Radius
\(R_P\)  Pocket Radius in Retainer
\(W_{BR}\)  Width of Bearing Raceway
\(W_{RR}\)  Width of Retainer Raceway
\(H_{BR}\)  Height of Bearing Raceway
\(H_{RR}\)  Height of Retainer Raceway (Incl. Solder)
\(H_{SO}\)  Stand-Off Height
\(H_R\)  Height of Retainer
\(\delta_\pm\)  Radial Play in Retainer Ball Pocket

**Acronyms**

ADC  Analog to Digital Converter
APC  Automatic Pressure Control
BR  Ball Riding (Cage Design)
BCB  Bisbenzocyclotene
CAD  Computer Aided Design
COF  Coefficient of Friction
DRIE  Deep Reactive Ion Etching
<table>
<thead>
<tr>
<th>Acronym</th>
<th>Full Form</th>
</tr>
</thead>
<tbody>
<tr>
<td>DSP</td>
<td>Double Side Polished</td>
</tr>
<tr>
<td>HF</td>
<td>Hydroflouric Acid</td>
</tr>
<tr>
<td>ICP</td>
<td>Inductively Coupled Plasma</td>
</tr>
<tr>
<td>IRL</td>
<td>Inner Ring Land (Cage Design)</td>
</tr>
<tr>
<td>KOH</td>
<td>Potassium Hydroxide</td>
</tr>
<tr>
<td>MEMS</td>
<td>Micro-Electro Mechanical Systems</td>
</tr>
<tr>
<td>ORL</td>
<td>Outer Ring Land (Cage Design)</td>
</tr>
<tr>
<td>RF</td>
<td>Radio Frequency</td>
</tr>
<tr>
<td>RIE</td>
<td>Reactive Ion Etching</td>
</tr>
<tr>
<td>RPM</td>
<td>Revolutions per minute</td>
</tr>
<tr>
<td>RF</td>
<td>Radio Frequency</td>
</tr>
<tr>
<td>SEM</td>
<td>Scanning Electron Microscope</td>
</tr>
<tr>
<td>SMT</td>
<td>Silicon Micro-Turbine</td>
</tr>
<tr>
<td>SPI</td>
<td>Serial Peripheral Interface</td>
</tr>
<tr>
<td>USB</td>
<td>Universal Serial Bus</td>
</tr>
<tr>
<td>UV</td>
<td>Ultra Violet</td>
</tr>
</tbody>
</table>
Chapter 1

Introduction

This chapter presents a brief background of MEMS bearings, the motivation for performing the present work, and a list of objectives. The layout of the thesis and a brief description of each chapter are also presented.

1.1 Background

The development of microengineered bearings capable of long-term operation would open up new applications for MEMS technology, particularly in the areas of micropower generation and microfluidics. Research on micro-scale bearings dates back to the late 1980s when the first silicon rotary micromachines were demonstrated [Mehregany 90]. These devices had simple journal or thrust bearings with sliding silicon contacts which showed high rates of friction and wear. Subsequent research on bearings of this type focused on the use of self-assembled monolayers [Maboudian 00] and dry coatings such as diamond-like carbon [Smallwood 06] to reduce sliding friction; however, such coatings suffer from degradation under load and do not allow long-term operation. More recently attention has turned to vapour- [Asay 08] and liquid-phase [Ku 12] lubrication methods for high-sliding contacts as these allow replenishment of the lubricant. Recently a liquid bearing was reported [Chan 11] which has a simple design and does not require external controls, but the performance is highly reliant on the thickness of the
1.1 Background

A fluid film in the bearing and this design could be difficult to encapsulate into a working MEMS device. In parallel with the above, levitation schemes based on electrostatic, magnetic or hydrostatic forces have been developed as presented in [Houlihan 02], [Komori 01], and [Livermore 04]. These are highly effective but require relatively complex control systems and an auxiliary supply of pressurised fluid or electrical power. Aerodynamic bearings as demonstrated in [Frechette 05] and [Lin 99] avoid this control complexity but can be made to work only at extremely high rotation speeds and have a very complex mechanical design.

Microengineered rolling element bearings, such as those presented in this thesis, could provide a viable alternative to the above approaches for applications involving low or moderate rotation speeds (up to around 50 000 RPM). Rolling element bearings achieve low friction and wear by reducing the degree of sliding at the load-bearing contacts; moreover they do not require any external control system or power supply and they can operate over a relatively wide range of speeds. Silicon MEMS rolling element micro-bearings have been under development since the mid 1990s, and in the last few years this technology has matured to the point where micro-ball bearings can be integrated into functional devices (see, for example [Waits 10]). The bearing raceways in these devices are formed by etching annular channels in a pair of silicon wafers. The micro-balls (typically stainless steel) are placed manually in the raceway channels on one wafer before the two wafers are bonded together; a final release etch is then used to free the moving part (inner ring) of the bearing.

This thesis extends the exploration of silicon MEMS micro-ball bearings in several new directions. The first is the integration of ball cage designs into devices with micro-ball bearings. Ball cages are used in macro-scale ball bearings to reduce friction, reduce the number of balls required, eliminate ball-to-ball collisions, and increase the stability of the bearing. This work is the first to explore the design and characteristics of ball cages in micro-scale bearings. A further micro-ball bearing improvement explored in this work is the fabrication of a curved ball race. This is also a common design aspect in macro-scale ball bearings and allows for the creation of self-centering and
angular contact ball bearings. The new bearing designs were fabricated and then tested using an existing micro-turbine design and a micro-tribometer for studying high-sliding contact MEMS devices. The ultimate goal of these micro-ball bearing design changes is to provide a platform for designing reliable and stable rotary MEMS bearings that can be incorporated into devices such as micro-pumps, gears and generators.

1.2 Motivation and Goals

The motivation behind this research was to explore the possibility of transferring common aspects found in macro-scale ball bearings to the design of micro-ball bearings, thereby creating a platform for designing reliable rotary devices for micro-scale applications. The primary goal was to introduce new design and fabrication techniques with broader applications in micro-motors, gears and pumps that will open new possibilities in the design of devices for the space, medical and mechanical engineering fields. This would bring the MEMS field closer to reaching the goal of achieving highly reliable micro-scale rotary devices.

1.3 Research Objectives

The aim of the project described in this thesis was to explore the integration of new geometries and fabrication techniques into the design of micro-ball bearings. The following tasks were performed during this exploration:

1. Characterize the performance of the devices designed for this study by modifying and improving, where possible, existing testing methodologies (Chapter 5)

2. Design and integrate a ball cage into a MEMS micro-ball bearing using the simplest technique possible (Chapter 6)

3. Design a fabrication technique for releasing moving parts from MEMS devices with multiple layers (Chapter 7)
4. Integrate a ball cage into a working MEMS device with a micro-ball bearing support mechanism (Chapter 8.)

5. Create a micro-ball bearing raceway with a curved geometry that will allow for the bearing to self-center and to better mimic race geometries found in conventional macro-scale bearings (Chapter 9.)

The aim of studying these bearing designs and geometries was to determine the feasibility of modifying the design and fabrication of micro-bearings to incorporate features commonly found in macro-scale bearings. Further the study was intended to determine of how these new designs affected the performance of the bearing designs. Though the new geometries explored here are not ideal for all micro-rotary applications, the designs presented have shown that it is possible to integrate many aspects of macro-scale bearings into the design of micro-scale bearings.

1.4 Description of Thesis

The remainder of this thesis is organised as follows:

Chapter 2: Bearings in MEMS Devices - Presents competing MEMS bearing technologies and previous work on MEMS micro-ball bearings.

Chapter 3: Micro-fabrication Techniques and Constraints - Describes the micro-fabrication techniques used to create the devices described in this thesis.

Chapter 4: Ball Bearing Design Consideration and Tribology - Describes the ball bearing design and tribological considerations important to the study presented in this thesis.

Chapter 5: Testing Methodologies - Describes the techniques used to characterize the devices created for this study.
Chapter 6: Proof of Concept of a Radial Ball Bearing with Integrated Ball Cage - Describes the fabrication and performance of the device with the simplest cage integration technique.

Chapter 7: In-Situ Fabrication of a Monolithic Silicon Ball Cage - Presents a new and novel technique for releasing moving silicon structures in devices made from stacking multiple wafers.

Chapter 8: Micro-Turbines with Integrated Silicon Ball Cage - Presents and compares 5 cage designs that were integrated into an existing micro-turbine device.

Chapter 9: Curved Raceway - Presents the integration of a curved raceway geometry, made with deep isotropic etching, into micro-ball bearings.

Chapter 10: Conclusions and Further Work - Presents the conclusions of the studies in this thesis and proposes suggestions for further work.

Bibliography Contains a full list of references cited in this thesis.
Chapter 2

Bears in MEMS Devices

Several MEMS bearings technologies have been explored over the years. This chapter will present several of the competing technologies that have emerged along with previous work on micro-ball bearings.

### 2.1 Competing Bearing Technologies

Research on micro-scale bearings has been reported since the late 1980s. This section will provide an overview of the dominant micro-bearing technologies that compete with micro-ball bearings in the design of rotary MEMS devices. Table 2.1 presents a comparison of the technologies presented in this section, including micro-ball bearings.

<table>
<thead>
<tr>
<th>Technology</th>
<th>Design Complexity</th>
<th>Speeds</th>
<th>Load Cap.</th>
<th>Stability</th>
<th>External Control</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pin Joint</td>
<td>Medium</td>
<td>Low-Medium</td>
<td>Low</td>
<td>High</td>
<td>None</td>
</tr>
<tr>
<td>Mag. Lev.</td>
<td>High</td>
<td>Medium-High</td>
<td>Low</td>
<td>Low</td>
<td>Complex</td>
</tr>
<tr>
<td>Jewel</td>
<td>Low</td>
<td>Low-High</td>
<td>Medium</td>
<td>Low</td>
<td>None</td>
</tr>
<tr>
<td>Air</td>
<td>High</td>
<td>High</td>
<td>Low</td>
<td>Low</td>
<td>Complex</td>
</tr>
<tr>
<td>Liquid</td>
<td>Medium</td>
<td>Low</td>
<td>Low</td>
<td>Low</td>
<td>None</td>
</tr>
<tr>
<td>Micro-Ball</td>
<td>Low</td>
<td>Low-Medium</td>
<td>High</td>
<td>High</td>
<td>None</td>
</tr>
</tbody>
</table>

Table 2.1: Comparison of micro-bearing technologies.
2.2 Pin Joint Bearing

One of the earliest reported rotary MEMS bearings was the pin joint bearing. In this design the rotor is fabricated around a "pin" that is attached to the stator. A cap is then fabricated on the top of the pin to keep the rotor in place. A typical fabrication technique was to fabricate the rotor in polysilicon on a sacrificial silicon dioxide layer as seen in [Mehregany 98] and [Garcia 95]. However the use of thick metal layers to form the bearing structures with a PMMA sacrificial layer has also been reported in [Guckel 98]. This design can only support very low loads due to the large surface area of the contacts at the interface of rotor and stator. The contact surface area limits the load due to the fact that high loads will cause the friction and wear at the interface to increase dramatically. The effects of wear have been observed in several applications using the pin joint bearing. In [Mehregany 90] it was found that the performance of a wobble motor increased over time due to wear increasing the bearing clearance; however, it was also observed that the particles from this wear process could seize the bearing. In this study no devices failed due to failure of the pin or the joint structures. Further, the observation of excessive wear causing failure due to rotor misalignment, making it impossible to actuate the device, in micro-turbines and gears after around 1 million revolutions was reported in [Gabriel 90].

Sandia National Labs has performed the most extensive tests of polysilicon pin joint structures with a rotor actuated by a set of capacitive linear drives connected by a pin joint bearing. In [Garcia 95], it was shown that a silicon nitride on polysilicon joint was superior to a polysilicon on polysilicon one. This was attributed to the formation at fewer particulates and improved tribological characteristics at the polysilicon to silicon nitride interface. The improved device was observed to survive over 2.8 million cycles at 1500 RPM. These devices also suffered from seizure due to wear particles collecting at the bearing interface. A further characterization of the failure modes of these devices, presented in [Miller 97], seems to indicate that asperities formed between sliding contacts cause sticking and ultimately failure in the motors. In other work [Paterson 97], it was difficult to observe wear on the bearing
contacts with SEM imaging and it was considered that failure modes often had more to do with electrostatics or drive failures, rather than failures in the bearing. A later study on loaded gears [Tanner 98] did show that the dominant failure mode in this type of bearing was in fact due to wear at the bearing interfaces. These results were further reported in [Tanner 00] and [van Spengen 03]. Figure 2.1 shows the design of the gear and the wear observed at the bearing interfaces. Later it was determined [Tanner 02] that the dominant design parameter for increasing the longevity of the bearing was an increased gap spacing in the bearing which reduces the contact time and friction between the rotor and stator hence reducing wear. The reduced wear also reduces the generation of particles that can seize the bearing and the loss of materials from the rotor a stator surfaces that can lead the ultimate mechanical failure of the parts.

Figure 2.1: A. Image of the gear arrangement in the polysilicon pin joint studies; B. Excessive wear found on drive gear pin hole; C. Undamaged pin hole on side gear to compare with above image; D. Cross section of the pin joint bearing interface showing unworn structure (top) and worn structures (bottom). Images reproduced from [Tanner 98]
2.3 Magnetic or Electrostatic Levitation

Magnetic and electrostatic levitation seem to be very compelling technologies for use in MEMS devices. With the low loads and forces experienced at the micro-scale levitation should be able to provide a reliable low friction bearing. The major challenges to using this technology, however, are incorporating permanent magnetic materials into the fabrication process (though this can be accomplished manually) or incorporating the control system required to keep the rotor suspended and stable during rotary operation. Examples of magnetic levitation systems in MEMS are described in [Komori 01] and [Houlihan 02]. These devices operate best at moderate to high speeds, and though the technology has been proven to be conceptually sound it has not received widespread acceptance as a MEMS bearing technology. A less complex method used to electrostatically levitate a simple proof mass in a power harvesting application has also been shown to be effective in eliminating stiction in [Suzuki 10]. Electrostatic levitation has also been demonstrated in MEMS gyroscopes in by using planar coils to levitate and stabilize an aluminium rotor as reported in [Williams 96] and [Shearwood 00]. This micro-motor was able to reach speeds of approximately 1000 RPM in air and was tested at to over 200 hours with no indication of degradation in performance. A final promising approach to creating levitation bearing is the use of diamagnetic levitation. This method uses static fields that require no external controls. A proof of concept design for a levitated accelerometer is presented in [Garmire 07] and a capacitive drive micro-motor is presented in [Liu 08]. A model for improving the performance of MEMS accelerometers using this technique is presented in [Pasquale 09]. Though these initial results are encouraging for broader application of this technique, there is no evidence yet of a fully integrated MEMS system using this technology.

2.4 Jewel Bearing

The jewel bearing has been widely used in watch making for centuries. This simple bearing requires only a pair of hard, smooth, cupped surfaces (jewels)
placed on stator platforms above and below a pin attached to the rotor. A conceptual drawing of this design is shown in Figure 2.2. This is a relatively simple design but requires at least 3 surfaces in order to work properly, which could increase the complexity of MEMS device design. The three surfaces involved would be the top jewel surface (stator), the rotating device in the center (rotor) and the bottom jewel surface (stator.) This design also can allow wobble and instability that is unfavourable in some MEMS applications. However, the simple design, wide range of acceptable speeds, low wear, and lack of external controls make this an attractive bearing technology. Some examples of this application in MEMS and centimeter scale applications can be found in [Lee 05], [Leivuo 06], [Romero 09] and [Bansal 09].

Figure 2.2: Conceptual drawing of a common jewel bearing design, with jewels forming a cup on the stator to hold a pin attached to the rotor.

2.5 Air Bearing

High-speed micro-turbines supported on an air bearing were first explored in the late 1990s by Massachusetts Institute of Technology. The stability of this type of bearing relies on carefully controlling the air pressure at several points in the device. Hydrostatic thrust bearings levitate the rotor at the center. Journal bearings at the periphery of the rotor are needed to keep the rotor
from colliding with the stator during rotation. A thrust plenum balance is needed to keep the bottom of the rotor from colliding with the stator under load. And finally there is the main inlet pressure which drives the turbine. Figure 2.3 shows the cross section of a silicon micro-turbine supported by an air bearing. Disadvantages of this approach are the complexity of the bearing design (requiring a 5 wafer stack), the need for closed loop control (the stability of the bearing is highly dependent on maintaining tight control over, and balancing, the air pressures in the device), and the fact that the rotor must necessarily collide with the stator when the air pressure is removed from the device.

Figure 2.3: Optical photograph (top) and schematic (bottom) of the cross section of a silicon micro-turbine supported on an air bearing. Diagram reproduced from [Frechette 05].

In the first reported results for a device of the type shown in Figure 2.3 in [Lin 99], the turbine was able to achieve speeds of up to 60,000 RPM. The upper speed and stable operation of the bearing were limited by the stiffness of the air bearing, and it was difficult to maintain rotation without the rotor touching down on the stator. With fabrication improvements and optimization of the bearing stiffness speeds of up to 1,400,000 RPM were achieved.
and reported in [Frechette 00]. In [Frechette 05], it is reported that several of the devices were still limited to speeds below 60,000 RPM, and all of the devices (high and low speed) could suffer from catastrophic failures typically attributed to rotor instability. The need for highly stable and precisely fabricated rotors required extensive optimization of the fabrication parameters, with special emphasis on the DRIE (deep reactive ion etching) processes. A further exploration of the DRIE optimization and tight fabrication tolerances is reported in [Kang 05] for a similar device used to explore MEMS turbochargers. Another study, [Hara 03], explored 3 types of journal bearing structures, finding that an asymmetrical multi-lobed journal bearing would be the best candidate for high-speed turbine operation. It was also observed that the air bearing operated in 3 distinct modes, as shown in Figure 2.4. Mode C is the most desirable mode due the fact that contact is made at the journal bearing and not at the top or bottom surface of the rotor allowing for the most stability and freedom of movement. Further in this mode only the journal pressure will need to be optimized to compensate for the centrifugal forces.

Work on creating working MEMS engines and generators struggled primarily with the rotor dynamics and operation of the air bearing support mechanism [Epstein 04]. Integration of inductive coils needed to create a generator or motor from these devices has been demonstrated in [Livermore 04]. However, optimization of the bearing stability and control still limit the application of this type of bearing.
A recent exploration of liquid bearings is presented in [Chan 11]. This bearing uses water or ethylene glycol as the bearing support mechanism. A hydrophobic (Cytop) layer is used to keep the bearing in a fixed location. This technique relies on the surface tension of the liquid to support the rotor of the device, making drag and load capacity dependent on the liquid film thickness. The results reported show a drag reduction of over 15 times compared to micro-ball bearing designs of similar dimensions. It is unclear, however, how a fully encapsulated liquid bearing could be incorporated into a MEMS device as the liquid would have to integrated into the device at
some point in the fabrication process and would have to be maintained at
the rotor and stator interfaces. This design method is attractive because it
requires no external control mechanisms and provides the reduced drag and
friction of air bearings with much less design complexity.

2.7 Previous Work in Micro-Ball Bearings

MEMS micro-ball bearings have been under investigation by the University
of Maryland since the late 1990s. Bearing manufacturers have also been pro-
viding micro-ball bearings, fabricated through more conventional methods,
for many years. This section will present an overview of the MEMS micro-
ball bearing research and will also some details on commercially available
micro-ball bearings.

2.7.1 Linear Micro-Ball Bearing

Linear ball bearings were the first type used to characterize micro-ball bear-
ing performance in MEMS devices, the first results appearing in the early
2000’s with [Lin 01] and [Lin 02]. In [Lin 04] linear bearings with V-groove
raceways, created by isotropic KOH etching, were studied in-situ using a
visual processing technique to measure the displacement of the slider in re-
lation to the stator. This first investigation showed a significant reduction in
the COF (coefficient of friction) between structures with micro-ball bearings
in relation to those with sliding contacts. The same system was also used to
further characterize the hysteresis present in this linear bearing system, as
described in [Tan 04]. A further study, leading to the development of a model
for the friction and hysteresis observed in the linear micro-ball bearings, was
presented in [Tan 06a].
2.7 Previous Work in Micro-Ball Bearings

A linear bearing was first integrated with an electrostatically driven micro-motor in [Modafe 06]. The bearing grooves had straight side walls, fabricated with DRIE, rather than the V-grooves used in earlier work due to an incompatibility between the KOH etching needed to create the V-grooves and the thick BCB used for electrical isolation in the electrostatic drive. Figure 2.5 shows a diagram of the linear electrostatic motor with the square wave driving signals. This device did perform as predicted when allowance was made for flaws relating to the fabrication process. A better agreement with theory was reported in [Ghalichechian 07]. Due to this understanding and characterization of dynamics of the bearing, a closed loop control system could be
2.7 Previous Work in Micro-Ball Bearings

integrated into the linear motor that provided positional control with a resolution of 120µm over a total travel of 2mm, as reported in [Beyaz 09]. This work provided initial confirmation of the performance and controllability of MEMS devices supported on micro-ball bearings.

2.7.2 Rotary Micro-Ball Bearings

The first exploration of a rotary micro-ball bearing was reported in [Waits 07a]. This was a simple bearing integrated into a square chip. The device was actuated by placing the outer corners of the device die into a flow of pressurized nitrogen. This first bearing suffered from ball jamming, extensive wear at the bonded wafer interface and wear on the steel balls, as shown in Figure 2.6. This design was further refined to produce a 6-phase capacitive micro-motor (similar in design to the linear motors mentioned in the previous section), as reported in [Ghalichechian 08]. These experiments showed that the performance of the bearing was in agreement with the previously developed models and proved that micro-ball bearings could be integrated and encapsulated for use in MEMS devices.

Further research involved integrating the micro-ball bearing into devices with micro-turbines in the rotor. The first turbine designs used a tangential flow to drive the turbine, see [Waits 07b]. This turbine still suffered from poor performance. A dramatic improvement in performance was realized when the turbine design was changed and the ball bearing raceway contact surface was changed to more closely resemble that of a thrust bearing, as shown in [McCarthy 09], reducing friction and wear. Moving the bond interface away from the center of the ball to reduce the wear at the bond interface increased the longevity and performance of the bearing, as reported in [Hanrahan 10]. These changes have finally enabled the micro-ball bearing to achieve a level of reliability and performance acceptable for integration into Power MEMS devices. Recently magnets and coils have been integrated into the SMT device for power generation [Beyaz 10].
2.7 Previous Work in Micro-Ball Bearings

Figure 2.6: A. Optical image of ball jamming in the bearing raceway; B. SEM image of ball without wear (top) and ball after 39 min. operation (bottom); C. SEM image of the unworn bonded silicon raceway interface (top) and the worn bonded silicon raceway interface. Images reproduced from [Waits 07a].

2.7.3 Conventional Micro-Ball Bearings

Conventional or commercially available micro-ball bearings require special manufacturing techniques, equipment and hand assembly. The manufacturing process can take between 8-12 weeks and must meet extremely tight tolerances. These bearings can be found in applications such as medical pumps, dentist drills, aerospace applications, watches and office equipment [McCann 09]. The complex manufacturing process could benefit from the batch fabrication techniques made available by using silicon as the raceway material, thereby reducing the overall time and cost of production. The extremely tight tolerance required for these bearings can also be met easily with the
fabrication techniques proposed in this and other MEMS micro-ball bearing research.

As an example of a commercially available ball bearing, one part listed by NSK Micro Precision Co., has an inner diameter of 0.6 mm, outer diameter of 2 mm, and a ball diameter of 0.3 mm [NSK 12]. NMB Technologies Corporation list a bearing with an outer diameter of 1.5 mm, inner bore of 0.5 mm and a ball diameter of 0.25 mm [NMB 12]. MPS, a member company of the Faulhaber Group, lists a bearing with an outer diameter of 1.6 mm, inner diameter of 0.3 mm, and ball diameter of 0.2 mm [MPS 12b]. MPS provides a large selection of micro-bearings for watches, medical equipment and other applications. The Timken S100 ball bearing, pictured in Figure 2.7, has an outer diameter of 2.5 mm, an inner diameter of 0.6 mm and a ball diameter of 0.6 mm. Information on Timken bearings and applications can be located at [Timken 12].

Figure 2.7: Picture of the Timken S100 micro-ball bearing, photo by Timken Co.
2.8 Discussion and Conclusions

Several bearing technologies have been proposed and tested for MEMS systems. Many of these techniques have limitations that would make them unacceptable for harsh environments or suffer from catastrophic failure over a short period of time. The pin joint bearing can only support light loads and fails due to wear at the rotor and stator/pin interface. Magnetic and electrostatic levitation bearing also support low loads and can require complex external controls. Though the use of diamagnetic materials could lead to levitation bearings that do not require any external controls the technique has not yet been demonstrated in a fully integrated system. Jewel bearings could provide a stable and low friction bearing with a long service life, however, this design has not yet been integrated in a MEMS device. Air bearings also offer a low friction bearing, but only work at higher speeds and suffer from low stability that can only be overcome with complex designs and external control systems. And finally, the liquid bearing which can also only support low loads and have not yet been fully integrated into a MEMS system.

Micro-ball bearings show a high stability over a wide range of operating speeds and applied loads with no external controls. The mechanical support in the bearing also allows devices to survive shock and vibration during operation that would cause catastrophic failures in several of the other bearing designs. The initial results have indicated that this low complexity design has broad application in rotary and linear MEMS actuators. Ball co-location, which resulted in the seizure of the bearing, was a problem in the early micro-ball bearing designs. The research this thesis will explore the use of ball cages or retainers, common in macro-scale bearings, to solve this problem. Further, a curved raceway will also be presented that will more closely match the geometry found in macro-scale ball bearings and allows the bearing to self-centre during operation. These improvements are intended to improve the longevity and performance of the bearings in MEMS applications.
Chapter 3

Micro-Fabrication Technologies

This chapter will outline the micro-fabrication techniques used to make the devices described in this thesis. A brief and general description of each technique will be presented with an explanation of the constraints of the technique as applied to this research. It should be noted that the techniques used were often chosen due to the availability of suitable equipment in our laboratory and not always because they were the first choice for accomplishing the step. Therefore, process and device design should be considered in the light of the restrictions of the processes described in this chapter. These restrictions inevitably lead to added complexity in some of the processing described in later chapters.

3.1 Patterning and Masking

The first step in a MEMS fabrication process involves creating a set of masks with the patterns required to define the desired device. The patterns on the masks are transferred to the substrate at different stages in the process flow using a photolithographic process. In the research described in this thesis all masks were created with 1:1 scale and the patterns transferred using a contact mask aligner. Other techniques such as proximity printing, projection and stepper printing were not used. For a more detailed description of photolithographic processes and technologies the reader is encouraged to
3.1 Patterning and Masking

review Chapter 2 in [Jaeger 02].

3.1.1 Photo-Masks

Two types of masks were used for the photolithography process: chrome/glass and polymer. Chrome/glass masks are made from a soda lime glass with a thin (typically 980 Å) layer of chrome which is patterned by direct-write lithography using either a laser or electron beam. These masks provide a high resolution, and are durable and easy to clean. The chrome/glass masks used in this work were printed by Delta Mask [DeltaMask 12] and had a printable resolution of 1 μm. In order to explore multiple designs, and to create the more complex multiple-mask designs at a lower cost, polymer masks were also used. These masks were printed by JD Photo-Tools [JDPhoto 12], with a photo-emulsion film on a polyester-based film. These masks must be handled carefully because the emulsion film is soft and easy to scratch and cannot be cleaned. The resolution of the masks can be as fine as 8 μm; however, they can exhibit dimensional instability under varying humidity and temperature conditions. The key advantage of polymer masks is that 24 masks for 100 mm diameter wafers can be printed for the same cost as 1 chrome/glass mask.

3.1.2 Photolithography

Photolithography is the process of transferring the pattern from the mask to a layer of photoresist on the substrate. In a mask aligner this is accomplished by aligning the mask to features on the substrate with the aid of a microscope, bringing the mask and substrate into contact, and then exposing the photoresist to UV (ultra violet) light incident through the mask. The exposure step produces a latent image in the photoresist layer which is converted to a topographic pattern when the wafer is immersed in a developer solution.

There are two common types of photoresist: positive and negative. In a negative resist the UV exposure results in cross-linking of the photoresist making it less soluble in developer. While in positive resists the UV exposure makes the photoresist more soluble in the developer. Therefore, after development, negative resists will remain in regions corresponding to the clear
areas of the mask, while positive resist will remain in regions corresponding
to the dark areas \cite{May06}. For this research positive photoresist was used
for all pattern transfer.

### 3.1.3 Masking Materials

Two types of masking materials were used for the processes described in this
thesis. Photoresist was used to define regions for electroplating, RIE (reactive
ion etching) of silicon dioxide and DRIE of silicon, while silicon dioxide was
used as an alternative mask material for RIE and DRIE etching of silicon.
These masking material will be described in the following sections.

#### 3.1.3.1 Photoresist

Photoresist is used to transfer the mask pattern to the substrate for sub-
sequent processing. For the fabrication processes described in this thesis, a
thick photoresist was required that could survive highly aggressive etching
and also act as a template or mold for thick (\(>1\ \mu\text{m}\)) metal electro-plating.
For all of these steps it was easiest to use one type of photoresist with an
optimized process for our specific laboratory equipment. As a result it was
determined that AZ9260 was the ideal photoresist for all of processes used
to fabricate the devices. This resist can be used to create thick coatings, and
it has a very high resistance to plasma etching (with selectivity of at least
80:1 observed in DRIE etching) as well as a high resistance to electroplating
bath chemistry; it also provides excellent photo-patterning capabilities. A
guide on recommended processing steps for AZ9260 positive photoresist can
be found in \cite{MicroChemicals09}. The standard process used for AZ9260 in
this thesis was provided by Dr. Werner Karl \cite{Karl08} and was optimized
to provide a coating with a thickness of approximately 10\(\mu\text{m}\) on the equip-
ment in the laboratory. Some minor optimizations were made to the coating
process over time to improve throughput and reduce processing time when
applying photoresist to multiple substrates. The AZ9260 coating process is
detailed in Appendix A.
3.1.3.2 Silicon Dioxide

Silicon dioxide was used as a resilient hard mask for several DRIE processing steps. Some of the advantages of silicon dioxide as a masking material are:

1. It can be grown on silicon by thermal oxidation. (A 1 µm layer can be grown in an oxidation furnace at 1100 °C, with 11 min⁻¹ Oxygen flow, for 48 h.)

2. It is able to survive high temperature processing.

3. It is unaffected by, or minimally affected by, most chemistry used in other processing steps (i.e. metal etching with acids, electroplating bath chemistry, photoresist stripping).

4. It offers high selectivity against silicon during RIE and DRIE processing, with a selectivity of 50:1 observed during DRIE etching.

5. It can be patterned by depositing a photoresist mask then etching the exposed oxide using HF wet etching or RIE dry etching. Mask resolution is limited only by the oxide thickness, the anisotropy of the etch process and the resolution of the photoresist mask.

In the process flow of the devices described in this thesis, a silicon dioxide mask is used during the final etching step. This is due to the fact that the mask can be defined on the wafer before it is broken into die for assembly. The hard mask can then survive handling, assembly and the high temperature wafer bonding. Different layers of silicon dioxide, photoresist and exposed silicon are also used in some processes to create stepped geometries in the silicon substrate.

The silicon dioxide masks used in this thesis were created by first defining a mask in AZ9260 and then transferring the pattern into the oxide by reactive ion etching the exposed silicon dioxide using an Oxford Instruments System 80 etch tool. The basic theory behind plasma etching can be found in Section 2.1.3.2 of [May 06]. In this process we used $CHF_3$ to create the etchant species in the plasma. Due to a characterization of the etching processes
found in [Kumar 07], it was determined that 2 min etch steps should be interspersed with 2 min cooling steps. This prevents resist damage induced by overheating during the etching process. This process provided excellent pattern transfer from AZ9260 mask to the oxide layer. Minimal undercut was observed and the photoresist patterns transferred within the required design tolerances. It was also determined that HF wet etching of the oxide would not provide the desired pattern resolution.

3.2 Deep Reaction Ion Etching (DRIE)

Deep Reactive Ion Etching (DRIE) is one of the key processes that has made this research possible. The DRIE process used for this research was the Bosch process, described below. Therefore all references DRIE in other sections of this thesis are referring to the Bosch process. This process allows for creating high aspect ratio, deep, anisotropic features in a silicon substrate. The section will give a brief overview of the principles of this process, processing considerations for through wafer etching, the use of halo masks to improve etch uniformity in through wafer etching, and the effects of etch lag.

3.2.1 Basic Principles

The Bosch DRIE silicon etching process was used in this research to define anisotropic trenches in the silicon substrates. Chapter 4 of [Kumar 07] has an excellent overview of the processes, design considerations and equipment used for the research presented here. The Bosch DRIE process achieves deep anisotropic etching using a highly reactive gas to create a high density plasma to etch the features in to the silicon while also interleaving passivation steps that protect the sidewalls from isotropic etching. Due to the inherent isotropic etching of the stable flourochemicals typically used in this process a polymer passivation layer is required to protect the sidewalls of the feature from the etchant species. In the Bosch process this is accomplished by time-multiplexed or pulsed etching, in which the substrate is first etched and then a passivation layer is deposited. [Schilp 96] An excellent overview of the Bosch
3.2 Deep Reaction Ion Etching (DRIE)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Dark3S</th>
<th>Dark4S</th>
</tr>
</thead>
<tbody>
<tr>
<td>Etch Step (s)</td>
<td>11</td>
<td>13</td>
</tr>
<tr>
<td>Passivation Step (s)</td>
<td>8</td>
<td>6</td>
</tr>
<tr>
<td>Etch Rate ($\mu$m min$^{-1}$)</td>
<td>3</td>
<td>4</td>
</tr>
<tr>
<td>$SF_6$ Flow Rate</td>
<td>130 sccm</td>
<td></td>
</tr>
<tr>
<td>$O_2$ Flow Rate</td>
<td>13 sccm</td>
<td></td>
</tr>
<tr>
<td>$C_4F_8$ Flow Rate</td>
<td>85 sccm</td>
<td></td>
</tr>
<tr>
<td>RF power</td>
<td>600 W</td>
<td></td>
</tr>
<tr>
<td>RF frequency</td>
<td>13.56 MHz</td>
<td></td>
</tr>
<tr>
<td>Platen Power</td>
<td>20 watt</td>
<td></td>
</tr>
<tr>
<td>APC setting</td>
<td>75% (manual)</td>
<td></td>
</tr>
</tbody>
</table>

Table 3.1: DRIE process parameters for anisotropic etching for recipes named Dark3S and Dark4S.

DRIE history and process considerations can also be found in [Laermer 10].

In this research a Surface Technologies Systems multiplex ICP etcher was used for the DRIE processing. $SF_6$ was used as the etchant gas and $C_4F_8$ was used to deposit the passivation layer. The recipes typically used for the DRIE processes described in this thesis are presented in Table 3.1. The Dark3S recipe has been optimized for etching 40$\mu$m-wide features through a wafer, and also performs well when all of the etched features are below 150$\mu$m. Dark4S performs well for features larger than 150$\mu$m, but does have a tendency to create footing or widening of the trench as the etch depth increases. The primary difference between the two recipes is the ratio of the etch step to the passivation step. The etch rates given are typically valid to a depth of approximately 250$\mu$m, at which point the etch rate seems to drop off markedly and at a rate that is highly dependent on the width of the etched feature.

3.2.2 Through-Wafer Etching

One method used to create the test suspension and ball cages presented in this research was to etch the structures from a silicon substrate. Etch lag becomes a significant factor during through-wafer etching, where larger features will etch at a higher rate than smaller features. This leads to the
need for using a halo mask design to ensure a near-constant etch rate for all of the features on the wafer during a through-wafer etch. In [Pike 03] the optimization of the DRIE etching parameters and mask design for through-wafer etching are discussed. The results of these optimization studies were used for through-wafer etching as presented in this thesis and also in [Ku 10b] and [Kumar 07].

The basic concept of the process involves patterning resist on the substrate with the desired features. The features are then etched to a depth that is typically greater than half of the thickness of the substrate. The substrate is then mounted on a backing wafer, typically a silicon substrate, using photoresist as a temporary bonding adhesive. The features are then etched the rest of the way through the substrate. The wafer stack is then soaked in acetone or photoresist stripper to detach the backing wafer and to remove any parts of the substrate that are no longer attached to the devices as a result of the etching.

3.2.3 Halo Mask for Through Wafer Etching

In order to overcome the effects of etch lag while trying to remove significant portions of silicon during a through-wafer etch a "halo" mask can be used. This type of mask is designed so that all of the features on the mask are of similar width. Large areas of silicon that need to be removed from the device are surrounded completely by a narrow trench so that they will drop out at the end of the process. This technique is used in, for example [Bayt 98], [Kumar 07], and [Pike 03]. Figure 3.1 shows the concept of this type of mask design.
3.2 Deep Reaction Ion Etching (DRIE)

Figure 3.1: Halo mask concept (left) with the etch trenches in white, the device structure in pink and the sacrificial substrate in green. SEM (right) of a device fabricated by through wafer etching using a halo mask.

3.2.4 Etch Lag

DRIE etch lag is an important factor that must be considered when designing devices that rely heavily on the geometries of high aspect ratio anisotropic trenches. Features with different widths will etch at different rates during DRIE processing. An illustration of this phenomenon is shown in Figure 3.2. In [Chung 04] it is shown that, regardless of the trench shape, the etch rate depends on the area of the trench. This effect is primarily due to ion and radical or inhibitor depletion during the DRIE process [Gottscho 92]. Differential charging of insulators, field curvature near conductors, image force deflection, and ion shadowing with ion angular distribution are responsible for ion depletion. Radical/inhibitor shadowing, molecular flow, bulk diffusion and surface diffusion are responsible for radical and inhibitor depletion [Jansen 97]. Optimization of DRIE parameters to accommodate differently sized high aspect ratio features has been explored in [Ayon 01] and this effect has also been used in the fabrication of 3-dimensional structures [Rao 04]. A model is also available in [Tan 06b] for predicting and simulating the effects of etch lag.
3.3 Electroplating

For many of the devices described in this thesis multiple DRIE steps were used to create stepped profiles in the silicon substrate. The design of the device features and etching process is further complicated, due to etch lag, because as the stepped profile in the silicon is etched deeper the effects of etch lag are enhanced on the narrower features patterned in the earlier layers. This lag requires that the narrower and deep features must be initially etched to a much greater depth in order to retain the desired final etch profile. A further challenge in most of the designs presented in this thesis is that many of the device features are of very different sizes. During some of the etching processes trench widths can range from 40 µm to 510 µm. These factors are critical considerations during the design of the masks and process flow for a device.

3.3 Electroplating

Electroplating or electrodeposition of metals onto a substrate is a common method of creating interconnects and depositing metals for wafer bonding.
3.3 Electroplating

<table>
<thead>
<tr>
<th>Plated Metal</th>
<th>Current Density ($mA/cm^2$)</th>
<th>Deposition Rate ($\mu m/min.$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nickel</td>
<td>10</td>
<td>0.25</td>
</tr>
<tr>
<td>Tin</td>
<td>5</td>
<td>0.2</td>
</tr>
<tr>
<td>Gold</td>
<td>2</td>
<td>0.1</td>
</tr>
</tbody>
</table>

Table 3.2: Electroplating parameters for the metals used in the fabrication processes in this research.

applications. A detailed description of the process can be found in Section 4.3.8.3 of [Gad-El-Hak 06]. For the work described in this thesis three metals were deposited using this technique: nickel (Ni), tin (Sn), and gold (Au). These metals were used to bond the die together as described below in Section 3.4.

In order to electroplate materials on a substrate with an insulating layer (in the case of this research a 1 $\mu$m thick layer of $SiO_2$) a conductive layer must be deposited on the substrate. For this application a 60 nm thick layer of Chrome was deposited as an adherence layer and a 100 nm thick layer of Copper was deposited as the conductive seed layer. These metals were deposited using a radio frequency (RF) sputter coater. The seed layer was then coated with AZ9260 photoresist and patterned with the desired bonding pad design. The metals (Ni, Au, Sn) were then electroplated into the patterned regions. Specific current densities were needed for each plated metal in order to control the surface quality and the plating rate of each metal. The optimal current density and plating rates were provided by Dr. Anisha Mukherjee [Mukherjee 09]; these were determined experimentally for the plating equipment available in the laboratory. The plating conditions for each metal are presented in Table 3.2.

After the patterned metals have been plated on the substrate the plating seed layer must be removed in order to perform additional processing on the wafers. In this work the seed layers were etched using a wet submersion etch. The etchants used for each metal are listed in Table 3.3. The best etchants for this process were determined experimentally and through detailed discussions with Dr. Munir Ahmad [Ahmad 09]. This process has been used to pattern
3.4 Eutectic and Solder Wafer Bonding

A method for bonding two or more silicon substrates together was needed for all of the devices described in this thesis. Tin-gold solder bonding was chosen as the bonding technique because of the availability and ease of electroplating the solder metals onto a silicon substrate. The solder layer was also able to withstand all of the subsequent processing steps, such as DRIE, plasma cleaning, resist stripping and masking steps. This was a key advantage that allowed for depositing the solder layer as the first step. This ensured that the solder was as planar as possible and was located at the wafer to wafer interface, while still allowing all of the processing steps to be performed using standard resist spinning and lithographic processes.

The concept behind eutectic solder bonding is that metal alloys will be formed between the eutectic metals, thereby providing advantages such as a higher or lower melting point, improved ductility, some self alignment capability during bonding, and enhanced reflow characteristics [Ramm 12]. All of these advantages are compounded by the advantageous processing parameters such as low to no bonding force, low temperature reflow (approximately 280°C for 20%/80% Sn/Au solder) and the ability to bond in a controlled or ambient air environment. Au/Sn solders also provide an excellent replacement for Pd based solders. Fluxless soldering of devices using Tin-rich solders has been demonstrated in air [Chuang 04]. Further a 20% Sn 80% Au is compared in hydrogen, nitrogen and air environments in [Kallmayer 96].

It has been determined that a nickel diffusion barrier between the solder and
the substrate can improve the quality and reflow of the Au/Sn solder pad over that of a copper barrier [Tsai 06], which led us to use a nickel diffusion barrier for all of our bonding processes. For further information on and characterizations of different types of Au/Sn bonds please refer to [Welch III 08].

For the devices presented in this thesis we were able to obtain a yield of 75% using a standard plated Au/Sn bond process in ambient air. The highest yields were obtained when a solder flux was applied to the bonding pads. To improve the yield Sn-3.0Ag-0.5Cu solder balls (the same as used in [Gu 09]) and solder flux were added to the solder stack. This increased the bonding yield to 95% when bonding in air.

3.5 Conclusion

The design of the devices presented in this thesis were based on the availability of the technologies described in this chapter. Design constraints and geometries were all determined by evaluating the capabilities and limitation of the micro-fabrication technologies. Several new fabrication techniques were devised using these base technologies that allowed for the creation of unique geometries and etching profiles. Examples include optimization of stepped etch profile geometries using multiple oxide masks, the creation of a curved annual bearing raceway and the in situ fabrication and release of embedded moving parts in a multiple wafer stack. The body of work presented here shows how flexible these basic techniques can be with an understanding of the underlying principles, process optimization and a bit of creativity.
Chapter 4

Ball Bearing Design Considerations

This chapter will cover the design parameters considered while creating the devices described in this thesis. The design of the bearings is based on general rules and suggestions found in earlier studies.

Though many models exist for ball bearings and for various facets of ball bearing designs, many of these models rely on conventional bearing designs and geometries that include curved raceways, demonstrated only at the end of this research, and lubrication which was not used in these studies. As the macro-geometry and design of the fabricated bearings do not match many of these design constraints in existing models it was difficult to use existing models to evaluate the bearing designs. Further, most models rely on the assumption that the balls are in pure rolling conditions, which will be shown not to be the case in the bearings studied here. A simplified model for the bearing loss is presented here. This model is used as an initial estimate for the bearing loss and will be used to evaluate and explain the empirical results presented later in this thesis.
4.1 Ball Cage

Ball cages are ubiquitous in traditional ball bearings. They help to ensure even load distribution, and prevent collisions between the balls which can lead to increased losses and wear. Some examples of ball cages that are used in macro-scale bearings can be seen in Figure 4.1. The cage design depends heavily on the bearing speed, application, temperature rating, cost and overall desired precision.

Ball cages should be less hard than the rolling elements, and common retainer materials are pressed steel (low-cost bearings), bronze or brass [Harris 06b]. Polymers are also used in bearings not intended for high temperature operation [Katagiri 04] and can offer advantages such as reduced friction and lower (cage-ball) collision noise, in addition to lower production costs. Polymeric cages can be fabricated into complex shapes and do not have the debris associated with metal cages. A wide range of polymers can be used including Nylon (polyamide) 6,6 and even high temperature polymers such as Peek [Harris 06b]. The greater mechanical flexibility of polymer cages can also be an advantage as it lowers the ball-pocket forces and the shape can assist in lubricating the rolling elements [Weinzapfel 09]. Unfortunately, in this research we violate the rule of having cage material that is softer than the rolling element. Silicon is used for the cages due to the ease of fabrication and design of silicon cage structures. Future works will explore the integration of polymers, metals and lubricating coatings into the cage designs. However, due to the added complexity of implementing these designs they have not been explored at this time.

In designing the cage geometries explored in this thesis we used certain design rules. The cage pocket clearance was limited to at most half of the diametral clearance of the bearing. In [Gupta 91] it was determined that higher ball pocket clearances resulted in the appearance of greater cage instability resulting in undesirable lateral and vertical movement. This design rule was also based on design recommendations found in [Khonsari 08]. A lower frictional coefficient between the cage and the ball is also expected to improve performance as shown in [Meeks 85b], [Meeks 85a] and [Kannel 78].
As no lubrication was used in our studies every attempt was made to reduce the cage contact area and ball forces by reducing the thickness of the cage when possible. This design approach also provides a more flexible cage which should reduce ball-pocket forces and improve the steady state operation of the bearing [Weinzapfel 09]. In general all of the cages and bearings were designed so that the maximum cage excursion would not allow the cage to collide with the inner or outer land during operation.

In addition to improving performance, cages could substantially lower the cost of silicon micro-ball bearings by reducing the number of expensive precision micro-balls required; in the absence of a retainer, the raceway needs to be substantially full in order to avoid excessive vibration and loss of load-bearing capacity if the balls become redistributed. Longer term the incorporation of cages could also facilitate the integration of intra-bearing sensors, opening up the possibility of new types of “smart” bearing that can provide real-time information about the bearing status, for example measurements of temperature and/or vibration. Such data might be useful both for tribological studies and for condition monitoring when the bearing is in service. Macro-scale smart bearings have been demonstrated, such as the one in [Holm-Hansen 00]. Here the outer ring of a bearing was modified to incorporate a piezoelectric sensor that could monitor dynamic load variations. In a silicon microengineered bearing, sensors with wireless power delivery and data transfer could potentially be integrated directly into the retainer, providing additional data on the moving parts.
4.1 Ball Cage

Figure 4.1: Examples of macro-scale cage designs. A. Nylon snap cage for ball bearing, B. Nylon cage for cylindrical bearing, C. Nylon cage of high-angular-contact bearing, D. Phenolic cage for precision ball bearing, E. Angular contact bearing with annular cage, F. deep-groove Conrad-assembly bearing with riveted cage, G. Dual row ball bearing with snap cage, H. Thrust ball bearing with annular cage design. Images reproduced from [Harris 06b].
4.2 Raceway and Roller Element Materials

Bearing components are made from various materials based on the bearing application and operating environment. In common lubricated bearings hardened stainless steels are often used for both the roller elements and the bearing raceway. However, in bearing lacking lubrication, such as those in space and other extreme environments, ceramics such as silicon nitride have been used to make the bearing raceway and rolling elements. It has been determined that bearings with ceramic components can operate for longer and at higher temperatures when deprived of lubrication than similar steel bearings. [Bhushan 01] Also for high speed applications ceramic rolling elements have a lower density and overall weight than steel components thus reducing internal bearing forces and improving the bearing performance. Ceramics have lower fracture toughness than steel, making ceramic components more susceptible to fracture from shock and they can rapidly crumble when any disruption occurs at the contact interface.

In this research we have chosen 440C stainless steel balls for all of the presented devices. This was primarily due to the lower cost and availability of steel balls over comparable balls made of other materials such as tungsten carbide or silicon nitride, though the latter materials will be explored in future research. All balls used in this research were Grade 5 and were sourced from Nanoball GmbH [Nanoball 12] or Micro Precision Systems AG [MPS 12a]. Ball Grade 5 ISO specifications are listed in Table 4.1.

Table 4.1: ISO 3290-1:2008 standard values for Grade 5 balls. The values represent the maximum deviation allowed for each parameter.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Max. Value</th>
</tr>
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<tbody>
<tr>
<td>Diameter Variation</td>
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</tr>
<tr>
<td>Deviation from Sphere</td>
<td>0.08µm</td>
</tr>
<tr>
<td>Surface Roughness</td>
<td>0.010µm</td>
</tr>
</tbody>
</table>

62
4.3 Raceway Geometry

Most of the devices described in this thesis have a rectangular raceway geometry. Though this geometry is acceptable and even ideal for roller bearings, it is not the ideal geometry for ball bearings. However, as this geometry is the easiest to create with DRIE processing, it has been used in all previous MEMS rotary ball bearing designs. In the research presented here, all of the designs except those presented in Chapter 9 were created with rectangular raceways. This was primarily due to the desire to first explore the viability of the ball cage designs without the added complication of also integrating the processing steps needed to create a curved raceway.

The curved raceway has many advantages such as self centering, improved load carrying capabilities and a greatly improved contact angle. The contact angle is that angle at which the roller element contacts the raceway under load. Figure 4.3 shows how the contact angle relates to the geometry of the ball bearing under thrust load. The curved raceway is also ideal for angular contact raceways that can support both radial and thrust loads. The larger contact surface will reduce the concentration of force on each roller element and the raceway. This could lead to a reduction in wear and fatigue, increasing the effective lifetime of the bearing. Figure 4.2 shows the contact regions of the ball with rectangular and the curved (angular) raceways under a thrust load. The rectangular raceway has a much smaller contact ellipse than the curved raceway.
4.3 Raceway Geometry

Figure 4.2: Examples of the 2 raceway geometries explored in this thesis. The contact region for each race is indicated by the red regions in the diagram, the rectangular raceway (left) provides smaller elliptical contacts, while the curved (angular) raceway (right) provides much larger elongated elliptical contact regions. A thrust load (W) is applied to the rotor as indicated by the blue arrow.

Figure 4.3: The contact angle (α) is shown for a ball bearing under thrust load. The contact angle is the angle at which the ball contacts the raceway under load or during operation. The contact angle can be different between at the rotor and the stator.
Table 4.2: Values used to calculate the contact areas of the rectangular and curved raceways. Steel values are from [Bhushan 01] and silicon values are from [Hull 99].

An example of the difference between the roller element loading in the two raceway geometries is presented below. With the following assumptions - The contact between the ball and the raceway conforms to the assumptions made for a classic Hertzian contact. The thrust load is static, centric and evenly distributed on all of the roller elements. The ball in the rectangular raceway will only transfer load to the contact at the bottom of the raceway. Further the curved raceway contact is also simplified by assuming a contact angle of 90° and neglecting the effects of diametral clearance. With these assumptions the forces will be the equal and opposite for contacts on the rotor and the stator. Table 4.2 contains the design parameters similar to those used in some of the designs in this thesis and used for this comparison.

A contact modulus $E$ is defined for both raceway types:

$$\frac{1}{E} = \frac{1 - \nu_{Si}^2}{E_{Si}} + \frac{1 - \nu_{St}^2}{E_{St}}$$  \hspace{1cm} (4.1)

$$E \approx 91 \text{ GPa}$$  \hspace{1cm} (4.2)

For the rectangular raceway the problem reduces to the Hertzian contact between a sphere and a plane [Bhushan 01]. In this case, the radius of the
4.3 Raceway Geometry

Contact area is given by:

\[
a = \left( \frac{3WR}{4E} \right)^{\frac{1}{3}} \approx 2.7 \mu m \tag{4.3}
\]

The reduced radius of curvature \( R \) is the radius of the ball due to the plane contact with the raceway:

\[
R = \frac{D}{2} = 250 \mu m \tag{4.4}
\]

The maximum pressure at the center of the contact is then:

\[
p_0 = \frac{3W}{2\pi a^2} \approx 0.66 \text{ GPa} \tag{4.5}
\]

For the curved raceway the calculation becomes more complex due to the need to perform elliptical integrals to find the contact ellipse major and minor axes. With the assumptions made above the calculation becomes [Harris 06b]:

\[
a = a^* \left[ \frac{3W}{2E\Sigma \rho} \right]^{\frac{1}{3}} \approx 15.2 \mu m \tag{4.6}
\]

\[
b = b^* \left[ \frac{3W}{2E\Sigma \rho} \right]^{\frac{1}{3}} \approx 1.3 \mu m \tag{4.7}
\]

\[
\Sigma \rho = \frac{1}{D} \left( 4 - \frac{D}{r} - \frac{2D}{d_m} \frac{d_m}{1 + \frac{D}{d_m}} \right) \approx 3715 \text{ m}^{-1} \tag{4.8}
\]

Contact parameters \( a^* = 4.395 \) and \( b^* = 0.3830 \) are approximate values taken from a table provided in [Harris 06b] based on a calculated value of \( F(\rho) \) = 0.958. The calculation of \( F(\rho) \), which represents the curvature difference between the ball and the race, has not been included but is based on Equation 2.31 in [Harris 06b]. \( F(\rho) \) is the curvature difference between the raceway and the ball.

The maximum pressure at the center of the contact is then:

\[
p_0 = \frac{3W}{2\pi ab} \approx 0.024 \text{ GPa} \tag{4.9}
\]
From the calculations above it can be seen that the maximum force concentration on the rectangular raceway for the design considered is almost 5 times greater than that of the curved raceway. This shows the maximum force at the center of the contact ellipse in the rectangular raceway is 27 times higher than that on the curved raceway. A lower maximum force reduces the amount of wear and vibration in the bearing leading to the ability to support higher loads with less wear and vibration. The curved raceway also has the advantage of allowing the bearing to self-center which can improve bearing load distribution, reduce vibration and improve bearing performance.

### 4.4 Raceway Tolerance

One of the key design parameters that was considered was the raceway tolerance or diametral clearance ($P_d$) of the bearing design. This parameter has many implications for performance of the ball bearing design. The stiffness of the bearing decreases with higher values of $P_d$. This will allow the bearing more freedom to spin, but will increase the free angle of misalignment and endplay allowing for more wobble in the bearing under eccentric loads. In bearings with a curved raceway this will also increase the contact angle. Depending on the application this can have advantages or disadvantages, as the value of $P_d$ increase the amount of eccentric load tolerance increase but this also means that the bearing will have more wobble and out of plane movement. For most MEMS applications a low value of $P_d$ is typically more desirable. The designs presented in this thesis typically have $P_d$ values between 10 µm to 40 µm.

To calculate $P_d$ for a bearing design refer to Figure 4.4 and use Equation 4.10 [Harris 06b]

$$P_d = d_o - d_i - 2D$$ (4.10)

$d_m$ is known as the bearing pitch diameter and is effectively the diameter of the center of the unloaded bearing raceway.
4.5 Raceway Fill

One of the factors that was explored was the number of balls that were used in the raceways of the bearing designs, otherwise known as raceway fill factor. In previous works on micro-ball bearings without ball cages it was determined that the best performance of the device was achieved with an 85% fill factor [McCarthy 08]. It has also been determined that when using a ball cage that an increase in the number of balls used in the design will reduce the effects of cage runout, such as vibration and instability [Nataraj 08]. Cage runout is an increase in the cage ball pocket size which is caused by wear between the ball and the cage. A higher fill factor will also reduce the load on each rolling element. However, in a high speed ball bearing it is often desirable to reduce the number of balls and thereby reduce the centrifugal force of the roller elements. It is also preferred to reduce the number of balls used in micro-ball bearings due to the high cost of the balls.

A ball bearing without a cage cannot have a fill factor below 50% as this would create an unstable situation in which the rotor could collide with the stator if all of the balls move to one side of the bearing. With a caged bearing, however, we have been able to demonstrate designs with fill factors below 50%.

\[ d_m = 1/2(d_i + d_o) \] (4.11)

Figure 4.4: Parameters for Calculating \( P_d \).
4.6 Wear

The fill factor \((F\%)\) is the percentage of the bearing raceway that is effectively filled with rolling elements. This can roughly be calculated as:

\[
F\% \approx \left( \frac{N_b D}{\pi d_m} \right) \times 100% \quad (4.12)
\]

Where \(N_b\) is the number of balls, \(D\) is the ball diameter, \(d_m\) is the bearing pitch diameter calculated with Equation 4.12.

4.6 Wear

Wear is the removal of material from a surface in the form of particles. The four fundamental modes of wear according to [Bhushan 01] are:

1. **Adhesive Wear**: This is caused by the strong adhesion of the two materials under elastic contact.

2. **Abrasive Wear**: This is caused by a harder material cutting into a softer material under elastic contact and is also considered 3-body wear as particles of material can also cause this mode.

3. **Fatigue Wear**: This is caused by repeated friction cycles that eventually cause fracturing at the contact surfaces.

4. **Corrosive**: This is caused by a chemical reaction at the contact surface and is also known as chemical wear. The surfaces act as transport mechanisms to remove the affected area.

In the devices described in this thesis we will see examples of the effects of the first 3 modes. The evidence of the wear on the devices will appear as micro-pitting, smearing and the removal of material at some of the interfaces on the bearings. Denting of the rolling elements is also possible, and is caused by particles or large asperities in the bearing raceway [Harris 06a]. Though this is not truly considered wear it is caused by the same mechanism as wear. Examples of wear and roller element denting on macro-scale bearings are shown in Figure [4.5]. Wear patterns is also used to determine the contact points and operating modes of the bearing designs.
Figure 4.5: Examples of wear and denting on macro-scale bearings. A. Smearing on the raceway of a tapered bearing, B. Extreme micropitting on ball bearing inner raceway, C. Advanced surface initiated fatigue on a thrust ball bearing, and D. Rolling element denting of a ball. Images from [Harris 06a].

4.7 Bearing Loss

The tribometer described in Section 5.1 is used to test the performance of bearing designs by measuring the torque applied to the outer (fixed) ring of the bearing. Two operating regimes were noticed in the testing of the ball bearing designs: sliding and rolling. Figure 4.6 shows how relative movements of the rotor, stator and the ball during each operating regime. The sliding regime of operation happens at low speeds and under low thrust loads. In [Hirano 65] a bearing with a magnetic ball was used to monitor the motion of rolling elements. It was determined that slip of the rolling elements in an angular contact ball bearing under the following conditions:
4.7 Bearing Loss

\[ \frac{N_b F_c}{W} < 0.1 \quad (4.13) \]

Where \( N_b \) is the number of balls, \( F_c \) is the centrifugal force, and \( W \) is the normal load on the bearing. This implies that under low loads that the centrifugal force has a significant impact on the behaviour of the ball. It is reasonable to assume that the low loads present in MEMS micro-ball bearings will conform to these conditions. For measurements presented in this thesis torque measurements related to slip will be denoted with a negative value. This is primarily due to the measurement technique used in the tribometer and the fact that the outer ring of the bearing moves in a different direction depending on whether the balls slip or roll (see Figure 4.6)

From [Harris 06b] the centrifugal force on a ball in the bearing design (assuming steel balls) can be calculated as:

\[ F_c = \frac{1}{2} md_m \omega_m^2 \quad (4.14) \]

\[ F_c = \frac{1}{8} \frac{4 \pi D^3}{3} \rho d_m \frac{4 \pi^2 n_m^2}{3600} \quad (4.15) \]

Figure 4.6: Diagram showing the net force on the stator from clockwise motion of the rotor during sliding (clockwise) and rolling (anticlockwise.)
4.7 Bearing Loss

\[ F_c = \frac{\pi^3}{10800} \rho d_m D^3 n_m^2 \]  

\[ F_c = 2.19 \times 10^{-8} d_m D^3 n_m^2 \]  

where \( D \) is the diameter of the ball, \( d_m \) is the bearing pitch diameter, and \( n_m \) is the orbital speed of the balls in RPM, \( \rho \) is the density of 440C steel \((7.65 \times 10^{-6} \text{ kg mm}^{-3})\). \( F_c \) expresses the centrifugal force in mN.

To convert the orbital speed from rad/s to rpm we use the relation:

\[ n_m = \frac{60\omega}{2\pi} \]  

where \( \omega \) is the rotational speed of the bearing in rad s\(^{-1}\).

As the rotational speed of the bearing increases the centrifugal force of the rolling elements becomes a more significant factor in the loss of the ball bearing. This is due to the change in the bearing loading as the balls are forced against the outer raceway.

The total measured frictional torque of a ball bearing will be defined as:

\[ M = M_c - (M_l + M_v) \]  

where, \( M_l \) is the frictional torque due to applied load and \( M_v \) is the torque due to lubrication viscous friction. For all of the bearings in this thesis lubrication is not used and the value of \( M_v \) is assumed to be 0 as the viscous drag of the balls moving through air is assumed to be negligible. \( M_c \) is the frictional torque due to the centrifugal force of the balls. \( M \) is expressed in \( \mu \text{N m} \).

\( M_l \) is defined as:

\[ M_l = 1000 \times f_l F_a d_m \]  

Where \( d_m \) is the bearing pitch diameter. The constant 1000 is used to change the units of \( M_l \) to \( \mu \text{N m} \). \( f_l \) is calculated as follows:
4.7 Bearing Loss

\[ f_l = z \left( \frac{F_a}{C_s} \right)^y \]  \hspace{1cm} (4.21)

\[ F_a \] is the load applied to each ball and in a thrust bearing can be calculated as:

\[ F_a = \frac{W}{N_b} \]  \hspace{1cm} (4.22)

\( W \) is the applied load, \( z = 0.0008 \) and \( y = 0.33 \) for thrust bearings (Table 10.1 [Harris 06b]). The coefficients \( z \) and \( y \) are determined empirically for different bearing geometries and therefore as we will discover do not accurately predict the performance of the bearings presented in this thesis. \( C_s \) is the static load at which the ball will permanently deform 0.0001\( D \) where \( D \) is the diameter of the ball. \( C_s \) is the highest static load at which the bearing will perform well without excessive vibration or noise. In a thrust bearing with a 90° contact angle \( C_s \) can be expressed as:

\[ C_s = N_b \times Q_{max} \]  \hspace{1cm} (4.23)

In this case \( Q_{max} \) is calculated for a ball deformation (\( \delta_s \)) of 0.0001\( D \) and in the case of the micro-ball bearings presented this can be calculated as follows:

\[ \delta_s = 2.52 \times 10^{-7} \left( \frac{Q_{max}}{D} \right)^2 \left( \frac{1}{0.5D} \right) \]  \hspace{1cm} (4.24)

\[ \delta_s = 0.0001D = 0.0001 \times 0.5 = 50 \text{ nm} \]  \hspace{1cm} (4.25)

\[ Q_{max} = D \sqrt{\frac{0.0000025D}{2.52 \times 10^{-7}}} = 3.52 \text{ N} \]  \hspace{1cm} (4.26)

This assumes that \( D = 0.5 \text{ mm} \) and that the balls are made of steel. This is the case for all of the bearings presented in this thesis so this value for \( Q_{max} \) is valid for all of the designs presented.

\( M_c \) is defined as:

\[ M_c = N_b F_c \frac{d_o}{2} \]  \hspace{1cm} (4.27)
4.8 Discussion and Conclusions

Where $N_b$ is the number of balls, $F_c$ is the centrifugal force on one ball, and $d_o$ is the outer diameter of the bearing raceway. The units of $M_c$ is $\mu$N m. This is a simplified model for the torque related to the centrifugal force and assumes that the resulting torque is applied to the outer radius of the raceway.

The total measured torque on the outer raceway can then be used to calculate the total power loss in the bearing. The total bearing loss is simply the force times velocity or the torque times the speed therefore power loss in Watts can be expressed as:

$$H = 1 \times 10^{-6} M \omega = 1.047 \times 10^{-7} n M$$ (4.28)

Where $H$ is the total power loss in Watts, $n$ is the rotational speed in RPM and $M$ is the total torque in $\mu$N m.

The torque measurements present in this thesis are shown with some negative measurement values. This is due to the fact that when the ball is sliding in the raceway it will force a displacement in the outer raceway in the direction of the rotor’s rotation. However, during pure rolling the outer ring will rotate in the opposite direction. This relationship is depicted in Figure 4.6. Therefore, it is convenient when presenting the data to have the sliding friction shown as a negative value.

It should be noted that this is the model for the bearing loss is highly simplified. No attempt was made to establish a comprehensive model of bearing losses or dynamics. A comprehensive model, that includes factors such as the gyroscopic moment, changes in contact angle, and the effects of the changes in the contact ellipse is beyond the scope of this thesis. The model presented here does provide a predictive curve that can be used to interpret the empirical data.

4.8 Discussion and Conclusions

This chapter has presented the parameters that were considered during the design of the bearings presented in this thesis. As Ball cages are incorpo-
rated in most macro-scale ball bearings to improve performance and evenly distribute the bearing load. This research investigates incorporating cages into micro-ball bearings the inspiration for the micro cage designs is presented with basic design guidelines. Further, a discussion of the differences between the curved and rectangular raceway geometry is presented as motivation for investigating the incorporation of the curved raceway into the micro-bearing design. The key difference being the self-centering and improved loading characteristics of the curved raceway. Raceway tolerance and raceway fill are also important considerations that not only drive the design and fabrication of the bearings but also affect the overall performance of the bearings. Finally the two parameters that are used to assess the bearing designs are presented: wear and power loss. Wear is damage to the bearing elements and can be evaluated visually. A simplified model is provided for evaluating the power loss in the bearing. This assumes that the centrifugal force on the balls in the bearing are the dominate source of the increased losses in the bearing at higher speeds. The use of macro-bearing designs, tolerance considerations, and models was critical to the successful design and evaluation of the micro-ball bearings presented in this work.
Chapter 5

Testing Methodologies

Two methods were chosen to test the micro-ball bearing designs described in this thesis. A tribometer was used to measure the torque on the outer raceway of the bearing under varying speeds and normal loads. The second method was to integrate the bearing designs into a Silicon Micro-Turbine (SMT) to evaluate the performance of the turbine under varying input power. This chapter will describe the theory and setup of each of these testing methods.

5.1 Tribometer

The tribometer used to evaluate the performance of the bearings was first proposed in [Holmes 03] as a method for evaluating sliding friction in MEMS devices with high sliding contacts, i.e. contacts where there is prolonged sliding and a large sliding distance. This platform was chosen because only slight modifications were needed to evaluate the torque on the outer raceway of the bearings presented here. This test platform has been used for several tests of high-sliding MEMS bearings and was described in [Ku 10a]. Later modifications of the setup allowed for testing liquid- and vapour-lubricated MEMS bearings as described in [Reddyhoff 11] and [Ku 12]. And finally a discussion of the wear from silicon on silicon contacts is found in [Ku 11]. These studies and the maturity of the test setup made it a desirable candidate for use in the studies of the micro-ball bearings discussed here.
5.1 Tribometer

A description of the measurement technique, the micro-machined test platforms and the test setup are presented in this section. An extended and detailed description of the tribometer and the results of the previously mentioned studies can be found in [Ku 10b].

5.1.1 Tribometer Measurement Technique

The tribometer is capable of measuring: the applied torque on a central platform, the normal load applied to the sample, and a change in fluid film thickness in a lubricated bearing testing. The last parameter does not apply to the devices described in this thesis. The bearings are mounted on silicon micro-machined platforms and are rotated using a DC electric motor. The rotational speed of the motor is controlled through the computerized test setup described in Section 5.1.3.

![Figure 5.1: Top view of the center test platform under applied torque.](image)

Torque is determined by measuring the angular displacement of a central platform. The platform is supported on silicon beams that will allow the platform to rotate a small angle ($\phi$), as illustrated in Figure 5.1. The dotted lines in the figure represent the position of the left-hand beam before the torque is applied to the platform. A laser source is aligned to a small prism mounted on the bottom of the test platform. By using the internal reflection of the prism the beam is redirected to a reflector. The laser beam is then directed to a detector along a 1.6 m path using a series of mirrors.
to increase the linear displacement of the beam associated with the small angular displacement caused by the applied torque. The first reflector has a piezoelectric actuator that is close-loop controlled to maintain the position of the beam spot at the center of the detector. The test rig is then calibrated to determine the relationship of the piezoelectric actuator voltage to known applied torque values.

The normal load is measured using an outer platform that has beams that are compliant to vertical forces. The displacement of the platform is shown in Figure 5.2 with $W$ representing the normal load, $\Delta z$ the vertical displacement of the platform, the dotted lines indicating the original position of the beams and the platform, and the blue boxes representing the bent support beams. The platform allows for a measurable displacement (in the order of $\mu$m) for the desired normal loads ($W$). An optical displacement sensor is used to measure the vertical displacement of the central platform as the normal load is applied to the sample. This platform is calibrated before testing to determine the relationship of the applied load to vertical displacement of the platform. Figure 5.2 shows a sideview of the outer platform with an applied normal load. The dotted lines indicate the position of the platform with no applied load.

![Figure 5.2: Side view of the outer platform under applied normal load.](image)

Section 5.1.2 shows the fully assembled test platform and describe the design considerations for both inner and outer platforms. The test setup and operation will be described in Section 5.1.3. The reader should refer to these sections for more detail.
5.1.2 Test Platform

Silicon has many favorable properties as a structural material as described in [Petersen 82] and can be fabricated into very precise structures using the micro-fabrication techniques described in Chapter 3. This makes it a desirable material for the test platforms used in this study. The torque measurement (or inner) platform and the normal load measurement (or outer) platform are fabricated separately and are then assembled by hand using cyanoacrylate adhesives (or super glue). A picture of a fully assembled test platform can be found in Figure 5.3. The labels indicate the various parts that will be covered in this section.

![Assembled and labeled test platform](image)

Figure 5.3: Assembled and labeled test platform

5.1.2.1 Torque Measurement Platform (Inner Platform)

The outer dimensions of the torque measurement platform are 10 mm by 10 mm; this allows the fabrication of up to 60 platforms on a 100 mm diameter wafer. The platforms are typically fabricated from a 100 mm diameter, p-type, <100> wafer with a thickness of 525±25 µm. Fabrication uses a single mask and through wafer etching (as described in Section 3.2.2). The mask is designed to use halo etching (as described in Section 3.2.3) which provides high quality beams as described in [Kumar 07]. The bearing sample holder and prism mount designs are found on the periphery of the wafer and are
5.1 Tribometer

etched at the same time as the test platforms. Test platforms were designed with either 22 or 50 beams to allow for configuring the maximum measurable torque, and also to increase the yield of the devices. The 50 beam design is shown in Figure 5.4. Extra beams can be removed from the platform, leaving only the beams needed for testing. Stoppers were incorporated into the design to restrict the allowable twist of the platform to 2°, thereby reducing the possibility of failure if too much force is applied. Figure 5.5 shows a top view of the platform and indicates the location of the stoppers and the sample holder. This picture also shows the platform with a typical configuration for testing, with all but 4 of the beams removed and indicates where beam length ($L$) and width ($w$) are measured.

Figure 5.4: SEM of the torque of a torque platform with all 50 beam (left) and a CAD rendering of the entire platform (right).
5.1 Tribometer

A full description and theoretical evaluation of the test platform design can be found in Section 3.3 of [Ku 10b]. This analysis and experimental work determined the design used for the micro-ball bearing tests. The platforms used for testing have the design parameters listed in Table 5.1. By limiting the angle of rotation to a maximum of 2° and by centering our expected measurement range below 1°, a linear approximation could be used for the relationship between the applied torque and the rotation angle of the inner platform. This is confirmed by calibrating each platform before testing. Theoretically evaluation of the non-linearity shown in the study of a similar device found in [Davis 04] found that the linear approximation below 1° is in good agreement with more rigorous higher order models. The following

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Beam Length (L)</td>
<td>2 mm</td>
</tr>
<tr>
<td>Beam Width (w)</td>
<td>80 μm</td>
</tr>
<tr>
<td>Platform Radius (R)</td>
<td>2.5 mm</td>
</tr>
<tr>
<td>Beam Depth (Thickness) (d)</td>
<td>0.5 mm</td>
</tr>
<tr>
<td>Maximum Rotation Angle</td>
<td>2°</td>
</tr>
<tr>
<td>Number of Beams (N)</td>
<td>4</td>
</tr>
</tbody>
</table>

Table 5.1: Typical design parameters for the torque measurement platform.
5.1 Tribometer

equations are used to approximate relationship of the applied torque to the 
rotational displacement of the platform, and are taken from Section 3.3.1 
of [Ku 10b]. The transverse shear force ($Y_T$) and the resulting couple ($M_1$) 
are given by:

$$Y_T = (aR + e)\phi$$  \hspace{1cm} (5.1)

$$M_1 = (eR + c)\phi$$  \hspace{1cm} (5.2)

Where:

$$a = \frac{12EI'}{L^3}$$

$$c = \frac{4EI'}{L}$$

$$e = \frac{6EI'}{L^2}$$

$$I' = \frac{w^3d}{12}$$

$R$ is the radius of the platform, $\phi$ is the rotational angle of the platform, 
$N$ is the number of beams in the platform and $E$ is the Young’s Modulus of 
silicon (from Table 4.2).

The torque is then given as:

$$T = N(R\cos\phi Y_T + M_1) \approx N(aR^2 + 2eR + c)\phi = k_\phi \phi$$  \hspace{1cm} (5.3)

$k_\phi$ is the torsion stiffness of the platform. Using the values from Ta-
bble 5.1 for the test platforms used for the testing gives a value of $k_\phi \approx 
8192 \mu\text{N m rad}^{-1}$.

5.1.2.2 Normal Load Measurement Platform (Outer Platform)

The normal load measurement platform was designed using a folded beam 
design to allow high lateral (or torsional) stability, compliance to applied 
normal loads, and maintain a compact design. The platforms are fabricated 
from a 100 mm diameter, p-type, <100>, double side polished (DSP) wafer.
with a thickness of 525±25 µm. The platform is 25 mm by 25 mm with 2.5 mm-diameter holes in the corners to mount the platform to the test setup. The torque platform is attached to a recess etched in the center of the normal load platform with adhesive. The Figure 5.6 shows a top view of the platform indicating the locations of the features mentioned. The platforms used in testing have the design parameters listed in Table 5.2. The linearity of displacement of the platform is confirmed by calibrating the platform before testing. Also, because the torque (inner) platform is calibrated while attached to this platform, and angular displacement resulting from twist of the folded beams is also accounted for during pre-testing calibration. A detailed evaluation of the torsional stiffness and the vertical compliance of the beams can be found in Section 3.3 of [Ku 10b]. The theoretical analysis of the design and the experimental results show a good agreement with devices described in [Pike 07].

The displacement of platform with four beams can be estimated by the following equation derived in [Ku 10b]:

\[
\delta = \frac{F_z}{k_z} = F_z \left( \frac{L^3}{6EI} + \frac{Lg^2}{2\beta bd^3G} \right)
\] (5.4)

In this equation \( \beta \) is a shape factor associated with the torsion of the beams [Timoshenko 51]. For beams with a 2:1 aspect ratio as in Table 5.2, \( \beta = 0.229 \). The parameter \( g \) is the separation of the center-lines of the beams. Putting \( E = 168 \) GPa (\(<110>\) direction), \( G = 61.7 \) GPa, \( g = 1.2 \) mm, \( I = 1.35 \times 10^{-14} \) m\(^4\) (second moment of area), and with other parameters as in Table 5.2, the total stiffness of each folded beam is 1773 N m\(^{-1}\) and the stiffness of a platform with 4 beams will be 7092 N m.
5.1 Tribometer

5.1.3 Tribometer Setup and Operation

A short description of the test setup is presented here; for a detailed description of the test set up and operation please refer to Section 3.8 of [Ku 10b].

The torque measurement platform is mounted on the top of the normal load measurement platform. A device holder is attached to the top of the torque platform and a prism mount is attached to the bottom of the platform. A small prism is attached to the prism mount and a small piece of silicon is attached to the bottom of the prism. This piece of silicon provides a better target for the optical displacement sensor used to measure normal load displacement. Figure 5.6 shows a CAD drawing of the assembled platform with labels indicating the parts of the platform.

Table 5.2: Design parameters for the normal load measurement platform.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total Folded</td>
<td>18 mm</td>
</tr>
<tr>
<td>Beam Length ((2L))</td>
<td></td>
</tr>
<tr>
<td>Beam Width ((w))</td>
<td>600 µm</td>
</tr>
<tr>
<td>Wafer Thickness ((d))</td>
<td>300 µm</td>
</tr>
</tbody>
</table>

Figure 5.6: Top view of normal load platform, \(w\) - beam width, \(L\) - half the folded beam Length.
The platform is mounted in the test rig. The normal measurement platform is then calibrated by placing weights onto center of the platform and recording the vertical displacement measured by a CCD laser displacement sensor (Keyence LKG-32) trained on the silicon reflector mounted to the bottom of the prism. The measurements are then used to provide a linear relationship between the applied normal load and the measured displacement. The laser used for measuring torque is then aligned with the prism. The piezo actuated reflector and fixed mirrors are used to center the beam spot on the beam sensor, as shown in Figure 5.8. The torque measurement platform is then calibrated by using a calibration tool that applies and measures a torque at the center of the platform. The torque measurement and the piezo actuator voltage are recorded by the LabView interface and used to provide a linear relationship between the applied torque and the actuator voltage.
Figure 5.8: Drawing of the underside of the test platform, depicting the torque measurement laser path from the source, reflected from the prism, and redirected to the sensor by the piezo actuated reflector and fixed mirrors.

A device is placed in a sample holder (see Figure 5.3) in the center of the platform and a device adapter is inserted into the center of the rotor of the bearing. The adapter is fabricated with the ball bearings and is designed to resemble a torque bit. The top or the adapter has holes to insert the pins of the motor adapter and a circular feature in the center that allows the ball on motor adapter to level the adapter during testing. Figure 5.9 shows a rendering of top and bottom of the adapter, a SEM image of the bottom and a picture of the adaptor inserted in the device which is in the sample holder and the motor coupler can be seen above the adaptor.
The device adapter is then coupled to the motor with the motor adapter. This adapter, as mentioned previously, had two pins that couple with the device adapter and a small (300 µm diameter) sapphire ball that levels the adapters of the device and the motor. The motor is attached to a platform that is moved with a precision linear actuator allowing for movements as small as 0.1 µm. The motor is moved down until the desired normal load is applied to the bearing.

A LabVIEW program, shown in Figure 5.10, is used to control the motor and to record the test data. The motor speed can be set to a specific RPM value, or can be programmed step through a set speed range. The piezo actuator voltage is converted to its calibrated torque value and recorded with the motor speed (in RPM) and the applied normal load.
5.1 Tribometer

Figure 5.10: Screen capture of the LabVIEW screen used for testing.

There is one difference in the measurements for micro-ball bearings compared to the planar contacts explored in previous studies using the micro-tribometer. As a ball bearing moves from the regime of ball slide to ball rolling the direction of the force applied to the stator reverses. When the ball is sliding in the bearing raceway, a clockwise motion of the rotor results in a net clockwise force on the stator, but when ball is rolling in the bearing raceway a clockwise rotation of the stator results in a net anticlockwise force on the stator. This is illustrated in Figure 5.11. The distinction between rolling and sliding regimes will be indicated in the results in the appropriate sections.
5.2 Silicon Micro-Turbine (SMT)

The silicon micro-turbine (SMT) used in these studies is relatively mature and has been used in many other studies. The most recent SMT advances by the University of Maryland are to incorporate an accelerometer to monitor the bearing properties as shown in [Hanrahan 12] and using the turbine in a micro-generator as described in [Beyaz 12]. Devices with rotor diameters of 10 mm and 5 mm were used in the studies and are pictured in Figure 5.12.

As the development of the devices was carried out with the University of Maryland and the test setup was based on an existing design, this chapter will briefly describe the turbine operation and design, and the test setup. The test setup used in these studies was modified to provide fully automated testing, not included in the University of Maryland setup. The test setup will be described below.
5.2 Silicon Micro-Turbine (SMT)

Figure 5.12: SMTs with 5 mm rotor diameter (left) and 10 mm (right).

5.2.1 Turbine Design and Operation

McCarthy 08 provides an overview of the radial in-flow turbine design parameters for the SMTs with 10 mm diameter rotors. The devices with 5 mm diameter are simply a scaled version of the 10 mm design. No effort was made to optimize the design of the turbine for these studies, as the primary concern was to explore bearing technologies. Fabrication of the turbines will be covered in the appropriate chapters as each study required slightly different fabrications techniques.

Figure 5.13 shows a cutaway view of the turbine in the test enclosure. The pressurized gas flow through the plumbing wafer and around the guide vanes to drive the turbine blades. Most of the air will exit the top of the test enclosure as shown in the figure, however, some of the gas will leak through the turbine bearing channel and provide a backside or thrust pressure on the rotor. This will provide a net upward force (normal load) on the rotor. The figure shows that some or the gas is vented from the bottom of the test enclosure, however, gas can be input from the thrust side of the device to
increase the thrust side pressure. In the studies covered in this work, gas was allowed to leak from the thrust side of the device, thereby reducing the normal load on the rotor. Figure 5.14 is a CAD drawing of the turbine, with half of the plumbing wafer cut away to reveal the locations of the speed tracking marks, the guide vanes and the turbine blades. A conceptualization of the gas flow, within a stalled rotor, is also included in the diagram.

Figure 5.13: A cutaway showing the turbine test setup. Pressurized gas is used to power the turbine. The input power is controlled by an electronically controlled proportional valve and input power is monitored by a flow sensor and inlet pressure sensor. The top of the turbine is vented to atmospheric pressure. Some of the inlet gas leaks through the bearing to the backside of the turbine. This gas applies a net upward force, monitored by the thrust pressure sensor, on the turbine allowing the bearing to function in the proper mode. In order to reduce this pressure a bleed valve is attached to the bottom side of the test enclosure. The bleed rate is controlled by an electronically controlled proportional solenoid.
5.2 Silicon Micro-Turbine (SMT)

5.2.2 Testing Setup

The test setup was designed to be fully automated. A diagram showing the parts and connections of the test setup is shown in Figure 5.15. Testing is performed by controlling the flow of pressurized gas into the SMT and measuring the speed of the turbine using an optical displacement sensor. The displacement sensor is positioned over the center of the device which has 12 features that are used to track the speed of the device. The input flow, input pressure, device speed and thrust side pressure of the device are recorded. The test setup is controlled by an Arduino Mega 2560 allowing for testing independent of an external computer system. All data is recorded in a tab delimited text file on a MicroSD card attached to the system, which allows easy import and analysis of the testing data.

Improvements over test setups reported in previous studies include au-
tomated flow control at the input and thrust side of the device and a new
method of measuring the speed of the device. The automation of flow con-
trol is accomplished by using proportional solenoid valves connected to a
controller that provides a constant current to drive the valves. The constant
current provided to the solenoid is windowed to conform to the desired oper-
ating range of the valve and has 4095 steps (12-bit control value) allowing for
a wide range of input power testing. Speed tracking has been improved by
converting the output of the optical displacement sensor into a digital signal.
This signal is then attached to a timer/counter on the Arduino controller
proving an asynchronous count based solely the converted optical pulses.
The number of pulses is then recorded every 500 ms and the counter is reset.
This provides an accurate count of the passing speed marks to within ±2
marks per 500 ms. The pulse counting method is base on a modified version
of Arduino pulse counting code found in [Margolis 11].

A discussion of each of the parts in the test setup follows:

Arduino Mega 2560: This is a micro-controller prototyping system
based on the Atmel ATmega2560 chipset, with an Open Source program-
ing environment. More information on Arduino systems can be found
at [Arduino 12b] for the Mega 2560 used in this system at [Arduino 12a].
Provides all control of the system. The SPI interface is used to communicate
with the Solenoid Controller and MicroSD Card. Analog input signals from
the Flow Sensor, Top Side Pressure Sensor, Thrust Side Pressure Sensor,
Manual Thrust Flow Control and Manual Input Control are converted to
10-bit digital values by the built in Analog to Digital Convert (ADC.) The
speed of the turbine is monitored by connecting the output of the Optical
Signal Digitizer to a clock input and counting the number of pulses every
500 ms. The systems is programmed through the USB interface, which can
also be used during operation to provide debugging information through the
Serial Monitor. However, the system is designed operate on its own and can
perform automated testing and data logging without being attached to an
external computer. Data is logged every 500 ms and includes: Record Num-
ber, Time in ms since the system was powered on, Flow Control Solenoid
digital control value, Thrust Control Solenoid digital control value, top side
5.2 Silicon Micro-Turbine (SMT)

Figure 5.15: Conceptual diagram showing the parts and connections in the SMT test setup.
pressure, thrust side pressure, gas flow rate, and the number of optical pulses for the previous 500 ms.

**Pressurized Gas:** This gas provides the air flow to actuate the turbine. Nitrogen and compressed air have both been used for testing. Compressed air is run through a drier to remove moisture before entering the test setup. The flow of gas entering the test setup is controlled by the Flow Control Proportional Solenoid. The Flow rate of the gas is measured by the Flow Sensor and the input at the turbine is measured by the Top Side Pressure Sensor.

**Flow Sensor:** The flow sensor used in the system is a microbridge mass airflow sensor with a Venturi type flow housing (AWM5104VN made by Honeywell.) It is a nitrogen calibrated sensor capable of measuring flow rates from 0-20 standard liters per minute (SLPM) and has a linearly proportional output from 1 V to 5 V. More information can be found in the data sheet [Hon-eywell 12]. The flow rate is recorded by Arduino Mega by attaching the output to a 10-bit ADC.

**Flow Control Proportional Solenoid:** The proportional solenoid is a SMC PVQ31-6G-23-01F. The valve requires a 12 VDC power supply and has a maximum flow rate of 100 SLPM. More information can be found in the data sheet [SMC 12]. This valve controls the flow of the pressurized gas input to the turbine test enclosure and is controlled by a constant current provided by one of channels of the Solenoid Controller. This is the key to providing fully automated testing of the turbines, as the input flow rate can be controlled electronically.

**Thrust Control Proportional Solenoid:** This valve is the same model of valve as used for the Flow Control Proportional Solenoid. The valve can be used to control an input pressure to the thrust side of the turbine or to allow for a leak off gas to reduce the thrust pressure. The solenoid controller provides a constant current to control the solenoid. This allows for automated control of the thrust side pressure of the turbine.

**Solenoid Controller:** This provides a constant current to the proportional solenoids over 2 analog channels. The current control has a resolution of 12-bits over the range required to operate the solenoids. The Arduino
provides 12-bit control signals for the 2 channels over the SPI interface. This allows full electronic control of the input flow rate and the thrust side pressure of the devices, thereby allowing for fully automated testing of the devices. The controller was designed, by the author, specifically for this application.

**Top Side Pressure Sensor:** The top side or input pressure sensor is a Freescale Semiconductor MPX5050 integrated silicon pressure sensor capable of measuring gauge pressures of 0 kPa to 50 kPa with proportional analog output range of 0.2 V to 4.7 V, more information can be found in [Freescale 10]. The output is measured by a 10-bit ADC channel on the Arduino controller. This provides a measure of the pressure of the gas input to turbine and is used to calculate the input power.

**Thrust Side Pressure Sensor:** The thrust side pressure sensor is the same model as the top side pressure sensor and the output is also measured by a 10-bit ADC channel on the Arduino controller. This provides a measure of the backside or thrust pressure of the device and can be used to calculate the normal load on the rotor of the device.

**Optical Displacement Sensor:** This sensor is used to measure the speed of the turbine by detecting the height difference between the turbine surface and the top of the speed tracking marks. This provides an analog signal, which is coupled through a capacitor to remove DC bias to the Optical Signal Digitizer. The sensor is a Philtec Model D6 reflectance dependent fiberoptic sensor with an analog output and a bandwidth of 20 kHz, more information can be found in the data sheet [Philtec 10]. The sensor tip is positioned over the speed tracking marks at the center of the turbine using the X,Y,Z positioner.

**Optical Signal Digitizer:** This is a comparator with a configurable reference (or threshold) voltage that is used to convert the analog signal from the Optical Displacement Sensor into a digital signal that can be recorded by the Arduino. The conversion threshold is controlled by a potentiometer attached to the control panel (the Digitizer Comparator Threshold.) Figure 5.16 shows the comparator circuit and a graph of the conversion from the optical signal to the digital signal as captured by a digital oscilloscope.
5.2 Silicon Micro-Turbine (SMT)

Figure 5.16: Optical Signal Digitizer comparator circuit (top) and a digital oscilloscope capture (bottom) of the conversion from the analog signal to digital.

**X,Y,Z Stage:** This is used to position the optical displacement sensor over the speed tracking marks at the center of the turbine. The Z height is adjusted to provide the maximum analog output from the optical displacement sensor. Manual 12.7 mm Linear Translation Stages (part number DT12XYZ/M) from Thorlabs are used in this test setup. A custom made Perspex arm was used to attach the displacement sensor tip to the translation stages.

**MicroSD Card:** This provides 2 GB of data storage. All testing data is written by the Arduino controller to the card through the SPI interface. The text file written by the Arduino is tab delimited for easy import into data analysis software.

**Turbine Test Enclosure:** The test enclosures are custom made Perspex enclosures fabricated for this test set up. Separate enclosures were made for the 10 mm and 5 mm rotor diameter devices. The top of the enclosure has two input ports for connecting the pressurized gas lines, two grooves for the seal o-rings, a recess for the turbine and an exhaust port in the center (which also serves as the hole in which the displacement sensor is inserted.) The bottom of the enclosure has the thrust side gas port and provides a contact for the
5.2 Silicon Micro-Turbine (SMT)

thrust side seal o-ring. The two parts are bolted together by 4 bolts and the entire enclosure is bolted down to an optical bench for stability during testing. Pictures of the test enclosure can be found in Figure 5.17.

![Control Panel](image)

**Control Panel:** This panel is used to turn on the test setup and control the manual and automated functions. It also provides the main power connection to the test setup. A green LED on the panel indicates that the power is on.

*Power Switch* - Turn on/off main power to the test setup.

*Manual/Auto Control Switch* - This switch selects between the Auto and Manual modes of operation. In Auto mode the control program will set the Thrust Control Proportional Solenoid to a predefined value and will step through a preset input flow range. The Manual Input Flow Control and Manual Thrust Flow Control are disabled in this mode. Initial values are set in the programmed memory (requiring reprogramming of the Arduino controller in order to change the start values.) In manual mode the Flow Controls are active and the user can manually adjust both the thrust and input flow rates. Manual mode also provides debugging information though
the USB serial monitor.

Manual Input Flow Control - This a potentiometer that provides an analog value from 0 V to 5 V to a 10-bit ADC channel on the Arduino. This value is converted into a 12-bit value and sent to the Solenoid Controller to change the flow rate of the input (pressurized) gas.

Manual Thrust Flow Control - This a potentiometer that provides an analog value from 0 V to 5 V to a 10-bit ADC channel on the Arduino. This value is converted into a 12-bit value and sent to the Solenoid Controller to change the flow rate of the thrust proportional solenoid.

Digitizer Comparator Threshold - This a potentiometer that provides an analog value from 0 V to 5 V to comparator in the Optical Signal Digitizer. This provides the digital conversion threshold for the optical signal input, when the the optical signal is greater than this input the comparator provides a high (5 V) signal otherwise a low (0 V) signal is provided.

5.3 Discussion and Conclusions

By choosing mature test platforms, that required only minor modifications or some improvements, it was possible to focus on the design and evaluation of the unique micro-ball bearing geometries. The advantages of using these measurement techniques are:

For the Tribometer: It is possible to resolve torque measurements down to 1 µN m with the current platform configuration with maximum measurable torque of 100 µN m. The maximum measurable torque is configurable by increasing the number of beams attached the to test platform. The normal load has a measurement resolution of 5 mN and can applied reliably in 10 mN increments. The speed can be set in increments of 1 RPM of the range of 0 RPM to 30 000 RPM. Measurement acquisition is configurable in 1 s increments. The testing range and measurement resolution of this system are sufficient provide insight into the losses in the micro-bearings. The system also has the added benefit of showing when the bearing is operating in rolling or sliding friction.

For the SMT: The turbine is a fully integrated MEMS solution for testing
5.3 Discussion and Conclusions

the bearing technologies. The input power can be measured and controlled in 5 mW increments. The force on backside of the turbine can be measured in 5 mN increments. Speed can be resolved up to 200,000 RPM with a resolution of 1/12 of a revolution. The data acquisition can also be configured to capture data in increments of 100 ms and with data storage of 2GB testing can be performed for multiple days without interruption. Also, testing can be performed in a fully automated mode including the ramping up and down of the input power.

These test platforms provide the means by which the bearing designs presented in this thesis can be analyzed and compared. Due to the automation of the test platforms it is possible to evaluate the repeatability of test results and to confirm performance on more than one device for each design.
Chapter 6

Proof of Concept of a Radial Ball Bearing with Integrated Ball Cage

This chapter will discuss the first micro-ball bearing with an integrated silicon cage. This was a simple design to explore fabrication and design techniques. This design was first reported in [Hergert 10] and is important in that it showed that a silicon cage could be fabricated into a MEMS bearing and that the cage could withstand the internal bearing forces. A CAD drawing of an exploded device is presented in Figure 6.1. The rotor, cage and stator are held together in the center with electroplated solder. The motivation for integrating a ball cage into a micro-ball bearing was to alleviate ball jamming as described in [Waits 07a]. Ball jamming is a bearing failure mode in which the balls in the raceway are collated in the raceway and seize the bearing. This becomes possible as the raceway and balls experience wear during operation. With a ball cage, seizure of the bearing is not possible due to ball jamming, because the balls are always physically separated by the cage. The use of the cage also allows for even spacing of the balls which improves the load distribution in the bearing.
6.1 Design

This bearing was designed for simplicity. The fabrication process required 3 lithographic masks and 5 fabrication steps. One DRIE etch step was required on each side of the wafer for fabrication. The design was also a first experiment with the idea of using a silicon dioxide mask to allow patterning the die, assembling the die at high temperature and then release the devices using DRIE.

Though there were many bearing designs on the masks with different cage thickness and raceway tolerances, only one design was successfully fabricated and tested. Figure 6.2 shows how the design parameters relate to the bearing design and Table 6.1 shows the values of each of the design parameters in the case of the successful design. The bearings were designed to use Grade 5, 440C steel balls with a diameter of 500 µm.

The primary goal of this design was to see if a ball cage could be fabricated into a MEMS ball bearing and if that cage could survive the bearing forces. The thickest cage \( C = 100 \mu m \) was the only design that was able to survive fabrication and testing. The tolerances chosen were sufficient to
6.1 Design

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$D_r$</td>
<td>Raceway inner diameter</td>
<td>1.18 mm</td>
</tr>
<tr>
<td>$D_b$</td>
<td>Raceway outer diameter</td>
<td>2.2 mm</td>
</tr>
<tr>
<td>$g$</td>
<td>Cage release gap</td>
<td>40 µm</td>
</tr>
<tr>
<td>$t$</td>
<td>Raceway tolerance ($0.5P_d$)</td>
<td>10 µm</td>
</tr>
<tr>
<td>$C$</td>
<td>Cage width</td>
<td>100 µm</td>
</tr>
<tr>
<td>$N_b$</td>
<td>Number of balls (not shown)</td>
<td>4</td>
</tr>
<tr>
<td>$F_%$</td>
<td>Raceway fill (not shown)</td>
<td>40%</td>
</tr>
</tbody>
</table>

Table 6.1: Values for the design parameters of the tested devices.

ensure smooth operation of the bearing while not permitting enough room for the cage to collide with the inner or outer lands during operation.

The bonding process for this design had the lowest yield of any of the designs presented in this thesis. This is primarily because the solder pads had significantly different areas which caused the pads to have different heights. This made bonding the devices difficult and unreliable. This may also have been the major cause of the cage failures, which always occurred at solder bond interface.

Figure 6.2: Depiction of the design parameters and how they relate to the bearing design.
6.2 Device Fabrication

The radial ball bearing with an integrated cage is fabricated using a 100 mm diameter, p-type, <100>, DSP silicon wafer with a thickness of 525±25 µm. The wafers also had 0.5 µm of thermally grown silicon dioxide. A description of the simple 3 mask fabrication follows. The steps are labeled with the letter corresponding to the depiction of the step in Figure 6.3.

Figure 6.3: Process flow for a radial ball bearing with an integrated ball cage. A) electroplate solder pads, B) etch cage and raceway features, C) define DRIE mask in backside oxide, D) insert steel balls and bond die, and E) release the rotor and cage using oxide mask from Step C.

6.2.1 Step A - Plate Solder Pads

The wafer is sputter coated with a chrome and copper seed layer (not shown in the process flow diagram.) The wafers are coated with photoresist and
patterned with the solder pad mask. The solder pads are electroplated with a 2\,\mu m nickel diffusion barrier, 3\,\mu m of tin and 300\,nm of gold. The photoresist is then stripped and the exposed seed layer (not shown in the diagram) is etched to expose the silicon dioxide below.

### 6.2.2 Step B - Etch Cage Arms and Bearing Raceway

After the solder pads are plated, photoresist is spun on the wafer. The bearing raceway and cage arms are patterned. The exposed oxide is etched using RIE. Then DRIE is used to etch the features to a depth of 250\,\mu m. The photoresist is then stripped from the wafer.

### 6.2.3 Step C - Pattern Cage Release Channels

The wafer is then flipped over and spun with photoresist. The test adaptor feature and the cage release channels are patterned into the oxide on the back of the wafer using RIE. The photoresist is then removed from the wafer. The wafer is broken into die for the assembly step.

### 6.2.4 Step D - Device Assembly

The steel balls are placed between the cage arms in the raceway of each device. The solder pads are then carefully painted with solder flux. The flux enhances the reflow of the eutectic metals by helping to reduce oxidation. The top die is then aligned to the bottom device using a custom bond aligner. Force is applied to the die and heat is applied using a butane torch. The bonded die are then removed from the bonder and placed on a metal surface to cool.

### 6.2.5 Step E - Release Etch

The bonded die are then attached to a carrier wafer and DRIE is used to etch the features patterned in the oxide on the backside of the wafer. The bearing and cage release channels are etched until they meet the raceway
feature on the other side of the wafer. This etching is performed on both sides of the assembled die. At the end of this processing step the individual devices drop out of the die. The bearing rotor and cage are released from the bearing stator and are free to rotate.

6.2.6 Fabricated Devices

The following images show the parts that have been fabricated with the process described above. In Figure 6.4 SEM images show the cage, rotor, stator and an assembled bearing. In Figure 6.5 the stator of a bearing is shown with raceway damage caused by over-etching the device during the release step. This damage helped to identify shortcomings in the fabrication process that were addressed in future design efforts. Finally, Figure 6.6 shows a bearing design having 8 balls with the stator removed to show the internal parts of the bearing. Though none of the 8 ball designs survived fabrication and testing this SEM does provide an excellent representation of the internals of the bearings.

Figure 6.4: SEM of the parts of the bearing: the ball cage (upper left), the rotor (upper right) the stator (lower left), and fully assembled bearing (lower right).
6.3 Testing

The devices were tested on the micro-tribometer covered in Section 5.1. The devices were tested by applying a thrust load to the rotor (center) of the bearing. Two devices were tested with a load of 60 mN and at speeds of
10 RPM to 20000 RPM. It should be noted that this is a radial bearing design. This means that the testing was carried out on the bearing in the most non-ideal orientation and provided a good stress test of the bearing and cage design.

6.3.1 Cage Bond Failure

One of the key failure modes in this bearing design was a break at the cage bond interface. As discussed in Section 3.4, the yield of the cage bond was relatively low due to the fact that the cage bond pads were significantly smaller than the other bond pads on the device. The cage features were more than 1000 times smaller than the surrounding features and typically had a height difference of 100 nm to 250 nm below the height of the surrounding features. This height difference had to be overcome by the reflow of the solder and was difficult to achieve. However, when a good bond was achieved the bond was very strong as can be seen in Figure 6.7, a good bond can remove silicon from the half of the cage during failure. The cage in Figure 6.7 had 3 cage arms that had poor solder bonds. These solder pads show evidence of some reflow, but solder was not able to bond to the other half of the cage. All cages that failed quickly during testing showed signs of poor solder reflow on one of more of the cage arms. The difficulty in getting this solder bond to work properly has motivated the exploration of single crystaline cages that do not require bonding to form the cage pockets.
6.3 Testing

Figure 6.7: SEM image of a cage that quickly during testing. The

6.3.2 Tribometer Results

Tribometer testing was carried out on the bearings for low and high speeds. The measurements were used to test the validity of the model presented in Section 4.7. Though the characteristic curve of the model is similar to the measured values the predicted values were lower than the measured torque. The model was modified in two stages. First the low speed measurements were used to assess the value of $M_l$ as the centrifugal forces would be negligible at the lower speed. Then, with the modified calculation for $M_l$ the model was compared to the high speed results to assess the effects of the centrifugal force. The results of these analysis steps are described below.

2 devices were measured at speed of 50 RPM to 550 RPM in increments of 50 RPM. The results of this testing is presented in Figure 6.8 with a plot of a modified model (Equation 6.1) for the torque. Measurement of the torque is less stable at lower speeds, therefore the spread of the torque measurements is shown on the graph as error bars. It should also be noted the torque is negative because the balls are sliding at lower speed rather than rolling, therefore the outer ring is moving in the direction of the motor which is represented as a negative value. It was found that the torque on the outer
platform predicted by Equation 4.19 was 3 orders of magnitude lower than the values measured during testing. As was stated in Section 4.7, the value of \( z \) and \( y \) used to calculate \( f_I \) are determined empirically for existing macro-scale bearing. It was determined by analyzing the measured torque that the value of \( z \) is not appropriate for this bearing design. If we substitute \( z = 0.8 \) instead of the value of \( z = 0.008 \) reported in [Harris 06b], the model closely fits the measured torque values for the low speed measurements. Centrifugal forces are negligible at these lower speeds so the frictional torque is nearly constant as would be expected from the model.

![Figure 6.8: The measured torque of the bearing at low speeds and the predicted torque from Equation 6.1. The error bars indicate the range of the measurement for each speed and the connected points are the average of the measurements.](image)

After modifying the model for to predict a more realistic value for \( M_f \) the model was compared to the high speed measurements. To obtain the high speed torque measurements, a bearing was tested at speeds of 1000 RPM to 20 000 RPM in increments of 200 RPM. An average of three tests is plotted in Figure ?? with the predicted torque values of the modified model.
The crossing at the x-axis does not represent that no torque was measured at the outer platform. In the range from approximately 6000 RPM to 8000 RPM the balls are transitioning from sliding to rolling and there is a mixture of both which results in reducing the measured displacement of the outer platform. It was found that the measured centrifugal torque was 3 times greater than the value predicted by Equation 4.19.

These observations led to the following modified model for the total frictional torque for the bearings presented in this chapter:

\[
M = 3 \times \left( 2.19 \times 10^{-8} \right) N b_d_m D^3 n_m^2 \frac{d_o}{2} - 0.8 \left( \frac{F_a}{C_a} \right)^{0.33} F_a d_m \quad (6.1)
\]

Figure 6.9: The average of the measured torque for the bearing over 3 test runs at speeds of 1000 RPM to 20 000 RPM with the predicted torque from Equation 6.1.

The measured results and the modified model indicate that the friction torque at higher speeds does change proportional to \( \omega^2 \); which indicates that the centrifugal torque does play a significant in the bearing loss at higher
speeds as predicted. The significant difference between the original model of the load torque and the measured load torque can likely be attributed to both geometric and scaling factors. Figure 6.10 shows the measured and predicted power loss which is proportional to $\omega^3$ as expected.

![Predicted and Measured Power Loss](image)

Figure 6.10: The graph shows the measured power loss in the bearing from 1000 RPM to 20000 RPM. At 1000 RPM the bearing loss is approximately 0.5 mW. As the speed increases and the centrifugal force plays a more dominate role and increases the bearing loss.

### 6.3.3 Wear

After 5 hours of testing the second tested device was soaked in chrome etchant to break the solder bonds. The parts of the bearing were then viewed in the SEM to evaluate any indication of wear. Minimal wear was noted on bearing interfaces. There was some indication of abrasive or shock wear in the ball pockets of the cage as seen in Figure 6.11. The edge of the ball contact with stator and the rotor both showed signs of abrasive wear in the form rounding of the interface. Figure 6.12 shows the wear on the rotor and also indicates the location of over-etching damage on the rotor. Figure 6.13 shows wear damage at the ball contact on the bottom half of the stator.
6.3 Testing

Figure 6.11: SEM image of the cage after testing. The circles indicate the locations of the wear on the ball pockets. The damage to the cage arms is due to wear as similar damage is not seen in Figure 6.7 on a cage that failed after very little testing.

Figure 6.12: SEM image of the top of the silicon rotor after testing. Fabrication damage is indicated in the boxes and wear damage is contained in the ellipse. The wear damage appears as a rounding at the edge of the rotor where the rotor and ball contact each other. This damage is not seen in untested devices.
6.4 Discussion and Conclusions

As a first proof of concept this bearing design provided an excellent test case for designing and fabricating ball bearings with an integrated ball cage. The solder bond at the center of the cage was the only point of failure in the device and this may have been due to the design of the solder pads and the bonding process. The fabrication process also caused damage to the raceway of the bearing, leading us to explore solutions to this damage when designing new devices. Overall, this design proved that the silicon cage could survive for several hours of continuous operation at speeds of up to 20,000 RPM with a thrust load of 60 mN. This design also showed a measurable difference when the balls were in the sliding and rolling regimes. Further the model for the friction torque presented in Section 4.7 was found to under estimate the frictional torque measured for these bearings. The model for the load torque predicted values 3 orders of magnitude below the measured values. This difference could be attributed to geometric and scaling differences between these bearings and the bearing that we used to create model. The centrifugal
torque was also under estimated but only be a factor of 3. The model has been modified to reflect these observations. The results do indicate that the frictional torque does increase proportional to $\omega^2$ which indicates that centrifugal force does play a significant role in the bearing loss at higher speeds.
Chapter 7

In-Situ Fabrication of a Monolithic Silicon Ball Cage

The dual row style cage was designed to demonstrate the fabrication of a retainer ring from a single piece of silicon. Due to the fact that the weakest point in the radial design presented in Chapter 6 was the solder bond in the ball cage, a geometry and fabrication process that would allow for creating a cage without the need for bonding together parts of the cage were explored. The focus of this exploration was to create a process that would allow for creating the desired geometry while keeping the silicon cage attached to the stator and rotor until after the device was assembled. This work was presented in [Hergert 13a].

The fabrication process consists of a novel method of using windows in the top and bottom wafers of the device to etch away sacrificial beams located on the center wafer that holds the cage and rotor in place, this is shown in Figure 7.1. This process could be adapted for any multi-wafer process in which the design requires that a part on one wafer be released after the wafer stack is assembled. Examples would be complex mechanics such as a watch assembly, the parts of power harvesters or, as in this case, a rotary device. A conceptual drawing of the bearing design can be found in Figure 7.2.
Figure 7.1: Conceptual drawing of the multi-wafer release etch technique (left), DRIE is used to etch a sacrificial beam through windows that have been etched in the upper and lower wafers of the 3 wafer stack. A CAD drawing of the bearing design with red arrows to indicate the location of the sacrificial beams and the rotor release channel before release etching (right A) and after release etching (right B.)

Figure 7.2: Conceptual drawing of the dual groove style bearing design. The cutaway (lower right) shows the orientation of the parts of the device under a thrust normal load.
7.1 Design

Several fabrication methods provide the possibility of creating a cage design from a single piece of material. Several initial ideas came from [Jaeger 02]. One option is to create structures by building the device on the surface of a bulk silicon substrate, then use an isotropic etching process for release as described in [Kovacs 98] or [Fedder 96]. However, this method would have presented design challenges and might have suffered from poor geometry control during the isotropic release process. Another option for creating the device would have been to create the cage on the surface of a silicon substrate with an intermediate sacrificial layer. This method has been shown to work for devices made of polysilicon using silicon dioxide as the sacrificial layer as in [Mehregany 98] or [Pister 92]. After etching away the silicon dioxide layers the structures are free to move. This is a very attractive option, however, we were not able to create the polysilicon layers needed for this technique in our laboratory. Also, it would not have been possible to use this method to create a cage over or within the deep trench of the bearing raceway. Another option would have been to create the entire bearing by defining the geometry in layers of electroplated metals formed in or on a polymer sacrificial layer as shown in [Guckel 98] and [Cohen 10]. This would have required multiple masks and it would have been challenging to incorporate the balls into the bearing design.

In [Frechette 05] a sacrificial silicon beam was used to hold a turbine in place during the fabrication process, and this tab was snapped out using micromachined silicon needles before testing. A further improvement is the use of a laser to remove the sacrificial tabs as described in [Lin 99]. A similar method is described in [Abraham 08]. Using a silicon beam to support the ball cage during device assembly seemed to be the most accessible design option with the processing techniques available. This method allows for creating the exact geometries needed for the device, uses standard DRIE processing to create the structures, and provides a rigid and reliable support mechanism to hold the cage in place during assembly. However, a method of removing the support beams was required. As a DRIE step is required to release the
7.1 Design

rotor of the bearing after assembly, it seemed natural to try to use this etch step to remove the sacrificial beams as well. This idea was further supported by the use of stencils as masks as reported in [Pang 88] and [Villanueva 09]. The beam release concept is illustrated in Figure 7.3.

![Figure 7.3: Conceptual drawing of the cage support release etching. This is done in two DRIE etch steps, one from the top and one from the bottom.](image)

The design of the bearing required that windows were etched all the way through the silicon of the top and bottom of the device to expose the silicon beams holding the cage in place on the middle wafer of the device. These windows had to be large enough to compensate for etch lag effects which are dependent on feature size and that become more dominant with increased feature depth. Etch lag is further explained in Section 3.2.4. The etch windows with and sacrificial beam can seen in Figure 7.4.

In our first investigation of the sacrificial beam release method we explored using 8 beams to secure the cage and the rotor. These beams were located above the balls and therefore only exposed to DRIE etching for one of the rotor release etch steps. The beam had a width of 20 µm and the etch window had a height (in the radial direction) of 40 µm and a width of 80 µm.
The perceived advantages of this design was that the beams were closer to the surface of the middle wafer and thus closer to the top of the etch window, 8 beams provided excellent support during assembly, and the beam was protected during one of the DRIE etching cycles by the ball which was intended to eliminate excessive etch damage to the bearing side wall. However, This design did not allow sufficient time for the beam to be etched all the way through on a single etch cycle; only 50 µm of the total 150 µm beam thickness was etched.

The final design had only 4 beams which were exposed to DRIE etching during both the top and bottom rotor release etch cycles. This was accomplished by designing the beams into the space between the top and bottom ball in the cage. The cage release windows were also increased in size to a height of 110 µm and width of 80 µm. The thickness of the cage support beams was kept at 20 µm. With these design changes the cage support beams were completely etched away during the rotor release etch steps. A diagram showing the window design with the beam is provided in 7.4 and a comparison of the two design iterations is provided in Table 7.1. The final layout design of the device top, raceway and middle are shown in Figure 7.5.
Figure 7.4: Drawing of the cage release window showing the cage support beam, the top and bottom balls, the release channel and measurement labels for the cage features.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>First Design</th>
<th>Final Design</th>
</tr>
</thead>
<tbody>
<tr>
<td>Beam Width ($W_b$)</td>
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<td>20 $\mu$m</td>
</tr>
<tr>
<td>Window Width ($W_w$)</td>
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<td>150 $\mu$m</td>
</tr>
<tr>
<td>Window Height ($H_w$)</td>
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<td>110 $\mu$m</td>
</tr>
<tr>
<td>Beam Thickness</td>
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<td>100 $\mu$m</td>
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<tr>
<td>Total Support Beams</td>
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<td>4</td>
</tr>
<tr>
<td>Release Etch Beam Exposure</td>
<td>Top Only</td>
<td>Top and Bottom</td>
</tr>
<tr>
<td>Successful Release Etch</td>
<td>No</td>
<td>Yes</td>
</tr>
</tbody>
</table>

Table 7.1: Design parameters for the cage support beam and etch windows for the first and second design iterations.
7.1 Design

Figure 7.5: Layouts on both sides of the top/bottom wafers, and on the center wafer of the dual row style design with labels indicating important features.
7.2 Device Fabrication

Due to the complexity of the device fabrication, the process has been broken up into 3 distinct sections: Top and Bottom Die Fabrication, Center Die Fabrication, and Device Assembly and Release Etching. For more information on the processes described in this section please refer to Chapter 3.

7.2.1 Top and Bottom Die Fabrication

The top and the bottom die are fabricated on the same wafer using a 100 mm diameter, p-type, <100>, DSP silicon wafer with a thickness of 525±25 μm. The wafers also had 1 μm of thermally grown silicon dioxide. A description of each processing step follows. The steps are labeled with the letter corresponding to the depiction of the step in Figure 7.6.
7.2 Device Fabrication

7.2.1.1 Step A (Front Side)- Plate Solder Pads

The wafer is sputter coated with a chrome and copper seed layer (not shown in the process flow diagram.) The wafers are coated with photoresist and patterned with the solder pad mask. The solder pads are electroplated with a 2\(\mu\)m nickel diffusion barrier, 3\(\mu\)m of tin and 300 nm of gold. The photoresist is then stripped and the exposed seed layer (not shown in the diagram) is etched to expose the silicon dioxide below.
7.2 Device Fabrication

7.2.1.2 Step B (Front Side) - Pattern Raceway Stand-off Trench

The wafer is coated with photo resist and is patterned with the raceway stand-off mask. The exposed oxide is then etched using RIE. The photoresist is then stripped from the wafer.

This feature will be etched below the level of the contact region of the ball bearing raceway. This allows for etching the cage release window all the through the wafer. Initial designs that did not have this recess showed damage to the contact region of raceway caused during the release etch steps. Devices with the recessed showed less damage to the raceway from the window and release etch.

7.2.1.3 Step C (Front Side) - Pattern the Raceway and Etch the Raceway Stand-Off

The wafer is coated with photoresist and the raceway is patterned on the wafer. The raceway stand off (patterned in the oxide in the previous step) is etched to a depth of 100 µm using DRIE. As the raceway is protected by oxide it is not etched. The photoresist is left on the wafer for the next processing step.

7.2.1.4 Step D (Front Side) - Etch Bearing Raceway

The exposed oxide is etched from the the wafer. The raceway is etched to a depth of approximately 225 µm using DRIE. The depth of the standoff also increased during this etch creating a stepped profile. After the raceway is etched, the photoresist is removed from the wafer. This completes all of the processing on the top side of the wafer.

7.2.1.5 Step E (Back Side) - Define Cage Release Window and Identification Marks

The back side of the wafer is coated with photoresist and the cage release windows are patterned in the oxide using RIE. The die identifier and the
device release lines are also etched in the oxide. After the oxide etch the photoresist is stripped from the wafer.

7.2.1.6 Step F (Back Side) - Define Testing Adaptor Feature and Bearing Release Channels; Pre-Etch Cage Release Window

Photoresist is coated on the wafer. The bearing release channel and the test adaptor feature are patterned in the resist. The cage release windows are etched to a depth of approximately \( 150 \mu m \). The photoresist is left on the wafer after this step.

7.2.1.7 Step G (Back Side) - Pre-Etch Testing Adaptor Feature and Bearing Release Channels; Etch Cage Release Window Through the Wafer

The exposed oxide is etched by RIE using the photoresist from the previous step as a mask. The test adaptor feature and the bearing release channel are etched to a depth of approximately \( 125 \mu m \) using DRIE. During this etch the cage release windows and the device edge are etched through the wafer (meeting the features etched on the front side of the wafer.) This leaves the windows needed to etch the beams holding the cage in place on the center die of the device.

7.2.1.8 Images of Fabricated Top and Bottom Die

Below are SEM images of completed top and bottom die used in the fabrication of the dual row style cage bearing. The devices depicted in the images were early prototypes and show signs of spires and the loss of solder pads (in Figure 7.7). These sample images do show the die with the features and etch depths used in the final device.
7.2 Device Fabrication

Figure 7.7: SEM image of the raceway of the bottom die for the dual row style cage device.

Figure 7.8: SEM of the raceway side of the bottom die for the dual row style cage device, the arrows indicate features that have been etched completely through the wafer.
7.2 Device Fabrication

Figure 7.9: SEM image of the top of the top die for the dual row style cage device. The device identification marks, test adaptor features, bearing release channels and the cage release windows are clearly visible.

7.2.2 Center Die Fabrication

The center die is fabricated using a 100 mm diameter, p-type, <100>, DSP silicon wafer with a thickness of 525±25 μm. The wafer also had 1 μm of thermally grown silicon dioxide. A description of each processing step follows. The steps are labeled with the letter corresponding to the depiction of the step in Figure 7.10.

The fabrication process flips from the top side to the back side and then back to the top side of the wafer. This is due to the fact that the wafer would often break during the second electroplating process if the top side had deep etched features. In order to eliminate the risk of breaking the wafers during the second plating process all of the DRIE etching was done after both sides of the wafer had been electroplated with the solder pads.
7.2 Device Fabrication

7.2.2.1 Step A (Top Side) - Plate Solder Pads

The wafer is sputter coated with a chrome and copper seed layer (not shown in the process flow diagram.) The wafers are coated with photoresist and patterned with the solder pad mask. The solder pads are electroplated with a 2 μm nickel diffusion barrier, 3 μm of tin and 300 nm of gold. The photoresist is then stripped and the exposed seed layer (not shown in the diagram) is etched to expose the silicon dioxide below.
7.2.2.2 Step B (Top Side) - Define Cage with Support Beams

Photoresist is deposited on the top side of the wafer. The cage with support beams is patterned in the photoresist and then RIE is used to etch the exposed silicon dioxide. The photoresist is then stripped from the wafer.

7.2.2.3 Step C (Back Side) - Plate Solder Pads

The top side of the wafer is coated with photoreist to protect the existing solder pads during the electroplating step. The back side of the wafer is then sputter coated with a chrome and copper seed layer (not shown in the process flow diagram.) The wafers are coated with photoresist and patterned with the solder pad mask. The solder pads are electroplated with a 2µm nickel diffusion barrier, 3µm of tin and 300nm of gold. The photoresist is then stripped and the exposed seed layer (not shown in the diagram) is etched to expose the silicon dioxide below.

7.2.2.4 Step D (Back Side) - Define Cage with Support Beams

Photoresist is deposited on the back side of the wafer. The cage with support beams is patterned in the photoresist and then RIE is used to etch the exposed silicon dioxide. The photoresist is then stripped from the wafer.

7.2.2.5 Step E (Back Side) - Pattern Cage Without Support Beams and Pre-Etch Cage With Support Beams

Photoresist is deposited on the wafer and is patterned with the cage design without support beams. The pattern from the previous step is the etched to a depth of approximately 50µm. This etch will establish the height of the beams that will support the rotor and the cage during the assembly of the device. The photoresist is left on the wafer for the next step.
7.2.2.6 Step F (Back Side) - Etch Cage to Final Depth and Reduce Beam Thickness

The exposed silicon dioxide is etched. This will expose the silicon at the top of the support beams. The cage and the beams are etched approximately 225 µm so that the ball pocket is at a final depth of 275 µm. This also brings the area around the cage to a depth below the center of the wafer thickness. The photoresist is stripped from the wafer and this completes the processing of the back side of the wafer.

7.2.2.7 Step G (Top Side) - Pattern Cage Without Support Beams and Pre-Etch Cage With Support Beams

Photoresist is deposited on the top side of the wafer. The photoresist is patterned with the cage design without the supporting beams. In Step B (Section 7.2.2.2) the cage with support beams was patterned on the top side of the wafer, therefore the oxide pattern that defines the support beams already exists on this side of the wafer. This pattern is etched to a depth of approximately 50 µm. This established the height of the beam that will support the rotor and the cage during device assembly. The photoresist is left on the wafer for the next step.

7.2.2.8 Step H (Top Side) Etch Area Around Cage Through the Wafer and Reduce the Beam Thickness

The exposed silicon dioxide is etched from the wafer, thus exposing the silicon on top of the support beams. DRIE is then used to etch the cage and the beams to a depth of 225 µm. This provides a final ball pocket depth of 275 µm and support beam thickness (total from both sides of the wafer) of approximately 100 µm. The regions between cage and the rotor and the stator are etched completely through the wafer. Thus the rotor and the stator are held in place by only the 4 support beams. The photoresist is stripped from the wafer and the wafer is broken into die for assembly.
7.2.2.9 Images of Fabricated Center Die

Below are SEM images of the center die made by following the process flow described above. From the top view, presented in Figure 7.11, it is possible to see the regions that have been etched through the wafer around the edges of the cage. The image also shows how the 4 beams that support the cage and the rotor by attaching them to the stator. The Angled view of the device clearly shows the varying step heights that have been achieved in the process flow. The reduced height of the support beams is also evident in the image.

![Image of center die](image.jpg)

Figure 7.11: SEM image of the center die used for the dual row style cage device. The cage can be seen at the center of the device supported by 4 beams. The black regions in the image have been etched through the wafer.
7.2.3 Device Assembly and Release Etching

After the top, bottom and center die are made the device must be assembled and the cage and the rotor must be released. The following sections will detail the steps that are required for the assembly and release. The steps are labeled with the letter corresponding to the depiction of the step in Figure 7.13.
7.2 Device Fabrication

Figure 7.13: Device fabrication process flow for assembly and release etching of the dual row style cage device. The red arrows show the location of the sacrificial beams and the bearing release channel before (A) and after (B) the release etches.

7.2.3.1 Step A - Device Assembly

The device is assembled by first placing steel balls in the ball pockets on one side of a center die. Solder flux is then painted onto the solder pads and
200 µm diameter solder balls are placed at the center of each pad. Steel alignment pins are inserted into the alignment holes located on either side of the die. The bottom die is then slid onto the alignment pins. The center/bottom die stack is then flipped over while pressing the stack together with tweezers. The alignment pins keep the stack, steel balls and solder balls in the proper position. Steel balls are then inserted into the ball pockets on the other side of the center die. The top die is then slid onto the alignment pins, completing the 3 wafer stack. The die stack is then transferred to a custom bond aligner where a low force is applied to the stack. A butane torch is used to rapidly head the stack and reflow the solder balls. During the heating process the steel alignment pins expand and creating a snug fit with the alignment holes thereby improving the overall alignment of the stack during solder reflow. After reflow the devices are ready for the final release etching.

7.2.3.2 Step B - Release Etching

After assembly the devices are mounted on a backing wafer with the bottom side facing up. The patterned silicon dioxide from the previous processing steps is used as a mask for the final release etch steps. The bottom side of the device is etched, using DRIE, until the bearing release channel reaches the bearing raceway. During this etch the support beams, holding the cage and rotor in place, are also etched part of the way through. This is due to the windows in the silicon located above the beams.

After the bottom of the device is etched the device is removed from the backing wafer and mounted on a new backing wafer with the top of the device facing up. The top bearing release channel is then etched, using DRIE, until it meets the raceway. The support beams are completely etched away during this step. Thus at the completion of this step, when the device is removed from the carrier wafer, the rotor and cage are completely released from the stator.
7.2.3.3 Images of Assembled and Released Devices

Figures 7.13 and 7.14 show SEM images of the top and bottom of assembled and released devices. The top view presented here is of a 6 ball design that was too unstable to survive testing. The diameters of the top and bottom release channels differ because these are thrust bearing designs and therefore have a load support surface on the rotor at the top of the device and on the stator at the bottom of the device. An image of the released cage can be found in Figure 7.18 and a released rotor can be seen in Figure 7.19. The supports beams are completely etched from the cage and the rotor with only a slight feature to indicate where the beams were located before etching.

Figure 7.14: SEM image of the top of an assembled and released dual row style cage device. The test adaptor, bearing release channel, cage release windows and identification marks are visible.
7.3 Testing

The devices were tested on the micro-tribometer discussed in Section 5.1. A thrust load was applied to the rotor (center) of the bearing. One device that contained 8 balls was tested successfully. Another device that contained only 6 balls was extremely unstable and failed catastrophically during testing, breaking the device and the silicon test platform.

The 8-ball device that was successfully tested had a cage width of 300 µm, a diametral play ($P_d$) of 40 µm. The raceway diameter ($d_m$) was 1.75 mm. A thrust load of 40 mN was applied to the bearing. It was tested at speeds ranging from 50 RPM to 5100 RPM.

7.4 Test Results

The measured torque of the dual row device is plotted in Figure 7.16. In this figure the torque is plotted as a function of the speed. As can be seen in the figure, the bearing did not show signs of ball sliding at any of the measured speeds. The measured frictional torque does not show the expected
relationship to centrifugal forces, as was observed and described in Section 6.3.2, there is also no indication of ball sliding at any of the measured speeds. The frictional torque is almost constant over the tested speed range, with the exception of a jump in the measured torque above 4000 RPM. The torque is also significantly higher than seen in the radial design. For example the frictional torque for this design at speeds below 4000 RPM is close to measured the fictional torque of the radial bearing at 18,000 RPM. The observed frictional torque implies that this design is much less efficient than the radial design. The power loss of the bearing is plotted in Figure 7.17. Even though the power loss varies with $\omega$ rather then $\omega^2$, as seen in the radial design, the power loss over the measured range is almost and order of magnitude greater for the dual row bearing.

![Dual Row Bearing Torque Measurements](image)

Figure 7.16: Torque measurements of 4 tests of the dual row ball bearing design.
7.5 Wear

Figure 7.17: Measured bearing loss of the dual row ball bearing design for all 4 tests plotted with the power loss for the radial bearing over the same range. Though the power loss varies with $\omega$ rather than $\omega^2$, as seen in the radial design, the loss of the dual row bearing is almost an order of magnitude greater than the radial design over the same range.

7.5 Wear

Though minimal wear was apparent on the bearing it should also be noted that the bearing was only tested for a total of 2 h. Abrasive wear is apparent at the top edges of the cage pockets. This wear is similar to seen at the edge of the cage pockets in the radial bearing design and due to abrasion from the ball and possible from ball/cage shock. The cage wear is shown in Figure 7.18. The ball contact surfaces on both the rotor and stator showed signs of abrasive and possible adhesive wear. The pattern on both raceway surfaces was consistent with that seen on the contacts of other thrust style bearings. Figure 7.19 shows the wear on the rotor raceway, while Figure 7.20 shows the wear on the stator raceway.
7.5 Wear

Figure 7.18: SEM image of the cage after testing. Wear at the edge of the ball pocket is indicated in the ellipse on the left. The location of the support beam is indicated in the ellipse on the right; there is very little indication of the location of the support beam after the release etch.

Figure 7.19: SEM image of the rotor after testing. The arrow on the left points to rotor wear which appears as a discolored region on raceway. The ellipse in the figure indicates where the support beam was located before etching.
Figure 7.20: SEM image the stator after testing. The arrow indicates the location of the wear, which appears as a discolored region on the raceway. Pillar defects are present at the edge of the raceway and are an artifact of the multi-step etching processes.

7.6 Discussion and Conclusions

This chapter presented a new fabrication technique that allows for the release of moving parts that are embedded in the middle of a multi-wafer stack. This is possible by using DRIE to etch windows through the top and bottom wafers and then have beams on the center wafer that are exposed to DRIE etching after assembly. The technique was applied to a ball bearing with a dual row style cage. Though the cage design is inefficient and unstable the fabrication technique has proven to be viable. In Figures 7.18 and 7.19 the regions where the cage support beams were located are visible. The beams are completely removed from the cage and rotor surfaces with only small features remaining to indicate where the beams were attached. It should also be noted that in Figure 7.19 the etch windows are visible and that the windows and the surrounding features have experienced minimal damage during the release etch process. This is a new release technique that could have broad application in the fabrication of gear trains, power MEMS devices, or any application in which a part with a large range of motion must
be released after assembly of a device.

Though the fabrication technique proved to be a valid design approach, it should be noted that the design of the ball cage is this chapter is significantly less efficient than the radial design presented in Chapter 6. The tests results show an almost constant frictional torque over the tested range of 50 RPM to 5000 RPM. And though the power loss of this design is proportional to $\omega$, rather than $\omega^2$ as seen in the radial design, the power loss for this design is almost an order of magnitude greater. Due to the inefficiency of and the complexity of this design further exploration of this cage geometry is not recommended. However, the fabrication technique could have much broader application in complex MEMS devices.
Chapter 8

Micro-Turbines with Integrated Silicon Ball Cage

This chapter describes the investigation of the effect of incorporating silicon retainers into silicon MEMS thrust micro-ball bearings reported in [Waits 10] and integrated in the SMT devices described in Section 5.2. Section 4.1 describes some of the inherent advantages of using a ball cage in ball bearing designs. The bearing investigated in this chapter is a thrust-style bearing which lends itself to incorporation of a monolithically fabricated silicon retainer. To allow functional testing the bearing is integrated with a silicon micro-turbine (SMT), following [McCarthy 09]. This platform was chosen as it provides a proven method for actuation of the bearing during characterisation. Several different retainer designs were investigated in turbines of two sizes, with 5 mm- and 10 mm-diameter turbine rotors respectively. Photographs of devices of both sizes, together with a cut-away schematic view showing the retainer, are shown in Figure 8.1. The different designs were compared in terms of losses, inferred from the variation of turbine input power with rotation speed; measurements of repeatability and longevity were also made on selected devices.

An early investigation of the retainer geometries in 5 mm-diameter SMTs was reported in [Hergert 11]. Since the publication of the early work, the fabrication process for the devices was improved to allow for successfully
fabricating and test devices with a 10 mm rotor diameter, this work was first reported in [Hergert 13b]. This was achieved by making the turbine with a 1 mm-thick silicon wafer rather than a 500 µm thick wafer and changing the etching steps to better account for etch lag. These changes allowed for an increase in the thickness of silicon above the bearing raceway, making it sufficient to withstand the forces of bearing operation in larger devices.

Figure 8.1: Photographs of the both the 5 mm and 10 mm devices with a British Pound coin for scale (top), and a cutaway view of the device showing the retainer ring (bottom).

8.1 Retainer Ring Design Considerations

Figure 8.2 shows a cut-away schematic view of a conventional thrust bearing incorporating a so-called ball-riding (BR) retainer [Harris 06b]. In this style of bearing, the retainer is supported entirely by the rolling elements which are held captive in suitably shaped pockets, and consequently there is no contact between the retainer and the bearing rings. Alternative designs for radial/mixed load bearings employ inner ring (IRL) or outer ring (ORL) land riding retainers which are sized to fit the cylindrical surface of either the inner or outer bearing ring [Harris 73]. IRL retainers are driven by a friction force between the retainer and the inner ring, and under optimal conditions this
can result in negligible loading of the rolling elements by the retainer. ORL retainers are guided by the outer ring and are therefore subject to a drag force; this type of retainer tends to be used in high-speed applications.

Figure 8.2: Cut-away schematic view of a conventional thrust bearing with a ball-riding retainer.

Five different designs of retainer ring were investigated for the work presented in this chapter, designated as Full Ring, Full Skeleton, Half Skeleton, Outer Open and Inner Open. Scanning electron microscope (SEM) images of all five retainer types are shown in Figure 8.3. Also shown in this figure are schematic cross-sections for the different retainer types. The left-hand schematic also illustrates how the bearing and micro-turbine are integrated; the device is assembled at die level as a bonded two-die stack, comprising a lower "thrust" die and an upper "turbine" die, with the steel balls and cages manually inserted between the die. The bearing is formed at the interface between the two dies, with the rotor blades and guide vanes of the turbine being defined in the upper surface of the turbine die.
Figure 8.3: (a) SEM images showing the 5 mm retainer ring designs: Full Ring (top left); Full Skeleton (top right); Half Skeleton (bottom left); Outer Open (bottom right); Inner Open (centre). (b) Schematic cross-sections of bearings with the different retainer types.

Comparing Figures 8.2 and 8.3, two important differences, both arising from micro-fabrication process constraints, can be seen between the MEMS thrust bearing and its conventional counterpart. Firstly, the raceways on the conventional bearing are curved in cross-section so that the balls have a single point of contact with each raceway and the bearing is inherently self-aligning when subject to an axial load. In contrast, the silicon raceways are rectangular, with the balls riding on essentially flat surfaces top and bottom. The balls are constrained to follow a nominally circular path by the
sidewalls of the bearing raceway, with which they make intermittent contact, and also by the retainer. Secondly, while the pockets in the conventional retainer are shaped to enable it to ride on the balls, this cannot be achieved with the silicon retainer because the pockets are cylindrical. Instead the Inner Open retainer rides on the land adjacent to the bearing raceway on the outer (stator) side, while the Outer Open retainer is supported by the land on the inner (rotor) side. The inner land is slightly elevated with respect to the outer, and consequently the Full and Skeleton retainers should ride in the same way as the Outer Open type. However, if the height difference is sufficiently small then bearing vibration and dynamic distortions of the retainer may bring it into contact also with the outer land. The different silicon retainers can reasonably be classified as either inner ring or outer ring land riding, though they differ from conventional IRL and ORL retainers in that they ride on flat surfaces in the plane of the bearing rather than on cylindrical surfaces. It should also be noted that interactions between the retainer and the balls when the bearing is operational may cause the retainer to ride up and make contact with the top of the retainer raceway; the top of the raceway is recessed to provide a stand-off and reduce the contact area under these conditions. Key design parameters for the bearings developed in this work are given in Table 8.1.

The width of the bearing raceway was set at 510 µm giving a clearance of 5 µm either side of the 500 µm-diameter steel balls. This clearance determines the lateral play in the rotor position, and the value chosen was the smallest that would guarantee easy insertion of the balls, taking into account fabrication tolerances. The width of the retainer raceway was chosen to give a clearance of 25 µm either side of a Full Ring retainer, ensuring that no retainer-raceway sidewall contact would occur with any of the designs. The bearing raceway was made slightly higher than the ball radius to ensure a planar contact between the ball and the raceway sidewall. The height of the retainer raceway was then set so that the combined height of the raceways, including the thickness of the solder bond between the wafers and the recess in the top of the retainer raceway, was nominally 10 µm larger than the ball diameter i.e.
### 8.1 Retainer Ring Design Considerations

#### Table 8.1: Key design parameters for large (10 mm) and small (5 mm) devices. All dimensions are in µm.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
<th>Large Device</th>
<th>Small Device</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R_{BP}$</td>
<td>Radius of ball path</td>
<td>5000</td>
<td>2600</td>
</tr>
<tr>
<td>$R_I$</td>
<td>Retainer inner radius</td>
<td>full, outer open</td>
<td>4647.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td>inner open</td>
<td>4787.5</td>
</tr>
<tr>
<td>$R_O$</td>
<td>Retainer outer radius</td>
<td>full, inner open</td>
<td>5352.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td>outer open</td>
<td>5212.5</td>
</tr>
<tr>
<td>$g$</td>
<td>Ball pocket opening</td>
<td>outer open</td>
<td>278</td>
</tr>
<tr>
<td></td>
<td></td>
<td>inner open</td>
<td>267</td>
</tr>
<tr>
<td>$R_B$</td>
<td>Ball radius</td>
<td></td>
<td>250</td>
</tr>
<tr>
<td>$R_P$</td>
<td>Pocket radius in retainer</td>
<td></td>
<td>252.5</td>
</tr>
<tr>
<td>$W_{BR}$</td>
<td>Width of bearing raceway</td>
<td></td>
<td>510</td>
</tr>
<tr>
<td>$W_{RR}$</td>
<td>Width of retainer raceway</td>
<td></td>
<td>755</td>
</tr>
<tr>
<td>$H_{BR}$</td>
<td>Height of bearing raceway</td>
<td></td>
<td>260</td>
</tr>
<tr>
<td>$H_{RR}$</td>
<td>Height of retainer raceway (incl. solder)</td>
<td></td>
<td>245</td>
</tr>
<tr>
<td>$H_{SO}$</td>
<td>Stand-off height</td>
<td></td>
<td>5</td>
</tr>
<tr>
<td>$H_R$</td>
<td>Height of retainer</td>
<td></td>
<td>225</td>
</tr>
</tbody>
</table>
Table 8.2: Comparison of retainer parameters by geometry.

<table>
<thead>
<tr>
<th>Cage Name</th>
<th>No. Balls 10 mm/5 mm</th>
<th>Ball Fill Factor</th>
<th>Contact Area</th>
<th>Stability</th>
<th>Mechanical Strength</th>
</tr>
</thead>
<tbody>
<tr>
<td>Full Ring</td>
<td>32 / 16</td>
<td>50%</td>
<td>High</td>
<td>High</td>
<td>High</td>
</tr>
<tr>
<td>Full Skeleton</td>
<td>32 / 16</td>
<td>50%</td>
<td>Medium</td>
<td>High</td>
<td>Medium</td>
</tr>
<tr>
<td>Half Skeleton</td>
<td>16 / 8</td>
<td>25%</td>
<td>Very Low</td>
<td>Low</td>
<td>Medium</td>
</tr>
<tr>
<td>Outer Open</td>
<td>32 / 16</td>
<td>50%</td>
<td>Low</td>
<td>Medium</td>
<td>Low</td>
</tr>
<tr>
<td>Inner Open</td>
<td>32 / 16</td>
<td>50%</td>
<td>Low</td>
<td>Medium</td>
<td>Low</td>
</tr>
</tbody>
</table>

\[ \Delta = H_{BR} + H_{RR} + H_{SO} - 2R_B = 10 \mu m \quad (8.1) \]

where all the variables are as defined in Table 8.1. In choosing the design value of \( \Delta \), tolerances in etch depth and solder bond thickness were taken into account. It is essential that \( \Delta > 0 \) so that there is clearance above the balls while the device is being assembled.

### 8.2 Retainer Ring Design Variations

The five retainer designs were chosen to explore the effects on the bearing performance of sliding friction, retainer rigidity, ball pocket shape and ball complement. The differences between the designs are summarised in Table 8.2. Considering first sliding friction, this is expected to be most significant in the Full Ring design which has the largest overlap area with the inner and outer lands. The Skeleton designs reduce the overlap area on either side by removing material between the ball pockets, while the Open designs eliminate it entirely on one side of the raceway. Removal of material from the Skeleton and Open designs will also reduce the mechanical rigidity of the retainer. This has been found to improve the performance in conventional bearings under some loading conditions [Weinzapfel 09].

In addition to distributing the balls uniformly around the raceway, the retainer will also limit the radial excursions of the balls, encouraging them to follow a stable circular trajectory and reducing the extent to which they interact with the raceway sidewalls; this is expected to be beneficial in terms
of friction and wear. The radial play allowed by the ball pockets will differ for the Full/Skeleton and Open designs. Considering first the Full/Skeleton retainers, and referring to the schematic in Table 8.1, the radial play in the cylindrical pockets is expected to be:

$$\delta = R_P - \sqrt{R_B^2 - h_R^2}$$

(8.2)

Where $R_P$ is the pocket radius, $R_B$ is the ball radius, and $h_R$ is the height of the retainer above the centre of the ball. The value of $h_R$ when the bearing is at rest depends on the height of the bearing raceway. However, during operation $h_R$ can lie anywhere in the range:

$$(H_{BR} - R_B) \leq h_R \leq (H_{BR} - R_B) + (H_{RR} - H_R - \Delta)$$

(8.3)

Using values from Table 8.1, Equation 8.3 gives $10 \mu m \leq h_R \leq 20 \mu m$. The maximum play predicted by Equation 8.2 is then $\delta = 3.3 \mu m$. This is larger than the difference between the pocket and ball radii because the retainer sits above the centre of the ball. The open pockets on the Inner Open and Outer Open designs will allow the balls more radial play. The generalisation of Equation 8.2 in the case where the pocket has an opening on one side is:

$$\delta = \sqrt{R_P^2 - (g/2)^2} - \sqrt{R_B^2 - h_R^2 - (g/2)^2}$$

(8.4)

where $g$ is either the opening width (open side of pocket) or zero (closed side). For example, for the large Inner Open retainer, $g = 267 \mu m$ and with $h_R = 20 \mu m$. Equation 8.4 gives $\delta_+ = 3.3 \mu m$ and $\delta_- = 3.9 \mu m$. The other Open designs have similar gaps and hence will yield similar results.

The above calculations ignore the effects of wear at the bottom of the retainer, and so apply only when the bearing is newly fabricated. Over time the ball pockets will become enlarged due to abrasion by the balls, and this will increase the ball pocket play for all designs. It is expected that this effect will be more pronounced for the Open designs, since the edges of the openings are likely to exhibit higher rates of wear, so that the
difference in play between the Full/Skeleton and Open designs will become more pronounced throughout the bearing lifetime.

The ball complements for the Full Ring, Open and Skeleton designs were 32 balls for the large devices and 16 balls for the small devices, corresponding in each case to a fill factor of about 50%. These values were halved in the Half Skeleton designs. Increasing the number of balls is generally beneficial for performance because it lessens the load per ball, stiffens the bearing and reduces vibration \cite{Harris73}. It was therefore expected that losses would be higher in devices with Half-Skeleton retainers.

\section*{8.3 SMT Design}

The micro-turbine design adopted for the 10 mm-diameter devices was as described in \cite{McCarthy09}, and this design was simply scaled down for the 5 mm-diameter devices, a description of the SMT can also be found in Section \ref{sec:SMT}. No attempts were made to optimize the turbine performance. However, the bearing housing was altered to accommodate larger 500 μm diameter balls, primarily to ease the assembly of the devices. Also the turbine wafer thickness was increased to 1 mm so that it could accommodate the retainer raceway on the back side and the rotor blades and guide vanes on the front side (see Figure \ref{fig:SMT}) while leaving a sufficient thickness of silicon in between to withstand the thrust force applied to the bearing during operation. The process flow reported in \cite{Hergert11} did not leave a sufficient silicon thickness above the bearing to support the normal load during operation, resulting in rapid failure of 10 mm-diameter devices due to silicon fracture.

\section*{8.4 Device Fabrication}

Due to the complex fabrication process; the device fabrication is broken into several sections. The retainer, the turbine and the thrust side of the device are all fabricated separately. The following sections will describe the fabrication process for each part of the device followed by the assembly and release
of the device.

8.4.1 Retainer Fabrication

The retainer rings are fabricated from a 225 µm-thick, 100 mm-diameter silicon wafer with a 1 µm-thick thermal oxide layer on both sides. Photoresist (PR) is spin-coated onto the wafer and patterned by photolithography, and reactive ion etching (RIE) is used to etch the exposed silicon dioxide. Deep reactive ion etching (DRIE) is then used to etch through the wafer to release the retainer rings. A halo mask design was used to provide better etch rate uniformity and side wall quality. The silicon dioxide protects the ring in the final stages of the SMT fabrication process when DRIE etches are used to release the turbine rotor. Previous designs lacking the silicon dioxide layer showed etch damage to the retainer caused during the release etch, an example of the damage is shown in Figure 8.4.

![Figure 8.4: SEM image of a SMT with the rotor removed to expose the retainer and balls, release etch damage to the retainer is indicated by the arrow.](image)

8.4.2 Turbine Wafer Fabrication

The turbine wafer is fabricated using a 100 mm diameter, p-type, <100>, DSP silicon wafer with a thickness of 1000±50 µm. The wafers also had 1 µm
8.4 Device Fabrication

of thermally grown silicon dioxide. A description of each processing step follows. The steps are labeled with the letter corresponding to the depiction of the step in Figure 8.5.

Figure 8.5: Process flow for the turbine wafer of the SMT with a silicon ball cage.
8.4 Device Fabrication

8.4.2.1 Step A (Top Side) - Electroplate Solder Pads

The wafer is sputter coated with a chrome and copper seed layer (not shown in the process flow diagram.) The wafers are coated with photoresist and patterned with the solder pad mask. The solder pads are electroplated with a $2 \mu m$ nickel diffusion barrier, $3 \mu m$ of tin and $300 \text{ nm}$ of gold. The photoresist is then stripped and the exposed seed layer (not shown in the diagram) is etched to expose the silicon dioxide below.

8.4.2.2 Step B (Top Side) - Pattern Turbine Release Channel

Photoresist is deposited on the wafer. The turbine release channel is patterned in the photoresist and then etched into the exposed oxide using RIE. The photoresist is then stripped from the wafer.

8.4.2.3 Step C (Top Side) - Pattern Cage Raceway Standoff

Photoresist is deposited on the wafer. The cage raceway standoff is then patterned in the photoresist. The $200 \text{ nm}$ of the exposed oxide is etched. This provides for a stepped oxide mask that will be used later to define a recess in the cage raceway. The photoresist is then stripped from the wafer.

8.4.2.4 Step D (Top Side) - Pattern Cage Raceway and Etch Bearing Release Channel

Photoresist is deposited on the wafer and patterned with the cage raceway. The multi layer pattern that is produced by this step is shown in Figure 8.6. This pattern will allow for the fabrication of 3 distinct steps in the raceway. The turbine release channel (the only region with fully exposed silicon) is etched to a depth of $200 \mu m$ using DRIE. At this point the thinned oxide mask of the cage raceway standoff will begin to fail. The photoresist is not stripped at the end of this step.
8.4 Device Fabrication

Figure 8.6: Photograph showing the different mask layers on the turbine raceway before the bearing release channel is etched. The solder pads are also covered with photoresist.

8.4.2.5 Step E (Top Side) - Etch Cage Raceway Stand Off

The oxide mask for the cage raceway standoff is etched using RIE. Care is taken to not remove the mask for the cage raceway. The cage raceway standoff is then etched to a depth of 3 µm. The Photoresist is not removed after this step.

8.4.2.6 Step F (Top Side) - Etch Cage Raceway

After the cage raceway standoff is etched, all of the exposed oxide on the wafer is etched using RIE. The cage raceway is then etched to a depth of 240 µm using DRIE. The photo resist is then stripped from the wafer and processing of the Top Side of the wafer is complete.
8.4.2.7 Step G (Back Side) - Pattern Turbine Release Channel

Photoresist is deposited on the Back Side of the wafer. The turbine release channel is patterned in the resist and then etched into the exposed oxide using RIE. The photoresist is then stripped from the wafer.

This step is performed on the Top Side and Back Side of the wafer because etch lag (see [Jansen 97]) reduces the depth of the release channel on both sides of the wafer. If the release channel is etched from only one side, the resulting silicon thickness on the turbine is not enough to support the bearing forces of the 10 mm diameter devices.

8.4.2.8 Step H (Back Side) - Pattern Turbine Blade and Etch Bearing Release Channel

Photo resist is deposited on the wafer and patterned with the turbine blades. The turbine release channel is etched to a depth of 250 µm at which point the exposed oxide mask begins to fail. The photo resist is not stripped at the end of this step.

8.4.2.9 Step I (Back Side) - Pre-Etch Turbine Blades

The exposed oxide is etched using RIE to pattern the turbine blades. The turbine blades are then etched to a depth of 300 µm. The photoresist is then stripped from the wafer and the wafer is broken into die for assembly.

8.4.3 Thrust Wafer Fabrication

The thrust wafer is fabricated using a 100 mm diameter, p-type, <100>, DSP silicon wafer with a thickness of 500±25 µm. The wafers also had 1 µm of thermally grown silicon dioxide. A description of each processing step follows. The steps are labeled with the letter corresponding to the depiction of the step in Figure 8.7.
8.4 Device Fabrication

8.4.3.1 Step A (Top Side) - Electroplate Solder Pads

The wafer is sputter coated with a chrome and copper seed layer (not shown in the process flow diagram.) The wafers are coated with photoresist and patterned with the solder pad mask. The solder pads are electroplated with a 2 µm nickel diffusion barrier, 3 µm of tin and 300 nm of gold. The photoresist is then stripped and the exposed seed layer (not shown in the diagram) is etched to expose the silicon dioxide below.

8.4.3.2 Step B (Top Side) - Etch Bearing Raceway

Photoresist is deposited on the wafer and the bearing raceway is patterned in the resist. The pattern is then etched into the exposed oxide. The raceway is etched to a depth of 260 µm. The photoresist is then stripped from the wafer.

Figure 8.7: Process flow for the thrust wafer of the SMT with a silicon ball cage.
8.4.3.3  Step C (Back Side) - Pattern and Pre-Etch Thrust Release Channel

Photo resist is deposited on the Back Side of the wafer and patterned with the thrust release channel. The channel is then etched into the exposed oxide using RIE and then to a depth of 200 µm using DRIE. The photoresist is stripped and wafer is broken into die for assembly.

8.4.4  Assembly and Release Etch

Figure 8.8: Process flow for the assembly and release etch of the SMT with a silicon ball cage.
8.4.4.1 Step A - Place Cage, Solder Balls, and Steel Balls in Thrust Die

A thrust side die is placed on a surface and the solder pads are painted with solder flux. 300 µm diameter solder balls are placed at the center of the solder pads and are held in place by the solder flux. A cage is then placed over the raceway. The ball pockets are then filled with the steel balls. A SEM image of the thrust die after this assembly process is shown in Figure 8.9.

![SEM Image](image_url)

Figure 8.9: SEM of the 5 mm diameter SMT after the all of the parts have been assembled on the thrust die but before the turbine die has been placed on top.

8.4.4.2 Step B - Align Turbine Die and Bond the Device

The turbine die is attached to the vacuum chuck of a custom bonder/aligner. The assembled thrust die is placed under the turbine die, the die are aligned and then brought into contact under light force. A butane torch is then used to rapidly heat the die, causing the solder to reflow and bonding the die. The bonded device is then removed from the bonder and allowed to rapidly cool on a metal surface.
8.4.4.3 Step C - Turbine Release Etch

The bonded die is mounted on a backing wafer with the turbine side facing up. This turbine side is etched, using DRIE, until the turbine release channel meets the turbine release channel patterned in the raceway on the other side of the die. The device is then removed from the backing wafer, flipped over and mounted on a new backing wafer with the thrust side facing up. The thrust side is etched, using DRIE, until the thrust release channel meets the raceway on patterned on the other side of the wafer. At this point the turbine is completely released from the stator.

8.4.4.4 Images of Fabricated SMT Devices with Integrated Cages

The following are images of the SMT with an integrated ball cage. Figure 8.10 shows the full 5mm device after the final release etch. Figure 8.11 shows the steel balls and the ball cage in the SMT stator with the rotor removed. Figure 8.12 shows a rotor, with severely damaged blades, and a cage removed from the stator.

![Figure 8.10: SEM of the 5 mm diameter device after release etching.](image)

Figure 8.10: SEM of the 5 mm diameter device after release etching.
8.5 Testing

The devices were tested using the setup described in Section 5.2.2 and was carried out by driving the SMT with a compressed nitrogen or air supply while at the same time bleeding gas from the back side of the device to provide...
some control over the normal load on the bearing. In initial performance tests
the input drive power (inlet gauge pressure X volume flow rate) was varied,
and rotation speed, gas flow and input pressure were recorded. This allowed a
performance curve of rotation speed versus input power to be plotted. These
tests could not provide an absolute measure of the bearing losses, since the
turbine efficiency was unknown; however, they did allow comparison between
the bearings with different cage designs. Performance tests were carried out
on two devices for each combination of retainer type and device size. In
addition, the repeatability of results was verified for one device, and longevity
testing was performed on a 5 mm device and a 10 mm device to evaluate wear
during extended operation.

To test each device, the bleed valve was first set to a known value. This
value was determined by running a Half Skeleton device at 2 W of input
power and adjusting the bleed valve until the device operated smoothly. This
calibration was performed separately for the 5 mm rotor and 10 mm devices
and the initial valve setting was maintained for all subsequent testing. The
calibration resulted in an upward normal force on the underside of the rotor
of 5 mN to 48 mN for the 5 mm devices and 4.5 mN to 40 mN for the 10 mm
devices over the input power range tested. Each device was then ramped
through a range of inlet pressures at least 3 times to evaluate the performance.
Longevity and wear were evaluated by operating a 5 mm and a 10 mm device
continuously for an extended period of time (up to 12 hours) while counting
the number of revolutions. Open retainer designs were used in these tests.

8.5.1 Comparison of Cage Design Performance

Each type of retainer was tested by increasing the flow of nitrogen in steps
until an input power of 2.5 W was achieved. The flow of nitrogen was then
cut and the ramping was repeated two further times. In order to gain an
adequate comparison for each retainer ring, two devices of each type were
tested from the same fabrication run.

Figures 8.13 and 8.14 show the measured variations of speed with input
power for the 5 mm and 10 mm devices respectively. Six measurements were
taken at each input power setting, and all data points are plotted to give an idea of the scatter in the measurements. The results for both the 5 mm and 10 mm devices indicate that the Full Ring and the Full Skeleton designs exhibit the best performance (i.e. highest rotation speed for a given input power) over a wider range of input power, with the Full Ring performing the best overall. The Half Skeleton, Outer Open and Inner Open designs perform consistently less well over the same range.

The Full Ring and Full Skeleton consistently showed the lowest losses. The Full Skeleton retainer performed less well than the Full Ring in both sizes of device, and the reasons for this are currently unclear. One possibility is that the lower rigidity of the Full Skeleton retainer makes it more susceptible to dynamic distortions that result in more contact with the raceways. This could also explain the drop in performance of the 10 mm Full Skeleton device at higher power levels. This effect would be expected to be more pronounced in the larger device where the retainer is less stiff. The relatively poor performance of the more flexible design does run counter to results from previous studies [Weinzapfel 09]. However, the flat raceways on the silicon bearings make them less constrained and more susceptible to vibration, and it is not clear that results relevant to conventional bearings should carry over.

Low rigidity may also be a contributing factor to the poor performance of the Inner Open and Outer Open retainers. These designs also allow the balls more radial play which is likely to increase interaction between the balls and the raceway sidewalls. Furthermore, interactions between the balls and the edges of the pocket openings may lead to increased frictional losses with the Open designs. Finally, the poor performance of the Half Skeleton designs is consistent with their having a lower ball fill factor in the raceways [Nataraj 08].
It should also be noted that, because the thrust bleed valve was maintained in a fixed position during ramp testing, the turbines may not have been operating in the most ideal regime. It was verified during tests that the...
bearings were operating in the correct mode, with a net upward thrust on the rotor. However, it is likely that better performance could be obtained by re-adjusting the bleed valve during testing to find the optimal performance regime, and this will be explored in future work.

8.5.2 Performance Repeatability

A 5 mm diameter Full Ring device was run through successive ramp tests, averaging 30,000 revolutions per ramp, to evaluate the repeatability of the results. For each ramp test, the measured variations of rotation speed with inlet power were averaged, and then simple linear interpolation was used to estimate the rotation speed at fixed power levels of 0.5 W, 1.0 W, and 1.5 W. Figure 8.15 shows how the rotation speed at each of these power levels varied over the first 12 runs. At all power levels, an initial improvement in performance was observed, followed by a fall-off up to round the 8th run, beyond which the rotation speed was stable to within ±10%. It is believed that the initial improvement is due to removal of asperities on the bearing surfaces, while the subsequent fall occurs as the sides of the bearing raceway become worn making the bearing less stable due to increased play in the rotor position.
8.6 Longevity and Wear

Longevity was evaluated by running a 10 mm diameter device with an Outer Open retainer and a 5 mm diameter device with an Inner Open retainer for extended periods of time. The retainers with the lowest mechanical strength were chosen for this test as one of the aims was to induce early failure and identify the failure mode. However, both devices survived the longevity testing without failure and only required minimal cleaning to resume operation if the bearing became locked. Just over 2.5 million revolutions were logged for the 10 mm diameter device and just over 2.98 million revolutions were logged for the 5 mm diameter device. The devices were disassembled for inspection by heating the tested device to above the melting point of the solder to break the solder bonds.

Wear effects on the bearing raceways after longevity testing were similar to those described in [McCarthy 09] and [Hanrahan 10]. There was some ball-induced wear on the sidewalls of the bearing raceway, and narrow wear tracks could be seen in the silicon raceway surfaces top and bottom, as shown in Figure 8.16. The retainers in both devices experienced minimal wear on their
upper surfaces during operation. For example, Figure 8.17 is an SEM image showing the upper surface of the 5 mm Inner Open retainer. An optical profilometer scan indicated that the contact wear on the retainer did not exceed 1µm on any portion of the ring. This was also confirmed using a microscope, by observing that silicon dioxide was still present in most of the wear regions. The edge of one of the ball pockets also shows some signs of damage likely attributable to shock during start up of the device.

The undersides of the retainers showed wear around the edges of the ball pockets due to abrasion by the balls, as shown in Figure 8.18. This has the effect of increasing the effective pocket size, and relaxing the constraints imposed on the balls by the retainer.

Figure 8.16: SEM images of the raceway wear on the rotor (top image) and the stator (bottom image) of the 10 mm diameter device after longevity testing.
8.7 Discussion and Conclusions

This chapter presented the exploration of several different geometries of silicon retainer rings integrated into a SMT with a thrust ball bearing, at rotor
8.7 Discussion and Conclusions

diameters of both 5 mm and 10 mm. Full Ring retainers, which are an annular design with cylindrical ball pockets, were found to perform better than designs with pockets that were open on one side, even though the latter had lower contact area with the bearing rings. Removal of material from the Full Ring design while retaining the cylindrical pocket shape was also found to degrade performance. The SMT with the Full Ring retainer could operate at over 20,000 RPM with less than 2 W of input power. Also 5 mm- and 10 mm-diameter devices were able to operate for over 2.5 million revolutions without device failure. Further investigation will need to be carried out to fully characterize the performance of each design and to understand the effects of retainer and raceway wear on the bearing performance. Future work will include measuring the bearing vibration, observing the lateral displacements of the turbines during operation and finding the optimal running conditions for the turbine by varying the thrust bleed rate. Design improvements will include exploring softer retainer materials such as polymers and metals; exploring new ball materials; and improving the bearing geometry by moving the retainer to the center of the ball and creating a curved raceway to allow for better self-center of the bearing during operation.
Chapter 9

Curved Raceway

A curved raceway can support higher normal loads than a similar rectangular raceway. This design can also be used to create self centering ball bearings similar to those found at the macro-scale. This chapter will present the design and fabrication of the devices used to explore the viability of integrating the angular contact raceway into silicon MEMS rotary ball bearings. The raceway is created by using the isotropic properties of inductively coupled plasma (ICP) as found in deep reactive ion etching (DRIE) systems when passivation cycles are removed from the processing. The micro-tribometer and silicon micro-turbines (SMTs) were both used in this study to evaluate the bearings. Results will be presented for the bearing performance in terms of power loss, as well as for the longevity and wear.

9.1 Device Design

Two classes of curved raceway bearing were developed for this study: one designed for evaluation using the micro-tribometer, and another that was integrated into an SMT with a 5 mm diameter rotor. The novelty in these devices lay in the fabrication of the bearing raceway. As there was no integrated ball cage, there were fewer design constraints and geometric considerations than seen in the radial design described in Chapter 6. The fabrication process for the SMTs was also greatly simplified over the devices presented
in Chapter 8 as the raceway did not have the stepped profile needed for an integrated ball cage.

Figure 9.1: SEM of micro-lens mold with a diameter of 116.7 μm created using HNA etching, picture from [Albero 09].

Figure 9.2: SEMs of the curved profile created by the ICP method, picture from [Larson 05].

The primary design challenge for these devices was creating a curved raceway profile that was suitable for a ball with a 500 μm diameter. Curved geometries have been created previously in silicon for micro-lens molds. For
example, in [Albero 09] a wet etch process is used to create spherical molds as shown in Figure 9.1. In [Larsen 05] an ICP etch is used to create the micro-lens shape. This is very similar to the desired profile of the angular contact raceway and is shown in Figure 9.2. The ICP process has the advantage of being easily integrated into existing fabrication process flows described in this thesis, therefore it was deemed the most efficient method for creating the angular contact raceways. The special requirements for this process step will be covered in the following section.

9.2 Device Fabrication

This section will describe the device fabrication process flows for each of the two classes of device. Simplified process flows are presented here; for more details on the processes used please refer to Chapter 3.

9.2.1 Fabrication of SMTs with Angular Contact Raceways

The SMTs with angular contact raceways were fabricated from 100 mm diameter, p-type, <100>, DSP silicon wafers with a thickness of 525±25 µm. The wafers also had 1 µm of thermally grown silicon dioxide. A seed layer of chrome and copper was deposited on the wafer (and is not shown in the process flow) for electroplating the solder pads. A description of each processing step follows. The steps are labeled with the letter corresponding to the depiction of the step in Figure 9.3.

9.2.1.1 Bearing Raceways

The wafer is plated with eutectic solder pads and the curved raceways are then etched into the silicon substrate. The steps for these processes are listed below with the letter corresponding to the step depicted in Figure 9.3

9.2.1.1.1 Step A - Plate Solder Pads

The wafers are coated with photoresist and patterned with the solder pad
Figure 9.3: Device fabrication process flow for SMTs with angular contact raceways.
mask. The solder pads are electroplated with a 2 \( \mu \)m nickel diffusion barrier, 3 \( \mu \)m of tin and 300 nm of gold. The photoresist is then stripped and the exposed seed layer (not shown in the diagram) is etched to expose the silicon dioxide below.

9.2.1.1.2 Step B - Define Raceway Width

The wafer is coated with photoresist and a mask is used to define the desired width of the raceway (505 \( \mu \)m) The exposed oxide is etched using RIE and the photoresist is stripped from the wafer. This pattern is used during the isotropic etch step to monitor the width of angular contact raceway. Ensuring that the raceway reaches the minimum desired width at the end of the etching cycle.

9.2.1.1.3 Step C - Pattern Raceway Isotropic Etch Mask

The wafer is coated with photoresist. The raceway isotropic etch mask is used to pattern a track that is half the width of the final raceway (250 \( \mu \)m). The pattern is half the desired width to account for the isotropic nature of the etch. The etch (described in the next step) will undercut the photoresist mask and producing a toroidal geometry that is 250 \( \mu \)m deep and 500 \( \mu \)m wide.

9.2.1.1.4 Step D - Raceway Isotropic Etch

The raceway feature patterned in the previous step is etched using a DRIE in an ICP configuration. The DRIE parameters used to etch the raceways can be found in Table 9.1. After etching the raceway, the photoresist is stripped from wafer.

It should be noted that the photoresist is the only mask used in this etch process. It was found that even with bending in the photoresist, from the excessive undercut, the etch maintained a toroidal geometry.

9.2.1.2 Turbine Side

After the solder pads are deposited and the raceway is etched, the turbine is defined on the other side of the wafer. The process steps for defining the
Table 9.1: DRIE process parameters for etching the angular contact raceway.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Etch Time</td>
<td>90 min</td>
</tr>
<tr>
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<tr>
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</tr>
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</table>

9.2.1.2.1 Step E - Define Turbine Blades in Oxide

Photoresist is deposited on the wafer and patterned with the turbine blades. The blade design is then transferred to the wafer by etching the exposed oxide. The photoresist is then stripped from the wafer.

This pattern will be used as an etch mask during the turbine blade etch and device release step (Section 9.2.1.4.2 or Step I.) An oxide mask is required because it can survive the assembly process and high temperatures of the solder reflow (Section 9.2.1.4.1 or Step H.)

9.2.1.2.2 Step F - Etch Turbine Journal

After the turbine blades are defined, a new layer of photoresist is deposited on the wafer. The turbine journal is patterned and etched to a depth of approximated 175 µm. This depth accounts for the etch lag during the turbine blade etch and device release step (Section 9.2.1.4.2 or Step I.) After this step, the photoresist is stripped and the wafer is broken into individual die for assembly.

9.2.1.3 Thrust Side

After the solder pads are deposited and the raceway is etched, the thrust journal is defined on the other side of the wafer. The process step for defining the thrust journal is listed below with the letter corresponding to the step
depicted in Figure 9.3. This will then become the bottom wafer in the bonded device stack.

9.2.1.3.1 Step G - Define Thrust Journal  Photoresist is deposited on the wafer and patterned with the thrust journal. The oxide is then etched using RIE to defined the thrust journal. The wafer is then loaded into the DRIE and the thrust journal is etched to a depth of 175 µm. After this step the photoresist is stripped and the wafer is broken into individual die for assembly. The oxide remaining on the thrust side of the wafer will be used as an etch mask during the final release etch (Section 9.2.1.4.2 or Step I.)

9.2.1.4 Device Assembly

After the turbine and the thrust sides of the device have been fabricated, the device can be assembled. After assembly the oxide masks created in previous steps are used to define the turbine blades and release the turbine rotor from the stator. The process steps for the device assembly and release are listed below with the letter corresponding to the step depicted in Figure 9.3.

9.2.1.4.1 Step H - Placement of Steel Balls, Solder Balls, Die Alignment and Solder Reflow

The device is assembled in a custom aligner/bonder. The raceway of thrust die of the device is filled with the desired number of steel balls. Solder flux is then painted on to the solder pads using a fine tip paint brush; this serves to inhibit oxidation of the solder pads and to hold the solder balls in place. 300 µm diameter solder balls are then placed on the centers of the solder pads. The turbine die of the device is attached to a vacuum chuck and aligned to the thrust side of the device in a custom bond aligner. A force is applied to the top of the turbine side to hold the alignment position of the die and encourage a uniform reflow of the solder. The device is heated with the flame from a butane torch for 30 s. The force is removed, the device is removed from the aligner/bonder and is cooled on a metal surface.
9.2 Device Fabrication

9.2.1.4.2 Step I - Release Etches

The bonded devices from the previous step (Section 9.2.1.4.1 or Step H) are mounted to a backing wafer with the turbine side facing up. The device is etched using DRIE until the turbine journal reaches the raceway below. This results in a turbine blade height of 150 µm to 175 µm. The devices are then detached from the backing wafer, cleaned and remounted to a backing wafer with the thrust side of the device facing up. The thrust journal is then etched until it reaches the raceway. For both of these steps the oxide patterned in previous steps is used as the etch mask. These final etch steps define the turbine blades and release the turbine rotor from the stator.

9.2.2 Fabrication of Devices for Tribometer Testing

The ball bearings with angular contact raceways for tribometer testing were fabricated from 100 mm diameter, p-type, <100>, DSP silicon wafers with a thickness of 525±25 µm. The wafers also had 1 µm of thermally grown silicon dioxide. A seed layer of chrome and copper was deposited on the wafer (not shown in the process flow) for electroplating the solder pads. A description of each processing step follows the step letter corresponding to the depiction of the step in Figure 9.4.

9.2.2.1 Bearing Raceway

The bearing raceway is fabricated in the same way as the raceway for the angular contact SMT devices. Please see Section 9.2.1.1 for more details. The process steps are depicted in Figure 9.4 and letter for each step corresponds with the label in the figure.

9.2.2.1.1 Step A - Plate Solder Pads

See Section 9.2.1.1.1 for more details.

9.2.2.1.2 Step B - Define Raceway Width

See Section 9.2.1.1.2 for more details.
Figure 9.4: Device fabrication process flow for devices with angular contact raceways for tribometer testing.
9.2 Device Fabrication

9.2.2.1.3 Step C - Pattern Raceway Isotropic Etch Mask
See Section 9.2.1.1.3 for more details.

9.2.2.1.4 Step D - Raceway Isotropic Etch
See Section 9.2.1.1.4 for more details.

9.2.2 Backside

The backside of the device, or the side of the device that does not connect to the testing adaptor is fabricated with the steps indicated below. These steps result in features of 2 different etch depths and leave an oxide mask for the final release etch after assembly. The letter for each step corresponds with the depiction of that step in Figure 9.4. For the DRIE steps the wafer is mounted to a backing wafer; this is not shown in Figure 9.4.

9.2.2.2.1 Step E - Define Device Edge
Photoresist is spun on the back of the wafer after the bearing raceway has been patterned. The Photoresist is patterned with the edge features of the device (the feature that will release the device from the die.) The exposed oxide is etched using RIE and the photoresist is stripped from the wafer.

9.2.2.2.2 Step F - Pre-Etch Device Edge
Photoresist is spun on the wafer. The wafer is patterned with the device edge and the bearing back side rotor release channel. The device edge is then etched to a depth of approximately 250 µm. The backside rotor release channel will not be etched due to the oxide mask. The photoresist is not stripped from the wafer after this step as it will continue to be used as a mask in the next step.

9.2.2.2.3 Step G - Define and Pre-Etch Rotor Release Channel
The exposed oxide (covering the rotor release channel in the previous step) is etched using RIE. The rotor release channel and the device edge are then etched using DRIE. The rotor release channel is etched to a depth of 200 µm, while the device edge typically reaches a final depth of around 400 µm due to
etch lag. The photoresist is stripped and the wafer is broken into individual die for device assembly.

9.2.2.3 Adaptor Side

The adaptor side of the device has an adaptor port which allows the device to be coupled to the tribometer. Further, in order to easily identify the thrust style and radial style bearings, the radial bearing has a label as shown in the process depicted in Figure 9.4. The letter for each step corresponds with the depiction of that step in Figure 9.4. For the DRIE steps the wafer is mounted to a backing wafer, this is not shown in Figure 9.4.

9.2.2.3.1 Step H - Define Device Edge and Device Label

Photoresist is spun on the wafer. The wafer is patterned with the device label and the device edge. The exposed oxide is etched using RIE and the photoresist is then removed from the wafer.

9.2.2.3.2 Step I - Pre-Etch Device Edge

Photoresist is spun on the wafer. The wafer is patterned with the device edge, the adaptor port feature and the rotor release channel. The device edge is then etched to a depth of 250 µm. The photoresist is not stripped after this step as it will be used in the next step.

It can be seen that the device label is covered by the photoresist in this step. This will allow us to pattern and etch the other features while protecting the label from etching until the final release etch. This allows us to create 3 different etch depths with only 2 masks.

9.2.2.3.3 Step J - Define and Pre-Etch Rotor Release Channel and Testing Adaptor Feature

The exposed oxide from the previous step is etched using RIE. The wafer is then etched using DRIE. The rotor release channel and the adaptor port are etched to a depth of approximately 200 µm, while the device edge typically reaches a final depth of around 400 µm due to etch lag. The photoresist is removed and the wafer is broken into die for assembly.
9.2.2.4 Device Assembly

9.2.2.4.1 Step K - Placement of Steel Balls, Solder Balls, Die Alignment and Solder Reflow

The device is assembled in a custom aligner/bonder. The raceway of adaptor side of the device is filled with 7 steel balls. Solder flux is then painted on to the solder pads using a fine tip paint brush, this serves the purpose of removing oxidation from the solder pads and to hold the solder ball in place. 300 μm diameter solder balls are then placed on the center of the solder pads. The Backside of the device is attached to a vacuum chuck and aligned to the adaptor side of the device in a custom bond aligner. A force is applied to the top of the backside of the device to hold the alignment position of the die and encourage a uniform reflow of the solder. The device is heated with the flame from a butane torch for 30 s. The force is removed, the device is removed from the aligner/bonder and is cooled on a metal surface.

9.2.2.4.2 Step L - Release Etch

Patterned oxide from previous steps will be used as the DRIE mask in this step. After assembly, the die are mounted to a backing wafer (not shown in Figure 9.4) with the adaptor side facing up. The device is then etched using DRIE until the rotor release channel reaches the raceway below, and the device edge reaches the back of the wafer. The device label will also be etched to a depth of approximately 150 μm and the adaptor feature is etched an additional depth of approximately 125 μm. After DRIE processing the device is removed from the backing wafer, flipped over to expose the backside of the device, and mounted to a backing wafer for DRIE processing of the backside. The device is etched until the rotor release channel reaches the raceway below and the device edge reaches the back of the wafer. The wafer is then removed from the backing wafer.

9.2.2.4.3 Step M - Remove Device from Die

After the final release etches, the device is removed from the die. The device is completely separated from the die at this point and can be easily lifted from the surrounding silicon. Five devices are present on each die,
and when separated from the die look like the device labeled $M$ in Figure 9.4. The device can now be inserted into the tribometer test platform for characterization.

### 9.2.3 Images of Fabricated Devices

This section contains images of the fabricated devices. Figure 9.5 shows a rotor from a device used for tribometer testing. A steel ball that has been through testing is located next to the device to show the raceway curvature in relation to the ball in the design. A similar SEM for the SMT rotor is shown in Figure 9.6. In the image some fabrication damage can be seen on the raceway of the rotor. This damage was present on all of the devices fabricated with the current process, and was caused by over-etching during the bearing release etch step. Figure 9.7 shows a close up of the worst of the damage on a SMT stator raceway. This damage is only present where the balls are not located in the raceway during the release etch. Further studies will make an effort to protect the raceway from this damage. Figure 9.8 shows how the raceway should look without any of the over-etching damage. The raceway is extremely smooth and has the desired curved geometry.

![Figure 9.5: SEM image of a rotor used for tribometer testing. A steel ball has been placed next to the rotor to show the curvature of the raceway in relation to the ball used in the device.](image-url)
9.2 Device Fabrication

Figure 9.6: SEM image of the SMT rotor with a ball next to the raceway to show the curvature of the raceway in relation to the ball.

Figure 9.7: SEM image of a SMT stator raceway. The large depression on the raceway are damage for over-etching during the release etch step.
9.3 Performance

The performance of the angular contact raceway was evaluated by testing the SMTs and the devices in the tribometer. The SMT devices were used to evaluate the longevity of the bearings at high speeds. The tribometer was used to evaluate the bearing loss under controlled conditions. The following sections will describe the results of these tests.

9.3.1 Performance of SMT with Angular Contact Raceway

The SMT was run for a total of 10 million revolutions. Ramp tests were performed on the SMT before any longevity testing and then after every 2 million revolutions. The longevity tests were run at speeds of up to 70,000 RPM. These speeds were achieved by adjusting the bleed valve and the input pressure until the turbine was able to run smoothly at the higher speeds. This was primarily to reduce the duration of the longevity tests and to rapidly stress test the bearing in the turbine. The graph in Figure 9.9 shows that after an initial improvement in performance after 2 million revolutions the bearing...
9.3 Performance

performance starts to decrease. As will be seen in Section 9.3.3 the raceway shows minimal wear after testing. When compared to the performance of the 5 mm diameter SMT with the Full Ring retainer, presented in Chapter 8, the curved raceway under performs the bearing with the retainer at lower speeds as shown in Figure 9.10. This would be the case if the centrifugal forces on the balls dominates the power loss in the bearings as the Full Ring design has only 16 ball and the curved raceway has 25 balls. The curved raceway, however, does out perform the Full Ring bearing at higher input powers which is contradictory to what we would expect. This performance difference may have more to do with the turbine design and the differences in the blade heights for devices. The blade height for the Full Ring turbine was approximately 650 µm where the blade height for the curved raceway was approximately 125 µm.

![Graph of the speed and input power of the SMT after each 2 million revolution longevity test. After an initial improvement in performance after 2 million revolution, the turbine performance begins to decline.](image)

Figure 9.9: Graph of the speed and input power of the SMT after each 2 million revolution longevity test. After an initial improvement in performance after 2 million revolution, the turbine performance begins to decline.
9.3 Performance

Figure 9.10: Comparison of the performance of the 5 mm SMT with the Full Ring and the curved raceway SMT.

9.3.2 Tribometer Measurements

The bearings tested in this study had a radial bearing design much like the bearings described in Chapter 6 but without a cage. The bearings also had 7 balls in the raceway to give a fill factor \(F_r\) of approximately 74\% and \(d_m\) of 1.25 mm. The bearings were tested in 50 RPM speed increments from 50 RPM to 2500 RPM. The normal load for these tests was set to 40 mN. Figure 9.11 shows the raw measurements for the 3 tests. As seen in the graph the measurements had a relatively wide spread.

If Equation 6.1 is used to predict the frictional torque we find the the calculation for \(M_I\) is a good estimate for this bearing. However, the predicted centrifugal torque 9 times lower than the measured torque. However, the frictional torque (even with the wide spread) does conform to a change proportional to \(\omega^2\) as is expected for an increase in frictional torque due to the centrifugal force of the balls.

A further observation is that if it is assumed that the only influence on increase in the bearing frictional torque is the number of balls the relation-
ship between the radial bearing and the curved raceway bearing could be expressed as:

\[
M = \frac{e^{0.75N_b}}{7} M_c - M_l
\]  

(9.1)

This is using \( z = 0.8 \) to calculate \( M_l \). However, since there are several other aspects that differ in the bearings such as the lack of a cage in the curved raceway bearing and the different raceway geometries, this is simply an observation of the relation of the two designs.

The power loss varies by \( \omega^3 \) as expected. The predicted value based using Equation 9.1 to calculate \( M \) is plotted in Figure 9.12 with the power loss calculated using the average of the measured data.

Figure 9.11: Graph of the tribometer results for the angular contact bearings. The line represents the expected torque value for the devices and the measured data points for 3 tests are scattered plotted on the graph.
9.3 Performance

Figure 9.12: The power loss predicted by using Equation 9.1 to calculate the frictional torque and the power loss calculated using the average of the measured frictional torque.

9.3.3 Wear

Both the SMT and the tribometer devices showed very little wear on the raceways after testing. The SMT survived over 10 million recorded revolutions with speed up to 70,000 RPM and still showed minimal wear. As the curved raceway allows the load on the ball and the raceway to be spread over a larger area less wear would be expected on the raceway. Figure 9.13 shows the stator of the tested turbine with arrows indicating the over-etch damage on the raceway and with the wear region indicated by a box. The wear is seen as lighter areas within the box. The wear, which is very difficult to see, appears as small lighter areas on the raceway. The raceway of the device tested with the tribometer showed no detectable wear pattern.
9.4 Discussion and Conclusions

The curved raceway was successfully incorporated into both SMT and tribometer designs. The SMT was able to survive over 10 million revolutions at prolonged speeds of up to 70 000 RPM and showed minimal signs of wear on the bearing raceway. However, the performance of the turbine does show a steady decline after 2 million revolutions, which may be attributable to excessive wear on the balls. The curved raceway SMT performed worse than the SMT with the Full Ring retainer at lower input power, as would be expected if the the centrifugal force on the balls is the dominant cause of losses in the bearing as the curved raceway had 25 balls and the SMT with the full ring had 16 balls. However, at higher input power the curved raceway did start to perform better than the SMT with the Full Ring retainer. This may have been due to the difference in the turbine blade heights or it may have been due to the geometry of the raceway, further testing on devices with similar blade heights will need to be carried out to determine the cause of this phenomenon.

Figure 9.13: SEM image of the tested SMT stator raceway. The wear pattern is enclosed in the box and the over-etch damage to the raceway is indicated by the arrows. The wear can be identified as the lighter regions on the raceway region enclosed by the box.
The tribometer measurements showed frictional torque 9 times greater than the torque that was estimated by the Equation 6.1 for the radial bearing in Chapter 6. A higher frictional torque was expected for this design as 7 balls were in the test device rather than the 4 in the radial design. A relationship is provided for the difference in measured torque relating only to the number of balls in Equation 9.1. However, this is a rather naive modification that ignores other factors such as the lack of a cage in the curved design and the new raceway geometry. However, estimate for $M_l$ as calculated in Equation 6.1 does conform to the measured values for this design.

Additional work will need to be done to remove the release etch damage from the curved raceway and to characterize any advantage of using the curved raceway geometry. The design does allow for the creation of self centering bearings and there appeared to be less wear on the curved raceways went compared to the rectangular raceways.
Chapter 10

Conclusions and Further Work

This chapter will present an overview of the contributions of the thesis and a review of the research objectives. A brief overview of some of the conclusions reached through completing the work will also be provided, and future work will be proposed.

10.1 Contributions of this Thesis

This thesis provides an investigation of new fabrication techniques and geometries for use in micro-ball bearings for MEMS applications. The work presented here showed the first integration of ball cages into MEMS ball bearing designs. The investigation of the fabrication and design constraints will be useful in guiding future designs. Further, the observation of the effects of centrifugal force on the performance and the overall loss within the bearing has not been previously presented for bearings on this scale. The frictional torque related to the bearing load was found to be 1000 times greater for micro-bearing compared to the predicted value for macro scale bearings. Also, the measured friction torque increased with \( \omega^2 \) confirming that the centrifugal force of the balls plays a significant role in the bearing loss at higher speeds. It should be noted that the equation used to predict the frictional torque was extremely simplistic and it under estimated the frictional torque for the bearings presented in this thesis. Further, this thesis
presents new fabrication techniques and designs that can be used in future micro-ball bearing designs.

10.2 Objectives Revisited

In the introduction a list of research objectives was presented to provide an overview of the overall goals of this thesis. Each objective will be revisited here with a brief discussion how the objective was met and the results of the investigation presented in the preceding chapters.

1. Characterize the performance of the devices designed for this study by modifying and improving, where possible, existing testing methodologies (Chapter 5). This objective was met by using the existing micro-tribometer and the SMT designs. Data acquisition and gas flow control were improved in the SMT setup to provide accurate and simple measurement techniques not available in earlier similar research. The micro-tribometer was used as designed. Devices were designed to allow coupling of the driving motor and mounting into existing test platforms. Test platforms were modified by the author to allow for configuration of the measurement range and to improve device yield. The capabilities of each of the test platforms is:

Tribometer: It is possible to resolve torque measurements down to 1 $\mu$N·m with the current platform configuration with maximum measurable torque of 100 $\mu$N·m. The maximum measurable torque is configurable by increasing the number of beams attached to the test platform. The normal load has a measurement resolution of 5 mN and can be applied reliably in 10 mN increments. The speed can be set in increments of 1 RPM of the range of 0 RPM to 30 000 RPM. Measurement acquisition is configurable in 1 s increments. The testing range and measurement resolution of this system are sufficient to provide insight into the losses in the micro-bearings. The system also has the added benefit of showing when the bearing is operating in rolling or sliding friction.
SMT Setup: The turbine is a fully integrated MEMS solution for testing the bearing technologies. The input power can be measured and controlled in 5 mW increments. The force on backside of the turbine can be measured in 5 mN increments. Speed can be resolved up to 200,000 RPM with a resolution of 1/12 of a revolution. The data acquisition can also be configured to capture data in increments of 100 ms and with data storage of 2 GB testing can be performed for multiple days without interruption. Also, testing can be performed in a fully automated mode including the ramping up and down of the input power.

The tribometer testing provided the measurement resolution needed to confirm that the centrifugal force is the dominant source of the losses in the ball bearings. While the SMT allowed for prolonged testing of the bearings at high speed, while demonstrating the bearing design as a fully integrated part of a MEMS device.

2. Design and integrate a ball cage into a MEMS micro-ball bearing using the simplest technique possible (Chapter 6). The design technique presented provides a simple method of incorporating a ball cage into a micro-ball bearing. This design reduced the overall bearing fill factor and the centrifugal force related bearing loss over comparably sized devices without a cage as presented in Chapter 9. The limitation of this design was primarily in the failure of the cage bond interface. This was due to poor reflow during the bond process and also due to failure of solder pad due to poor adhesion with the silicon. This cage weakness led to the further studies of single crystalline silicon cages. However, this design provided insight into the power loss in the bearings. As it was observed that the frictional torque due to the applied load was 1000 times higher than that predicted for macro-scale bearings. It was also observed that the frictional torque increased with $\omega^2$ as would be expected if the centrifugal force of the balls played a dominant role in the bearing loss. When comparing the measured frictional torque to the simplified model it was discovered that the measured values were 3 times greater than the value predicted by the model.
3. Design a fabrication technique for releasing moving parts from MEMS devices with multiple layers (Chapter 7). This fabrication technique - using DRIE to remove sacrificial beams embedded in a multi-wafer stack - was intended to improve the micro-ball bearing performance by allowing for the in situ fabrication of a single crystalline ball cage. The dual row cage design was not as efficient as the previous design. Though the cage design was not optimal, the fabrication technique does prove to be viable for aligned multi-level MEMS application such as gear stacks and other complex mechanics.

4. Integrate a ball cage into a working MEMS device with a micro-ball bearing support mechanism (Chapter 8). The SMT was used to explore 5 different single crystalline ball cages. The cages were inserted manually in this application but could be integrated using the technique demonstrated in Chapter 7. The hope was to demonstrate a lower bearing loss than present in a cageless bearing by reducing the centrifugal force of the balls. The devices did not necessarily demonstrate an improved performance. However, testing indicated that the Full Ring ball cage provided the best performance and should likely be used in further investigations. The results of this study indicate that the rigidity of the cage is the most important factor in the design.

5. Create a micro-ball bearing raceway with a curved geometry that will allow for the bearing to self-center and to better mimic race geometries found in conventional macro-scale bearings (Chapter 9). The curved raceway was created with an optimized isotropic etching recipe. The tribometer tests provided frictional torque measurements that confirmed the modified calculation of $M_i$ using $z=0.8$ (rather than 0.0008 as in macro-scale bearings.) However, the centrifugal torque was 9 times greater than values predicted by the modified model. These results point to the need to further improve the model. However, the measurements did confirm that the frictional varied with $\omega^2$, further confirming the role of the centrifugal force as a dominant factor in the bearing loss. The SMT tests showed that the bearing could survive speeds of up to
70,000 RPM for prolonged periods and that after 10 million revolutions the bearing was still able to reach speeds of 25,000 RPM with an input power of 2.5 W. It was also observed that the performance of the curved raceway SMT under performed the SMT with the Full Ring retainer for input powers below 2 W, which would be expected if the centrifugal force from the balls in the bearings was the most significant factor in the power loss in the bearing. However at higher input powers the curved raceway started to perform better than the SMT with the Full Ring retainer which requires more exploration.

10.3 Conclusions

This work has shown that it is possible to integrate a ball cage into MEMS micro-ball bearings. This will be useful for future bearing designs by helping to reduce bearing losses due to centrifugal forces at intermediate speeds. The observation of the effects of centrifugal force on the bearing performance provides insight into the limitations of the micro-ball bearing for high speed applications. However, the micro-ball bearing has the advantage of providing high stability; it can also support higher loads and can be integrated into a device with a simple design with no need for external control systems. The ability to integrate the bearing in complex micro-mechanical has also been shown by developing fabrication techniques that can allow for multiple layers of moving parts. The designs and fabrication techniques presented in this thesis can be extended for use in multiple applications such as medical devices, micro-robotics and micro-motors. The research presented in this thesis provide a basis for creating truly intricate and complex micro-machines.

10.4 Further Work

There were several aspects of the dynamics and the design of the micro-ball bearings that were not explored in this thesis. Some of these aspects warrant additional exploration. The following is a list of additional aspects of the
10.4 Further Work

bearings that could be explored with the designs that are presented here:

**Hysteresis** This is a commonly observed phenomenon in ball bearing systems. Due to the configuration of the test equipment for the microtribometer and the SMT test setup, hysteresis was not studied in the bearings presented here. Positional, frictional and torque hysteresis could be characterized for the devices in future studies which would help to provide models for speed and positional control systems for systems with micro-ball bearings.

**Curved Raceway with Ball Cage** A thrust or radial ball bearing could be created with both a curved raceway and a ball cage. Though the design of such a bearing would be much more complex than the design presented in Chapter 6 it would have the advantages of being self-centering and would have a reduction in the number of balls thereby reducing centrifugal force losses.

**Different Ball Materials** Alternative ball materials could improve the performance and the longevity of the bearing. Silicon nitride balls would reduce the mass and centrifugal force loss in the bearing. Silicon nitride and tungsten carbide could increase the longevity of the balls in the bearing as balls made from these materials would be stronger than the 440C steel balls used for the bearings presented in this thesis. Balls made from these materials are readily available in the sizes needed for micro-ball bearing designs, but are much more expensive and more difficult to locate for purchase.

**New Cage materials** An exploration of new cage materials such as polymers, metals or even glasses could prove to improve cage fabrication techniques and reduce losses and frictional ball damage associated with the silicon cages.

**Exploration of New Bearing Types** With the technique described in Chapter 7 it would be possible to design and fabricate micro-jewel and magnetic levitation bearing. The exploration of these bearing types could
provide simple designs that operate at higher speeds without the centrifugal losses inherent in the micro-ball bearing designs. The technique could also be used to create complex gear sets that use either micro-ball bearings or other bearing designs.
Chapter 11

List of Published Works

The following works were published during the course of this research:


Bibliography


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Appendix A

AZ9260 Photoresist Coating Process

The following is the coating process that was used for depositing a 10 μm thick layer of AZ9260 photoresist on a substrate.

- **Clean and Dehydrate Substrate**
  1. Plasma clean substrate in barrel asher or RIE with Argon and Oxygen (2:4) plasma for 5 min to 10 min.
  2. Bake wafer in oven at 150 °C for a minimum of 5 min.

- **Apply Adhesion Promoter (HMDS)**
  1. Remove substrate from 150 °C oven and immediately transfer to spinner.
  2. Pour HMDS on the surface of the warm wafer and spin the wafer at with an a speed 1000 RPM for 90 s, with an acceleration of 1000 RPM/s.
  3. Return the substrate to the oven at 150 °C for 5 minute.

- **AZ9260 Photoresist Spin Coat**
  1. Remove substrate from the 150 °C oven and allow to cool on metal surface for at least 1 minute.
2. Pour at a AZ9260 resist from the bottle onto the center of the wafer until approximately half of the area of the wafer is covered with photoresist.

3. Spin the photo resists with the following spin steps:
   a. Speed: 500 RPM Acceleration: 500 RPM/s Time: 10 s
   c. Speed: 3000 RPM Acceleration: 3000 RPM/s Time: 2 s

4. Allow the substrate to rest on the spinner chuck for 5 min to 10 min.

- AZ9260 Coating Soft Bake
  1. Bake coated substrate in contact with a 60°C hotplate for 5 min.
  2. Move coated substrate to 100°C hotplate for 15 min.
  3. Remove substrate from heat and place on a cool metal surface for at least 2 minute.

- Rehydrate AZ9260 Coating
  1. Place coated substrate in a storage container and cover to avoid exposure to light.
  2. Allow the substrate to rest in the ambient environment for at least 90 min. Superior results were observed for electro-plating applications when the coating was allowed to rehydrate for at least 8 h.