Mechanism Approach for Enhancing the Dynamic Range and Linearity of MEMS Optical Force Sensing

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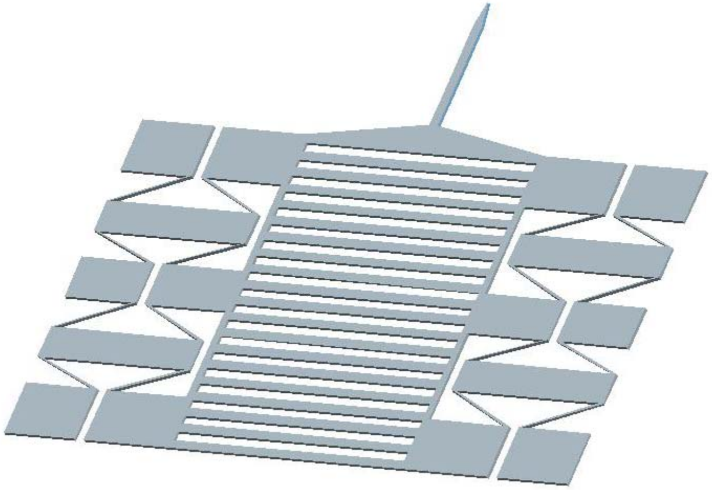
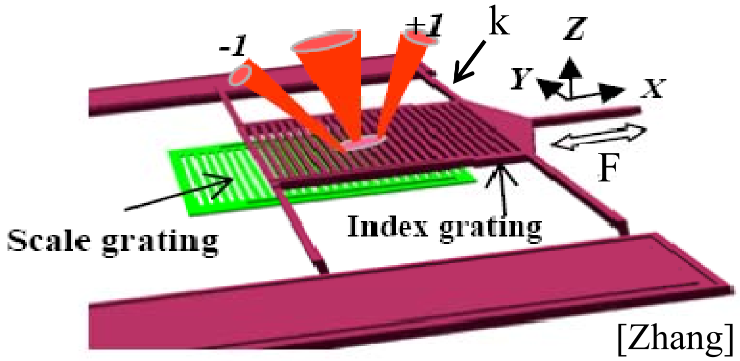
OUTLINE

- Introduction and background of MEMS force sensors and interferometry
- Motivation – Why are these devices beneficial
- A look at a current device – focusing on limitations and drawbacks
- A new design that builds upon previous devices
  - Analysis techniques
    - Analytical and Pseudo-Rigid-Body Model (PRBM)
    - FEA
    - Optimization
    - Latin hypercube design of experiments
  - Comparison to other designs
- Integration into systems
- Conclusions
  - Discussion of results
  - Future work



OPTICAL FORCE SENSOR

- Device is designed so that input loads are applied to a movable structure
- The stiffness of the structure is calculated
- Interferometry determines the displacement of the structure
- Force is computed using Hooke’s law: F = kx
- Many methods to implement interferometry
  - Michelson
  - Fabry-Perot
  - Sagnac
  - Diffraction based
    (linear optical encoder)
- Diffraction method – 2 types
  - Amplitude diffraction
  - Phase diffraction
    - preferred due to higher optical efficiency



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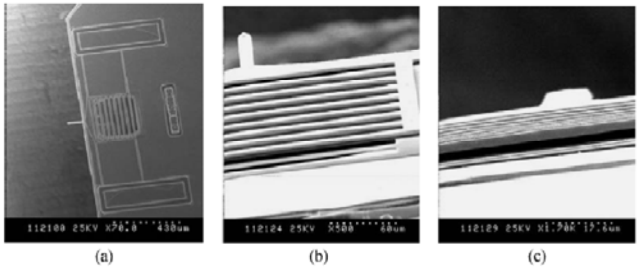
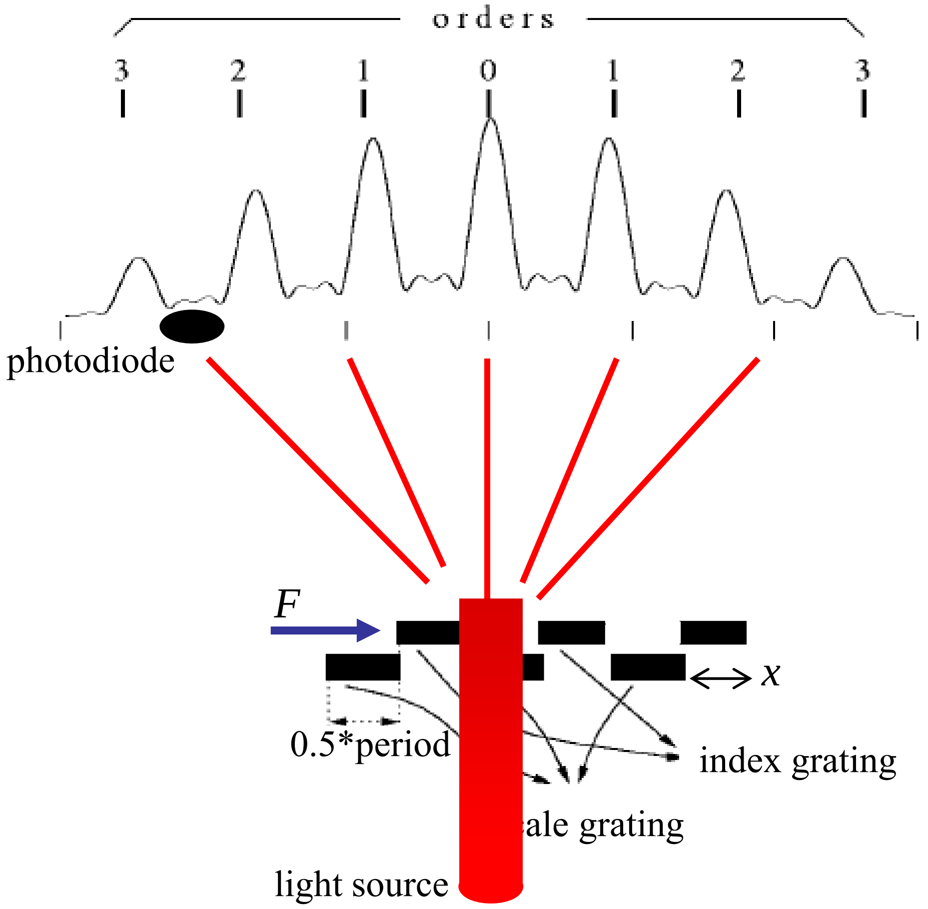


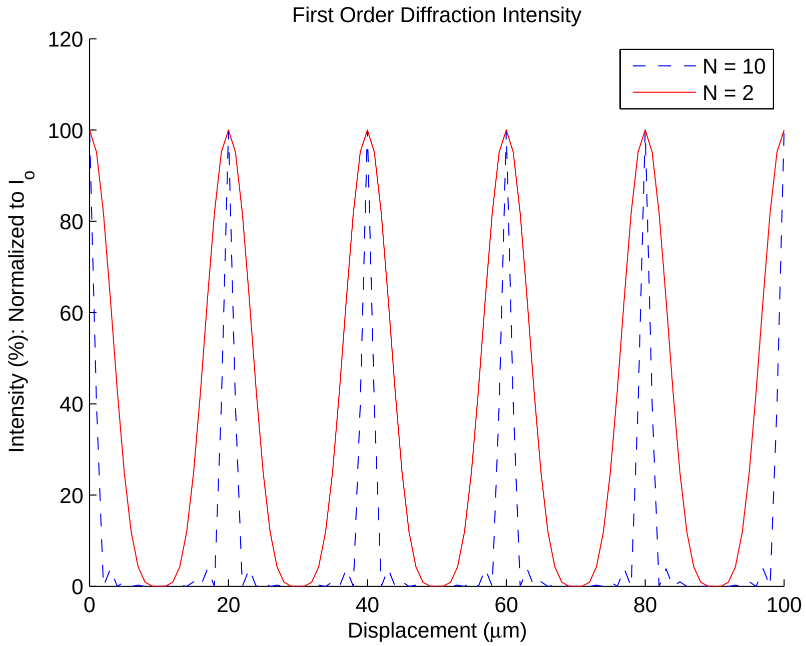
OPTICAL DIFFRACTION

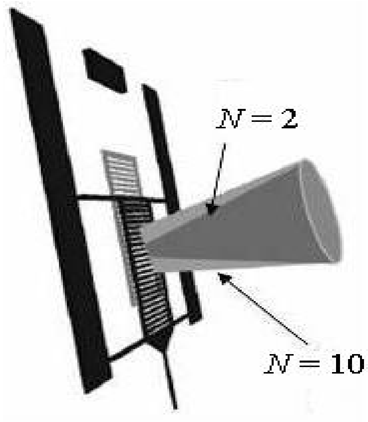
- Uses 2 constant period gratings to change the intensity of a light source
- The scale grating is fixed, while the index grating is free to move
- The index grating is fabricated above the scale grating
- While no input is applied the gratings are aligned
- Displacements cause the index grating to translate and vary the intensity of the diffracted orders
- Changes in intensity measured via photodiodes



OPTICAL SENSOR CHARACTERISTICS

- Sensitivity: change in intensity with respect to a unit displacement
- Dynamic Range: total range of motion for which the position can be determined
- Trade off between the two
- Both are determined by the number the grating periods under illumination
- Controlled by grating pitch and laser light source diameter
- Sensitivity can be enhanced mechanically

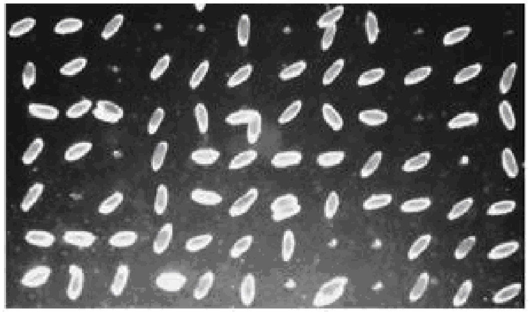


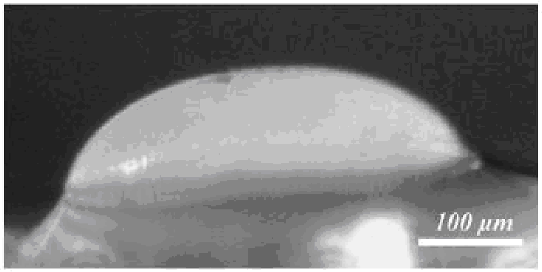


[Zhang]

MOTIVATION

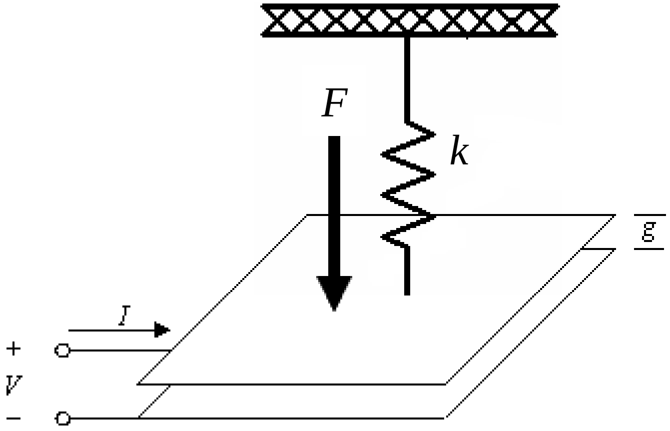
- Biomedical research
  - *In vivo* experiments (i.e. RNA interference)
    - Determination of required injection forces to penetrate membrane
    - Minimizing cell damage and preserving specimens
  - Cancer cell research
    - Investigate mechanical properties of cancer cells
    - Compare with healthy cells to distinguish
- Microassembly
  - Fabrication yields numerous small parts that require assembly
  - Assembly forces can range from *mN* to *µN*
- MEMS sensors provide a solution
  - Small device and feature sizes
  - High sensitivity for very small sensing ranges





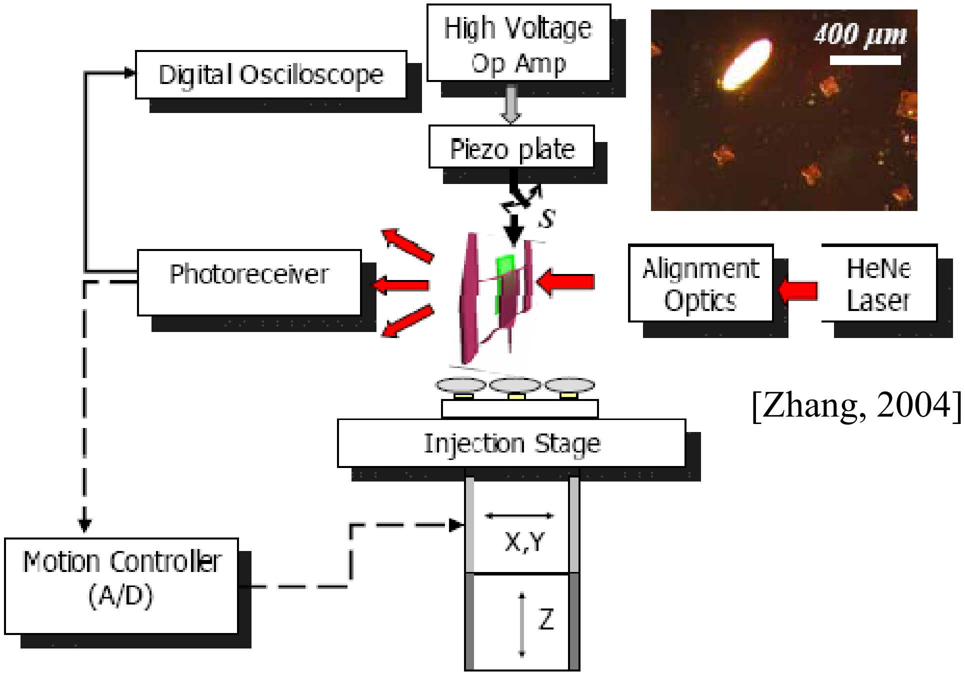
WHY OPTICAL MEMS SENSORS

- Capacitive force sensors are available with high resolution and on the chip integration
- Sun *et al*. have developed an capacitive force sensor with sub-μN resolution and range of about a half a mN
- Some applications may require same μN resolution over a range of tens or hundreds of mN
- Conflicting design goals in the electrical domain are a drawback for capacitive sensors
  - Sensitivity increases with decreasing gap height and increasing bias voltage
  - With small gap heights or large bias voltages, pull-in becomes a problem
  - The voltage-displacement relation is non-linear near pull-in
- Optical interferometry provides a means to decouple conflicting design goals
- Allows incorporation of capacitive elements for self-calibration

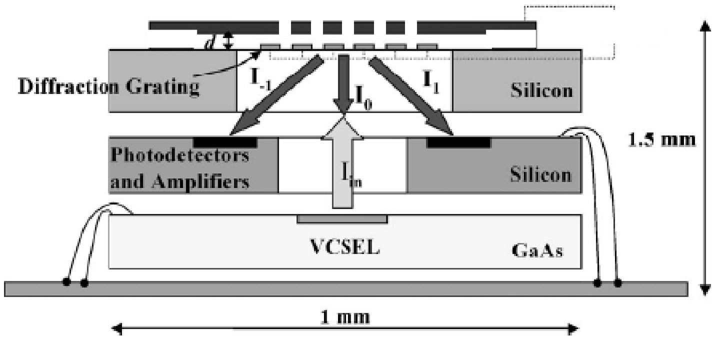


INTEGRATION OF OPTICAL COMPONENTS

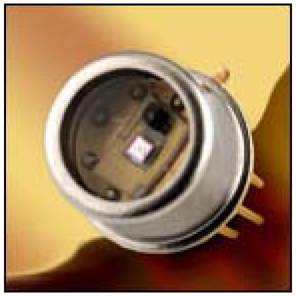
- A majority of optical sensors use “off the chip” components
- Emerging technologies can be applied to realize integrated systems
  - VCSEL (vertical cavity surface emitting laser)
  - Silicon p-n junction photodiodes
  - Short distances eliminate needs for large lenses



[Zhang, 2004]

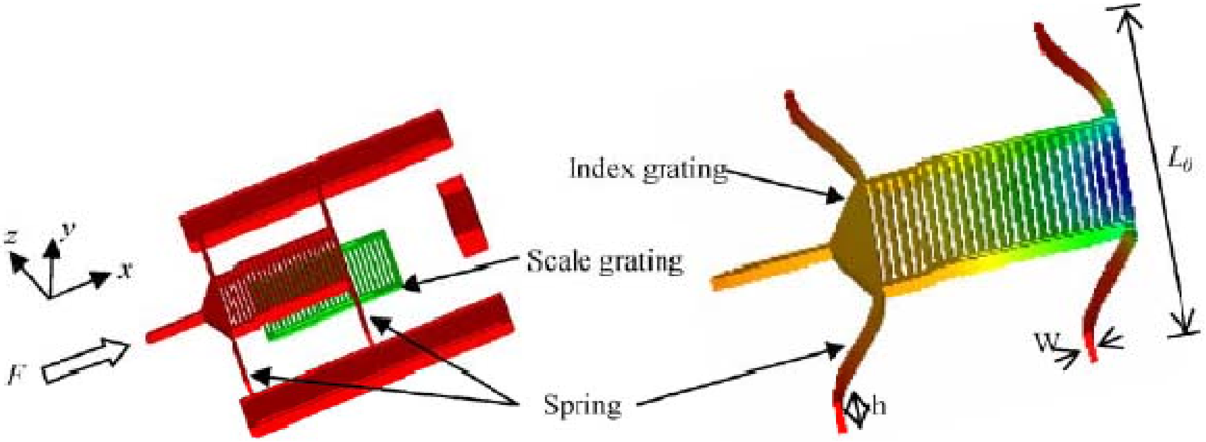


Advanced Photonix Inc.



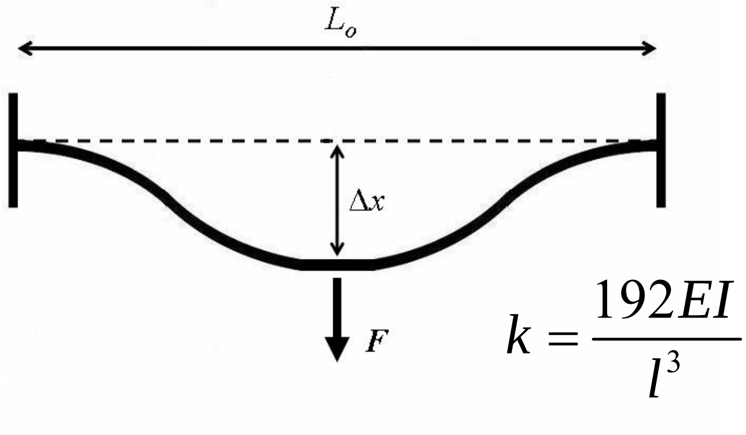
CURRENT STATE-OF-THE-ART DESIGN

- Index grating is suspended by 4 simple beams
- Force-displacement relation is linear for small displacements: ~ 10% of L
- During small deflections bending is the dominant mode
  - Function of area moment of inertia (*I*): $f(w^3)$
- Beyond small deflections axial stretching becomes dominant
  - Function of cross-sectional area (*A*): $f(w)$
- Thicker beams have more linear characteristics than thinner beam, but are also much stiffer



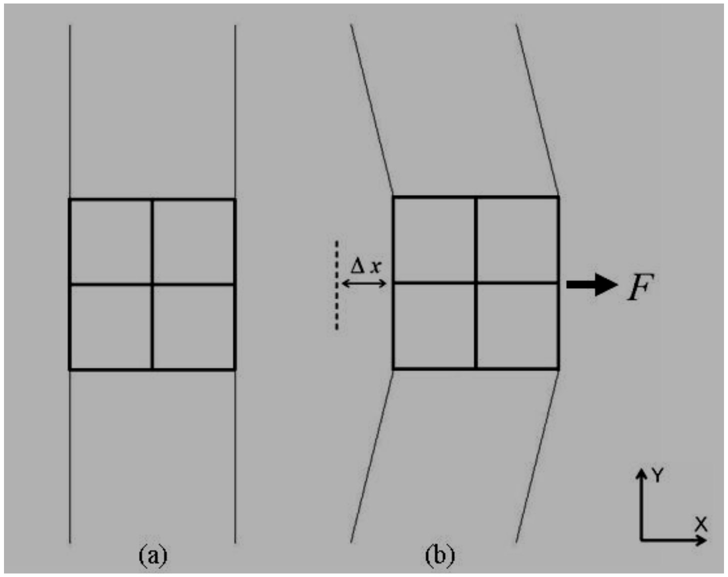
ANALYSIS OF 4-BEAM DESIGN

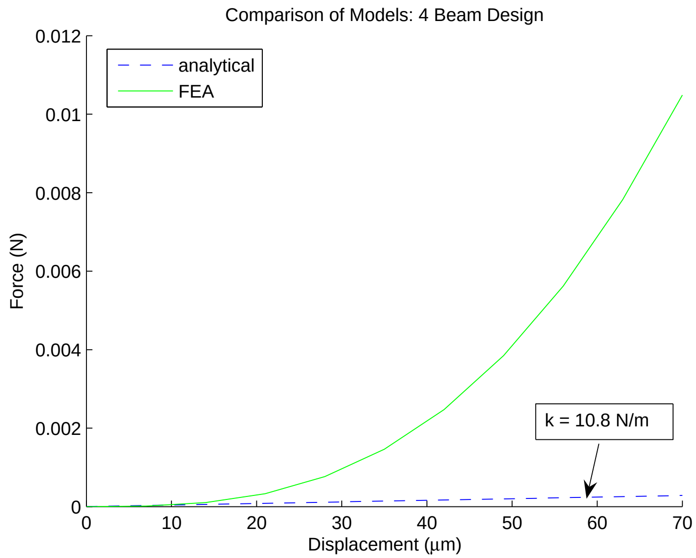
- Analytical
  - Hooke's law: F = kx



$$k = \frac{192EI}{l^3}$$

-
  - Elastic beam elements
  - Zero mass, DC or very low frequency operation
  - Nonlinear solver used
    - Able to handle large deflections
    - Geometric non-linearities
  - Static analysis using load steps

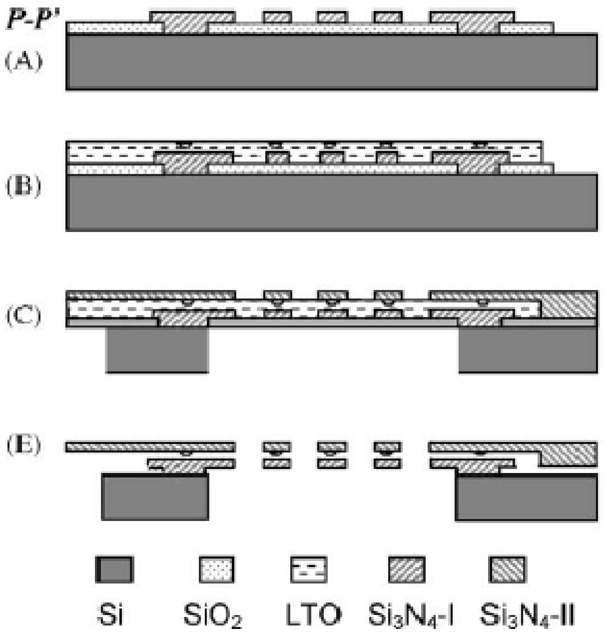


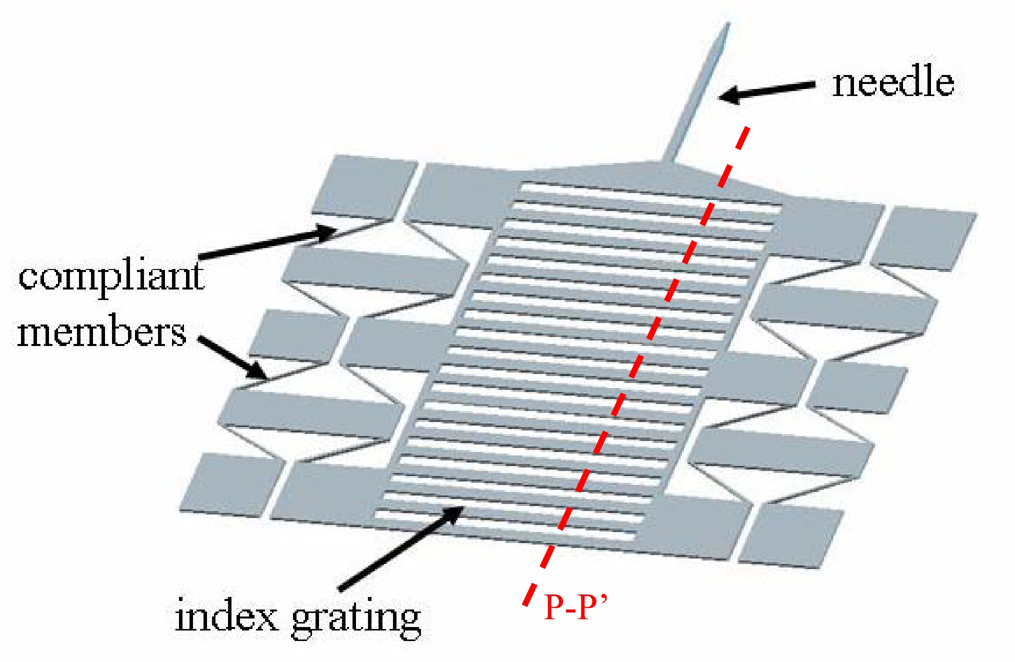


A COMPLIANT SOLUTION

- Unchangeable parameters
  - Structural material: LPCVD silicon nitride (Si_3N_4)
    - Good optical and stress qualities
  - In-plane thickness: 1.5 µm
    - Refraction properties

$$\phi_{shift} = \frac{4\pi}{\lambda} n_1 t$$





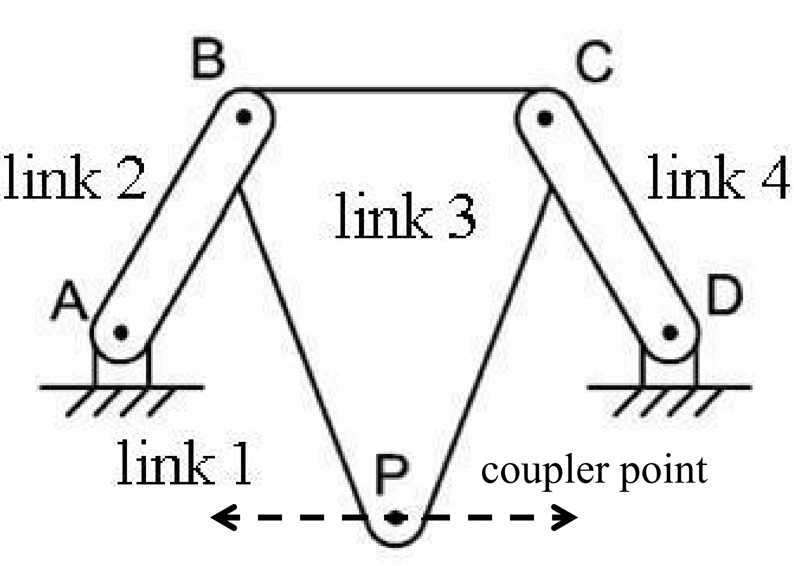
*not shown: substrate and scale grating

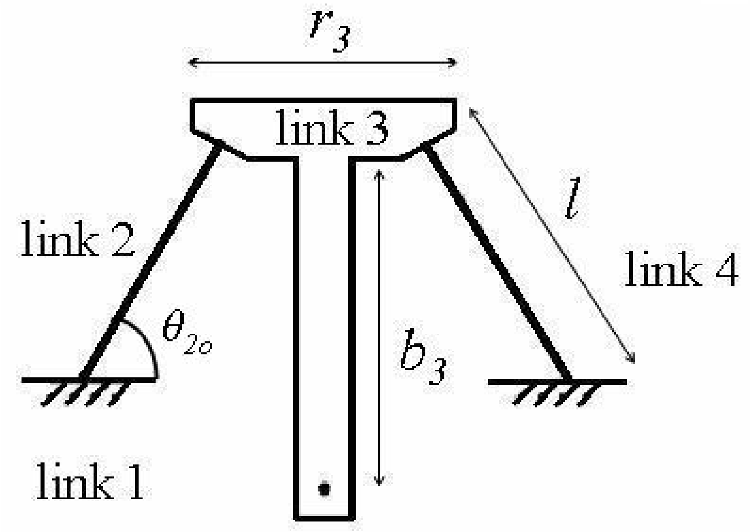
ROBERT’S MECHANISM

- 4 bar mechanism designed for straight line motion
- Certain geometric constraints are necessary:

$$\overline{AB} = \overline{DC} \qquad \overline{BP} = \overline{CP} \qquad \measuredangle \overline{BC} = 0$$

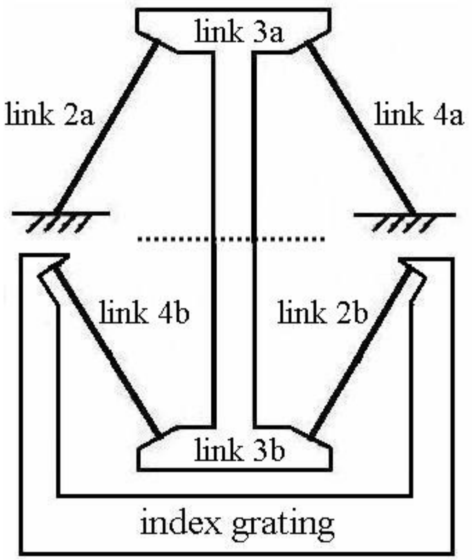
- Compliant version compatible with surface micromaching
  - Revolute joints difficult to implement in surface micromachining
  - Issues include
    - Alignment
    - Wear
    - Debris

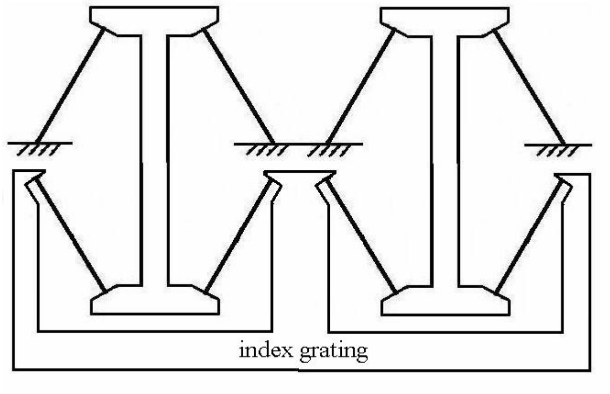


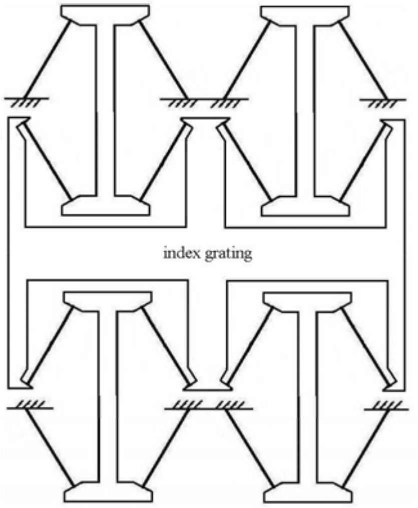


COMBINATIONS OF THE MECHANISM

- Combination in series eliminates need for revolute connection at coupler point
- Allows entire device to be monolithically fabricated
- Index grating can now rotate and translate
- Adding mechanism in parallel eliminates rotational degree of freedom
- Mirroring device reduces errors in straight-line motion caused by structural errors

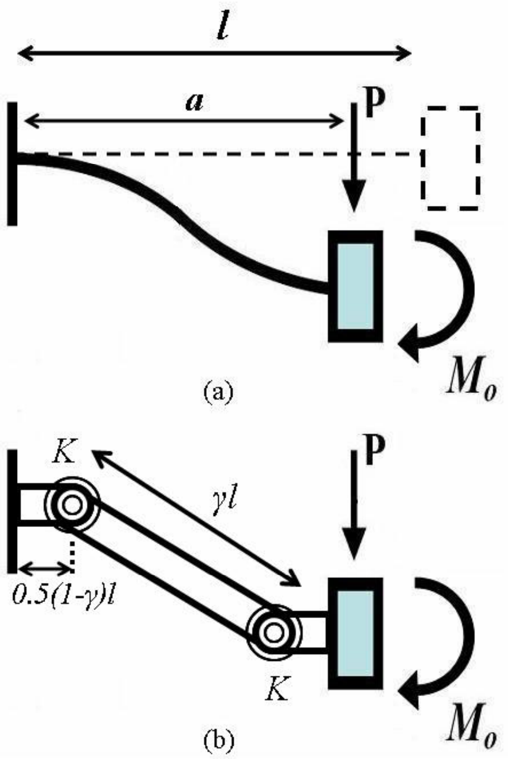






PSEUDO RIGID BODY MODEL

- Developed to bridge rigid-body mechanism theory to compliant mechanisms
- Analytical method to model compliant mechanisms using typical rigid-body kinematics
- Replaces compliant members with equivalent system of rigid links, revolute joints and torsional springs
- Resulting mechanism has the same force-displacement relation
- PRBM coefficients
  - Characteristic radius factor (γ)
  - Stiffness coefficient (K_θ)
  - Dependent boundary conditions of compliant beam



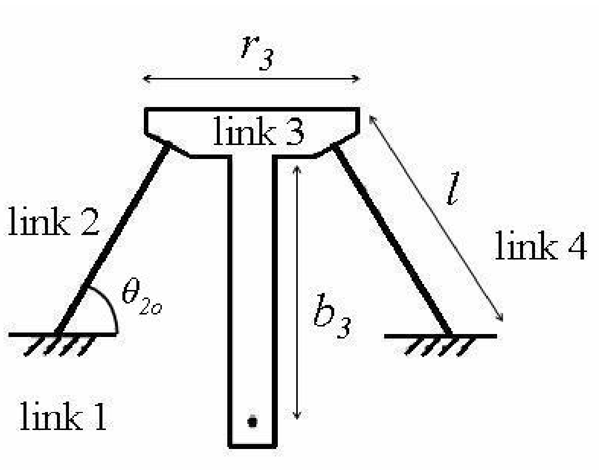
$$K = 2\gamma K_\Theta \frac{EI}{l}$$

[Howell]

APPLYING PRBM

$$x = N_S\left[r_2\left(\cos\theta_2 - \cos\theta_{2o}\right) + b_3\sin\theta_3 + \frac{r_3}{2}\left(\cos\theta_3 - 1\right)\right]$$

$$F = -N_T K\frac{\left(2 - h_{32}\right)\Delta\theta_2 + \left(2h_{42} - h_{32}\right)\Delta\theta_4 - \left(1 + h_{42} - 2h_{32}\right)\Delta\theta_3}{N_S\left[r_2\sin\theta_2 + h_{32}\left(\dfrac{r_3}{2}\sin\theta_3 - b_3\cos\theta_3\right)\right]}$$



N_S number of Robert's mechanisms in series (2 in this case)

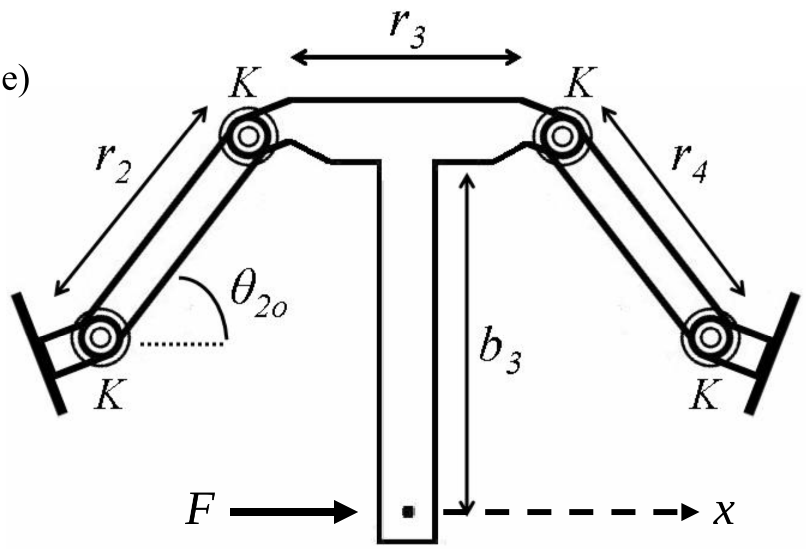
N_T total number of Robert's mechanisms (8 in this case)

θ_i final angle of link i ($i = 2,3,4$)

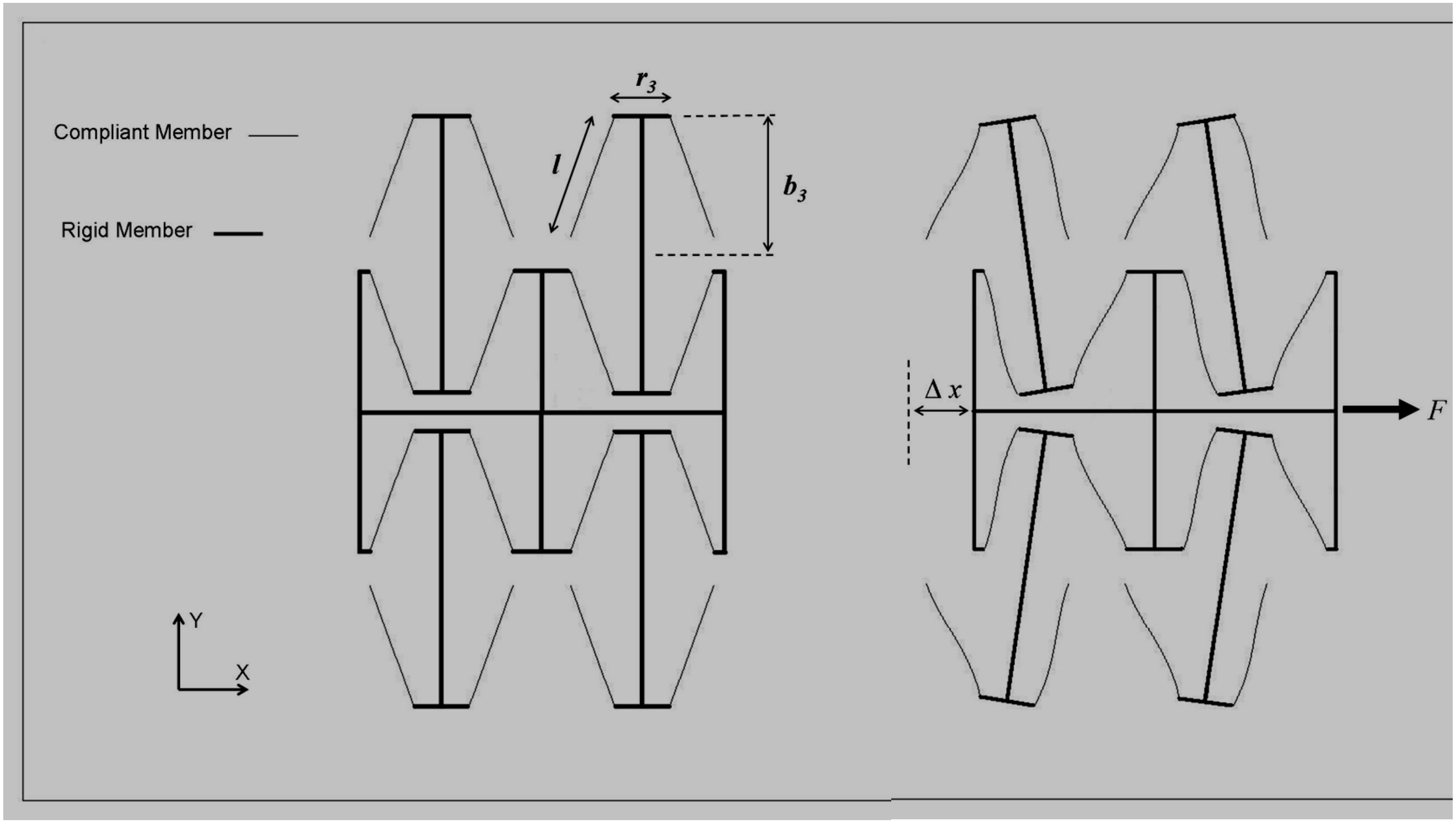
θ_{io} initial angle of link i ($i = 2,3,4$)

$\Delta\theta_i$ change in i^{th} link angle from initial position ($i = 2,3,4$)

$$h_{42} = \frac{r_2\sin\left(\theta_3 - \theta_2\right)}{r_4\sin\left(\theta_3 - \theta_4\right)} \qquad h_{32} = \frac{r_2\sin\left(\theta_4 - \theta_2\right)}{r_3\sin\left(\theta_3 - \theta_4\right)}$$

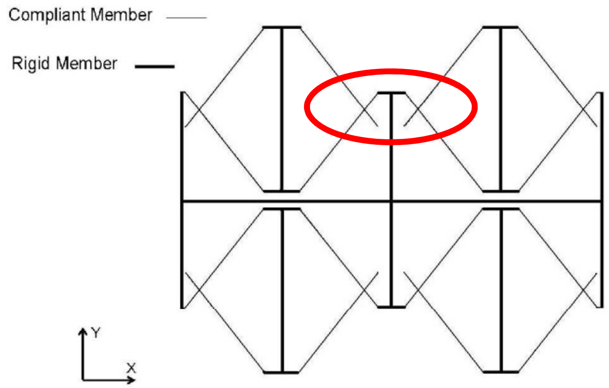


FEA MODELS

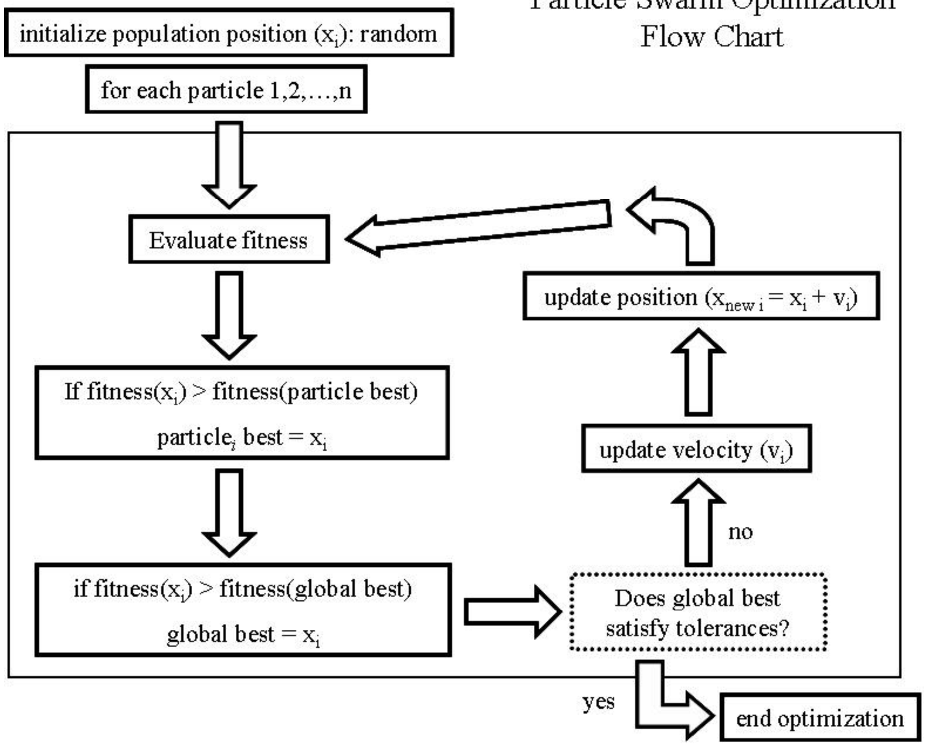


OPTIMIZATION

- Particle swarm optimization (PSO) is a population based stochastic method
- Objectives to minimize
  - Stiffness: slope of the force-displacement curve
  - Non-linearity: adjusted R-square value (ARS), best fit metric
- Constraints
  - Max stress < yield stress
  - Feasible configuration
  - Geometric



Particle Swarm Optimization Flow Chart



initialize population position (x_i): random

for each particle 1,2,…,n

Evaluate fitness

If fitness(x_i) > fitness(particle best)
particle$_i$ best = x_i

if fitness(x_i) > fitness(global best)
global best = x_i

Does global best satisfy tolerances?

no

update velocity (v_i)

update position ($x_{new\,i} = x_i + v_i$)

yes

end optimization

$$f_{obj}(\overline{X}) = \alpha * slope + (1-\alpha)*(1-ARS)$$

$$\overline{X} = [w, l, r_3, b_3, \theta_{2o}] \qquad l, r_3, b_3 \in [1\mu m, 300\mu m]$$

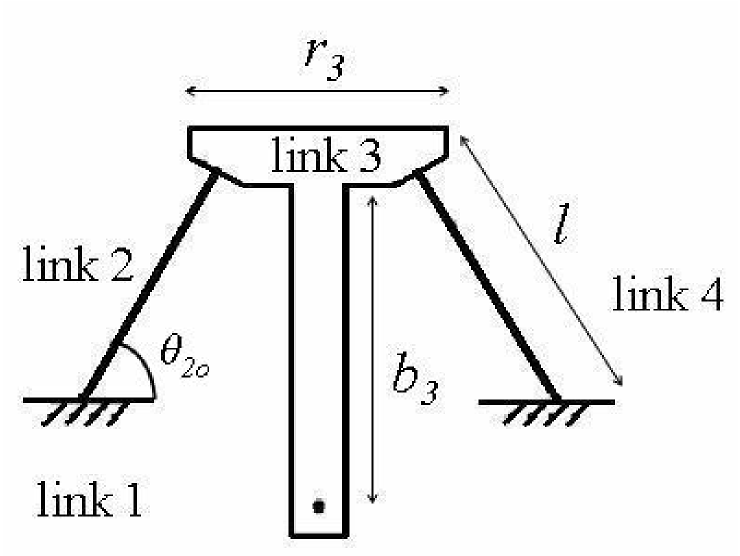
$$w \in [1\mu m, 10\mu m]$$

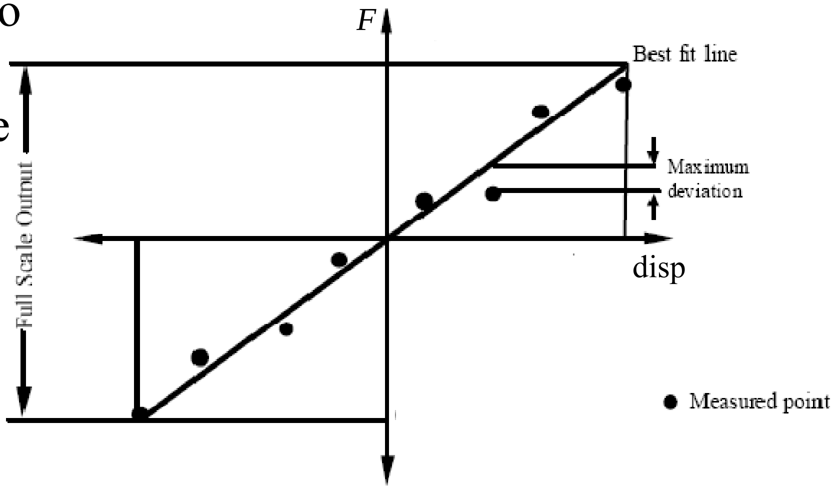
$$\theta_{2o} \in [20^\circ, 90^\circ]$$

OPTIMIZATION RESULTS

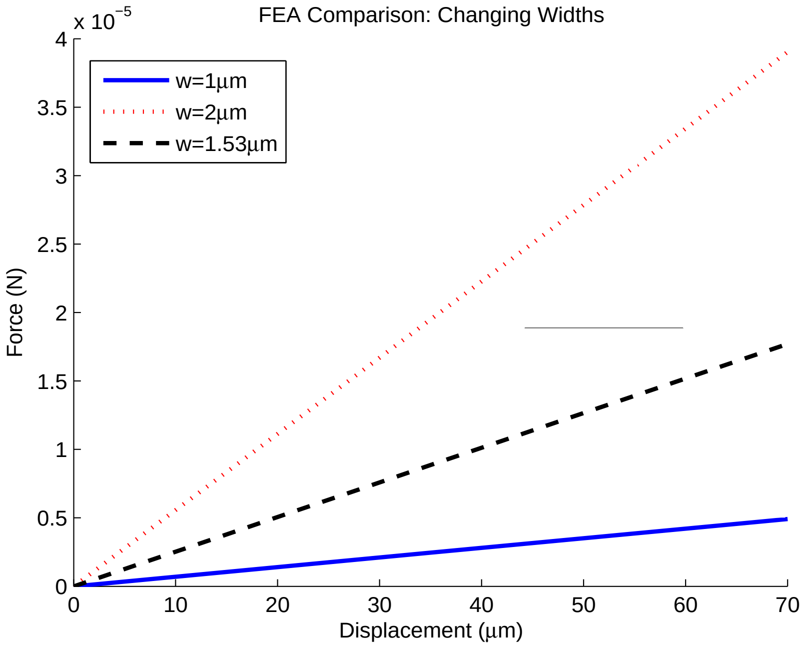
- Initial optimization conducted for 70 μm displacement range
  - 1.53, 222.0, 108.3, and 261.0 μm for w, l, $r3$ and $b3$, respectively
  - Initial angle for link 2 (θ_{2o}) = 72.5°
  - Stiffness: 0.26 N/m
  - ARS: 0.9997
  - Max displacement before failure: 200 μm
- Non-linearity
  - Measured in % of full scale output (*%FSO*)
  - Ratio of maximum deviation compared to overall range for a set of data
  - Generally 3%FSO or below is acceptable

$$NL = \frac{maximum\, deviation}{full\, scale\, output} \times 100\%$$





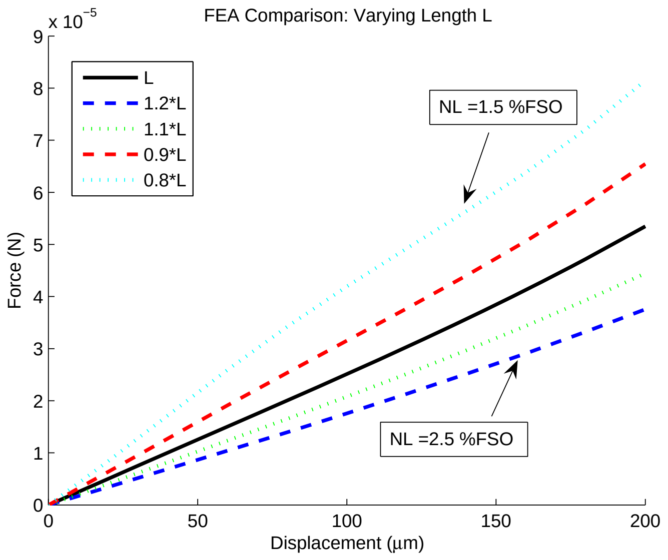
EFFECTS OF GEOMETRY


FEA Comparison: Changing Widths

w (*μm*)	**Max displacement (*μm*)**	**Max Force (*μN*)**	**FEA stiffness (*N/m*)**	***NL* (*%FSO*)**
1.53	200	53.4	0.26	2.8
2.0	170	98.6	0.57	1.73
2.5	138	153	1.10	0.98
3.0	118	222	1.88	0.74

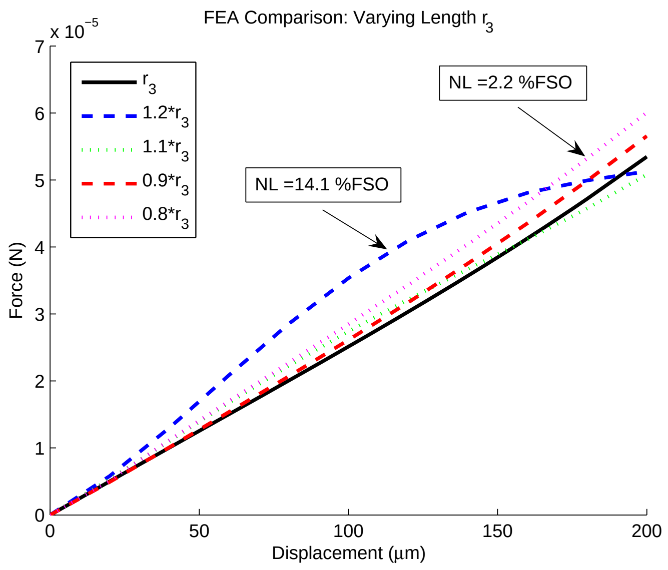
	change variable	**change stiffness**
w	−	−
l	+	−
r_3	−	−
b_3	+	−

PARAMETRIC STUDY


FEA Comparison: Varying Length L

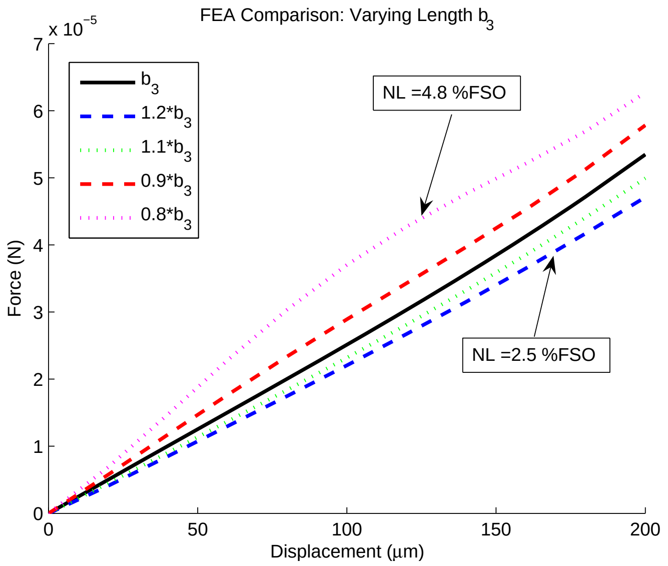
NL =1.5 %FSO

NL =2.5 %FSO


FEA Comparison: Varying Length r_3

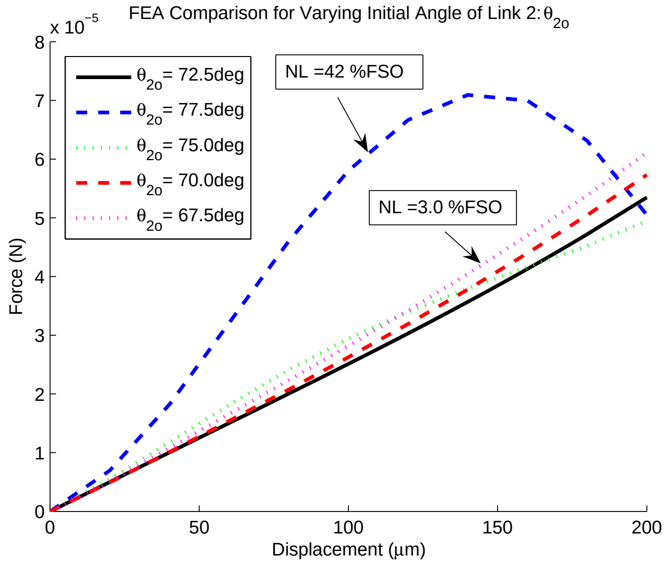
NL =2.2 %FSO

NL =14.1 %FSO


FEA Comparison: Varying Length b_3

NL =4.8 %FSO

NL =2.5 %FSO


FEA Comparison for Varying Initial Angle of Link 2: θ_{2o}

NL =42 %FSO

NL =3.0 %FSO

REVISED OPTIMIZATION

- Parametric study revealed optimal design dependent on displacement range
- Refinement of optimization algorithm
  - Initial PSO employed FEA within algorithm
  - Expensive function and time consuming
  - Latin hypercube design of experiments used to eliminate FEA and approximate *NL*
    - Define a cubic polynomial in 4 variables
    - 3 link lengths and width selected
    - 35 unknown coefficients, 70 FEA simulations
    - Polynomial ARS = 0.95
- Adjustment to problem statement
  - Modified objective function
  - Placed non-linearity as constraint
  - Changing constraint tolerance generates Pareto chart

$$\min k_{sensor}$$

$$l, r_3, b_3 \in [1\mu m, 300\mu m]$$

$$w \in [1\mu m, 10\mu m]$$

$$\theta_{2o} = 72.5^{\circ}$$

$$\sigma_{calc} < \sigma_{failure}$$

$$NL \le 3\%$$

PARETO OPTIMAL RESULTS

- New optimization conducted for 200 μm displacement range
- Non-linear constraint varied from 3 to 0.5%*FSO* in 0.5 increments
- Non-linear constraint: 3 to 1%*FSO*
  - 1, 300, 80, and 300 μm for w, l, r_3 and b_3, respectively
  - All geometric constraints active
  - *NL* and max stress constraints not active
  - Stiffness: 0.029 *N/m*
- Non-linear constraint: < 1%*FSO*
  - 1, 295, 116, and 300 μm for w, l, r_3 and b_3, respectively
  - *NL* constraint has become active
  - Stiffness: 0.033 *N/m*

$$\min k_{sensor}$$

$$l, r_3, b_3 \in [1\mu m, 300\mu m]$$

$$w \in [1\mu m, 10\mu m]$$

$$\theta_{2o} = 72.5^\circ$$

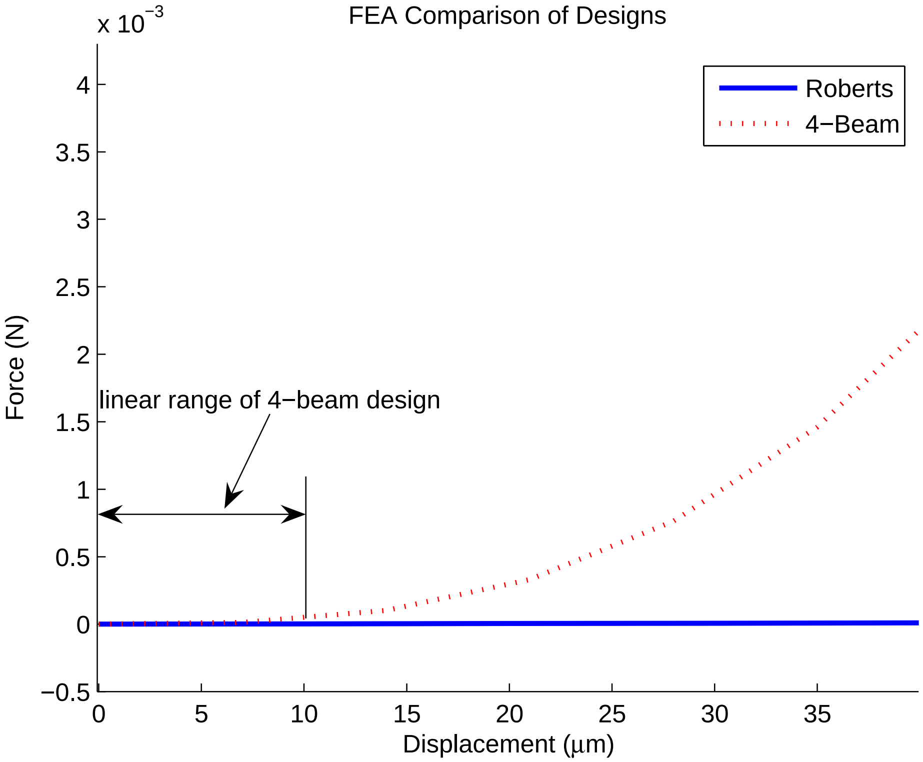
$$\sigma_{calc} < \sigma_{failure}$$

$$NL \leq 3\%$$

COMPARISON OF DESIGNS

NEW DESIGN ATTRIBUTES

- Larger linear range
- Very small stiffness without compromise of linearity
  - Higher sensitivity
- Minimizes compliance in remaining 5 DOF
  - Reduces optical errors
- Combination of mechanisms reduces measurement error due to structural variations
- Over etching of compliant members can be compensated via experimental calibration



CONCLUSIONS

- Optical force sensor
  - Compliant mechanisms enhances sensor characteristics
    - Increased linear range
    - High sensitivity via smaller stiffness
    - Allows optics to be setup for greater dynamic range
    - Robustness with respect to fabrication errors
    - Reduction in cross-axis sensitivity
  - Versatile platform
    - Selection of geometric parameters can yield devices with varying characteristics
    - Foundation for potential use in other areas such as microassembly

FUTURE WORK

- Experimental validation
  - Verify FEA results
  - Test effects of fabrication process
- Dynamic analysis
  - Update model to include mass properties
  - Determine natural frequency and resonance modes
  - Feasibility for high frequency applications and actuation
- Integration of components
  - Eliminate large laser source and lenses
  - Single or multi-chip design
  - Allow for inclusion in a lab-on-chip setup
- Explore higher degree of freedom sensors
  - 2DOF, 3DOF or 6DOF
- Patent Application in progress

System Integration of Sensing

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DEVELOPMENT OF A COMPLIANT ROTARY-TO-TRANSLATIONAL TRANSMISSION MECHANISM FOR A MICRO-PARALLEL KINEMATIC MECHANISM

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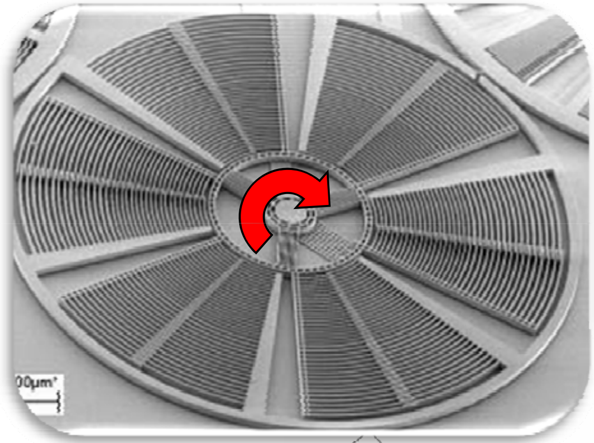
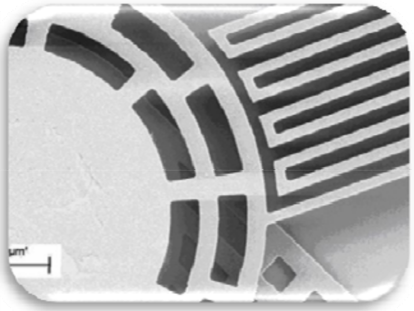
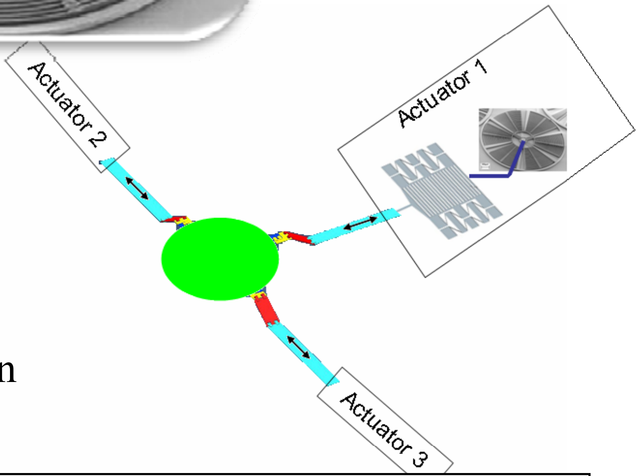
3DOF Micro-PKM





- 1DOF electrostatic actuator
- 3DOF PKM
- Compliant linkage design



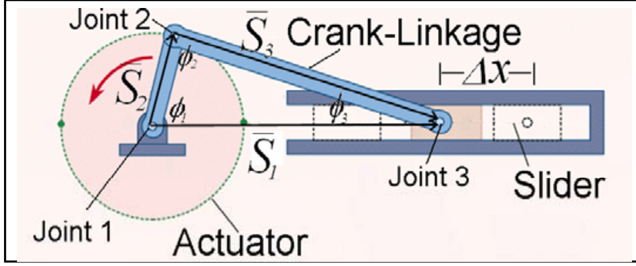
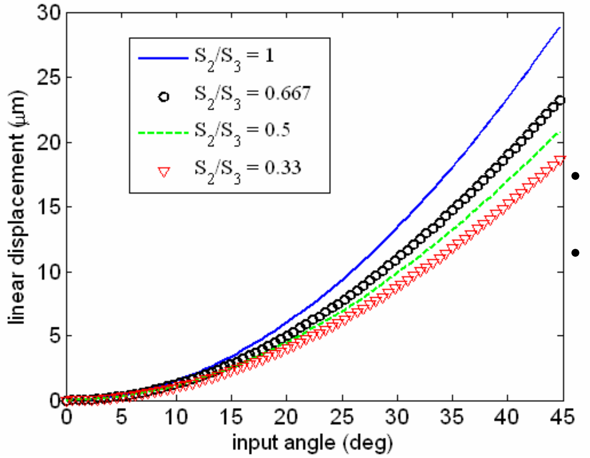


Fig 4. Rigid body slider-crank transmission mechanism.

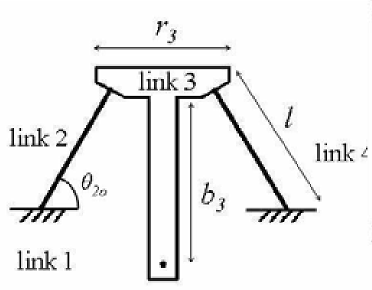
- Goal: 3DOF micromanipulator for precise micro/nano positioning
- Applications: manufacturing, assembly, teleoperation, biomedical devices, and optical positioning
- Challenges: adhesion, friction, dimensional inaccuracies and binding in the joints
- Approach: Integrate electrostatic rotary actuator into micro-PKM device using a rotary-to-translation transmission based on a compliant slider-crank mechanism.

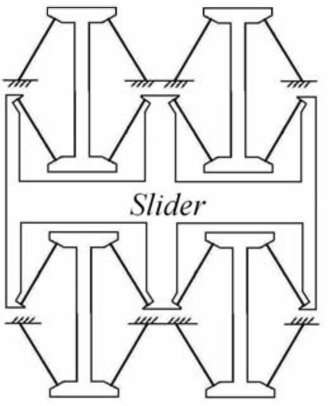
Transmission Design

Compliant Slider-Crank Design



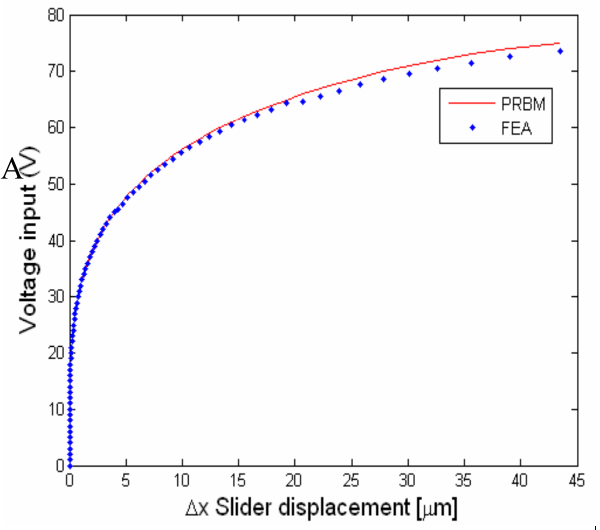
- Transmission ratio for crank-linkage
- Robert's mechanism slider



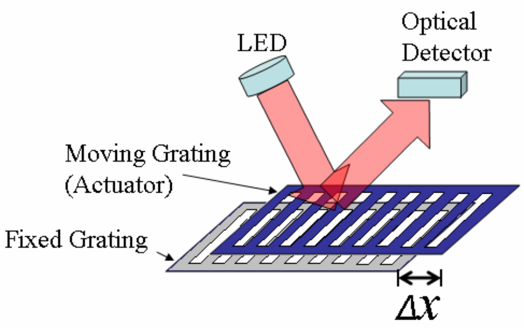


Actuator Performance and Sensing

- Evaluate actuator system performance using PRBM and FEA



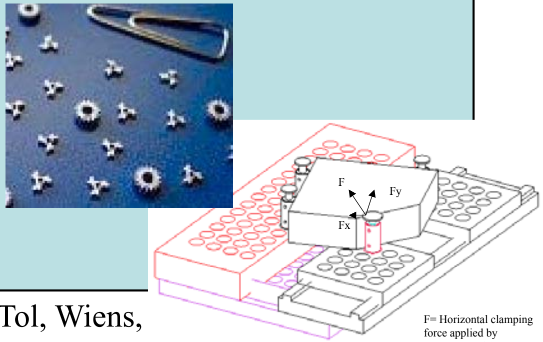
- Incorporate optical sensing into slider for control feedback



Design of Fixturing and/or Manipulation Devices

THRUST: Development of a technology suite for micro/mesoscale manufacturing utilizing micromanipulation and positioning devices – mechanisms / robotics

- **Material Handling and Fixturing**
  - Precision Positioning and Manipulation
  - Mechanically Adaptive Tooling – Dynamic Fixturing
  - Integration of MEMS Devices



F= Horizontal clamping force applied by Locating pin/clamp

Tol, Wiens, Schueller, 2003

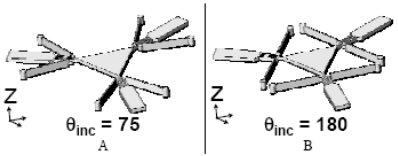
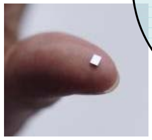
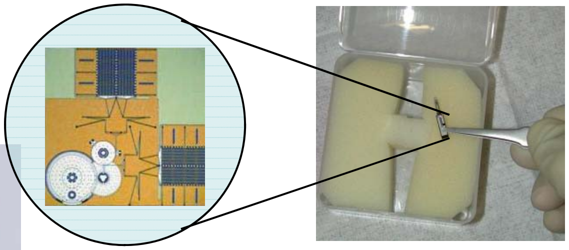


Fig. 5: HexFlex configured in two different stages

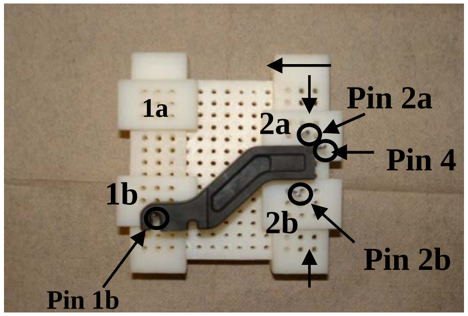
Culpepper

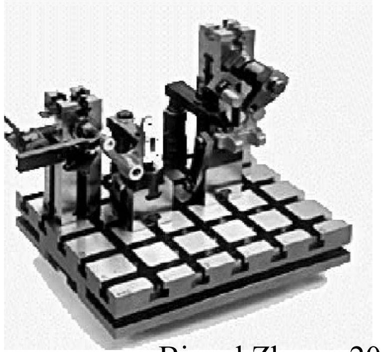




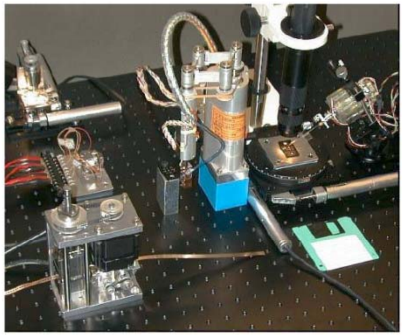
0.8 mm x 0.8 mm

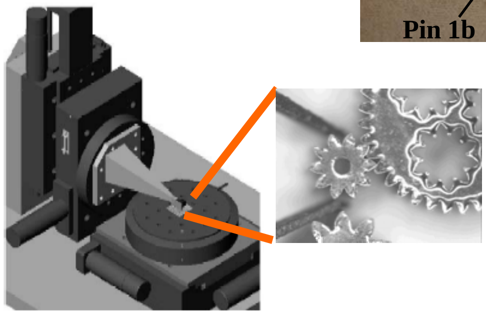
Memspi.com



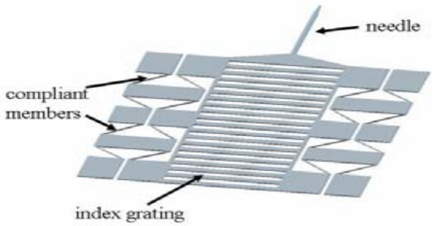


Bi and Zhang, 2001



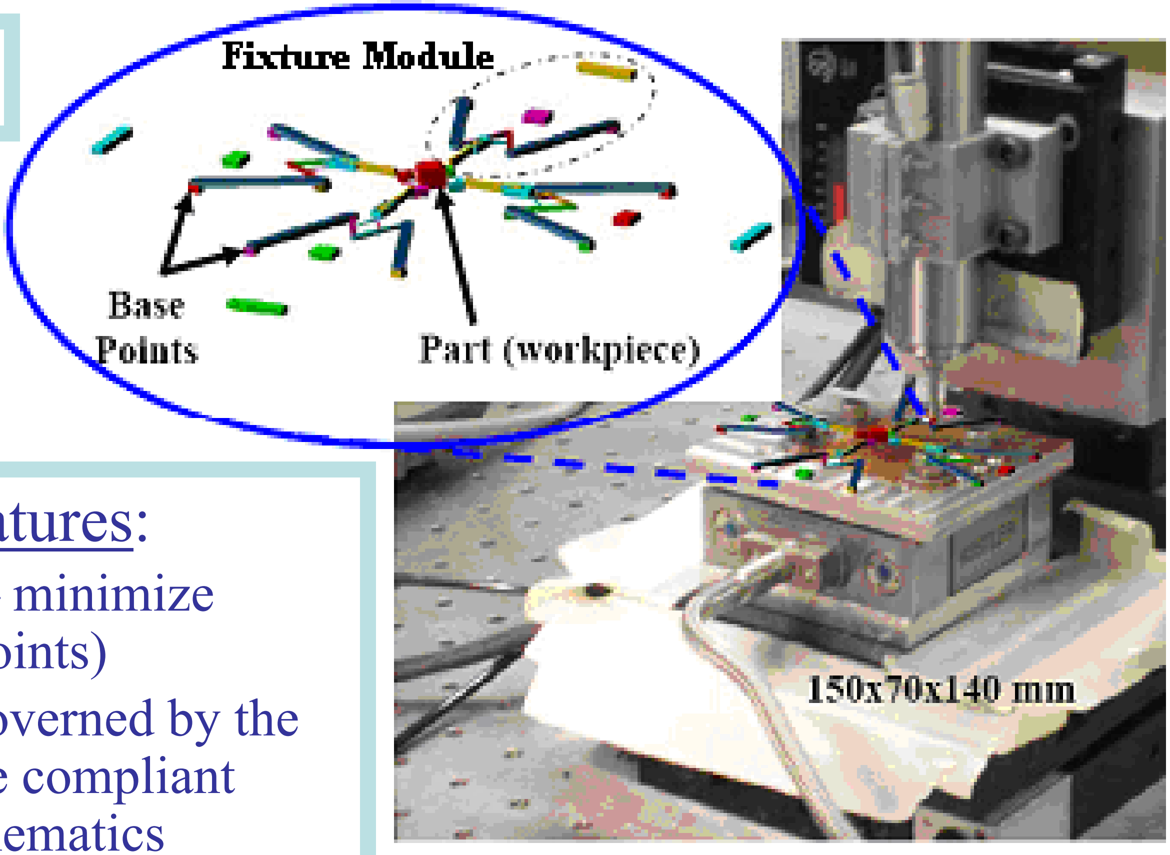


[Wiens & Gustavo, 2007]



Mechanically Adaptive Tooling – Monolithic

Micro-manipulator



→ → Micro machine tool stage courtesy of Jun Ni
S.M. Wu Manufacturing Research Center
The University of Michigan, Ann Arbor, MI

Macro-manipulator

Monolithic Design Features:

Modular fixture elements – minimize fabrication errors (e.g., joints)

Mechanically adaptive – governed by the underlying physics of the compliant elements and parallel kinematics

Mechanically adjusting the fixture device enables tuning of the kinematics and dynamics → passive and active control of tool-part interface dynamics