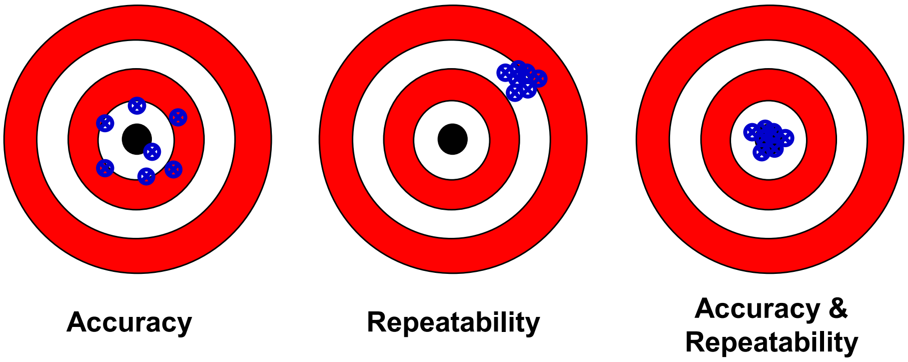
ADJUSTABLE GEOMETRIC CONSTRAINTS

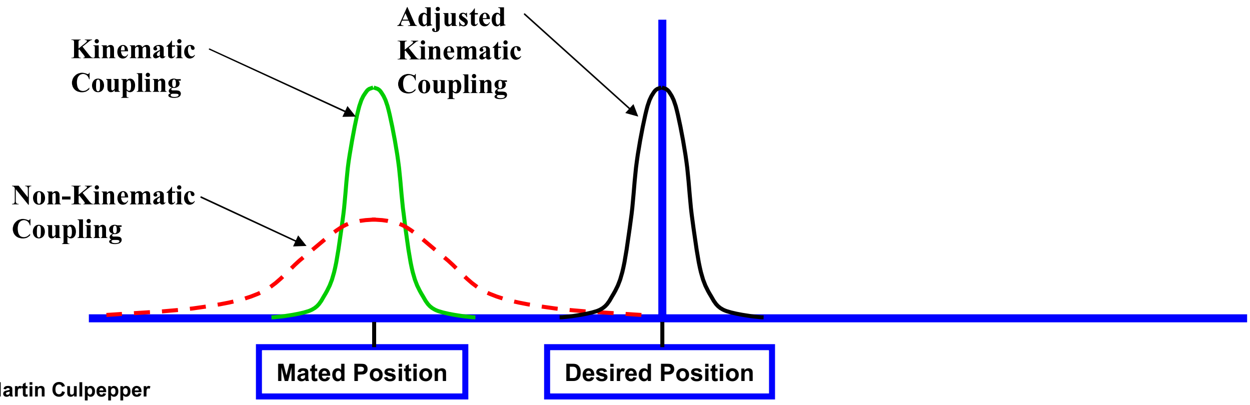
Hundreds of years of use/development and the &#@!*&% thing is not yet accurate!?!?!?

Why adjust kinematic couplings?

KC Repeatability is orders of magnitude better than accuracy

Accuracy = *f* (manufacture and assemble)

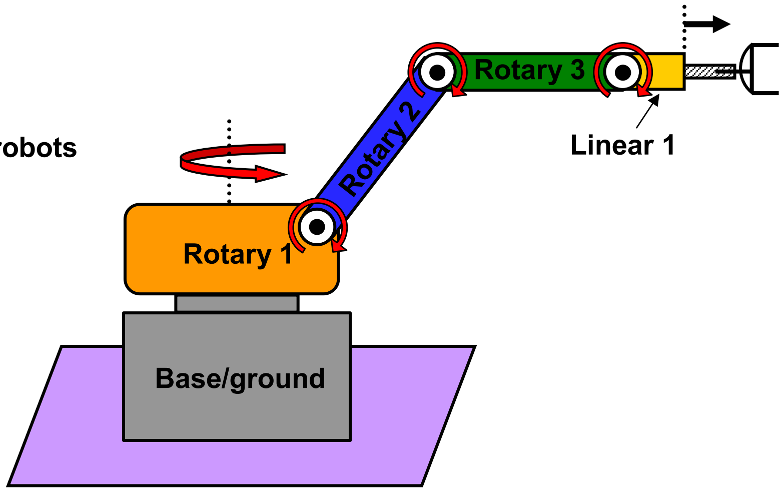




Serial and parallel kinematic machines/mechanisms

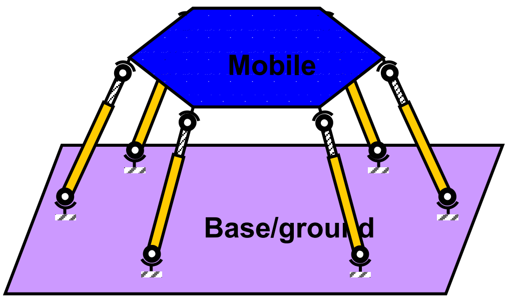
SERIAL MECHANISMS

- Structure takes form of open loop
- I.e. Most mills, lathes, “stacked” axis robots
- Kinematics analysis typically easy



PARALLEL MECHANISMS

- Structure of closed loop chain(s)
- I.e. Stuart platforms & hexapods
- Kinematics analysis usually difficult
- 6 DOF mechanism/machine
- Multiple variations on this theme



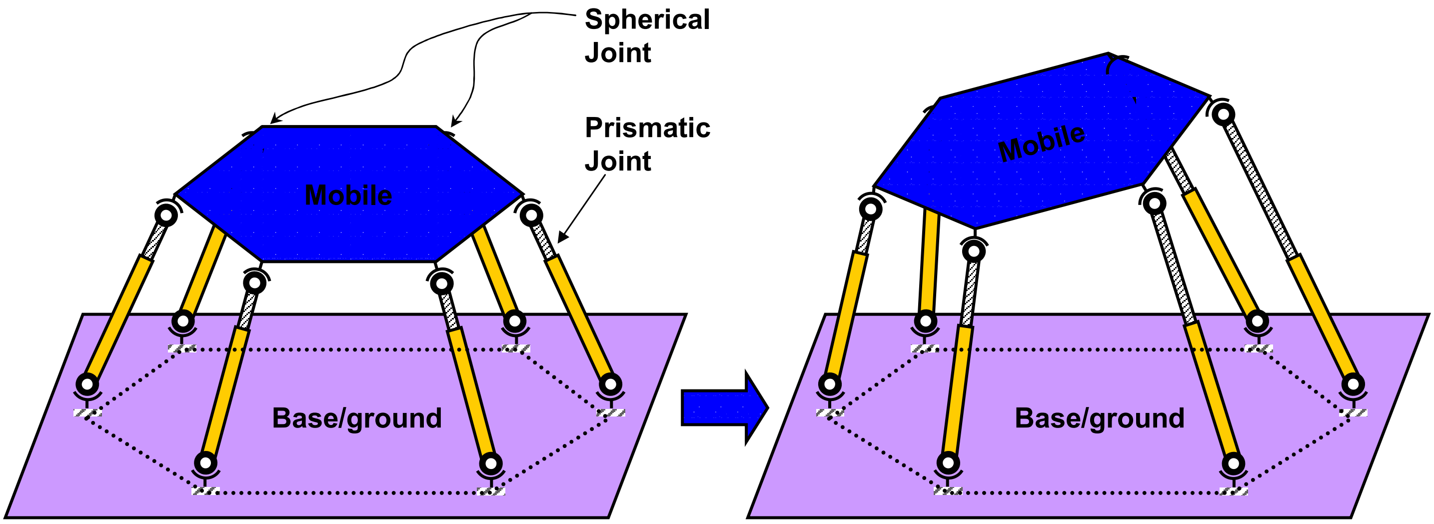
Parallel mechanism: Stewart-Gough platform

6 DOF mechanism/machine

Multiple variations on this theme with different joints:

Spherical joints:	3 Cs	Permits 3 rotary DOF	Ball Joint
Prismatic joints:	5 Cs	Permits one linear DOF	Sliding piston
Planar:	3 Cs	Permits two linear, one rotary DOF	Roller on plane

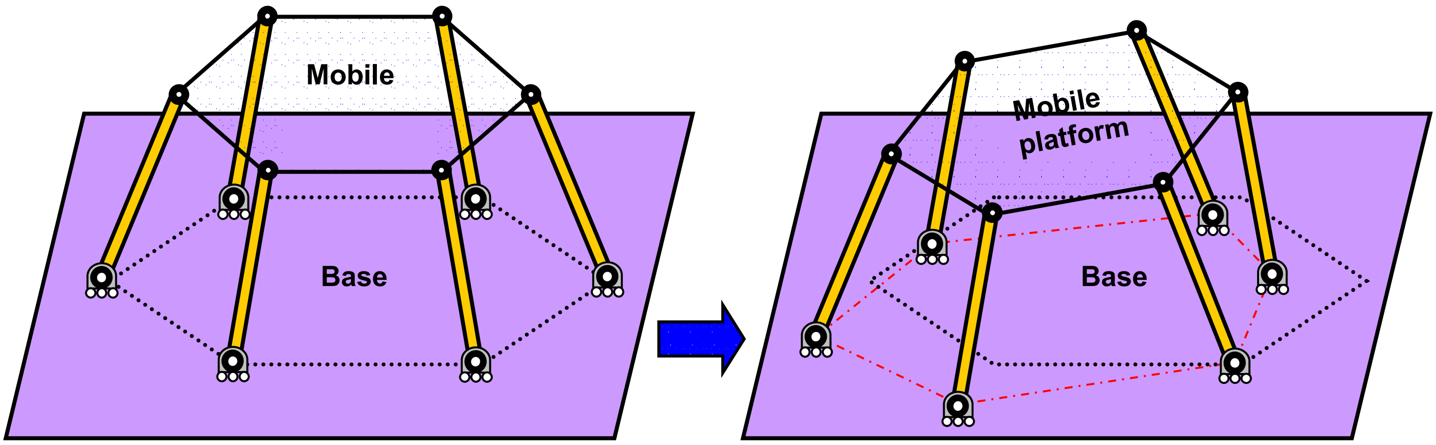
E.g. Changing length of “legs”



Parallel mechanism: Variation

6 DOF mechanism/machine by changing position of joints

Can have a combination of position and length changes



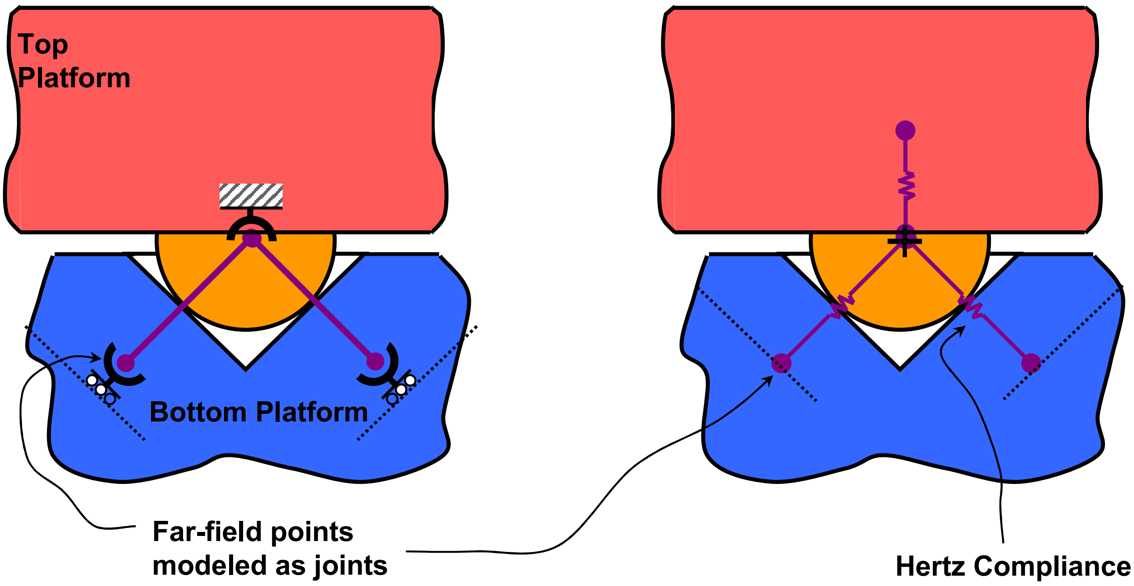
Kinematic couplings as mechanisms

Ideally, kinematic couplings are static parallel mechanisms

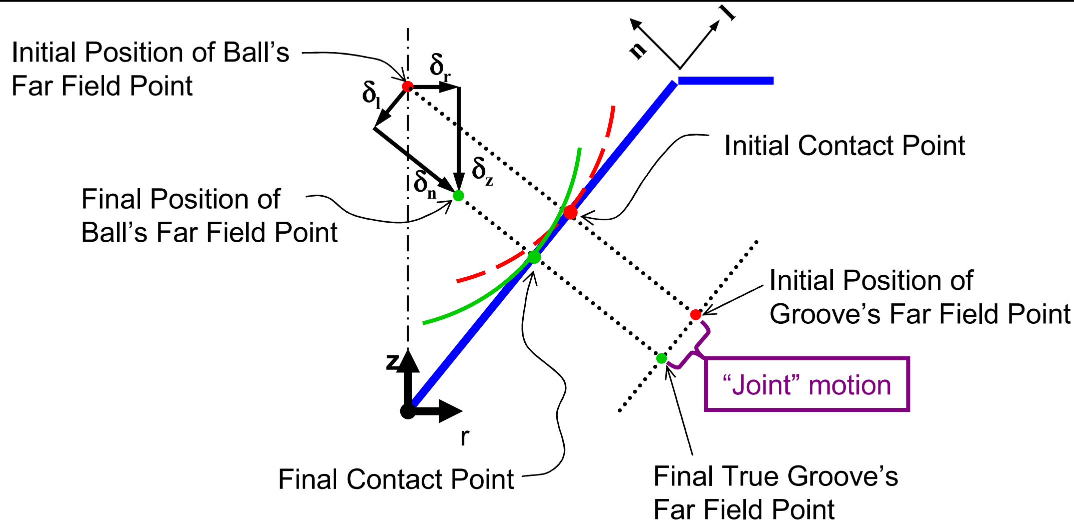
IRL, deflection(s) = mobile parallel kinematic mechanisms

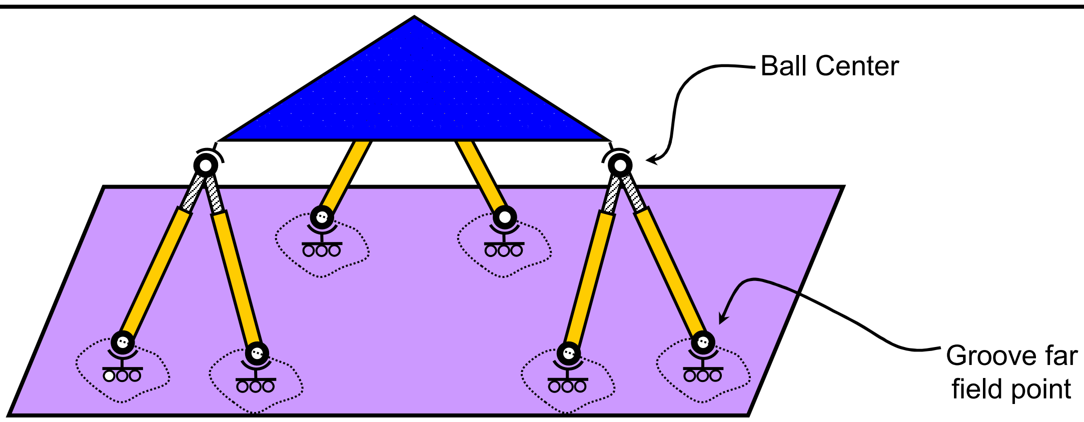
How are they "mobile"?

- Hertz normal distance of approach ~ length change of leg
- Far field points in bottom platform moves as ball center moves ~ joint motions



Model of kinematic coupling mechanism





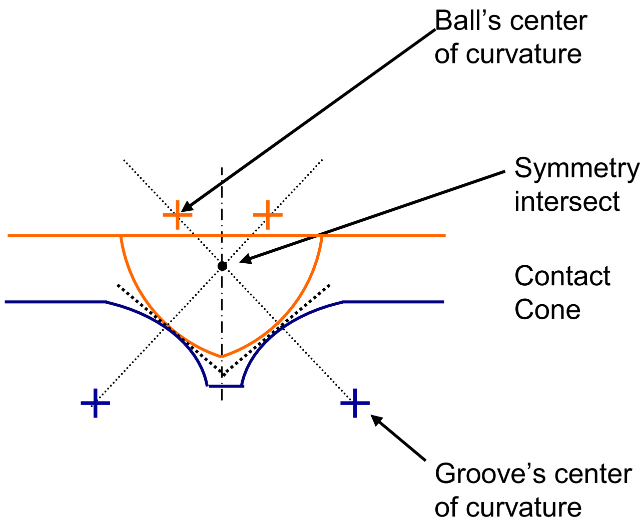
Accuracy of kinematic mechanisms

Since location of platform depends on length of legs and position of base and platform joints, accuracy is a function of mfg and assembly

Parameters affecting coupling centroid (platform) location:

- Ball center of curvature location
- Ball orientation (i.e. canoe ball)
- Ball centerline intersect position (joint)
- Ball radii

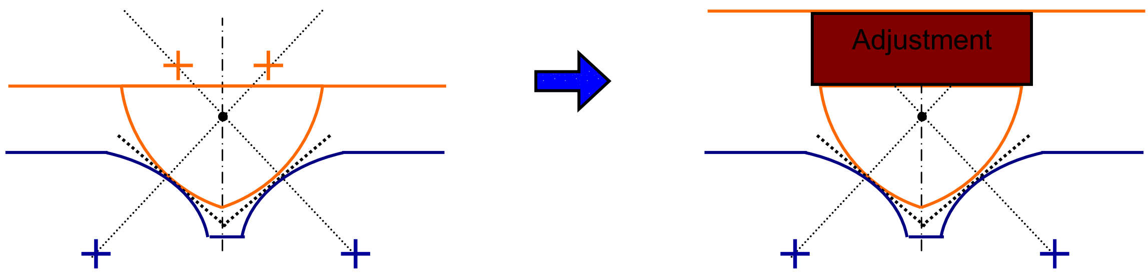
- Groove center of curvature location
- Groove orientation
- Groove depth
- Groove radii



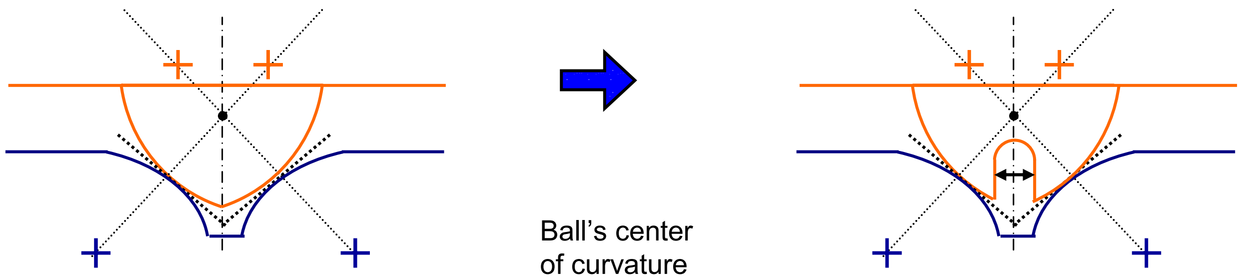
Utilizing the parallel nature of kinematic couplings

Add components that adjust or change link position/size, i.e.:

- **Place adjustment between kinematic elements and platforms (joint position)**



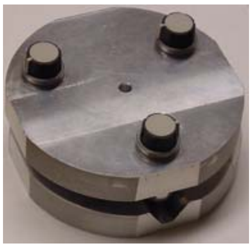
- **Strain kinematic elements to correct inaccuracy (element size)**

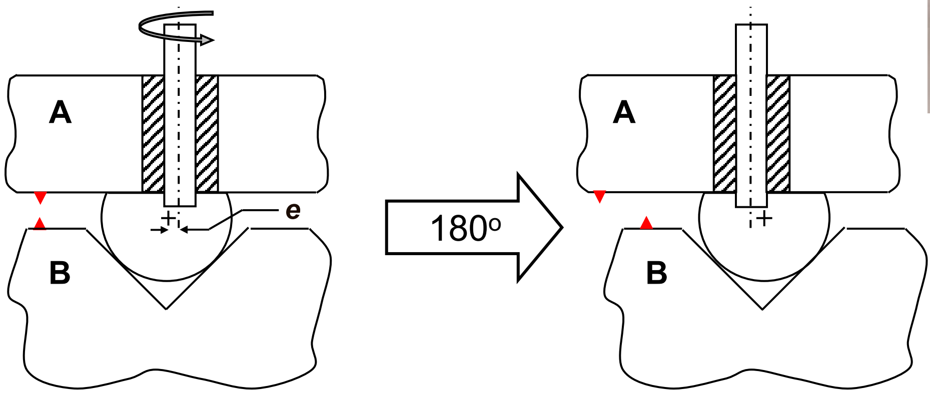


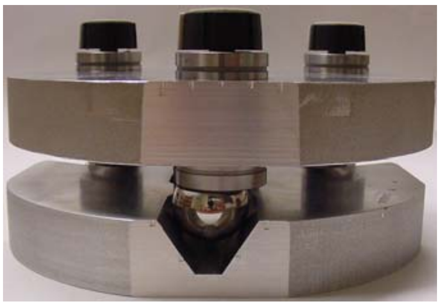
Example: Adjusting planar motion

Position control in x, y, θ_z:

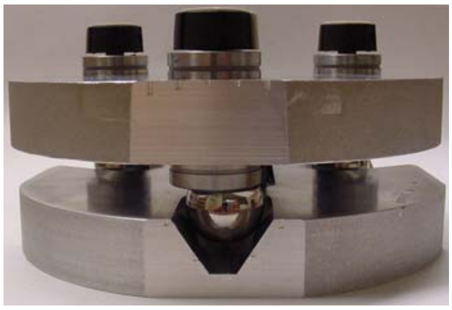
- Rotation axis offset from the center of the ball







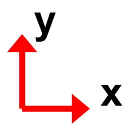
Eccentric left

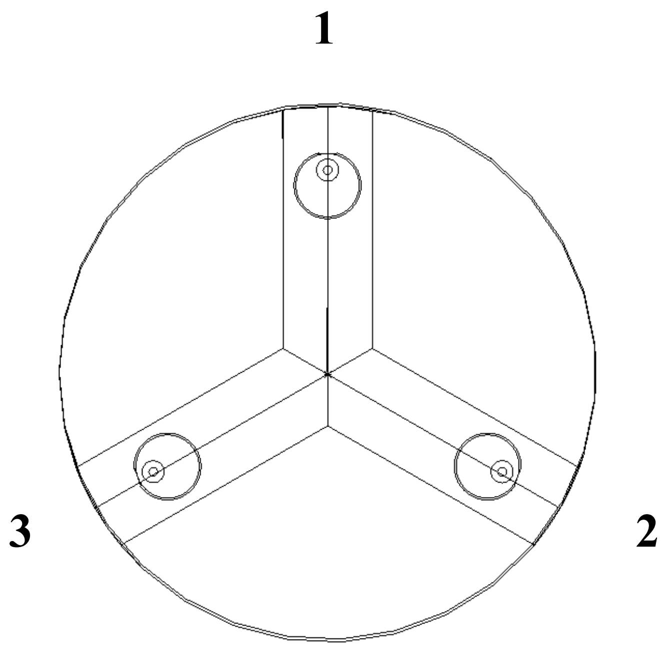


Eccentric right

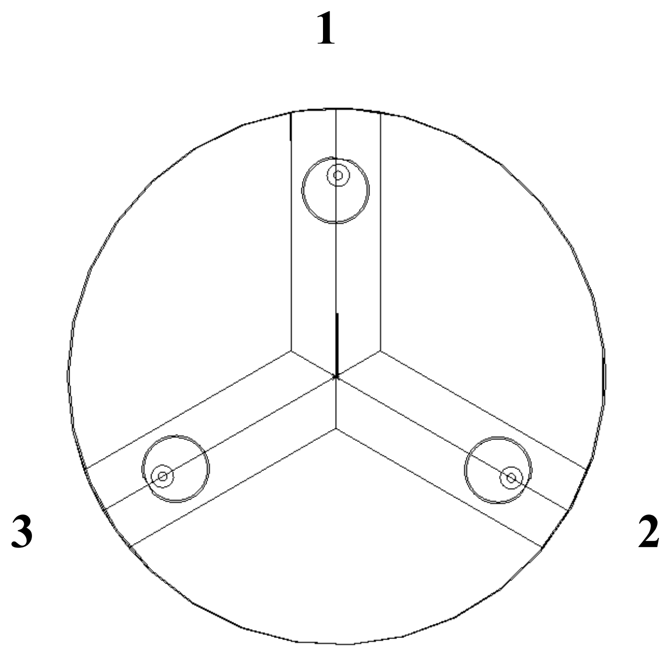
Patent Pending

ARKC demo animation





Input: Actuate Balls 2 & 3
Output: Δy

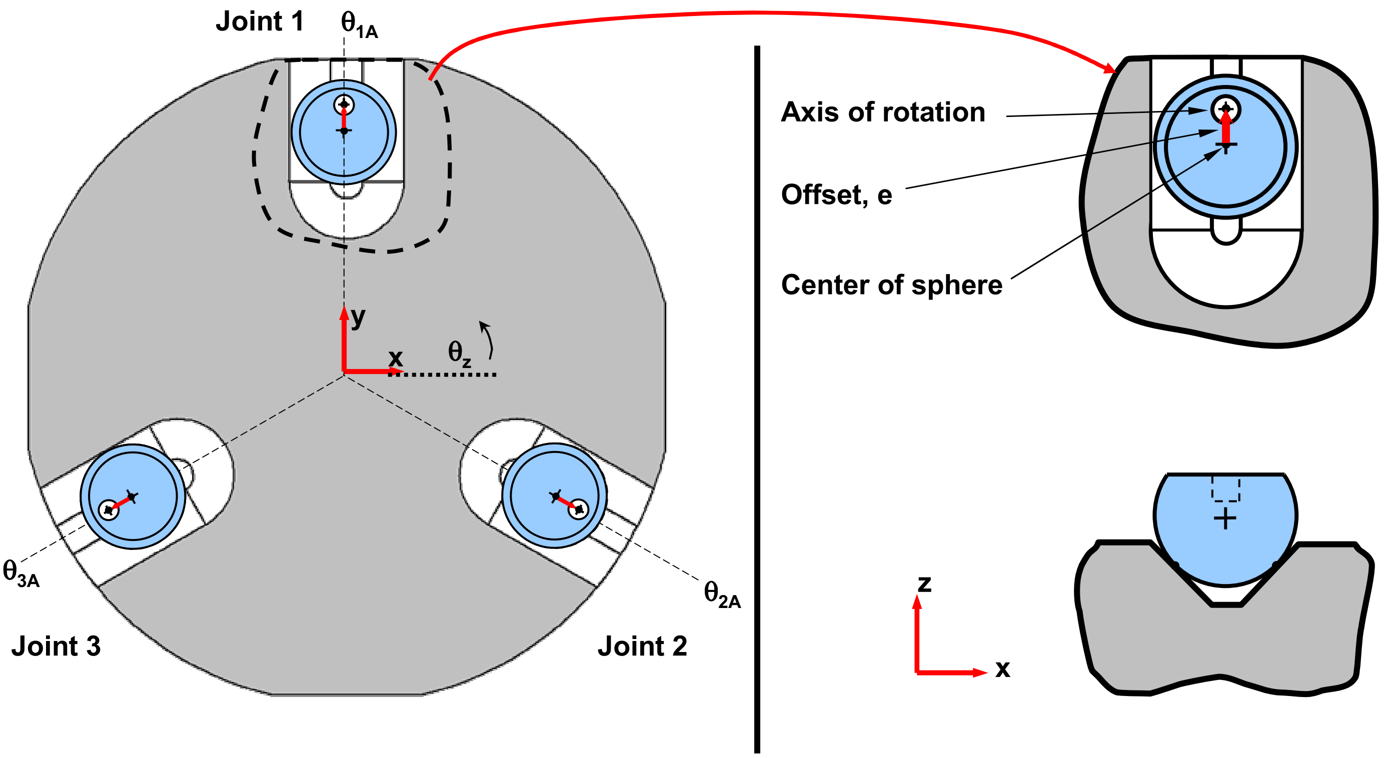


Input: Actuate Ball 1
Output: Δx and $\Delta\theta_z$

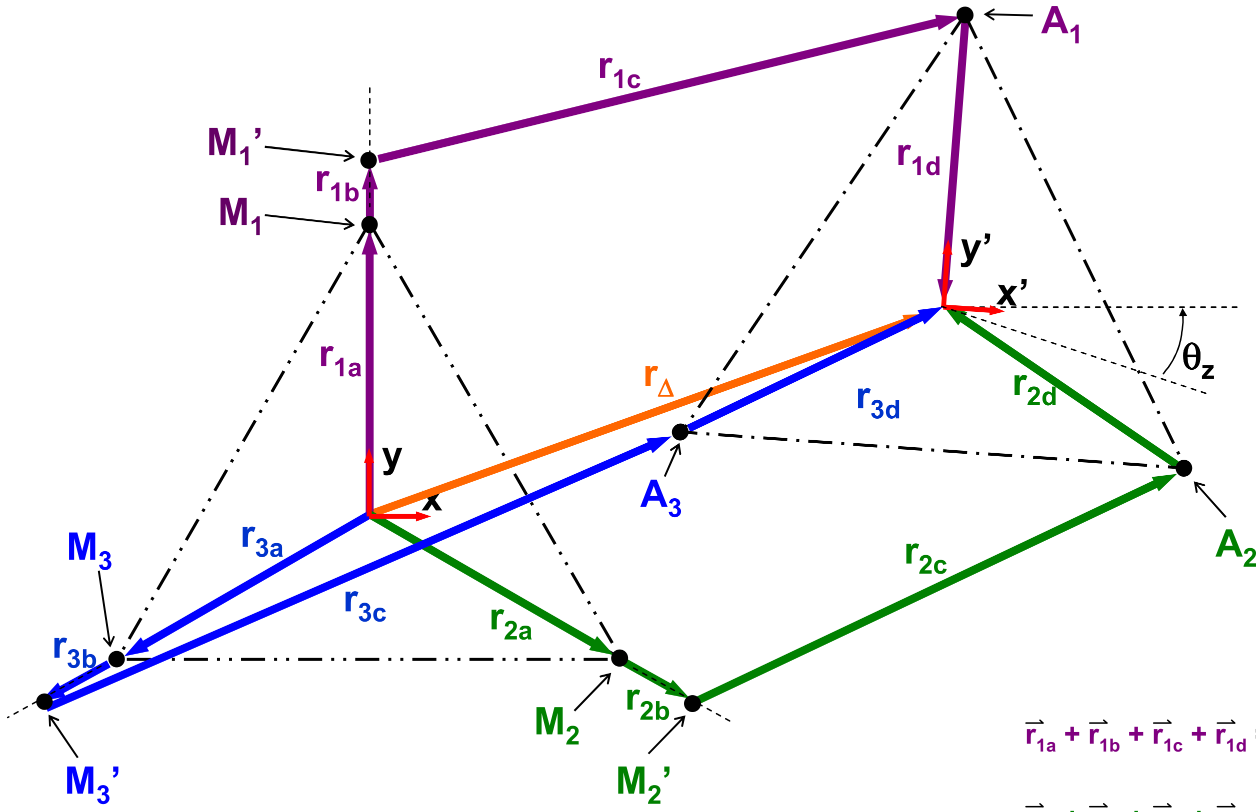
Planar kinematic model

Equipping each joint provides control of 3 degrees of freedom

View of kinematic coupling with balls in grooves (top platform removed)



Vector model for planar adjustment of KC



$$\vec{r}_{1a} + \vec{r}_{1b} + \vec{r}_{1c} + \vec{r}_{1d} = \vec{r}_{\Delta}$$

$$\vec{r}_{2a} + \vec{r}_{2b} + \vec{r}_{2c} + \vec{r}_{2d} = \vec{r}_{\Delta}$$

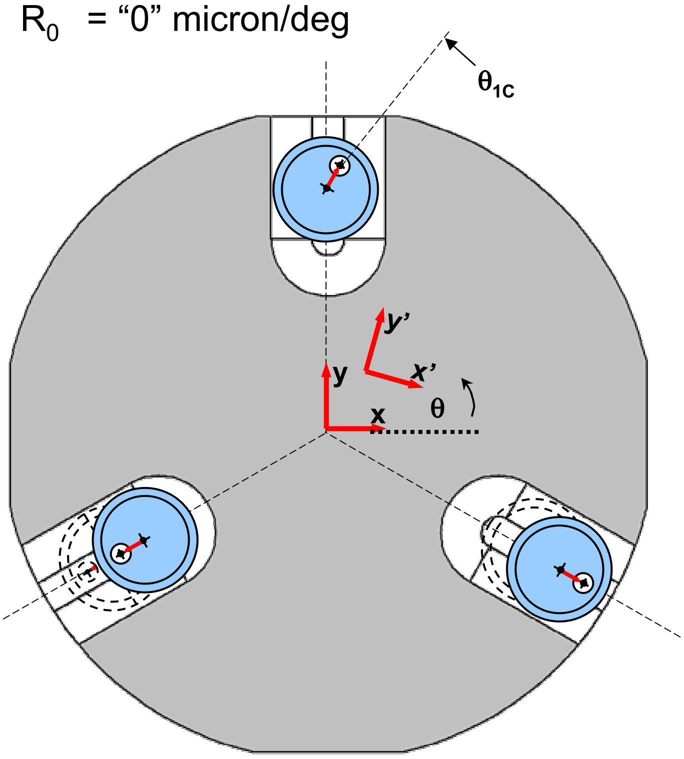
$$\vec{r}_{3a} + \vec{r}_{3b} + \vec{r}_{3c} + \vec{r}_{3d} = \vec{r}_{\Delta}$$

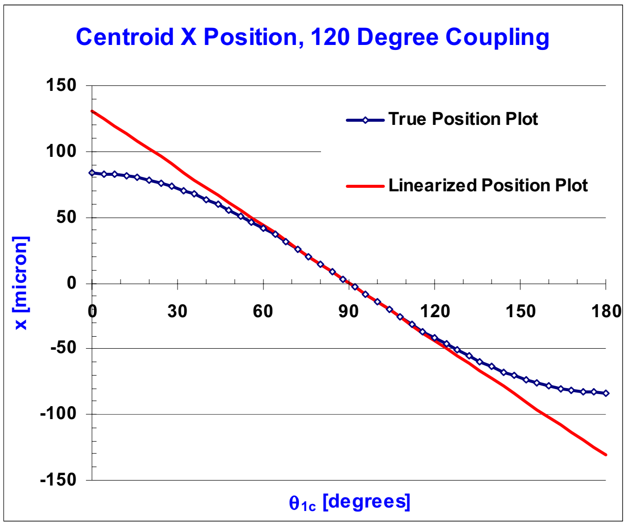
ARKC resolution analysis

For E = 125 microns

R_{90} = -1.5 micron/deg

R_0 = "0" micron/deg





Limits on Linear Resolution Assumptions

	θ_{1C}		
% Error	**Lower Limit** [Degree]	**Upper Limit** [Degree]	**Half Range** [Degree]
1	75	105	+/- 15
2	70	110	+/- 20
5	60	120	+/- 30
10	47	133	+/- 43

Forward and reverse kinematic solutions

ARKC Kinematic Analysis Spread Sheet

Do not change cells in red, only change cells in blue

PART I: COUPLING CHARACTERISTICS

Use this to specify groove angles and input ball rotations and heights used to calculate coupling position in part II

θ_{1A}	**90**	degrees	z_1	**5**	microns
	1.571	radians		0.0001969	inches
θ_{2A}	**330**	degrees			
	5.760	radians	z_2	**500**	microns
$\theta 3_A$	**210**	degrees		0.0196850	inches
	3.665	radians			
			z_3	**200**	microns
θ_{1C}	**87.7073**	degrees		0.0078740	inches
	1.531	radians			
θ_{2C}	**331.1455**	degrees			
	5.780	radians			
$\theta 3_C$	**211.1467**	degrees			
	3.685	radians			
E	**125**	microns			
	0.0049	inches			
R_T	**57150**	microns	Note: $R_T = L_{iD}$		
	2.2500	inches			

PART II: CALCULATED MOVEMENTS

This takes input from part I to calculate the position of the top part of the coupling.

θ_Z	0.0000	radians	θ_X	-0.004024	radians
	0.0006	µradians		-4024.4889	µradians
	0.000000	degrees		-0.230586	degrees
x	5.0005	microns	θ_Y	-0.003031	radians
	0.000197	inches		-3030.7040	µradians
				-0.173647	degrees
y	-0.0014	microns	z	235.0000	microns
	0.000000	inches		0.009252	inches

PART III: REVERSE SOLUTION

Use this to input a desired position and goal seek to solve for the position of the balls

	DESIRED POSITION	POSITION ERROR		
θ_Z	**0.000**	0.00	0.0	µrad
x	**5.000**	-0.001		microns
y	**0.000**	0.001		microns

BALL SETTINGS

See part I for modified ball angles, the following angles are the difference between groove and ball angles

θ_1	-2.2927	degrees
θ_2	1.1455	degrees
$\theta 3$	1.1467	degrees

ERROR SUM **2698.1** <- x, y, θ_z <- Use the solver on this value to set it to a value (ideally zero) that is small, but greater than or equal to 0.

Low-cost adjustment (10 μm)

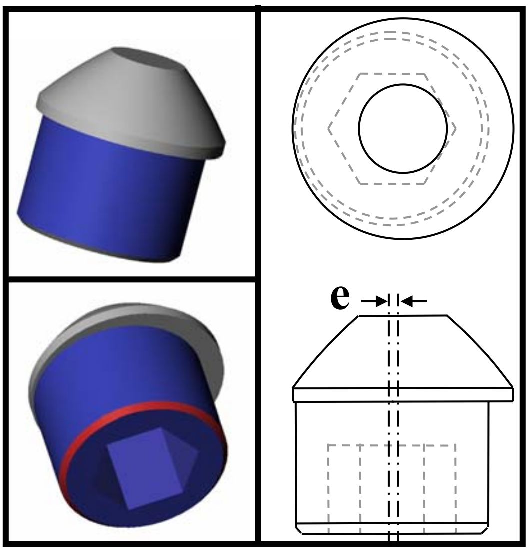
Peg shank and convex crown are offset

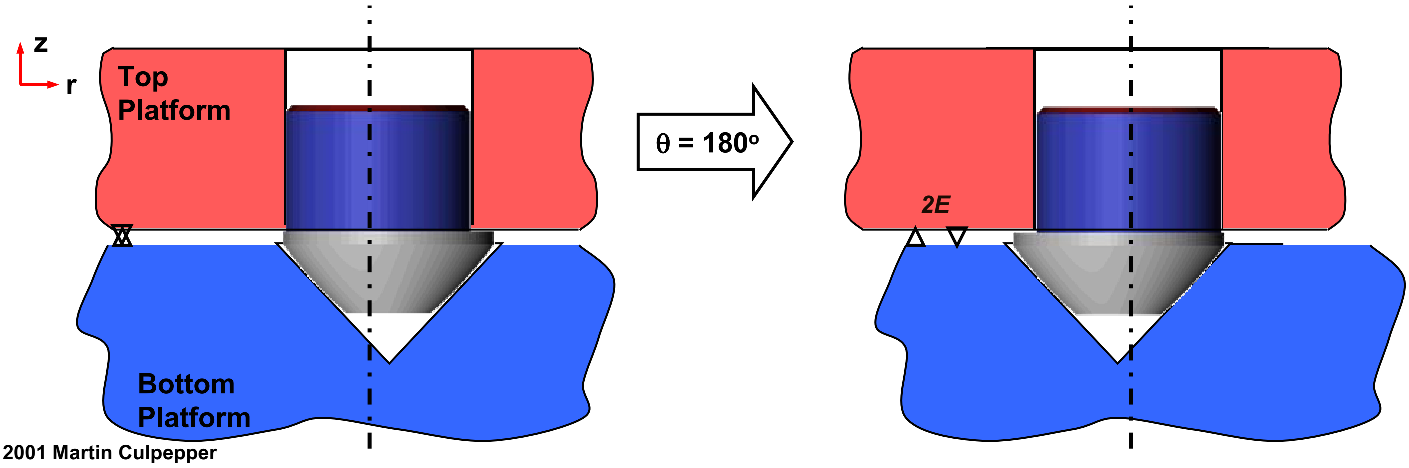
Light press between peg and bore in plate

Adjustment with allen wrench

Epoxy or spreading to set in place

Friction (of press fit) must be minimized…



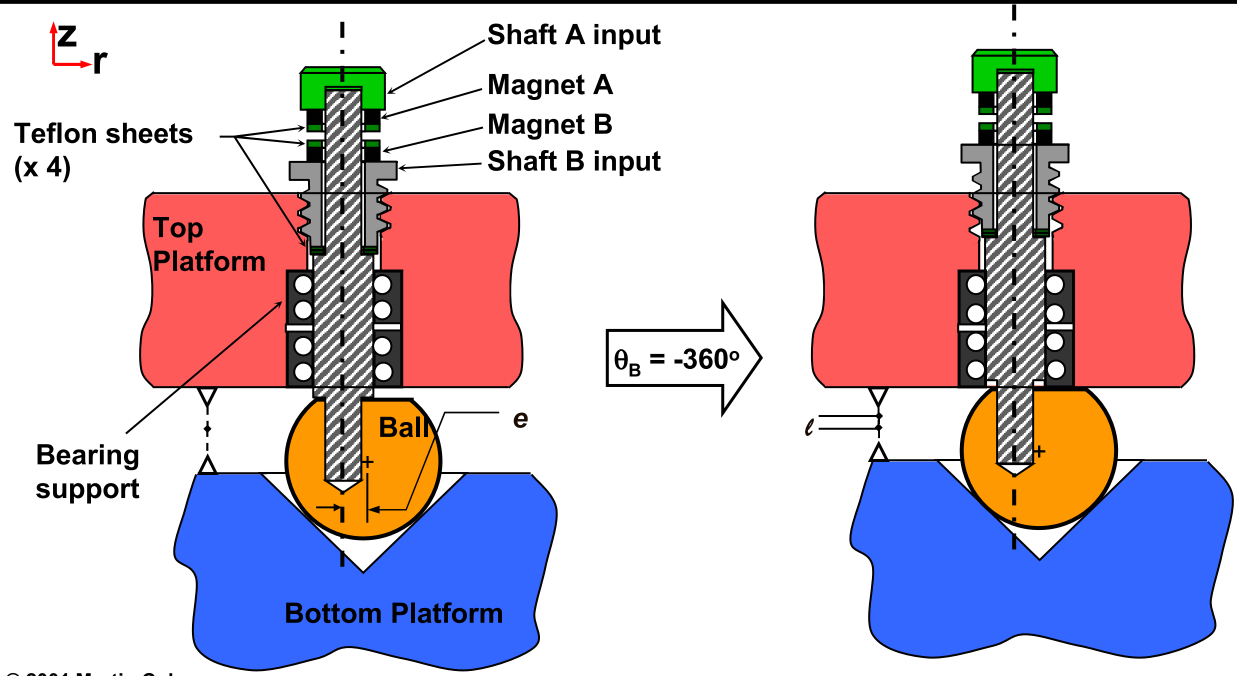


Moderate-cost adjustment (3 micron)

Shaft B positions z height of shaft A [z, θ_x, θ_y]

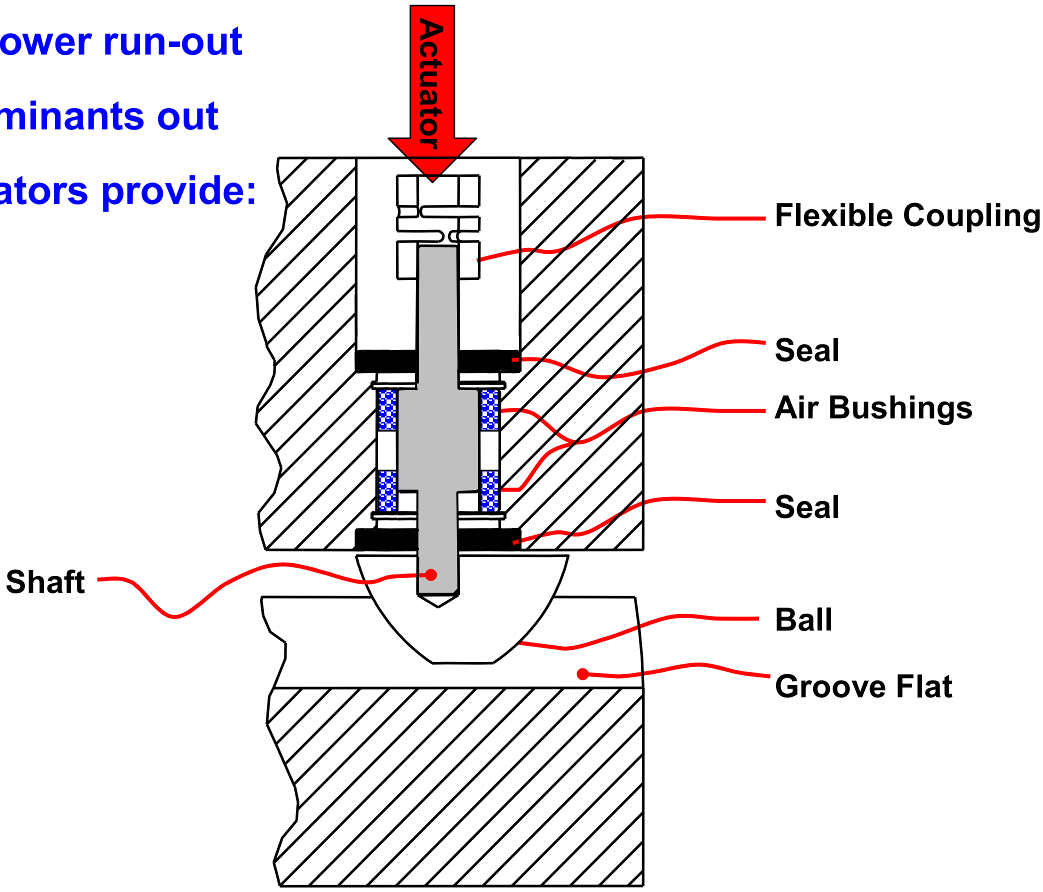
Shaft A positions as before [x, y, θ_z]

Force source preload I.e. magnets, cams, etc..



“Premium” adjustment (sub-micron)

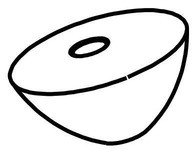
- Run-out is a major cause of error
- Air bushings for lower run-out
- Seals keep contaminants out
- Dual motion actuators provide:
  - Linear motion
  - Rotary motion



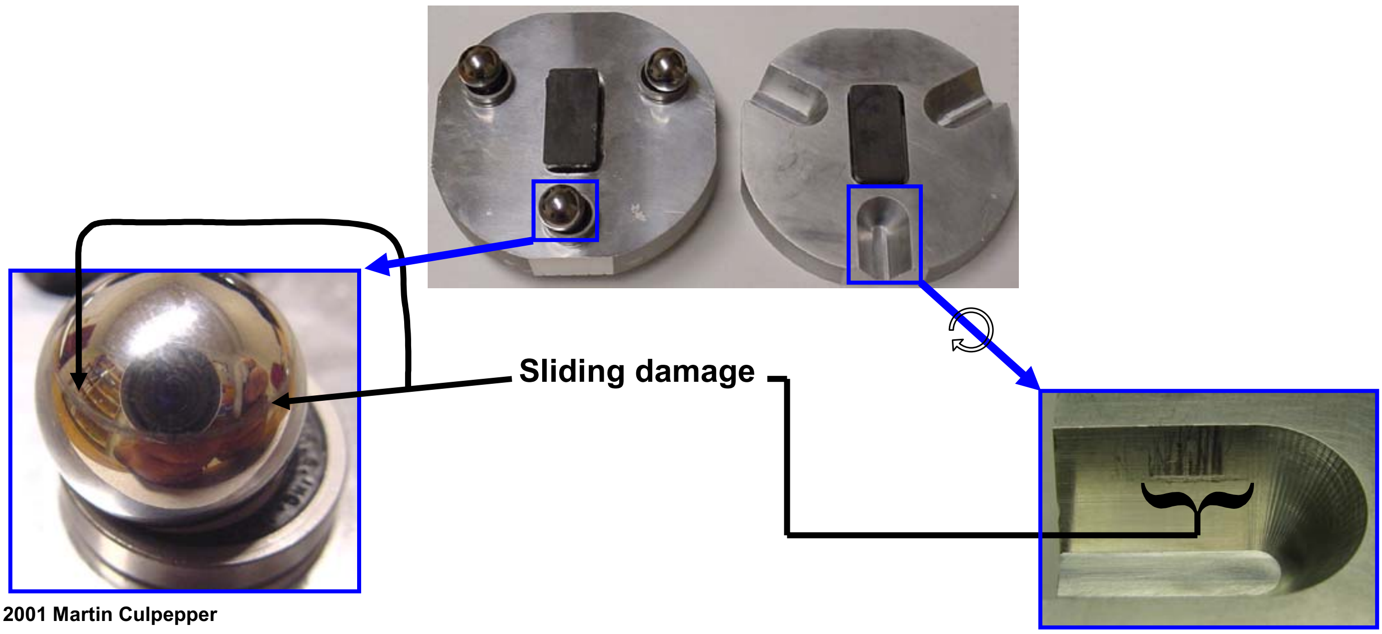
Mechanical interface wear management

Wear and particle generation are unknowns. Must investigate:

- **Coatings [minimize friction, maximize surface energy]**
- **Surface geometry, minimize contact forces**
- **Alternate means of force/constraint generation**



At present, must uncouple before actuation



PARTIAL CONSTRAINT

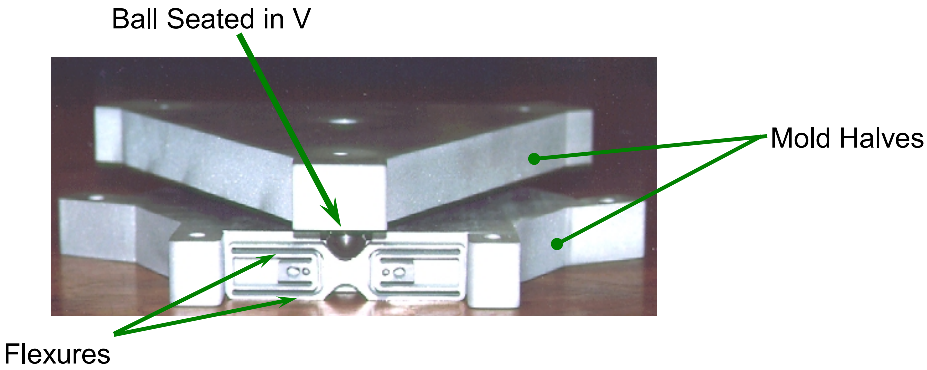
Motivated by coupling envy……

Adding and taking away constraints

It may be helpful to add/remove DOF in coupling applications

For instance, KCs can not form seals

- **We can add compliance to KCs to allow this to happen**
- **This is equivalent to adding a Degree of Freedom**



Care must be taken to make sure

- **compliant direction is not in a sensitive direction**
- **Parasitic errors in sensitive directions are acceptable**

Stiffness ratio

Actuation loads should be:

- Applied through center of stiffness
- In compliant direction

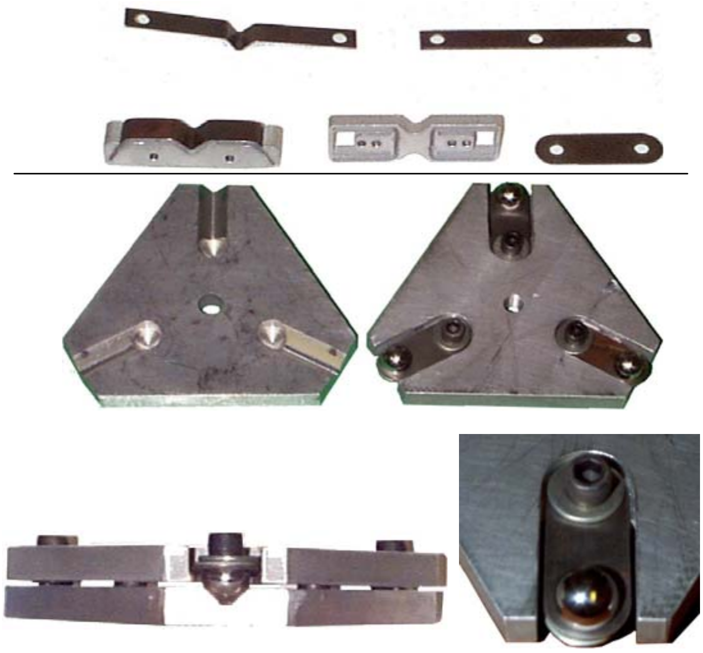
Error loads are often proportional to applied loads

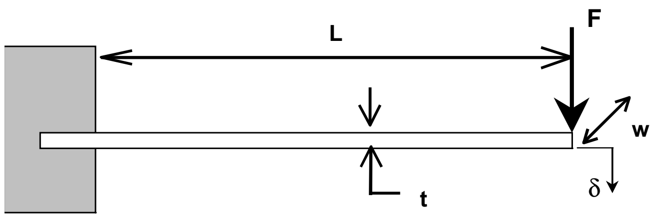
- Example: Bolt head friction
- $T_B \sim F_B R_B \mu$
- Design for $k_{sensitive} >> k_{non\text{-}sensitive}$

Practical metric is stiffness ratio:

$$\frac{k_{sensitive}}{k_{non\text{-}sensitive}} >> 1$$

Stamped compliant kinematic couplings





Characteristics

Stroke ≤ 0.25 inches

Repeatability 5 -10 microns

Ball movement in non-sens. direction

Applications/Processes

1. Assembly
2. Casting

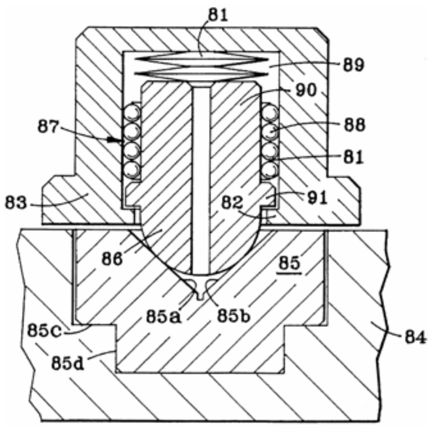
Design Issues (flexure)

1. $K_r \sim \frac{w^2}{t^2}$
2. Tolerances affect K_r

Cost

$ 10 - 200

Integral spring compliant kinematic couplings



Characteristics

1. Repeatability (2.5 micron)
2. Stroke ~ 0.5 inches

Applications/Processes

1. Assembly
2. Casting
3. Fixtures

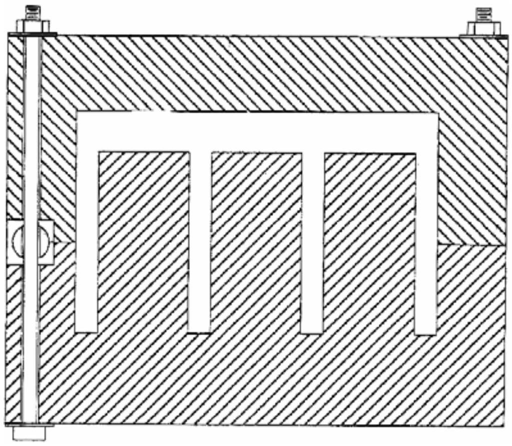
Design Issues (flexures)

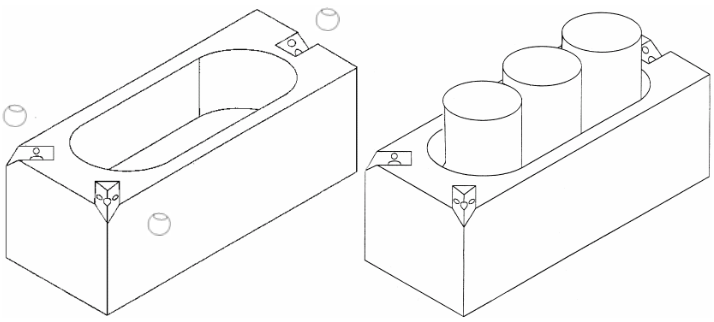
1. $K_r = \dfrac{K_{guide}}{K_{spring}}$
2. Press fit tolerances

Cost

$ 2000

Plastic compliant kinematic couplings





Characteristics

1. 180 microns
2. ~ 0.125 inches
3. 1 Time Use

Applications/Processes

1. Sand Casting

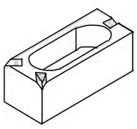
Design Issues

1. Loose Sand
2. K_r application specific

Cost

1. Modify Pattern
2. Purchase Balls
3. Tie Rods

Experimental results

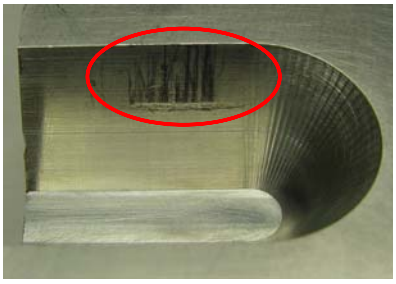
		Coupling Type	Prototype	Average Repeatability				
				Radial	Standard Deviation	Angular	Standard Deviation	Manufacturing Cost
				μinch (microns)	μinch (microns)	μ radians	μ radians	$
Stamped	1.		Water Jet (mate-ball)	300 (7.6)	100 (2.5)	35	21	23
	2.		Stamped (mate-ball)	400 (10.2)	200 (5.1)	120	70	10
	3.		Stamped (mates-V)	200 (5.1)	100 (2.5)	52	30	10
	4.		Stamped (mates-V)	300 (7.6)	100 (2.5)	57	23	10
Springs	5.		Ultra Die Set Coupling	100 (2.5)	N/A	N/A	N/A	2000
Plastic	6.		Grooves made in mold sand	19,000 (480)	7000 (180)	3900	1300	(cost to modify pattern)

USING CONSTRAINTS IN MECHANISM DESIGN

Alternatives to motion with physical contact

Problems you can not avoid with contact:

- Surface topology (finish)
- Wear and Fretting
- Friction
- Limited resolution, at best on order of microns….



Wear on Groove

Next generation applications require nanometer level fixtures, i.e.:

- Fiber optics
- Photolithography

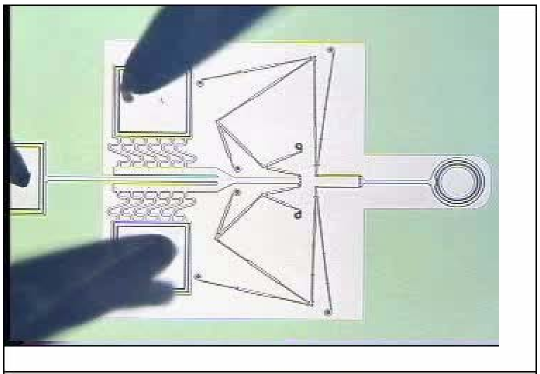
Compliant mechanisms:

- Mechanical reduction to interface with larger scale actuators
- Motion through strain
- Small and moderately sized motions in comparison to mechanism size
- Can be made to emulate machines

Compliant mechanism examples

University of Michigan: Prof. Sridhar Kota

- http://www.engin.umich.edu/labs/csdl/index.htm

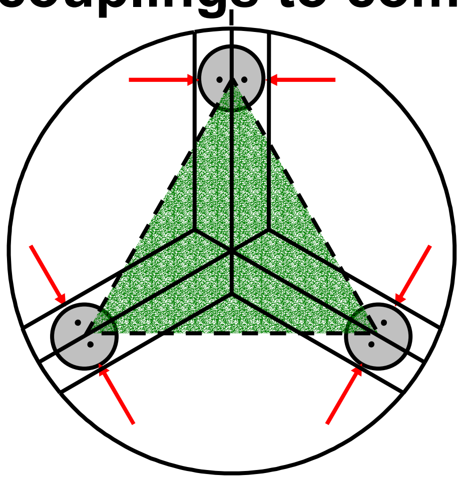


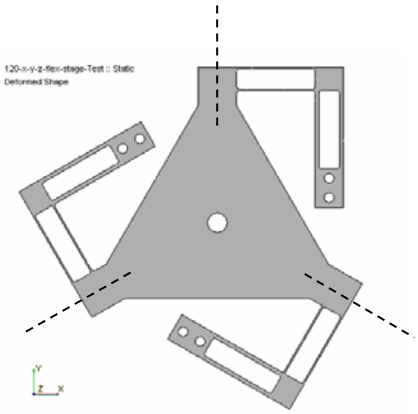
thermal1[1].mpeg

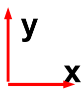
Why compliant mechanisms in precision fixtures

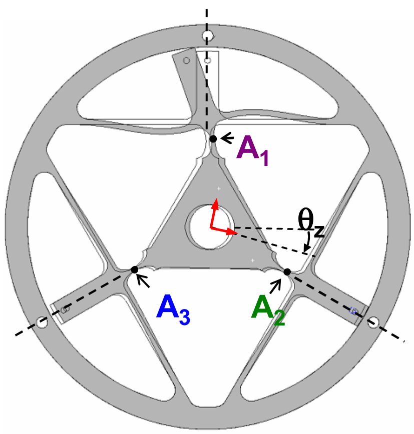
- Repeatable/low hysteresis
- No assembly
- No contact

From kinematic couplings to compliant stages

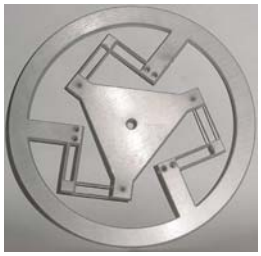








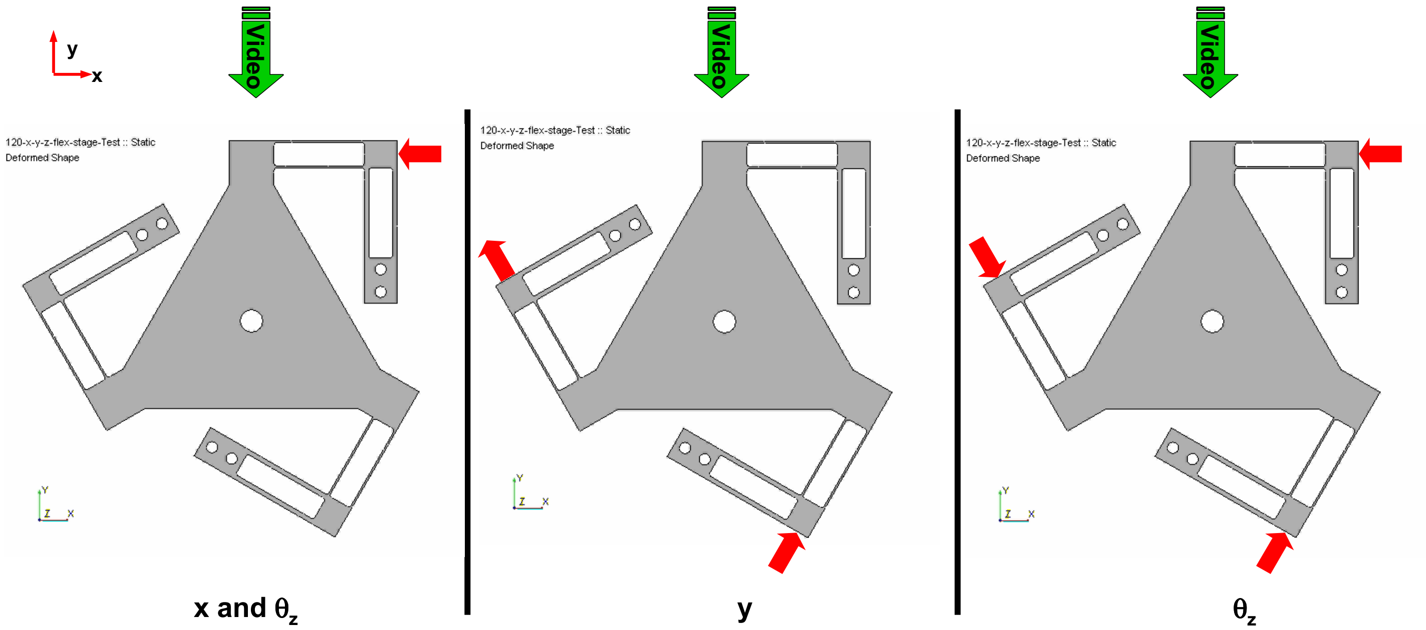
Constraint based compliant mechanisms



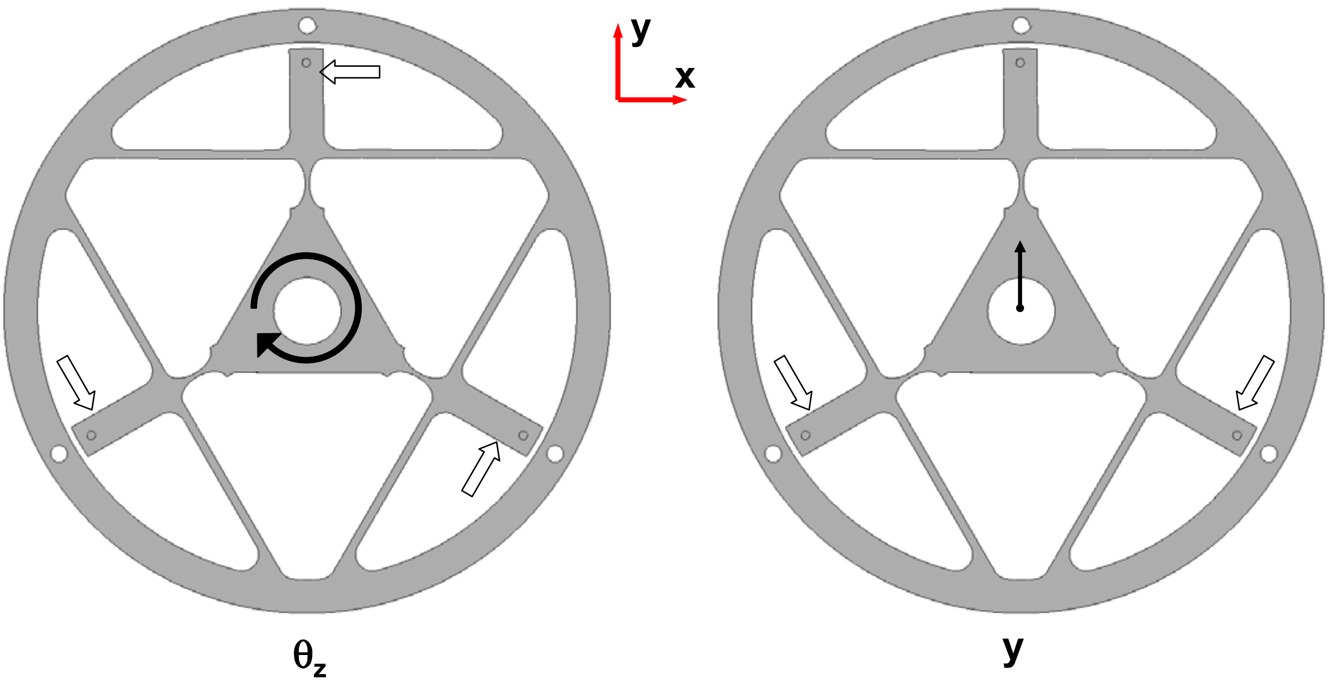
High volume, low cost, multi-degree of freedom alignment

Example 3 DOF flexure system:

Target applications: Opto-electronic packaging/alignment



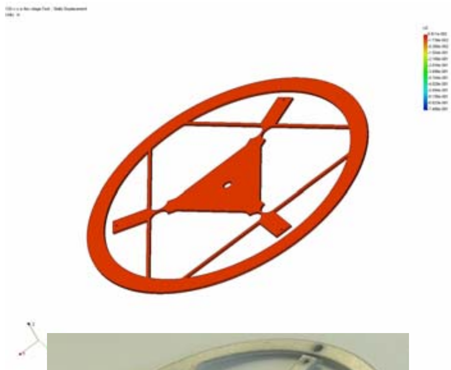
Constraint based compliant mechanisms cont.



Constraint based compliant mechanisms cont.

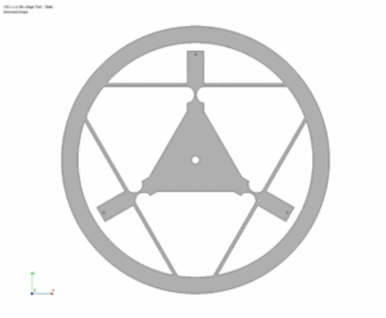
Example 6 DOF alignment capability

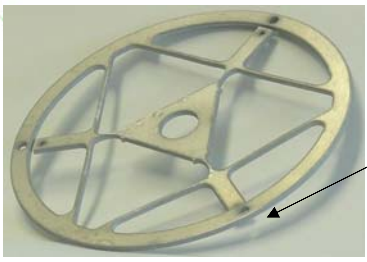
Target app.: Micro and meso scale positioning (I.e. opto-electronics)



Video

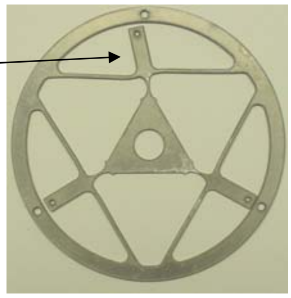
Video





Permanent set due to plastic deformation

Have option to rely on elastic deformation



z , θ_x , θ_y

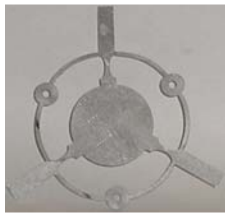
x and θ_z

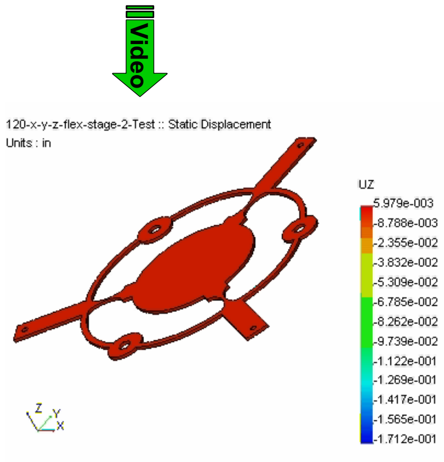
Patent Pending

Constraint based compliant mechanisms cont.

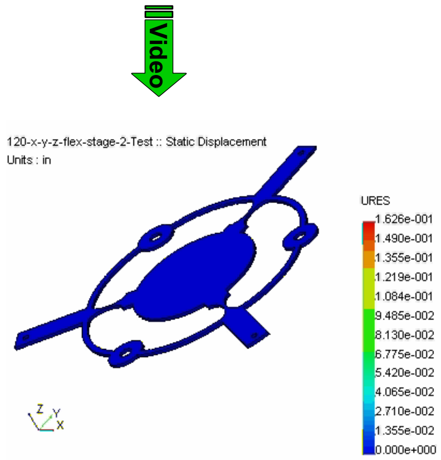
Example 6 DOF alignment capability

Target app.: Micro/meso scale positioning (I.e. opto-electronics)

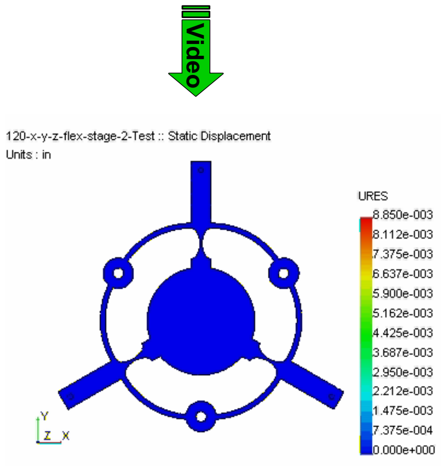




z , θ_x , θ_y



z



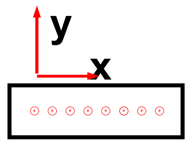
θ_z

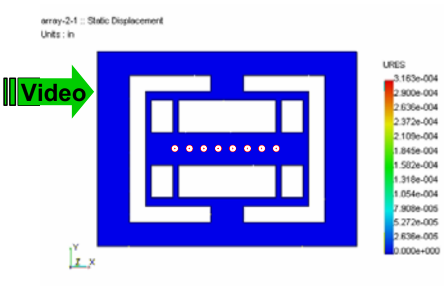
Patent Pending

Constraint based compliant mechanisms cont.

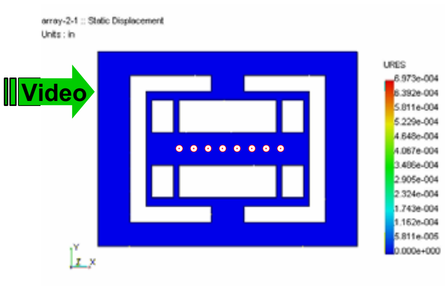
3 DOF active alignment [x, y, z]   &   2 DOF passive alignment [z, θ_y]

Good fit for wire-EDM (stacked sheets) ~ order of $ 1 - 10

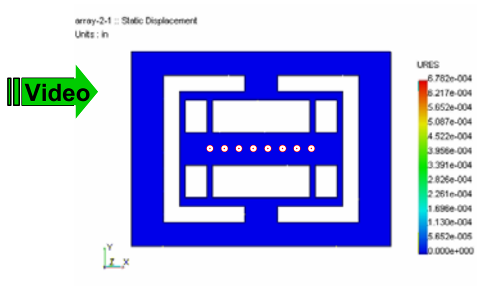




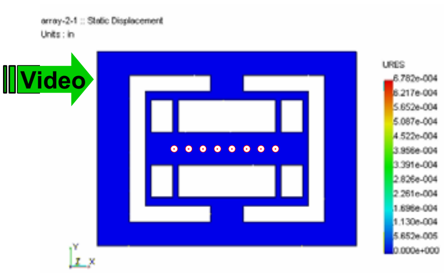
Δx



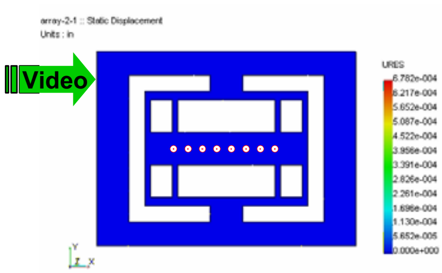
Δy



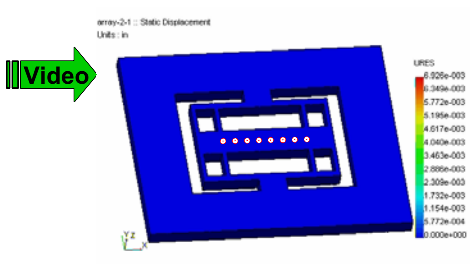
$\Delta \theta_z$



Δx & $\Delta \theta_z$



Δx & Δy

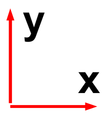


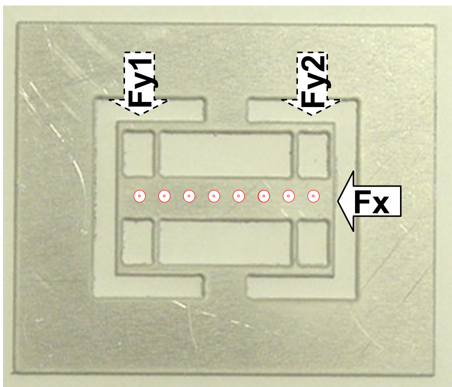
$\Delta \theta_y$

Constraint based compliant mechanisms cont.

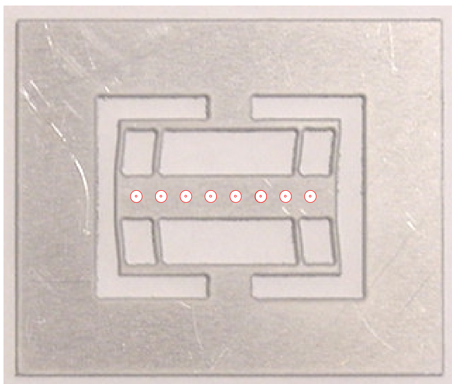
Plastic deformation can be utilized for position keeping

Device should be potted in place to avoid stress relief





Initial position



Plastically flexed

Constraint based compliant mechanisms cont.

Static or flexible kinematic coupling

Components biased toward each other

Flexure takes up bias, provides mating force in z direction