2.75
CONSTRAINT
LECTURE

Design of Constraints in Precision Systems

Background

- History
- Reasons
- Requirements
- Problems

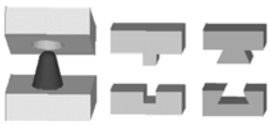
Classes of Constraint

- Kinematic
- Quasi-Kinematic
- Variable Geometry
- Partial/Compliance

Hardware/discussion time

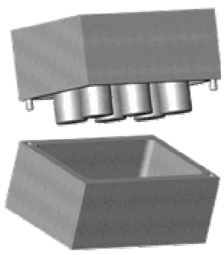
Elastic Averaging will be done next lecture

Common Coupling Methods



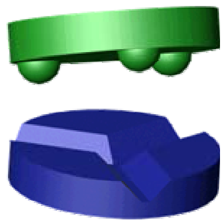
Elastic Averaging

Non-Deterministic



Pinned Joints

No Unique Position

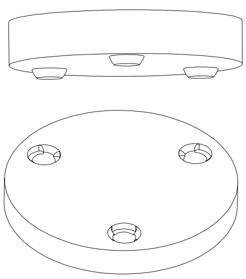


Kinematic Couplings

Kinematic Constraint

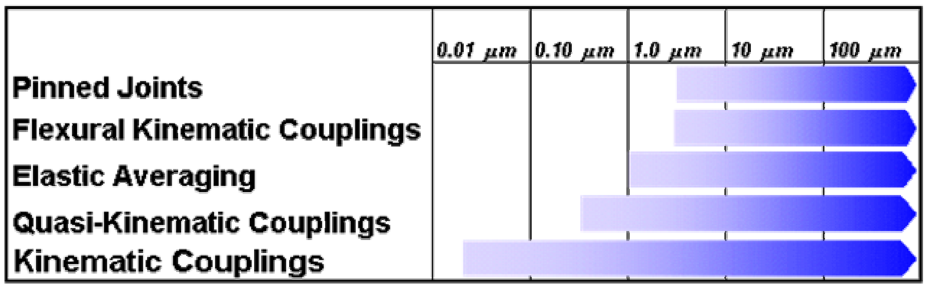
Flexural Kin. Couplings

Kinematic Constraint



Quasi-Kinematic Couplings

Near Kinematic Constraint



Perspective: What the coupling designer faces…

APPLICATION	SYSTEM SIZE	REQ'D PRECISION
Fiber Optics	Meso	Nano
Optical Resonators	Meso	Nano
Large array telescopes	60 ft diam.	Angstrom
Automotive	3 ft	1 micron

Problems due to strain affects

⊙ Thermal Affects	Air, hands, sunlight
⊙ Gravity	Sagging
⊙ Stress Relief	Time variable assemblies
⊙ Loads	Stiffness

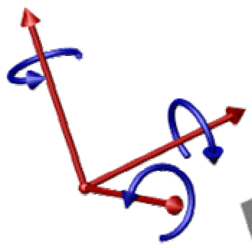
Problems due to sub-optimal designers

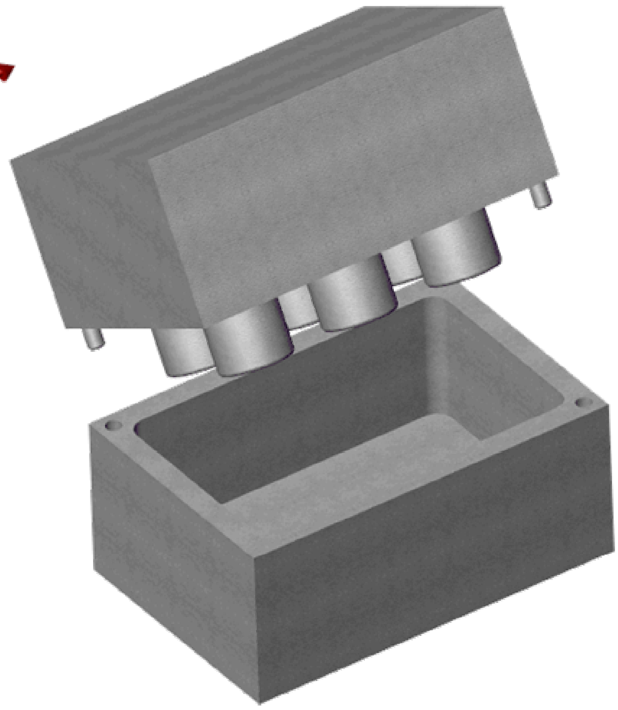
- Competing cost vs performance
- Automotive (no temperature control, large parts, resistance to change)

General Service Requirements & Applications

Ideal couplings:

- Inexpensive
- Accurate & Repeatable
- High Stiffness
- Handle Load Capacity
- Sealing Interfaces
- Well Damping





Example Applications:

- Grinding
- Optic Mounts
- Robotics
- Automotive

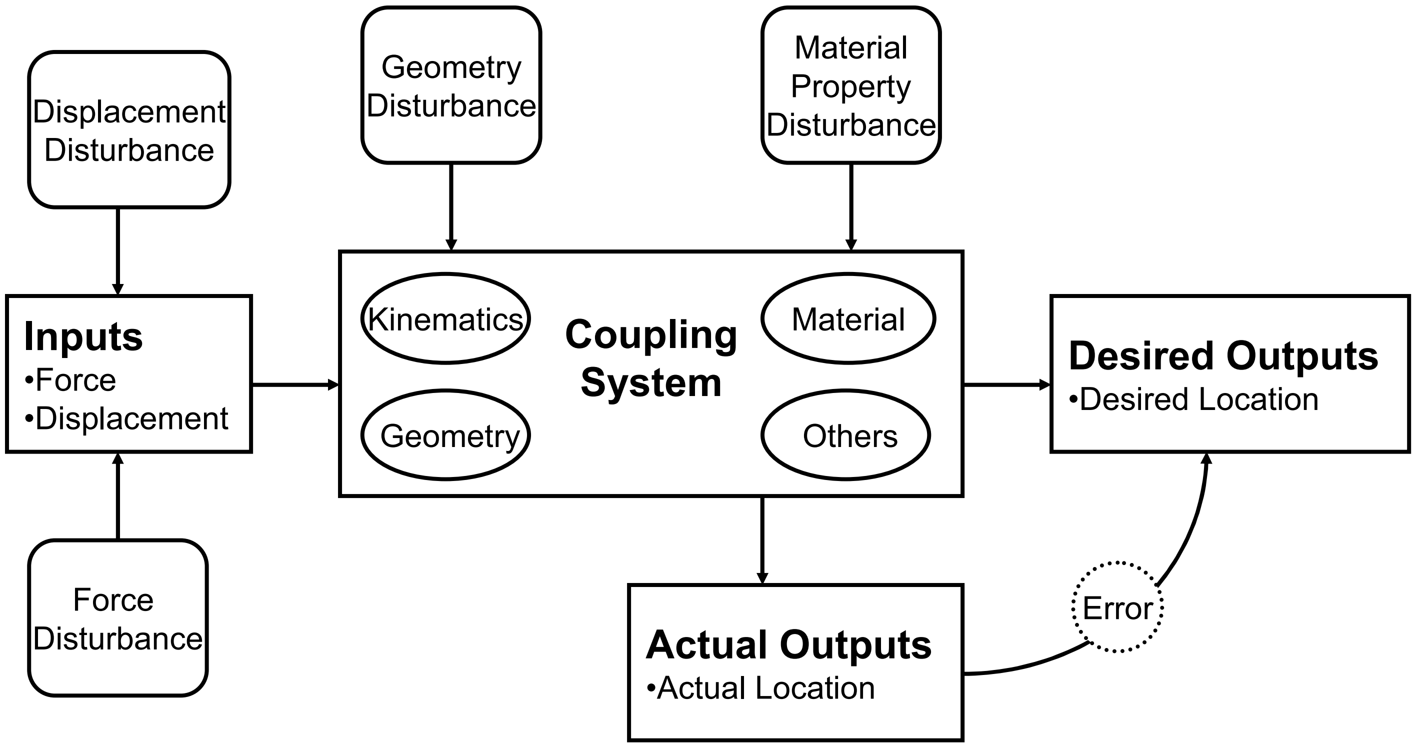
Sensitivity

- What are the sensitive directions?!?!?!?!?

Couplings Are Designed as Systems

You must know what is going on (loads, environment, thermal)!

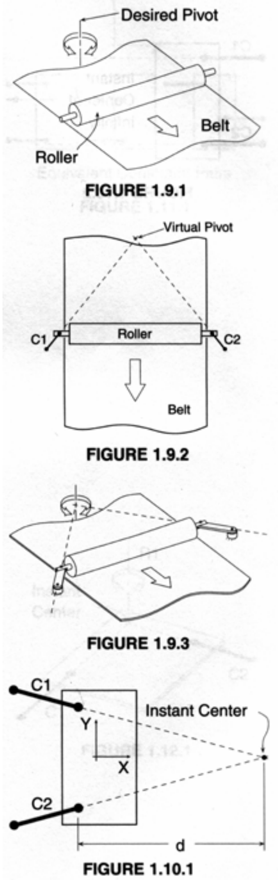
Shoot for determinism or it will "suck to be you"



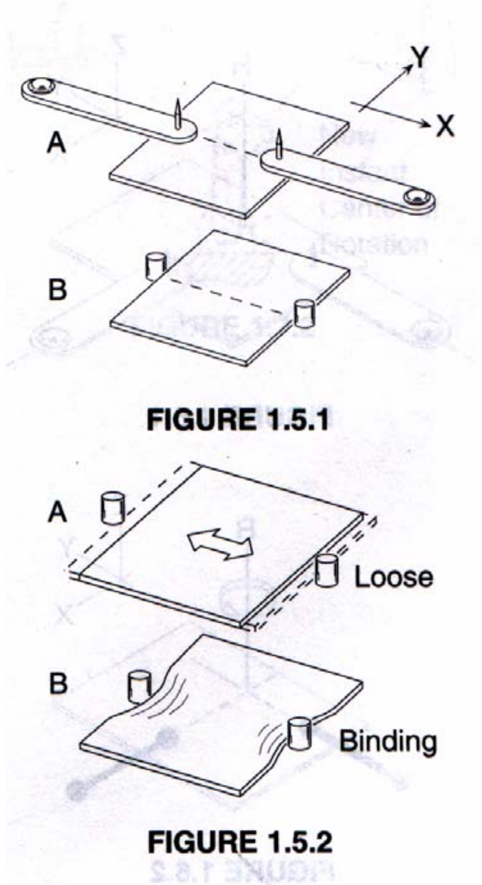
KINEMATIC COUPLINGS

The good, the bad, the ugly….

Defining Constraint



Clever use of constraint



Penalties for over constraint

Exact Constraint (Kinematic) Design

Exact Constraint: Number of constraint points = DOF to be constrained

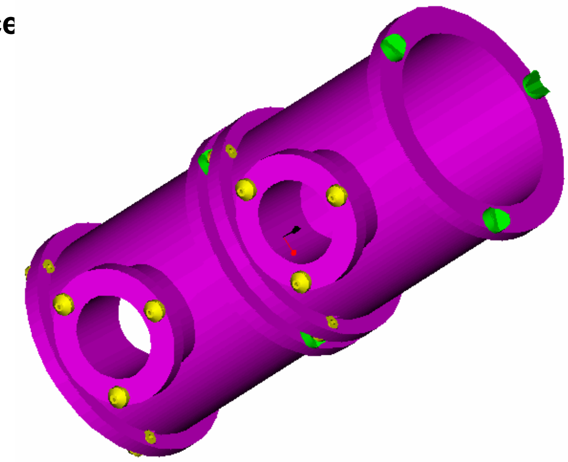
These constraints must be independent!!!

Assuming couplings have rigid bodies, equations can be written to describe movements

Design is deterministic, saves design and development $

KCs provide repeatability on the order of parts' surface finish

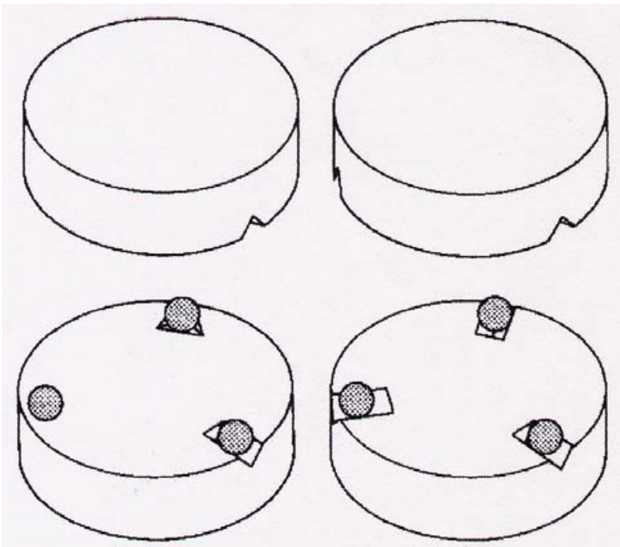
- ¼ micron repeatability is common
- Managing contact stresses are the key to succe



Making Life Easier

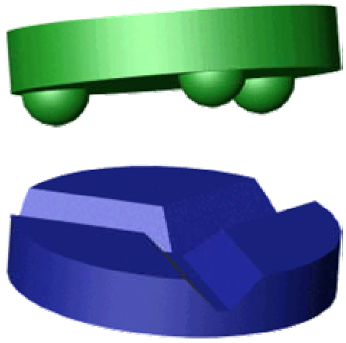
“Kinematic Design”, “Exact Constraint Design”…..the issues are:

- KNOW what is happening in the system
- Manage forces and deflections
- Minimize stored energy in the coupling
- Know when “Kinematic Design” should be used
- Know when “Elastic Averaging” should be used (next week)



Kelvin Clamp　　Boyes Clamp

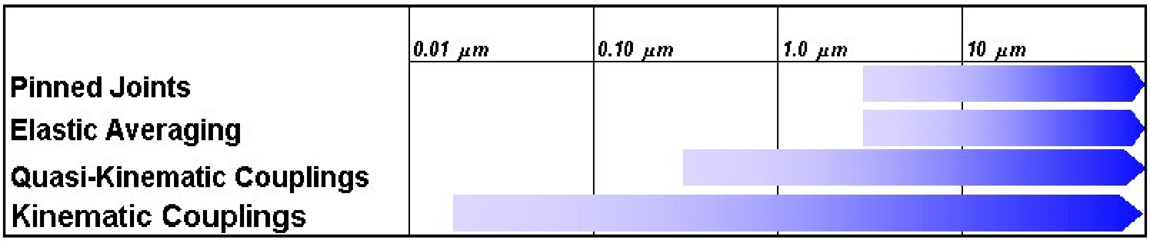
Kinematic couplings



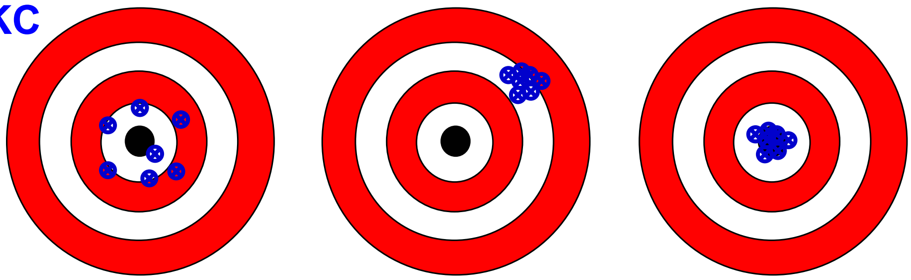
Kinematic Couplings:

- Deterministic Coupling
- # POC = # DOF
- Do Not Allow Sealing Contact
- Excellent Repeatability

Performance



Power of the KC

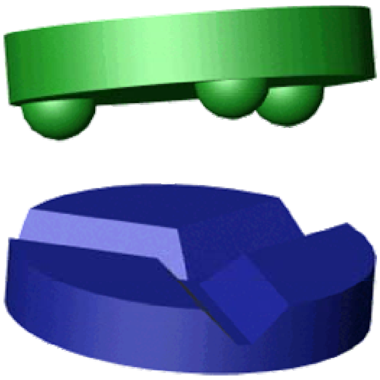


Accuracy

Repeatability

Accuracy & Repeatability

Modeling Kinematic Coupling Error Motions



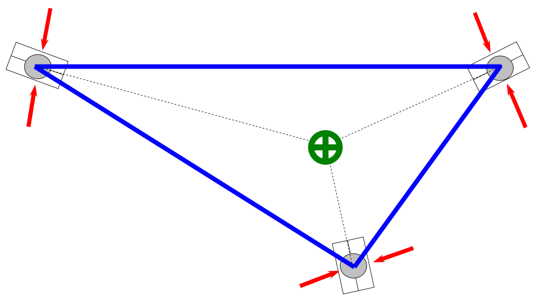
Geometry Material n_i → **Applied Loads [Fp & Mp]** → **Interface Forces [Fi]** → **Deflections δ -> Δr** → **Relative Error**

6 Unknown Forces & 6 Equilibrium Equations

$$\Sigma F_i = F_P \qquad \Sigma M_i = M_P$$

Hertzian Point Contact for Local Displacements

$$\delta_i = f(E_B, E_G, \nu_B, \nu_G, R_B, R_G)$$



Kinematic Coupling Groove

Mating Spherical Element

Contact Force

Coupling Centroid

Angle Bisectors

Coupling Triangle

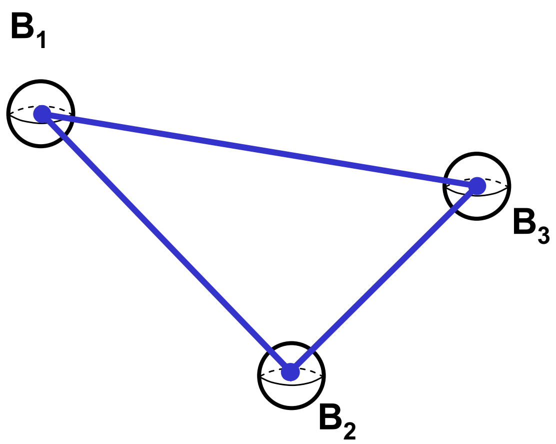
KC Error Motion Analysis

Need δ_x, δ_y, δ_z, ε_x, ε_y , ε_z to predict effect of non-repeatability

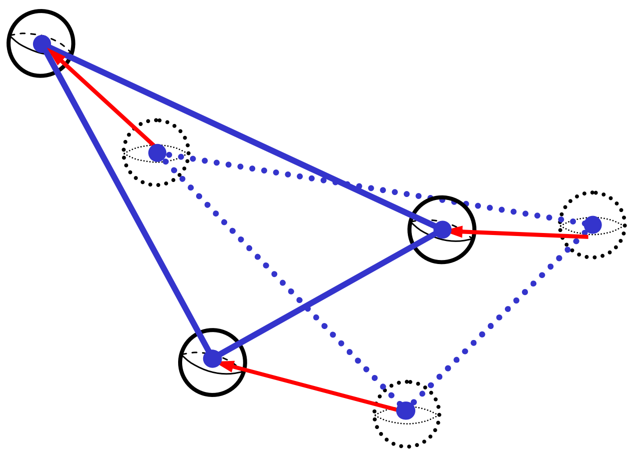
Hertz deflections -> displacements of ball centers

Three ball centers form a plane

Analyze relative position of "before" and "after" planes for error motions



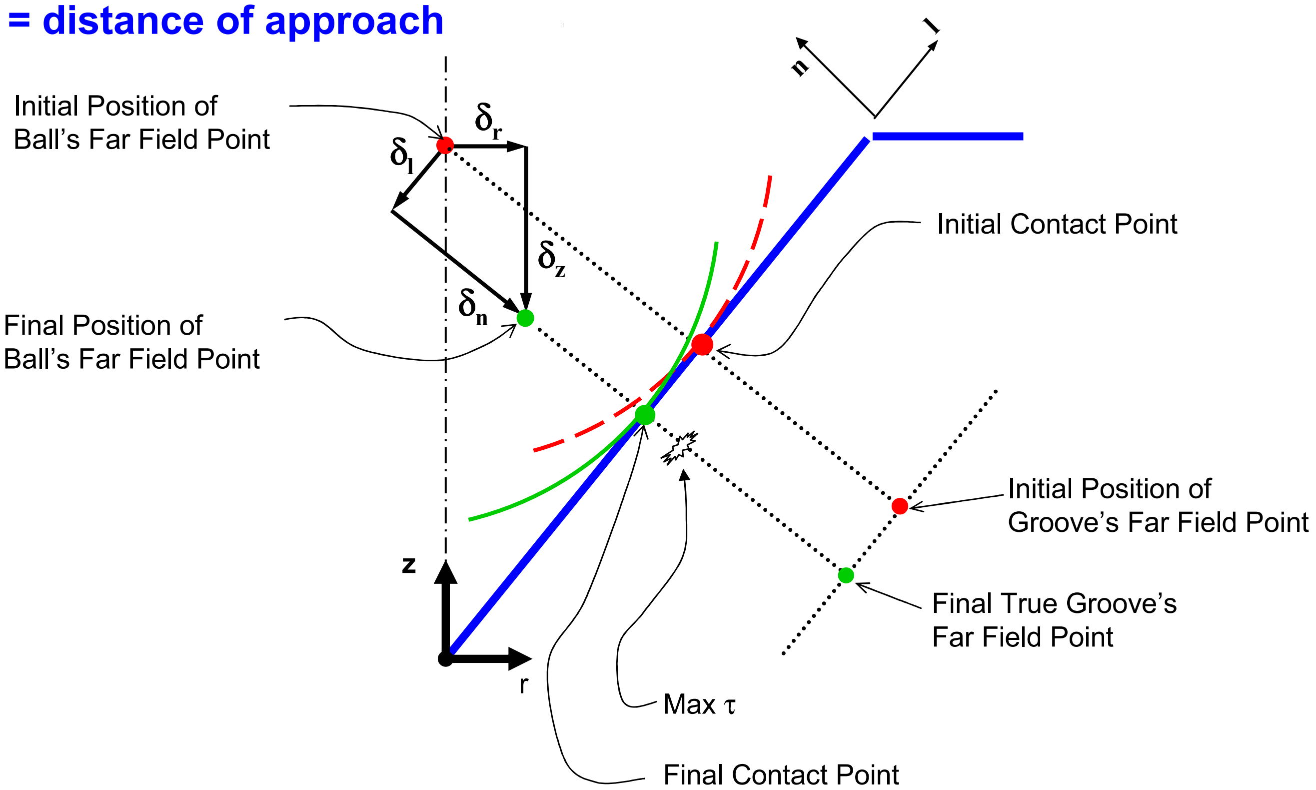
Original Positions


Final Positions

Kinematic Couplings and Distance of Approach

How do we characterize motions of the ball centers?

δ_n = distance of approach



Max shear stress occurs below surface, in the member with larges R

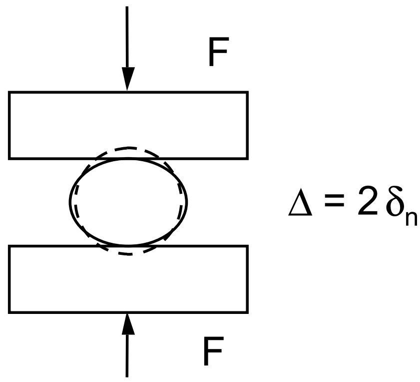
Contact Mechanics – Hertz Contact

Heinrich Hertz – 1st analytic solution for "near" point contact

KC contacts are modeled as Hertz Contacts

Enables us to determine stress and distance of approach, δ_n

Radii →	Ronemaj	**1.00E+06**
	Ronemin	**0.06250**
	Rtwomaj	**0.25000**
	Rtwomin	**-0.06500**
Load →	Applied load F	**13**
	Phi (degrees)	**0**
	Max contact stress	**10,000**
Modulus →	Elastic modulus Eone	**3.00E+07**
	Elastic modulus Etwo	**4.40E+05**
ν Ratio →	Poisson's ratio vone	**0.3**
	Poisson's ratio vtwo	**0.3**
	Equivalent modulus Ee	4.77E+05
	Equivalent radius Re	0.2167
Stress →	Contact pressure	12,162
	Stress ratio (must be less than 1)	1.22
	Deflection at the one contact interface	
Deflection →	Deflection (μunits)	829



Key Hertz Physical Relations

Equivalent radius and modulus:

$$R_e = \frac{1}{\frac{1}{R_{1major}} + \frac{1}{R_{1minor}} + \frac{1}{R_{2major}} + \frac{1}{R_{2minor}}} \qquad E_e = \frac{1}{\frac{1-\eta_1^2}{E_1} + \frac{1-\eta_2^2}{E_2}}$$

$\cos(\theta)$ function (ϕ is the angle between the planes of principal curvature of the two bodies)

$$\cos\theta = R_e\left[\left(\frac{1}{R_{1major}} - \frac{1}{R_{1minor}}\right)^2 + \left(\frac{1}{R_{2major}} - \frac{1}{R_{2minor}}\right)^2 + 2\left(\frac{1}{R_{1major}} - \frac{1}{R_{1minor}}\right)\left(\frac{1}{R_{2major}} - \frac{1}{R_{2minor}}\right)\cos 2\phi\right]^{1/2}$$

Solution to elliptic integrals estimated with curve fits

$$\alpha = 1.939e^{-5.26\theta} + 1.78e^{-1.09\theta} + 0.723/\theta + 0.221$$

$$\beta = 35.228e^{-0.98\theta} - 32.424e^{-1.0475\theta} + 1.486\theta - 2.634$$

$$\lambda = -0.214e^{-4.95\theta} - 0.179\theta^2 + 0.555\theta + 0.319$$

Contact Pressure	Distance of Approach	Major Contact Axis	Minor Contact Axis
$q = \frac{3F}{2\pi cd} \approx 1.5\sigma_{\text{tensile for metals}}$	$\delta = \lambda\left(\frac{2F^2}{3R_eE_e^2}\right)^{1/3}$	$c = \alpha\left(\frac{3FR_e}{2E_e}\right)^{1/3}$	$\beta\left(\frac{3FR_e}{2E_e}\right)^{1/3}$

KEY Hertz Relations

Contact Pressure is proportional to:

- Force to the 1/3rd power
- Radius to the –2/3rd power
- Modulus to the 2/3rd power

Distance of approach is proportional to:

- Force to the 2/3rd power
- Radius to the –1/3rd power
- Modulus to the –2/3rd power

Contact ellipse diameter is proportional to:

- Force to the 1/3rd power
- Radius to the 1/3rd power
- Modulus to the –1/3rd power

DO NOT ALLOW THE CONTACT ELLIPSE TO BE WITHIN ONE DIAMETER OF THE EDGE OF A SURFACE!

Calculating Errors Motions in Kinematic Couplings

Motion of ball centers -> Centroid motion in 6 DOF -> Δx, Δy, Δz at X, Y, Z

- **Coupling Centroid Translation Errors**

$$\delta_{\zeta c} = \left(\frac{\delta_{1\zeta}}{L_{1c}} + \frac{\delta_{2\zeta}}{L_{2c}} + \frac{\delta_{3\zeta}}{L_{3c}}\right)\cdot\frac{L_{1c} + L_{2c} + L_{3c}}{3}$$

- **Rotations**

$$\varepsilon_x = \frac{\delta_{z1}}{L_{1,23}}\cdot\cos(\theta_{23}) + \frac{\delta_{z2}}{L_{2,31}}\cdot\cos(\theta_{31}) + \frac{\delta_{z3}}{L_{3,12}}\cdot\cos(\theta_{12})$$

$$\varepsilon_y = \frac{\delta_{z1}}{L_{1,23}}\cdot\sin(\theta_{23}) + \frac{\delta_{z2}}{L_{2,31}}\cdot\sin(\theta_{31}) + \frac{\delta_{z3}}{L_{3,12}}\cdot\sin(\theta_{12})$$

$$\varepsilon_{z1} = \frac{\sqrt{\left(\alpha_{B1}\cdot\delta_1 + \alpha_{B2}\cdot\delta_2\right)^2 + \left(\beta_{B1}\cdot\delta_1 + \beta_{B2}\cdot\delta_2\right)^2}}{\sqrt{\left(x_1 - x_c\right)^2 + \left(y_1 - y_c\right)^2}}\cdot\mathrm{SIGN}\left(\alpha_{B1}\cdot\delta_1 - \alpha_{B2}\cdot\delta_2\right) \longrightarrow \varepsilon_z = \frac{\varepsilon_{z1} + \varepsilon_{z2} + \varepsilon_{z3}}{3}$$

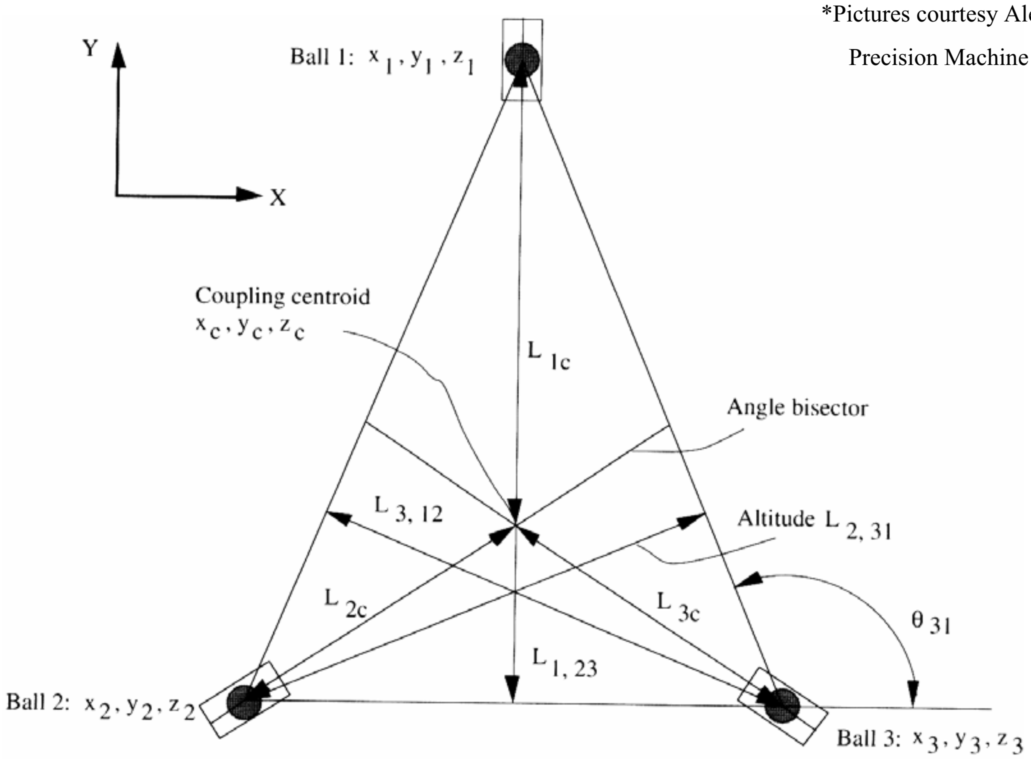
- **Error At X, Y, Z (includes translation and sine errors)**

$$\begin{pmatrix} \Delta_x \\ \Delta_y \\ \Delta_z \\ 1 \end{pmatrix} = \begin{pmatrix} 1 & -\varepsilon_z & \varepsilon_y & \delta_x \\ \varepsilon_z & 1 & -\varepsilon_x & \delta_y \\ -\varepsilon_y & \varepsilon_x & 1 & \delta_z \\ 0 & 0 & 0 & 1 \end{pmatrix}\cdot\begin{pmatrix} X - x_c \\ Y - y_c \\ Z - z_c \\ 1 \end{pmatrix} - \begin{pmatrix} X - x_c \\ Y - y_c \\ Z - z_c \\ 1 \end{pmatrix}$$

Kinematic Coupling Centroid Displacement

*Pictures courtesy Alex Slocum

Precision Machine Design



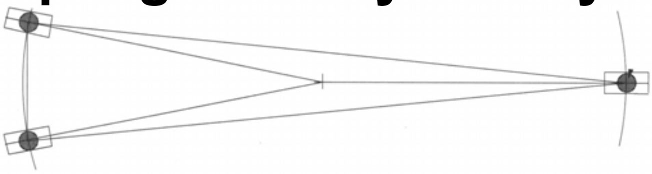
Centroid Displacement: $\delta_{\xi c} = \delta_{1\xi} \dfrac{L_{1c}}{L_{1,23}} + \delta_{2\xi} \dfrac{L_{2c}}{L_{2,31}} + \delta_{3\xi} \dfrac{L_{3c}}{L_{3,12}}$

General Design Guidelines

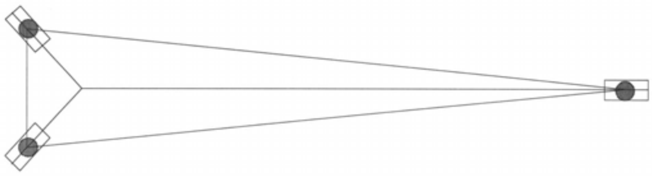
1. Location of the coupling plane is important to avoid sine errors
2. For good stability, normals to planes containing contact for vectors should bisect angles of coupling triangle
3. Coupling triangle centroid lies at center circle that coincides with the three ball centers
4. Coupling centroid is at intersection of angle bisectors
5. These are only coincident for equilateral triangles
6. Mounting the balls at different radii makes crash-proof
7. Non-symmetric grooves make coupling idiot-proof

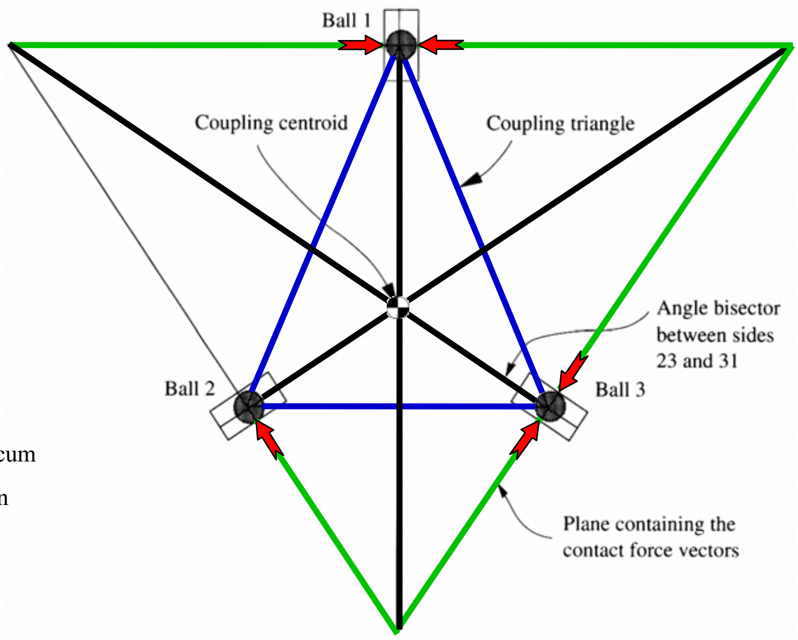
Kinematic Coupling Stability Theory

Poor Design



Good Design

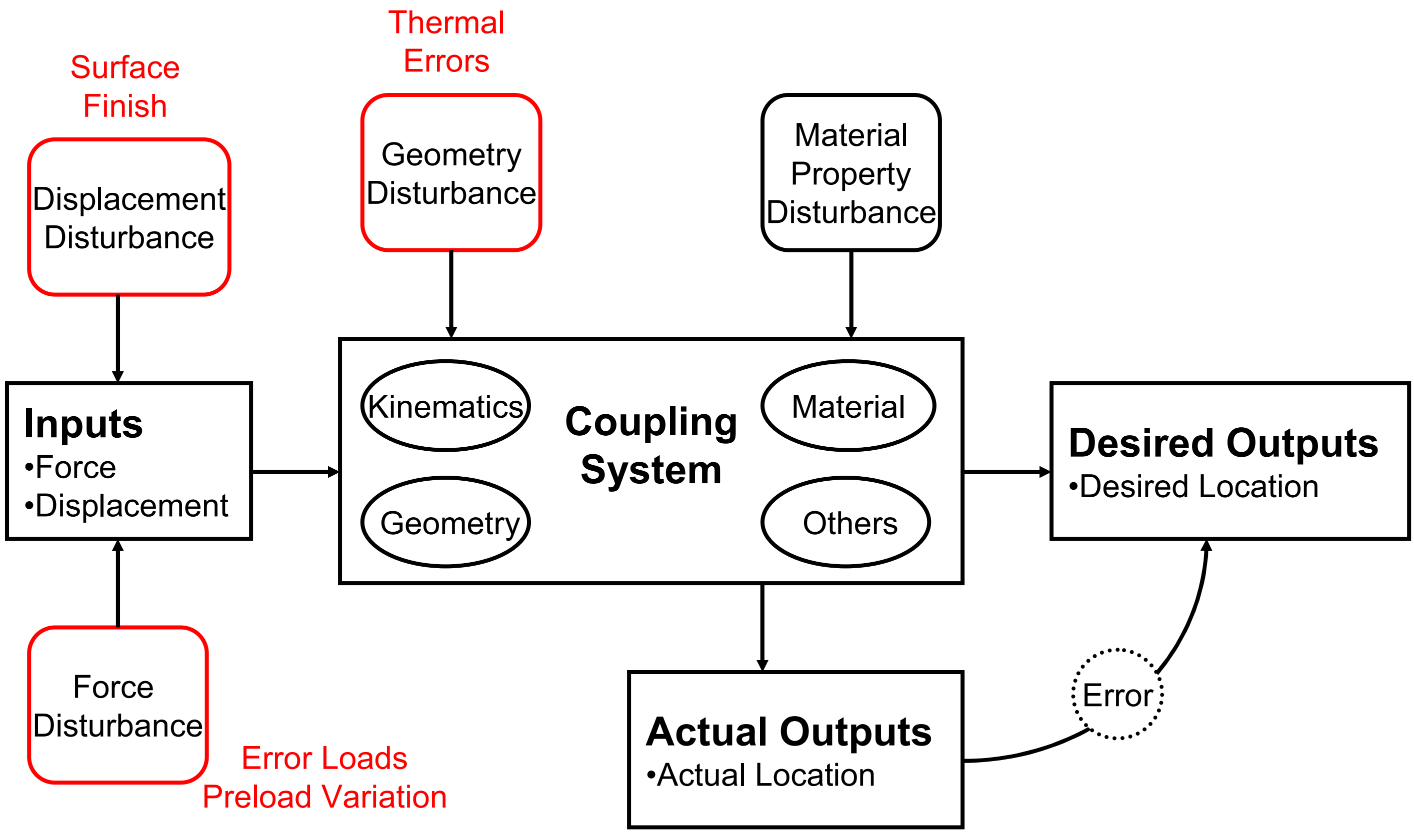




*Pictures courtesy Alex Slocum

Precision Machine Design

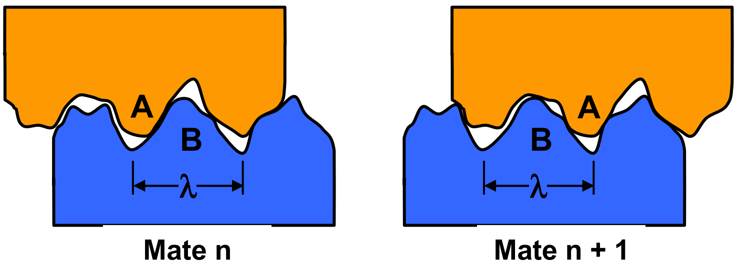
Sources of Errors in KCs



Problems With Physical Contact (and solutions)

Surface topology (finish):

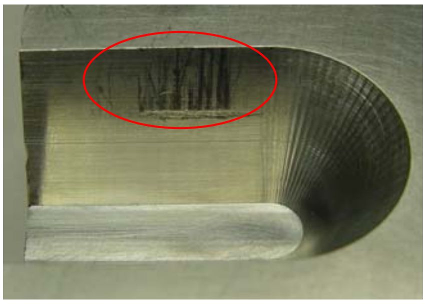
- 50 cycle repeatability ~ 1/3 μm Ra
- Friction depends on surface finish!
- Finish should be a design spec
- Surface may be brinelled if possible



Wear and Fretting:

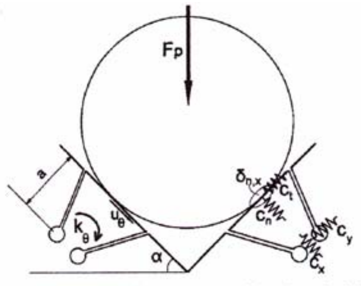
- High stress + sliding = wear
- Metallic surfaces = fretting
- Use ceramics if possible (low μ and high strength)
- Dissimilar metals avoids "snowballing"

Wear on Groove



Friction:

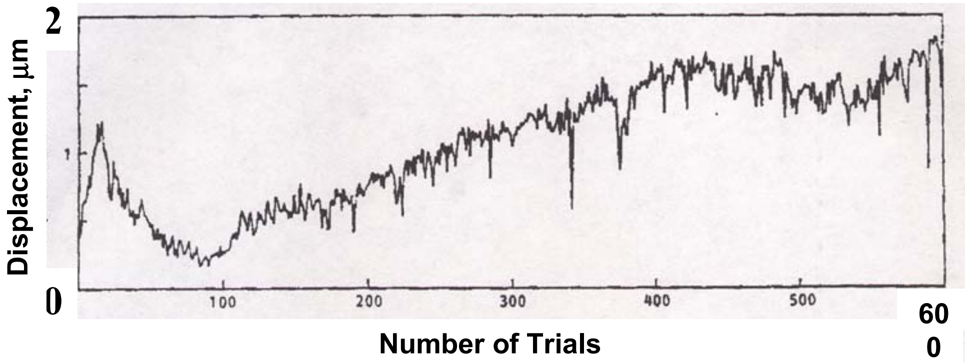
- Friction = Hysteresis, stored energy, overconstraint
- Flexures can help (see right)
- Lubrication (high pressure grease) helps
  - Beware settling time and particles
- Tapping can help if you have the "magic touch"



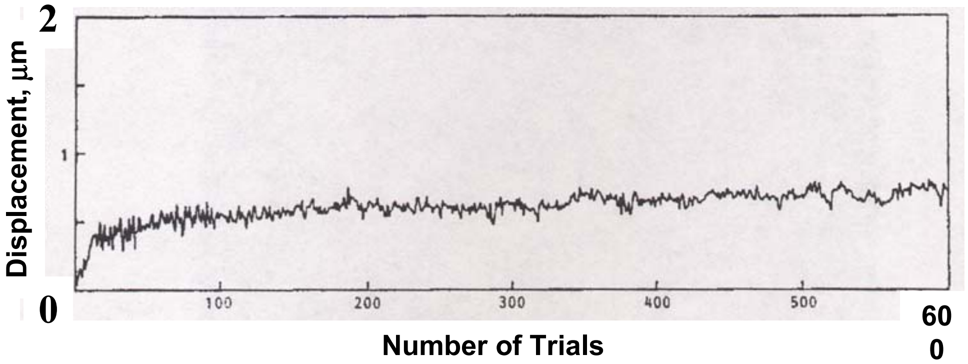
Ball in V-Groove with Elastic Hinges

Experimental Results –Repeatability & Lubrication

Radial Repeatability (Unlubricated)



Radial Repeatability (Lubricated)



Practical Design of Kinematic Couplings

Design

- Specify surface finish or brinell on contacting surfaces
- Normal to contact forces bisect angles of coupling triangle!!!

Manufacturing & Performance

- Repeatability = f (friction, surface, error loads, preload variation, stiffness)
- Accuracy = f (assembly) unless using and ARKC

Precision Balls (ubiquitous, easy to buy)

- Baltec sells hardened, polished kinematic coupling balls or…..

Grooves (more difficult to make than balls)

- May be integral or inserts. Inserts should be potted with thin layer of epoxy

Materials

- Ceramics = low friction, high stiffness, and small contact points
- If using metals, harden
- Use dissimilar materials for ball and groove

Preparation and Assembly

- Clean with oil mist
- Lubricate grooves if needed

Example: Servo-Controlled Kinematic Couplings

- Location & automatic leveling of precision electronic test equipment
- Teradyne has shipped over 500 systems

Example: Canoe-Ball Kinematic Interface Element

The "Canoe Ball" shape is the secret to a highly repeatable design

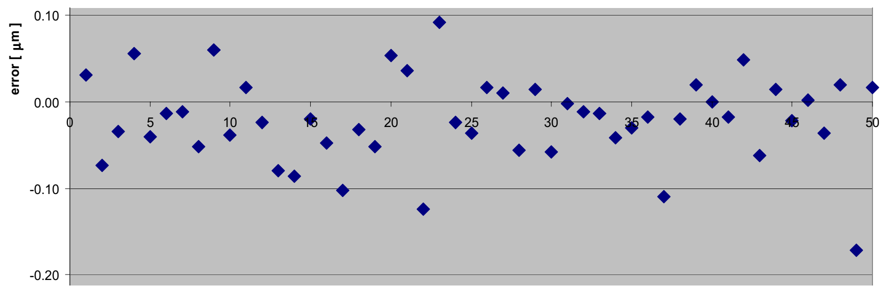
- **It acts like a ball 1 meter in diameter**
- **It has 100 times the stiffness and load capacity of a *normal* 1" ball**

Large, shallow Hertzian zone is *very (i.e. < 0.1 microns)* repeatable

Canoe Ball Repeatability Measurements

Test Setup

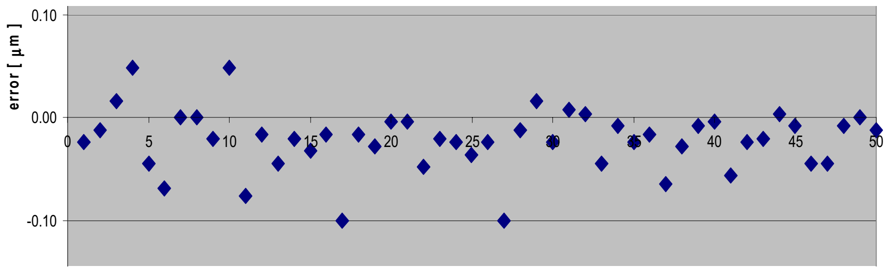
Coupling



0.1 µm

-0.2 µm

Meas. system



0.1 µm

-0.1 µm

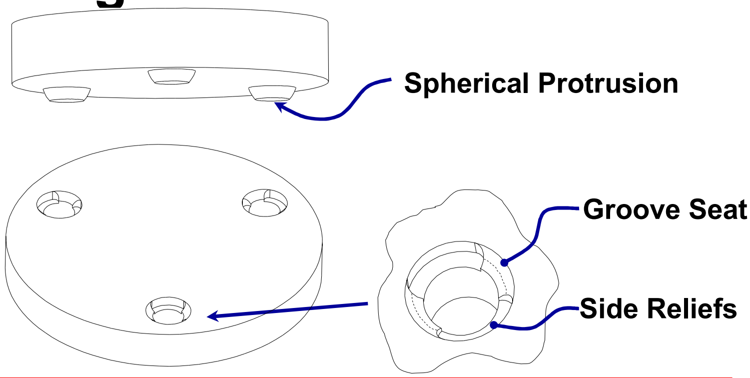
QKCs

Why do it the easy way when you can do it the lazy way?

Quasi-Kinematic (QKC) Alignment

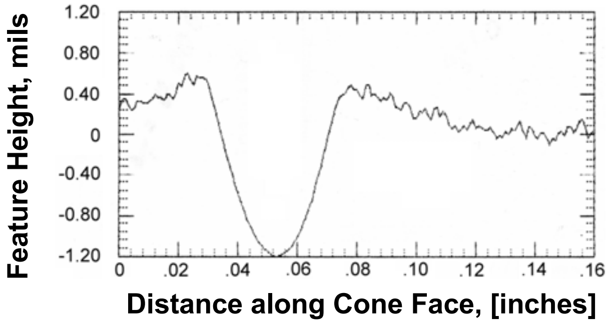
QKC characteristics:

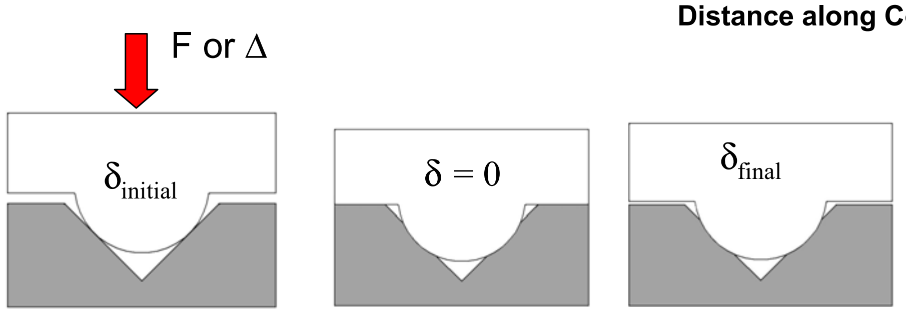
- Arc contact
- Submicron repeatability
- Stiff, sealing contact
- Less expensive than KCs
- Easier to make than KCs



QKC Function:

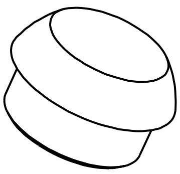
- Ball & groove comply
- Burnish surface irregularities
- Elastic recovery restores gap





Details of QKC Element Geometry

PAIRS OF QKC ELEMENTS



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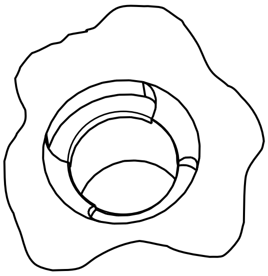

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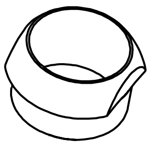

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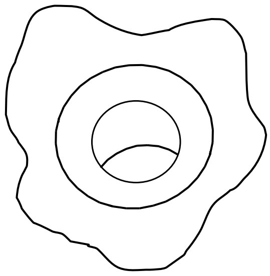
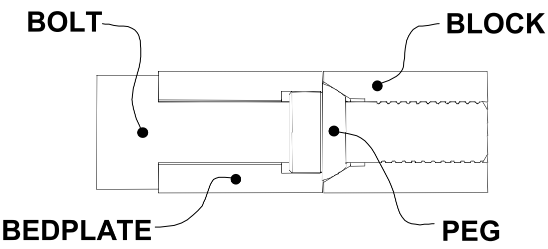


TYPE 2 GROOVE MFG.

+ =

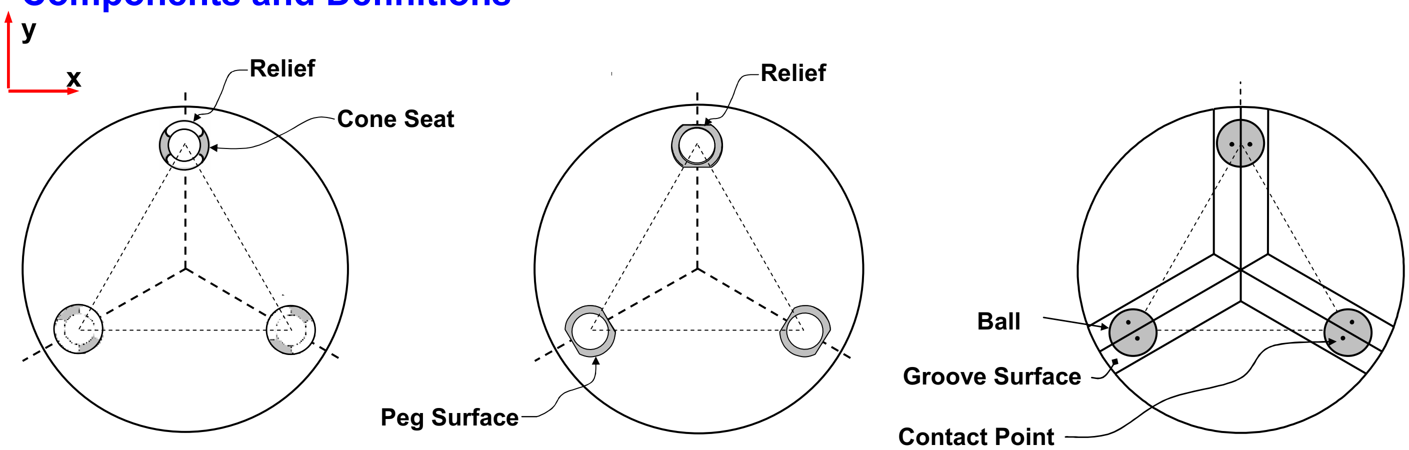
CAST FORM TOOL FINISHED

ASSEMBLED JOINT

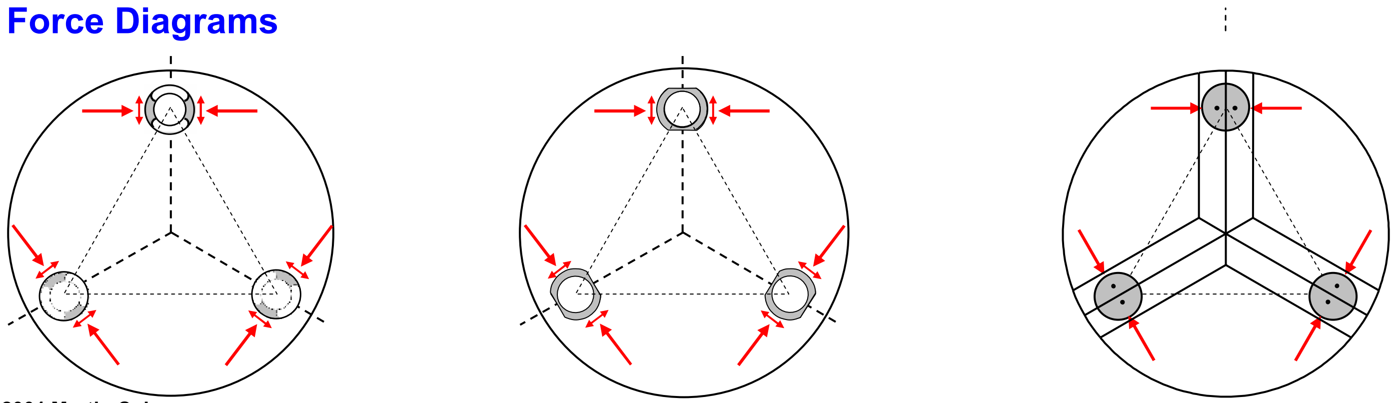


QKC Methods vs Kinematic Method

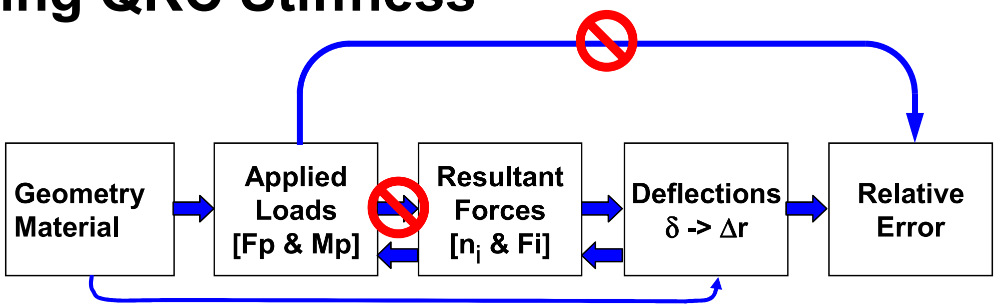
Components and Definitions



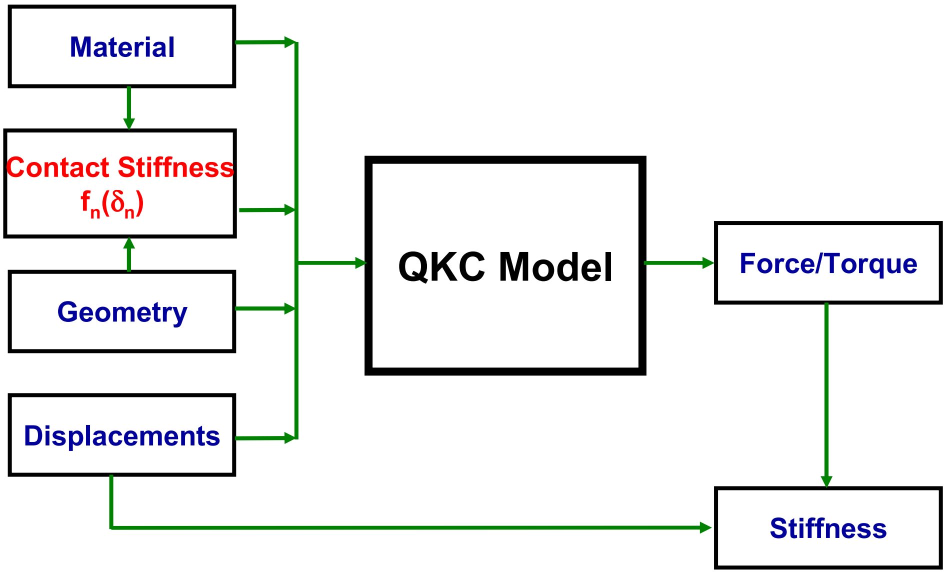
Force Diagrams



Modeling QKC Stiffness



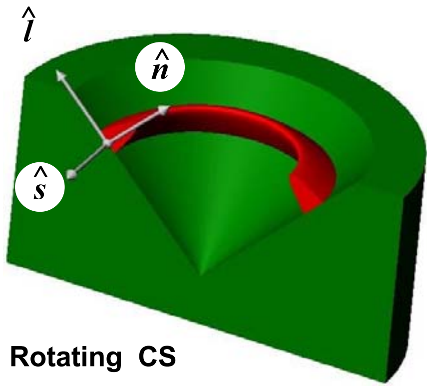
24 Unknowns & 7 Eqxns.



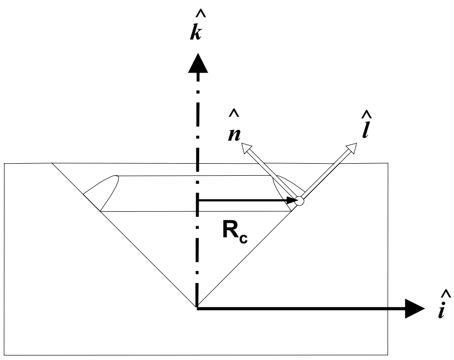
Contact Mechanics

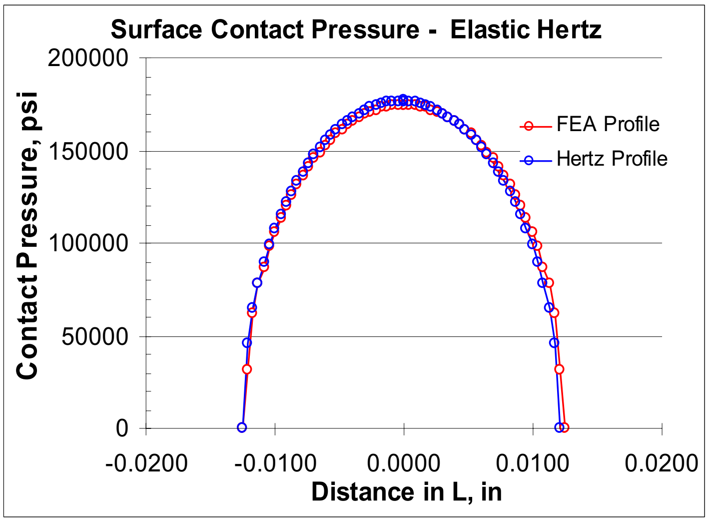
MECHANICS:

- **Use Rotating Coordinate System**
- **Assume Sinusoidal Normal Distance of Approach**
- **Obtain Contact Stress Profile as Function of Above**
- **Integrate Stress Profile in Rotating CS thru contact**



Rotating CS

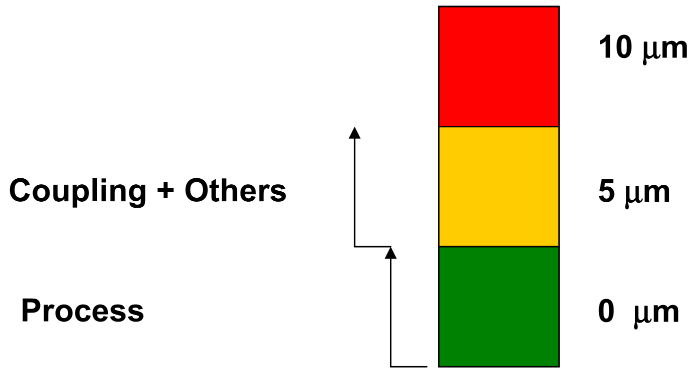




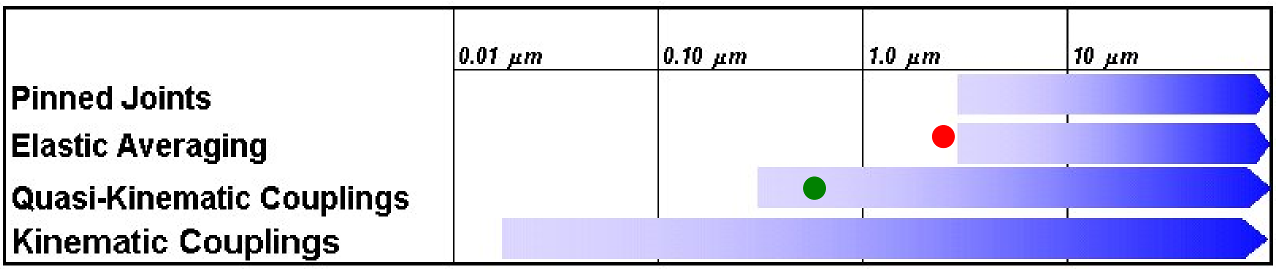
Example: Duratec Assembly

Characteristics:

- Ford 2.5 & 3.0 L V6
- > 300,000 Units / Year
- Cycle Time: < 30 s



Rough Error Budget



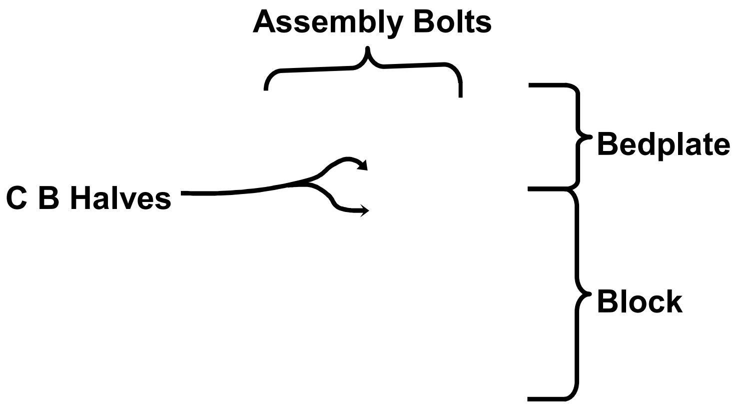
Example: Assembly of Duratec Block and Bedplate

COMPONENTS

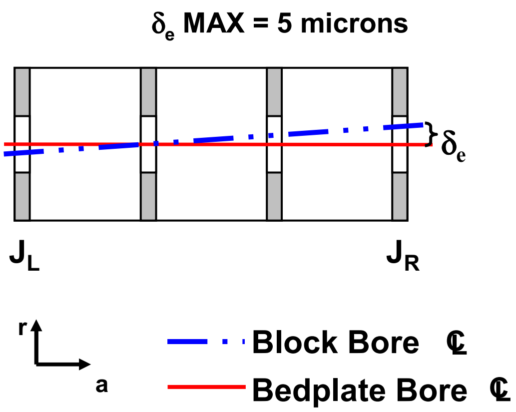
Block

Bedplate

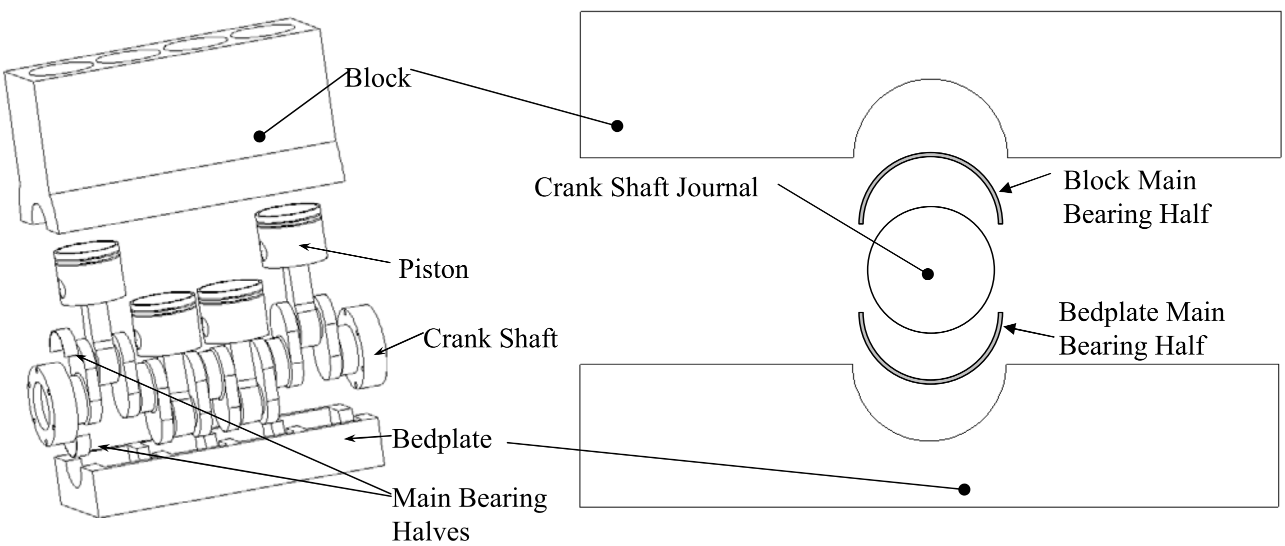
ASSEMBLY



ERROR



Bearing Assemblies in Engines



Results of Duratec QKC Research

MANUFACTURING:

Engine Manufacturing Process With Pinned Joint

Op. #10
- Mill Joint Face
- Drill/Bore **16 Holes**
- Drill Bolt Holes

→

Op. #30
- Drill Bolt Holes

→

Op. #50
- Press in **8 Dowels**
- Assemble
- Load Bolts
- Torque Bolts

→

Op. #100
- Semi-finish crank bores
- Finish crank bores

Modified Engine Manufacturing Process Using Kinni-Mate Coupling

Op. #10
- Mill Joint Face
- Drill/Bore **3 Peg** Holes
- Drill Bolt Holes & Form **3 Conical Grooves**

→

Op. #30
- Drill Bolt Holes

→

Op. #50
- Press **3 Pegs in BP**
- Assemble
- Load Bolts
- Torque Bolts

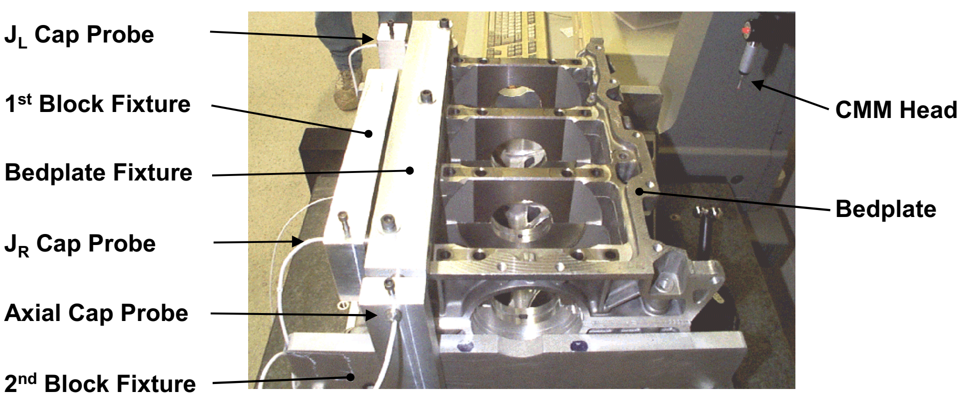
→

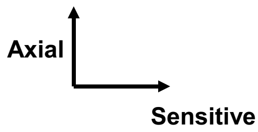
Op. #100
- Semi-finish crank bores
- Finish crank bores

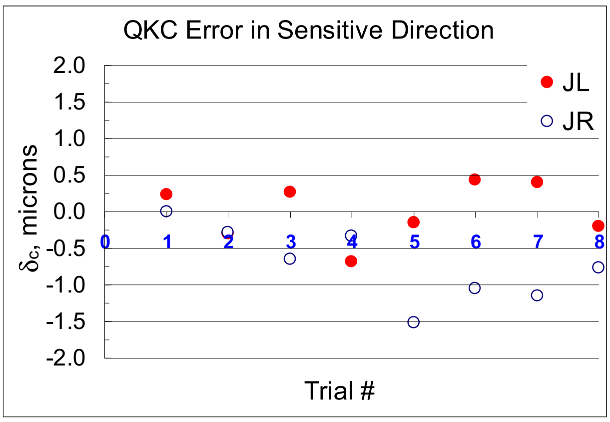
DESIGN:

ITEM	QKC	Pinned Joints
# Precision Pieces	3	8
# Precision Features	3	16
Feature Placement Tolerance	+/- 0.08mm	+/- 0.04mm
Average Centerline Repeatability	0.65 µm	4.85 µm
Normalized $/Engine	0.64	1

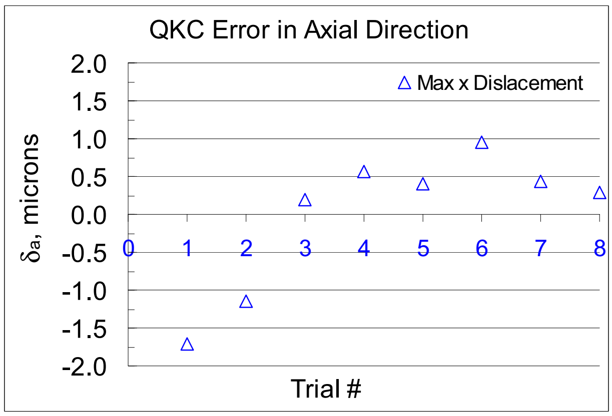
Engine Assembly Performance







$(^{Range}/_{2})|_{AVG} = 0.65\ \mu m$



$(^{Range}/_{2}) = 1.35\ \mu m$

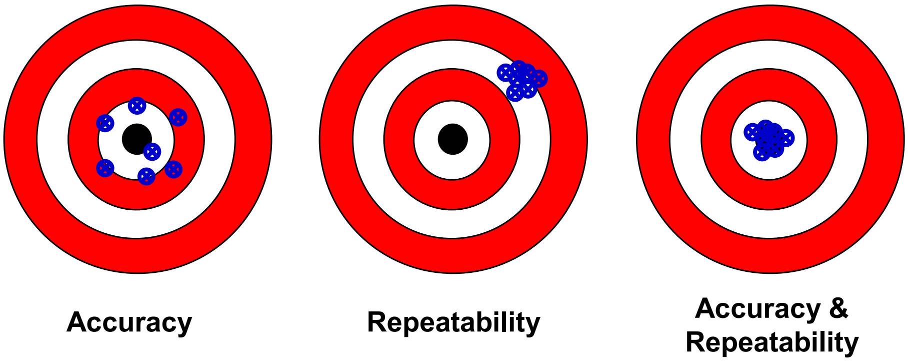
ADJUSTABLE GEOMETRIC CONSTRAINTS

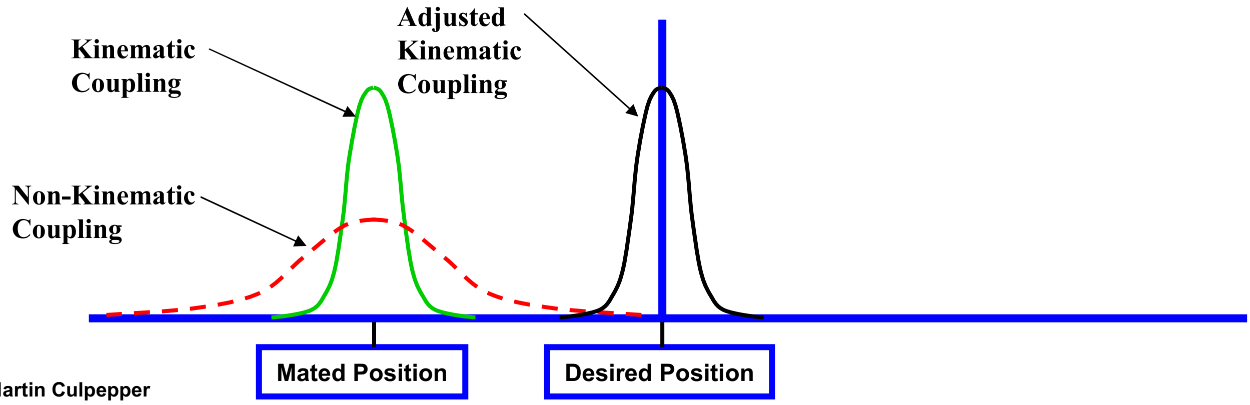
Hundreds of years of use/development and the &#@!*&% thing is not yet accurate!?!?!?

Why adjust kinematic couplings?

KC Repeatability is orders of magnitude better than accuracy

Accuracy = *f* (manufacture and assemble)

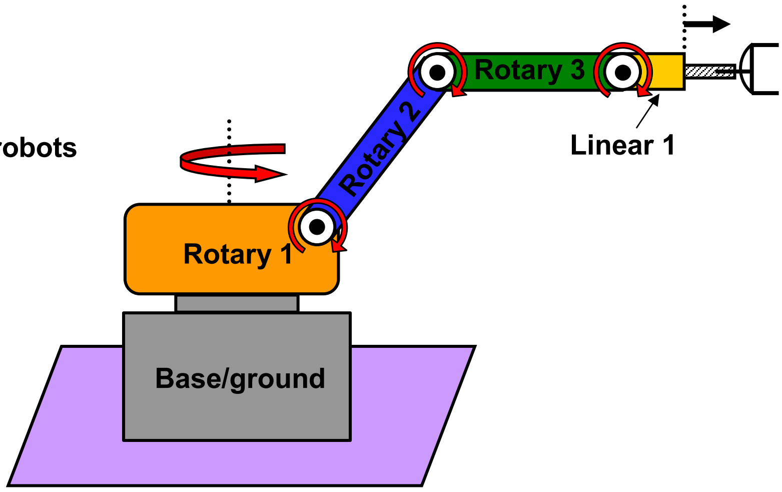




Serial and parallel kinematic machines/mechanisms

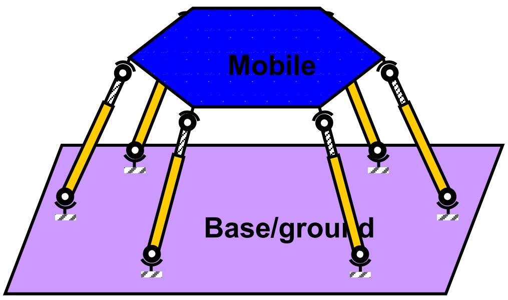
SERIAL MECHANISMS

- 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



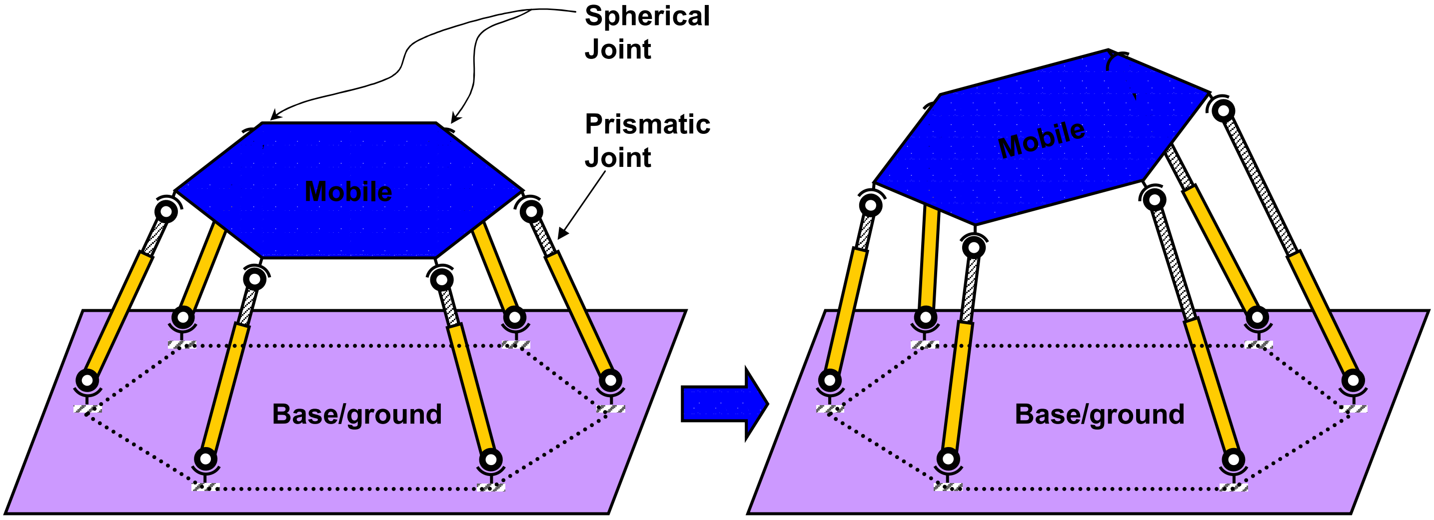
Parallel mechanism: Stewart-Gough platform

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

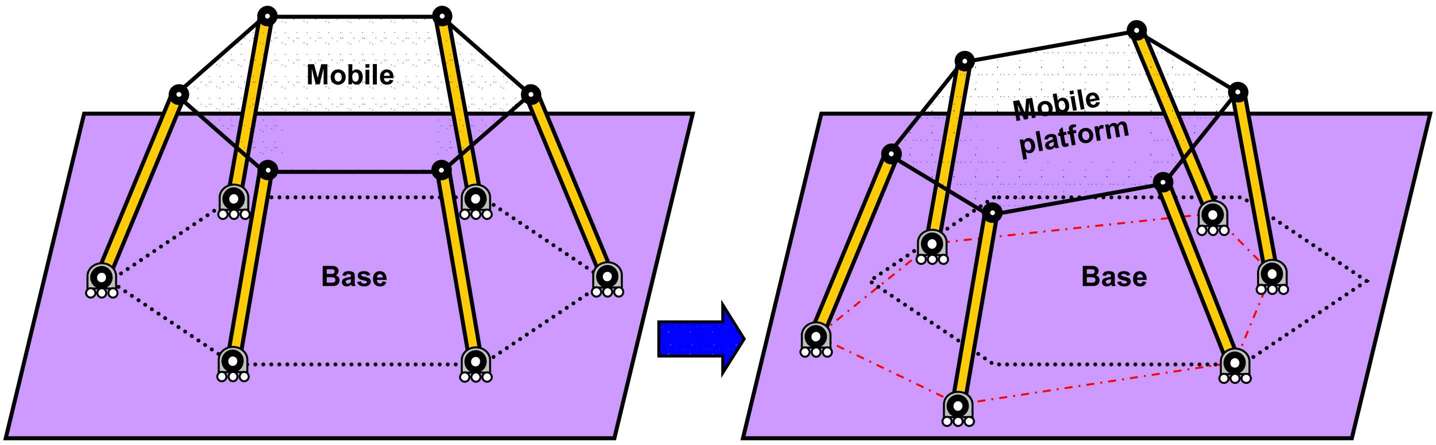
E.g. Changing length of "legs"



Parallel mechanism: Variation

6 DOF mechanism/machine by changing position of joints

Can have a combination of position and length changes



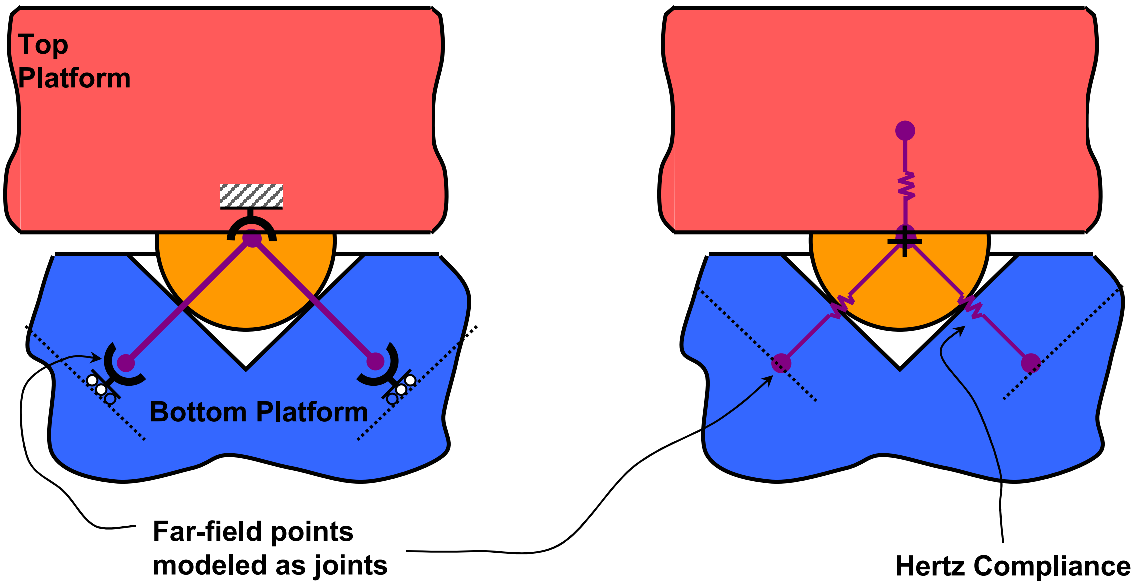
Kinematic couplings as mechanisms

Ideally, kinematic couplings are static parallel mechanisms

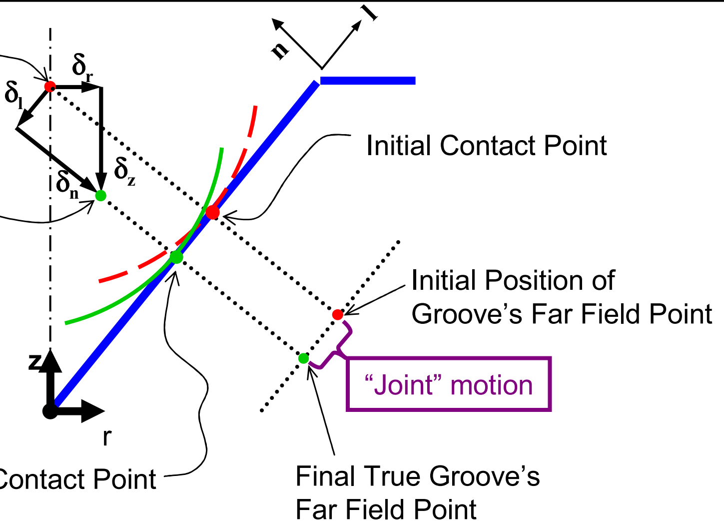
IRL, deflection(s) = mobile parallel kinematic mechanisms

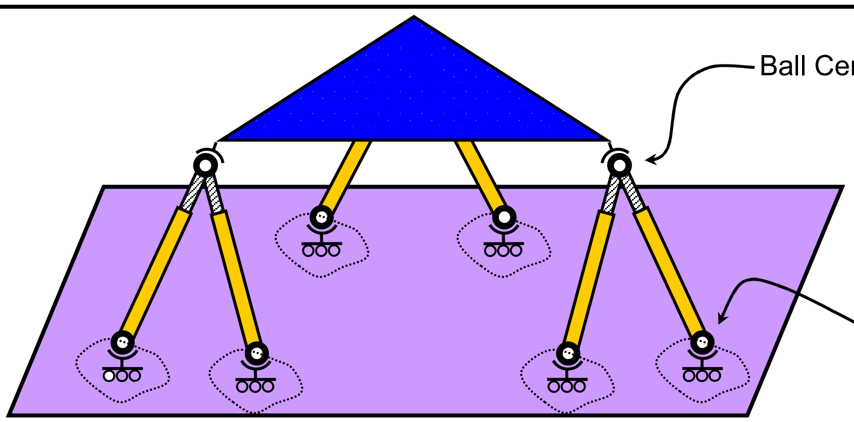
How are they "mobile"?

- 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





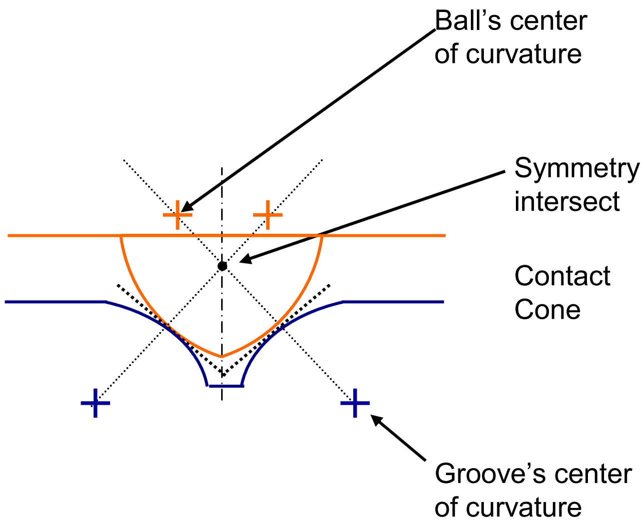
Accuracy of kinematic mechanisms

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

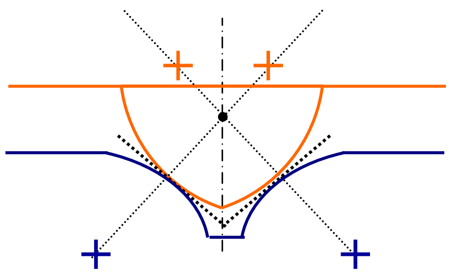
- Groove center of curvature location
- Groove orientation
- Groove depth
- Groove radii

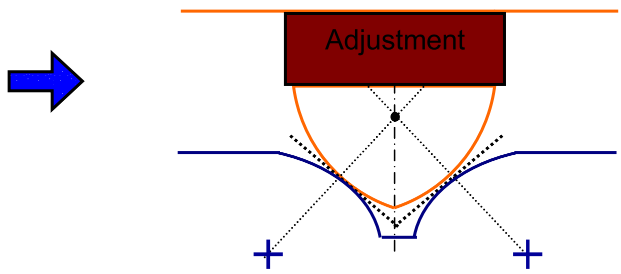


Utilizing the parallel nature of kinematic couplings

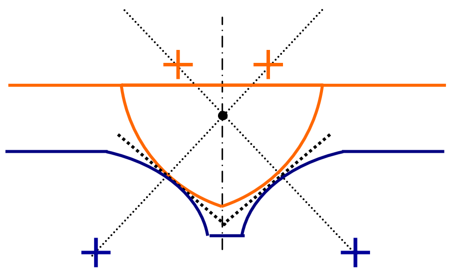
Add components that adjust or change link position/size, i.e.:

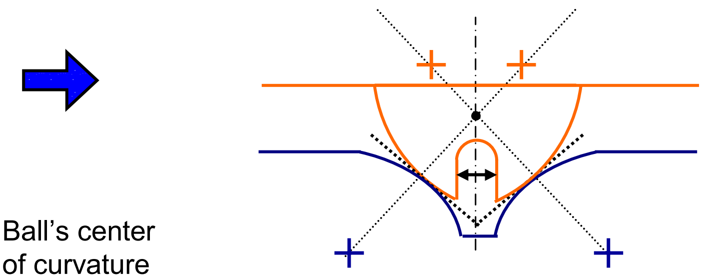
- Place adjustment between kinematic elements and platforms (joint position)





- Strain kinematic elements to correct inaccuracy (element size)



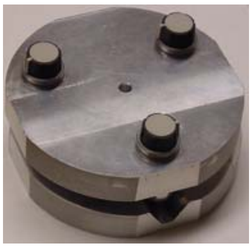


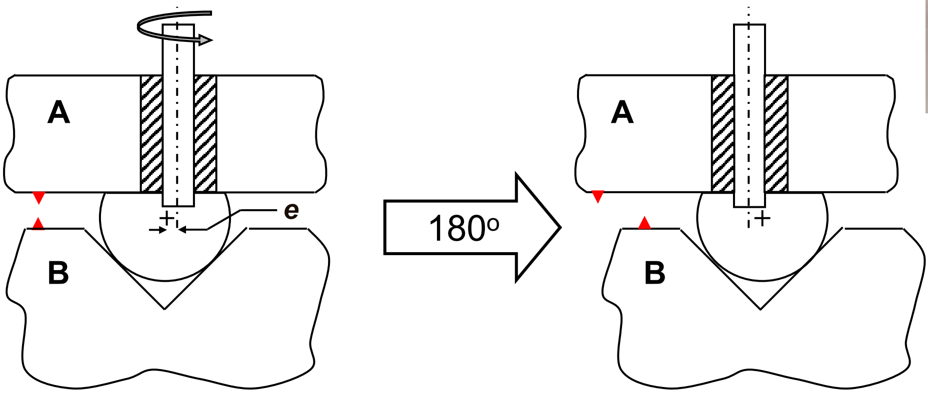
Ball's center
of curvature

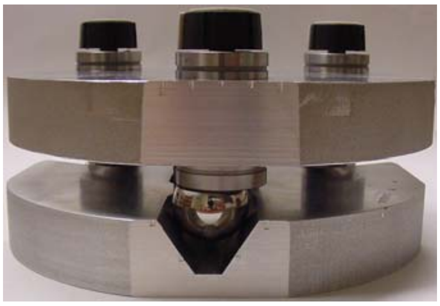
Example: Adjusting planar motion

Position control in x, y, θ_z:

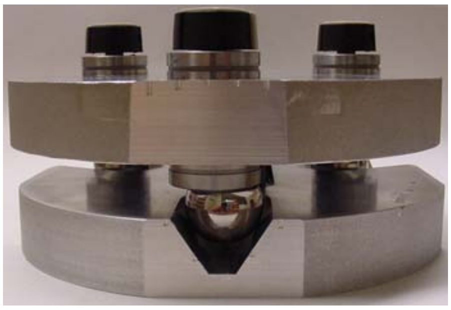
- Rotation axis offset from the center of the ball







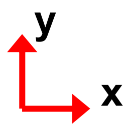
Eccentric left

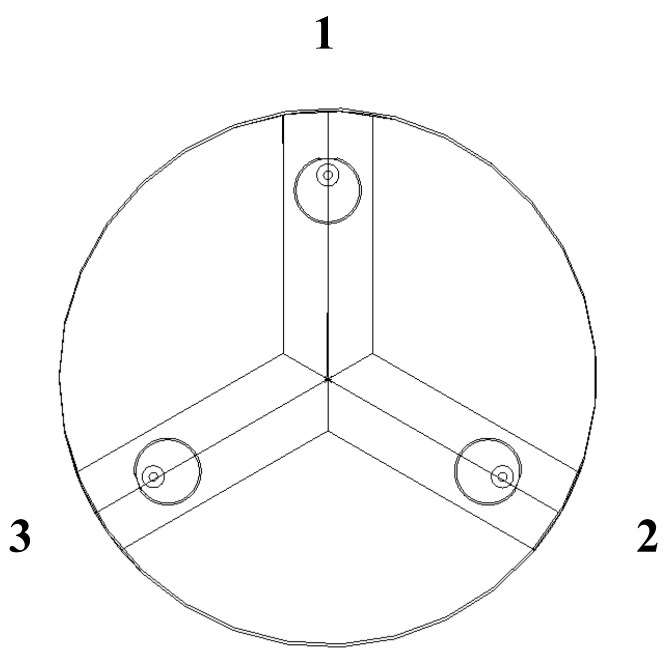


Eccentric right

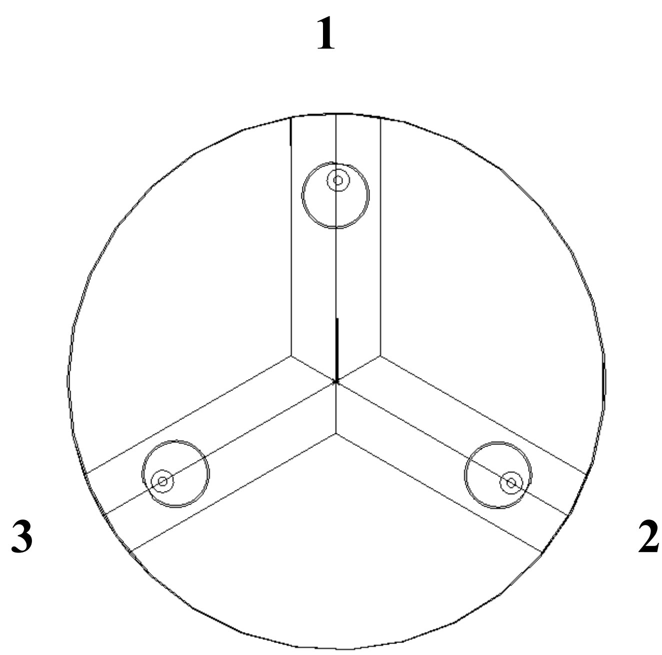
Patent Pending

ARKC demo animation





Input: Actuate Balls 2 & 3
Output: Δy

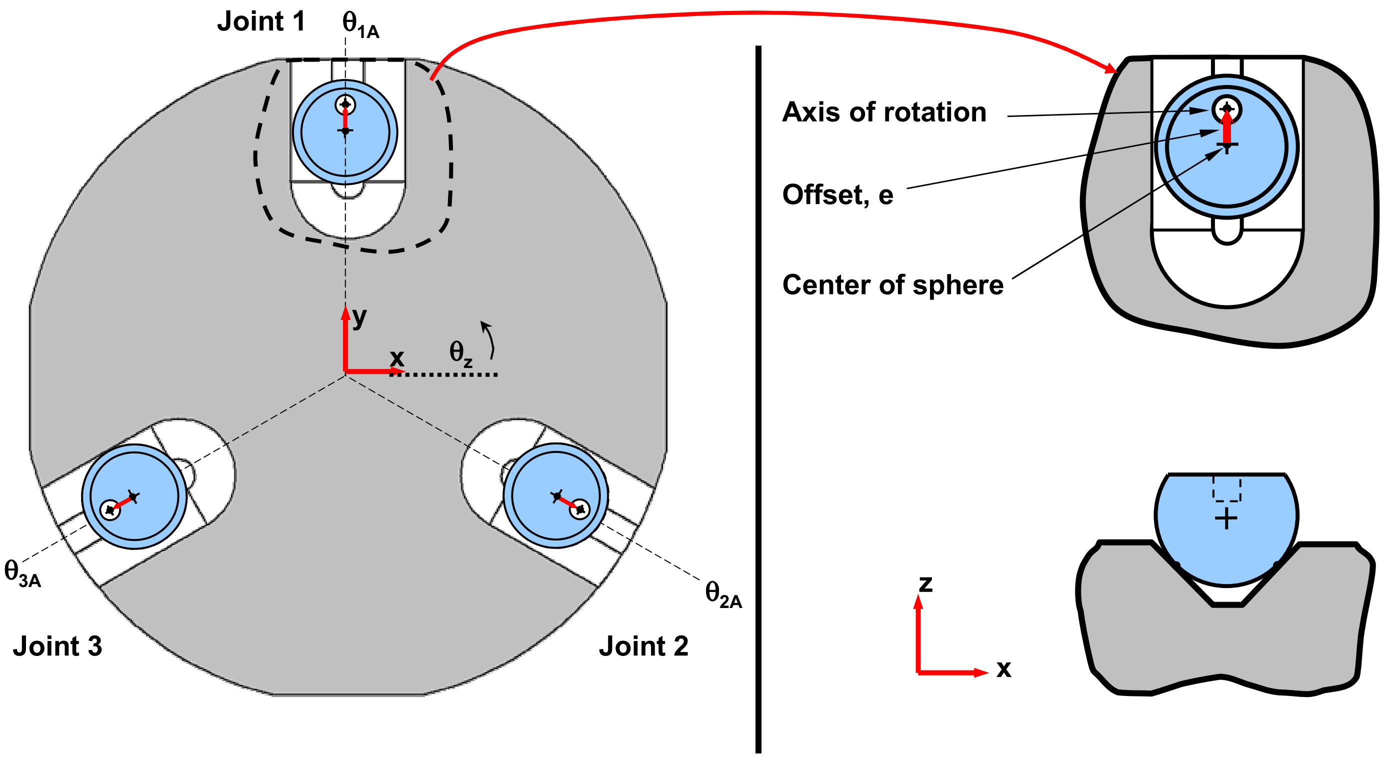


Input: Actuate Ball 1
Output: Δx and $\Delta\theta_z$

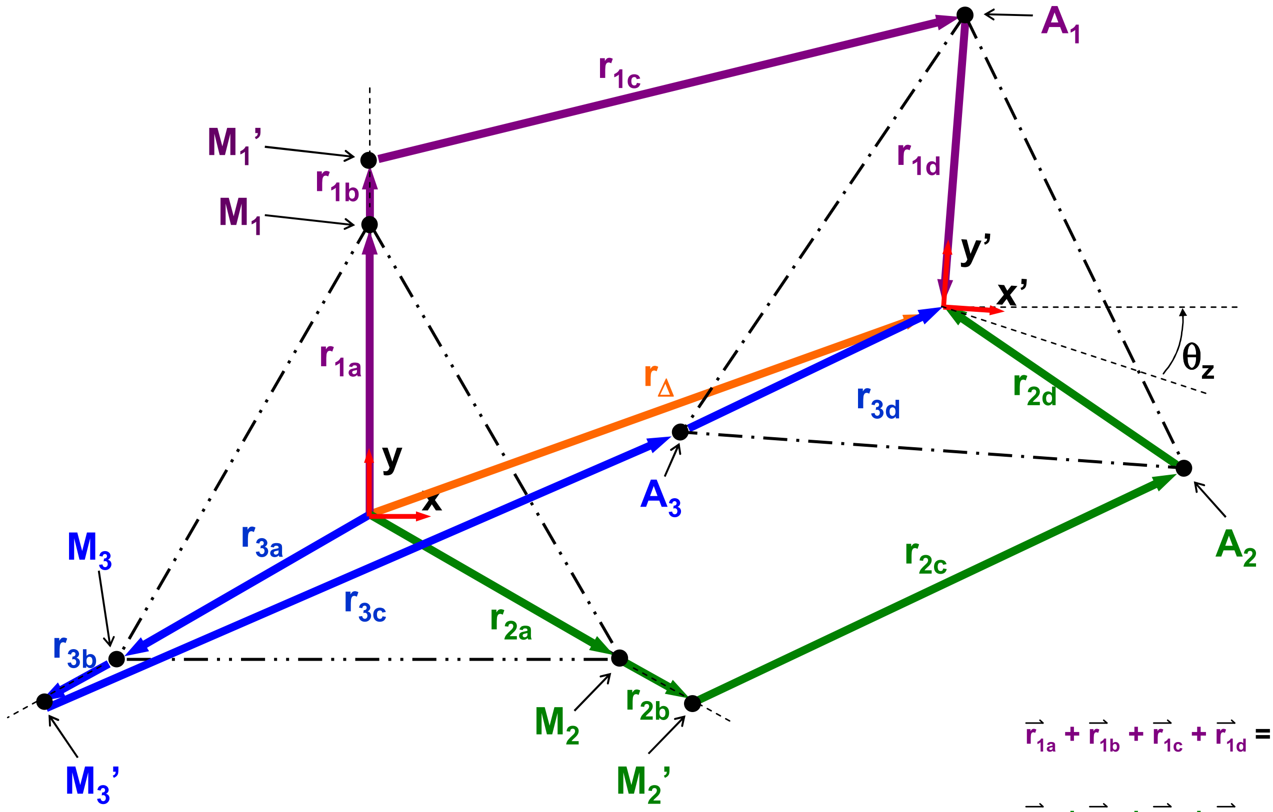
Planar kinematic model

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



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

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

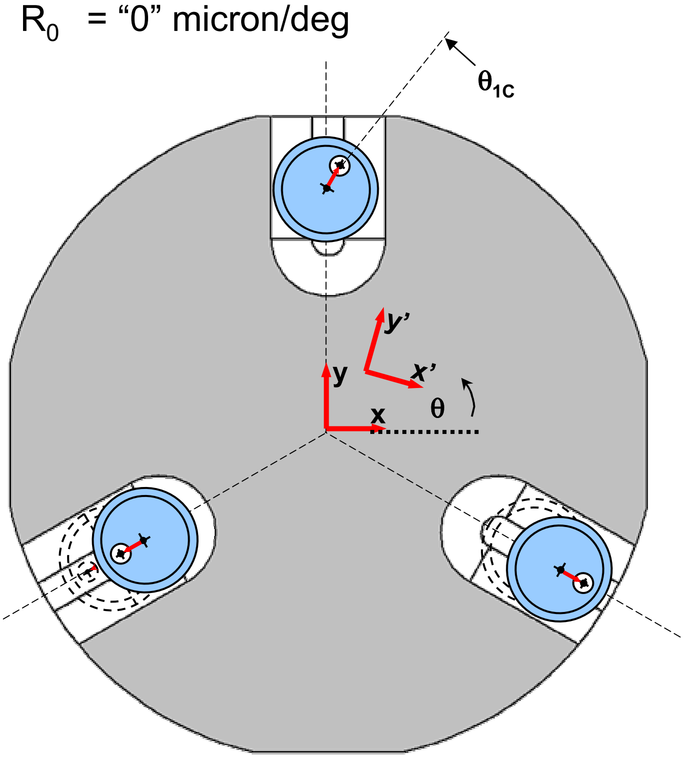
$$\vec{r}_{3a} + \vec{r}_{3b} + \vec{r}_{3c} + \vec{r}_{3d} = \vec{r}_{\Delta}$$

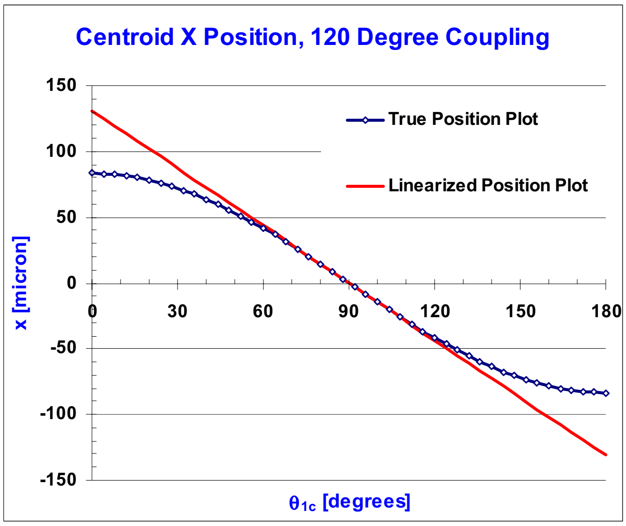
ARKC resolution analysis

For E = 125 microns

R_{90} = -1.5 micron/deg

R_0 = “0” micron/deg





Limits on Linear Resolution Assumptions

% Error	θ_{1C} 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)

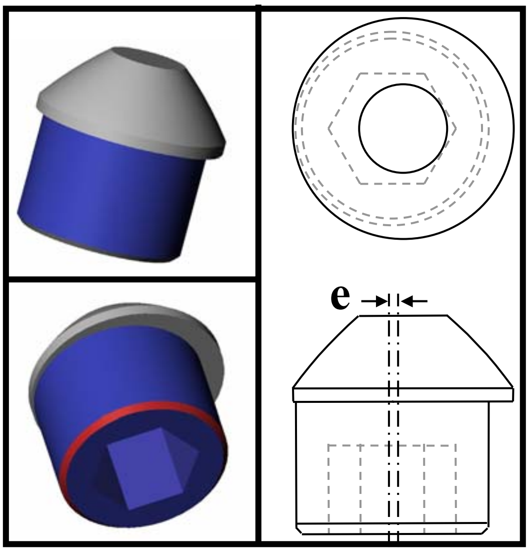
Peg shank and convex crown are offset

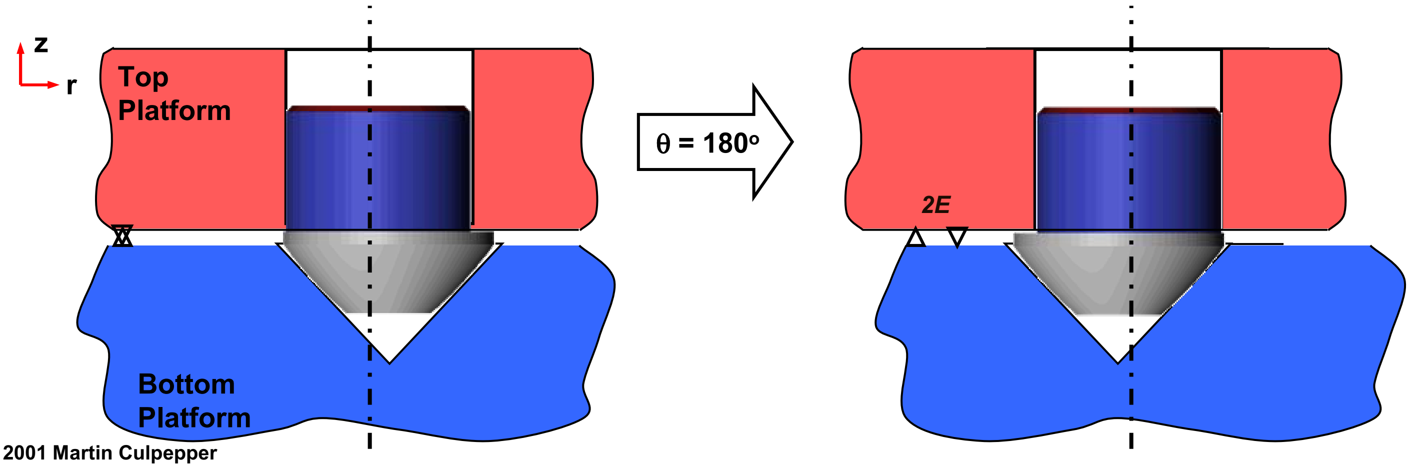
Light press between peg and bore in plate

Adjustment with allen wrench

Epoxy or spreading to set in place

Friction (of press fit) must be minimized…



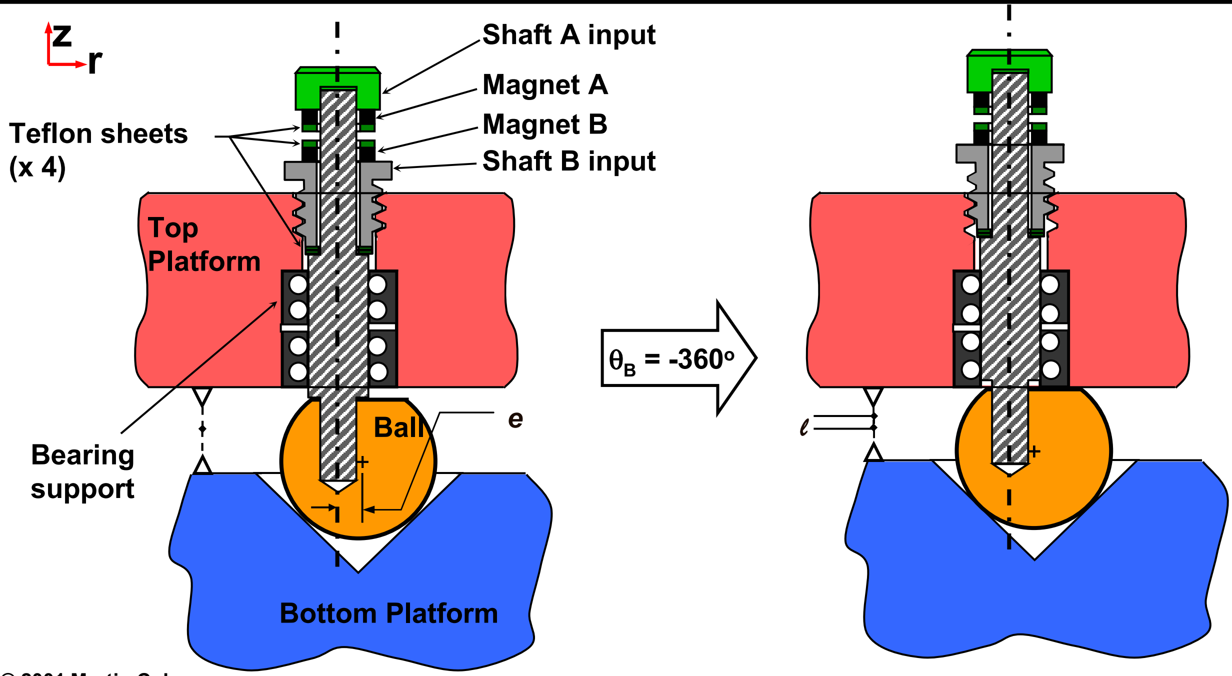


Moderate-cost adjustment (3 micron)

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

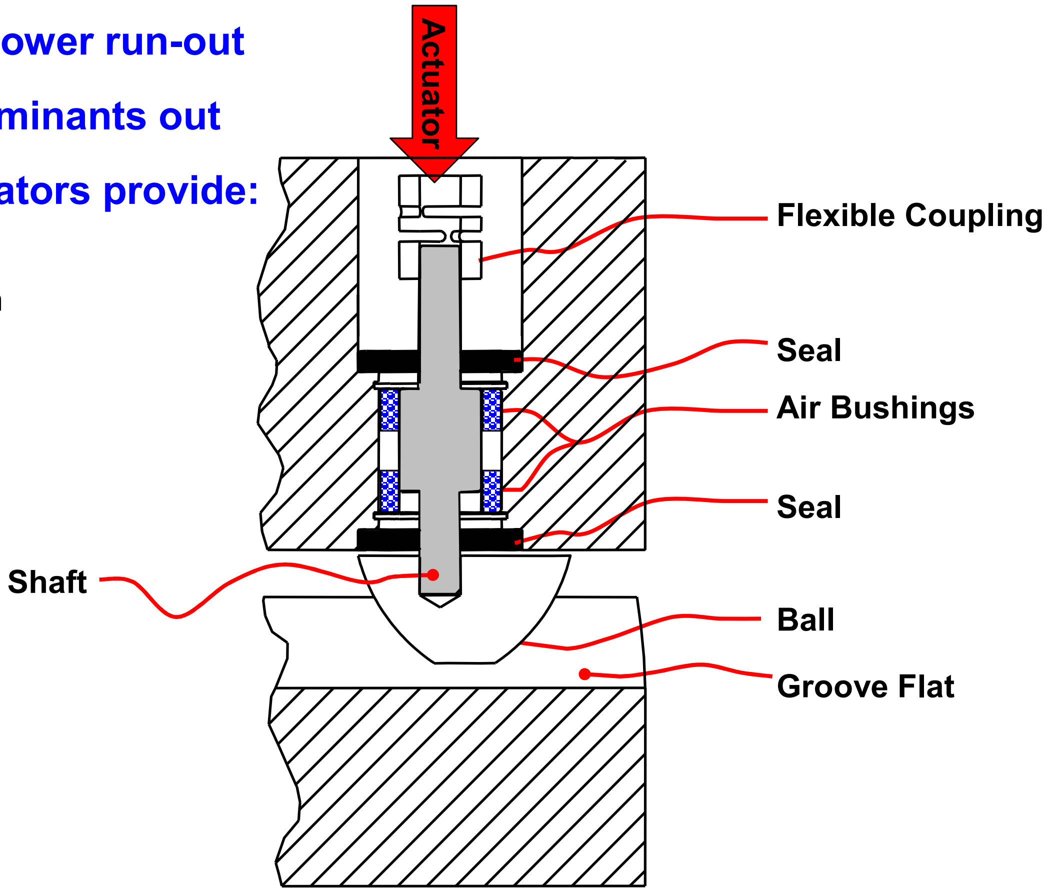
Shaft A positions as before [x, y, θ_z]

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



“Premium” adjustment (sub-micron)

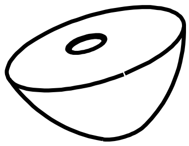
- Run-out is a major cause of error
- Air bushings for lower run-out
- Seals keep contaminants out
- Dual motion actuators provide:
  - Linear motion
  - Rotary motion



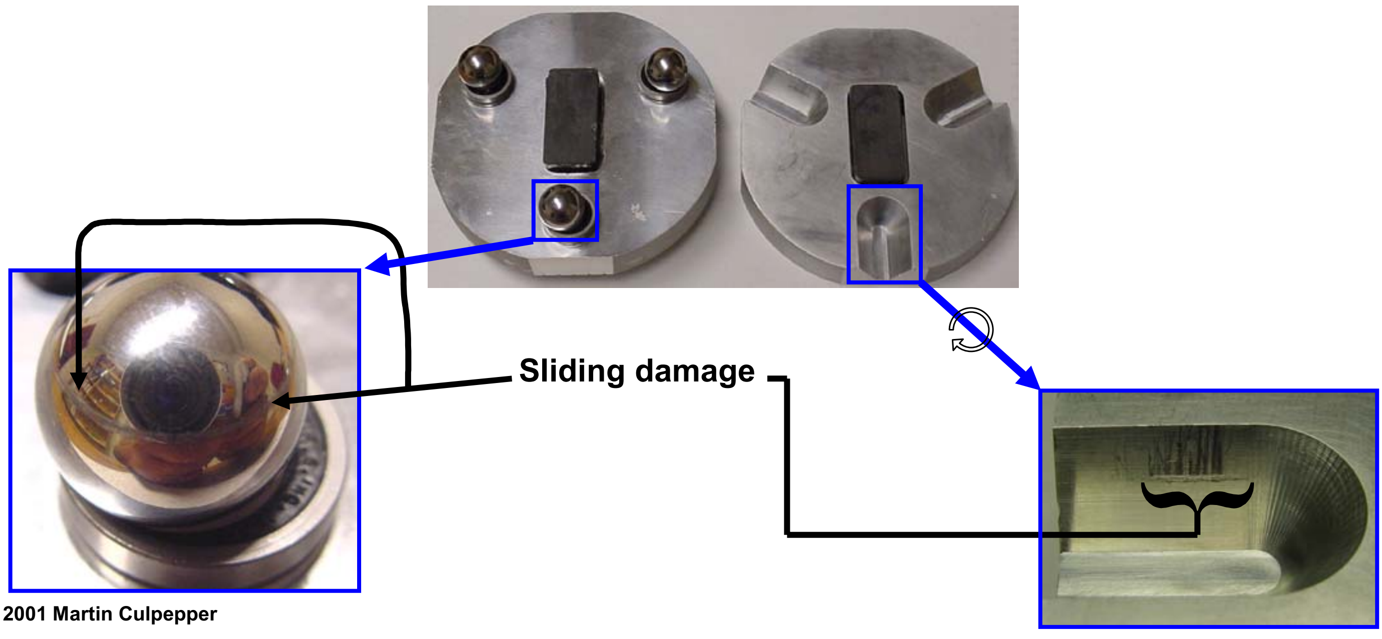
Mechanical interface wear management

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

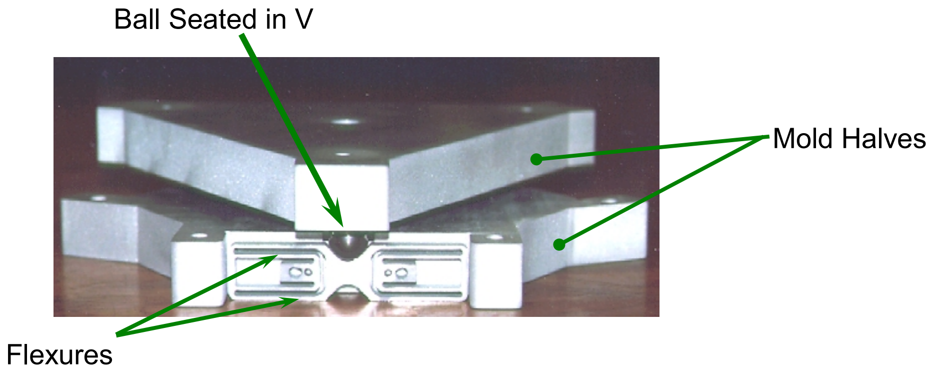
Motivated by coupling envy……

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

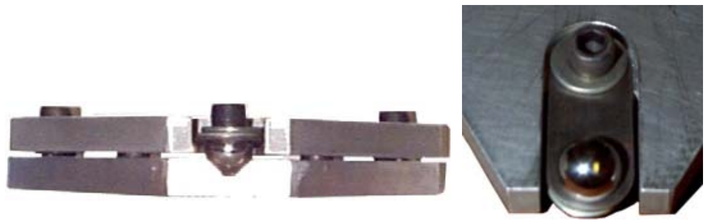
Error loads are often proportional to applied loads

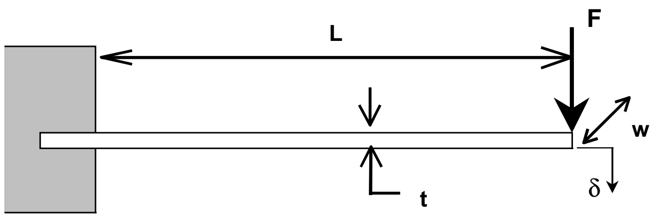
- Example: Bolt head friction
- $T_B \sim F_B R_B \mu$
- Design for $k_{sensitive} >> k_{non\text{-}sensitive}$

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

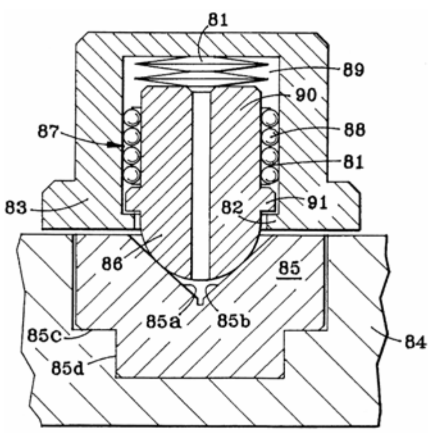
Design Issues (flexure)

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

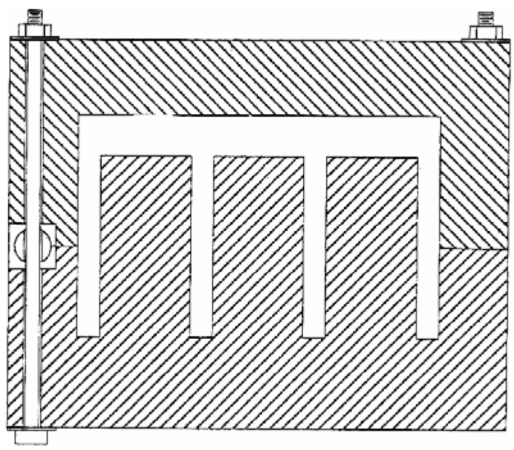
Design Issues (flexures)

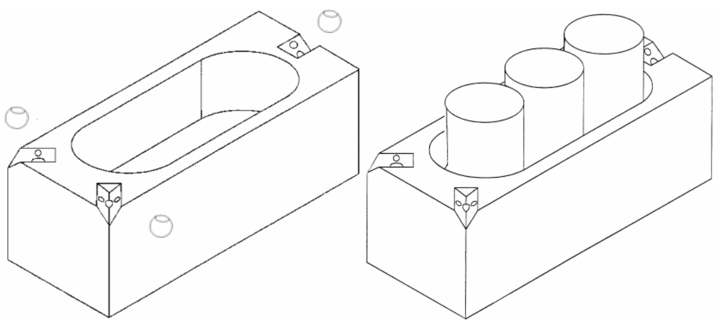
1. $K_r = \dfrac{K_{guide}}{K_{spring}}$
2. Press fit tolerances

Cost

$ 2000

Plastic compliant kinematic couplings





Characteristics

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

Applications/Processes

1. Sand Casting

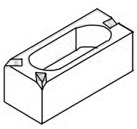
Design Issues

1. Loose Sand
2. K_r application specific

Cost

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

Experimental results

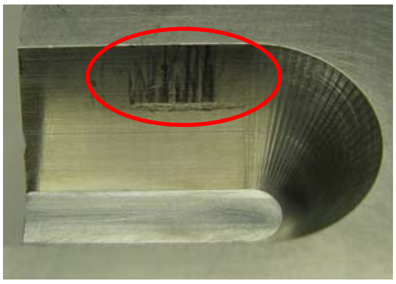
		Coupling Type	Prototype	Average Repeatability Radial μinch (microns)	 Standard Deviation μinch (microns)	 Angular μ radians	 Standard Deviation μ radians	Manufacturing Cost $
Stamped	1.		Water Jet (mate-ball)	300 (7.6)	100 (2.5)	35	21	23
	2.		Stamped (mate-ball)	400 (10.2)	200 (5.1)	120	70	10
	3.		Stamped (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

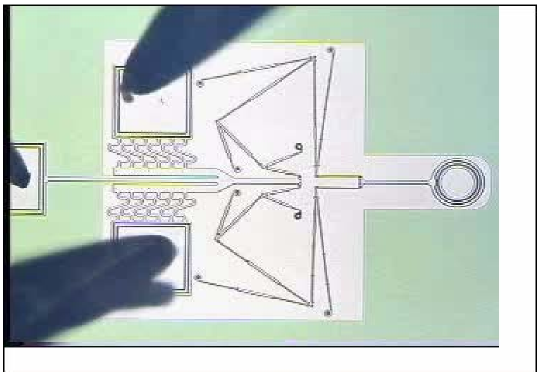
Compliant mechanisms:

- 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

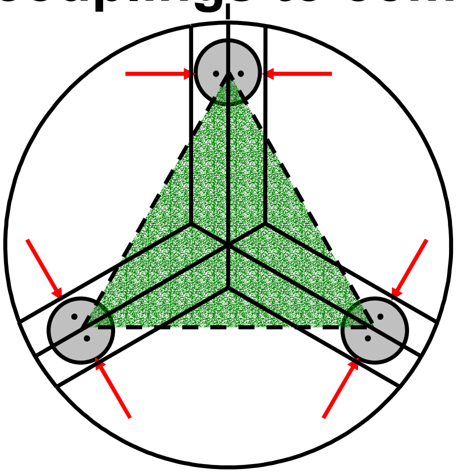


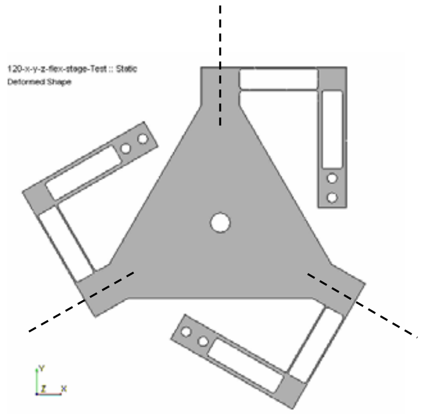
thermal1[1].mpeg

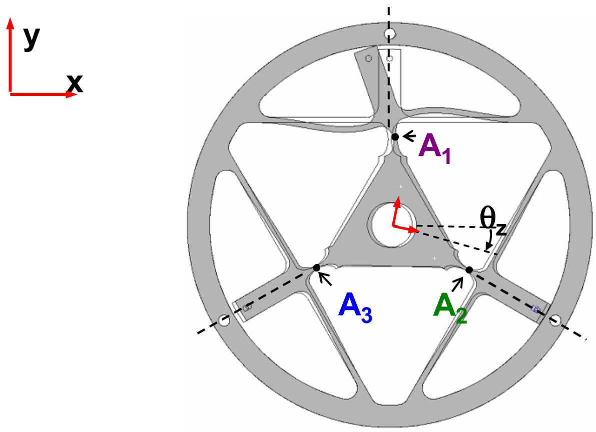
Why compliant mechanisms in precision fixtures

- Repeatable/low hysteresis
- No assembly
- No contact

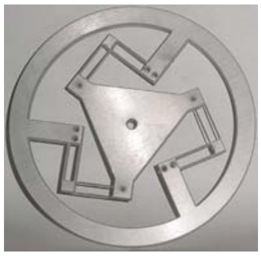
From kinematic couplings to compliant stages







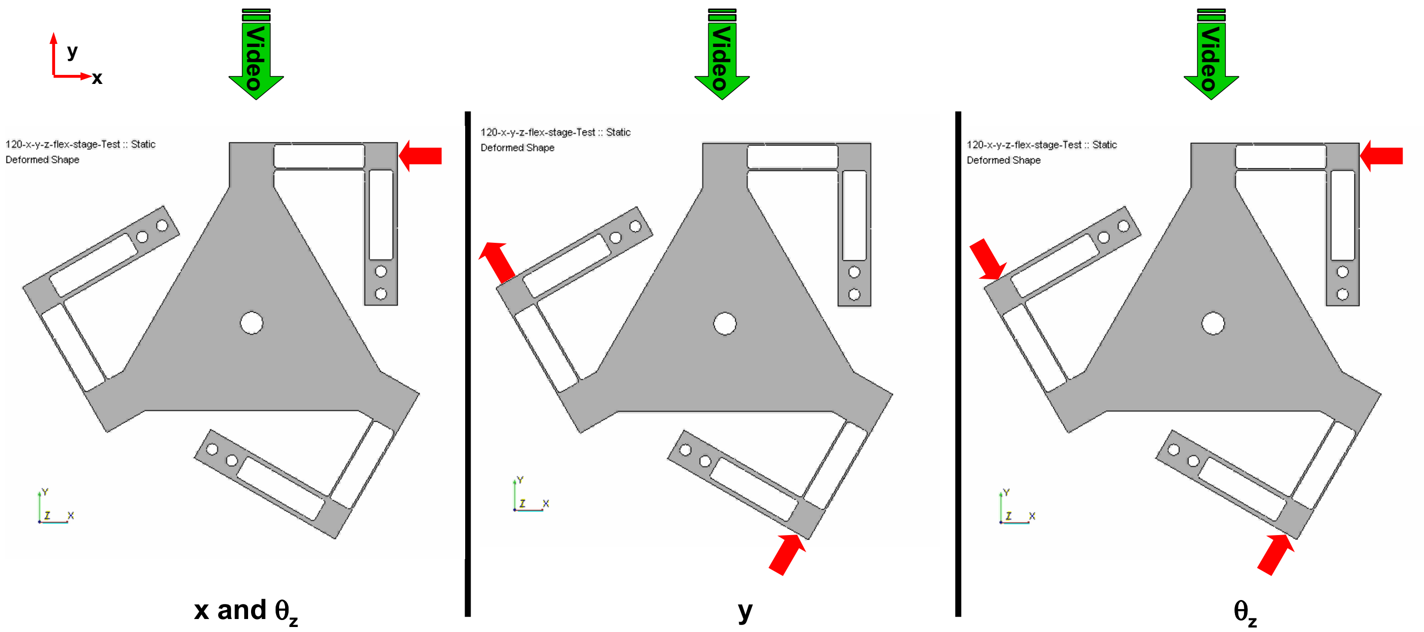
Constraint based compliant mechanisms



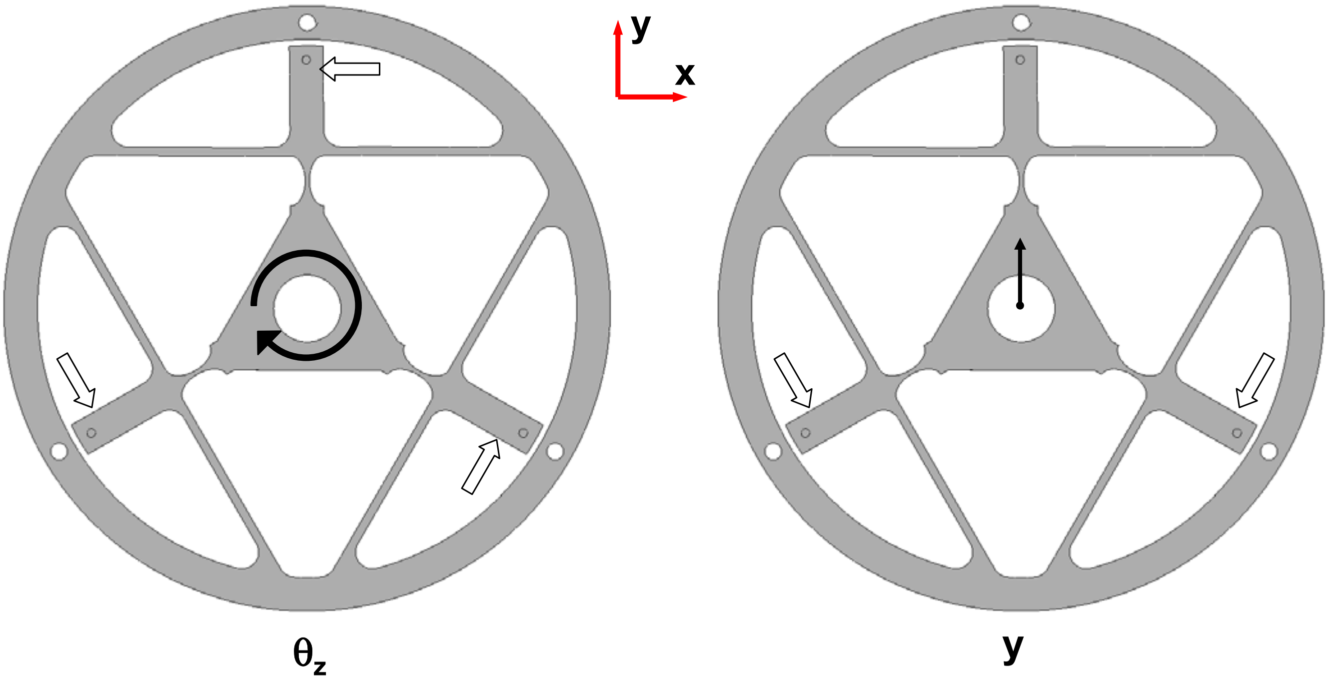
High volume, low cost, multi-degree of freedom alignment

Example 3 DOF flexure system:

Target applications: Opto-electronic packaging/alignment



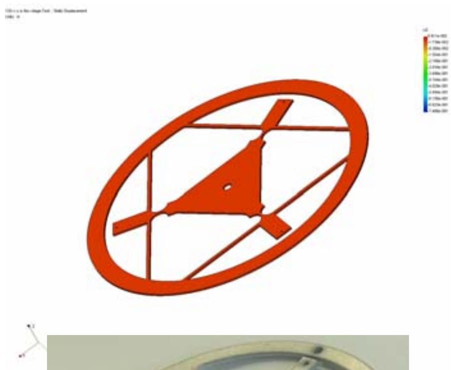
Constraint based compliant mechanisms cont.



Constraint based compliant mechanisms cont.

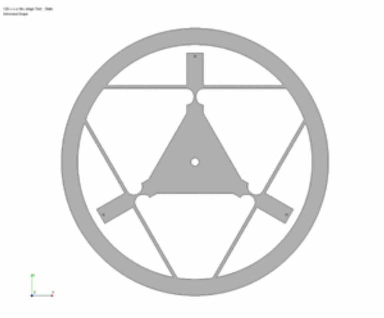
Example 6 DOF alignment capability

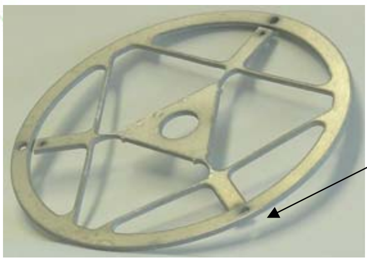
Target app.: Micro and meso scale positioning (I.e. opto-electronics)



Video

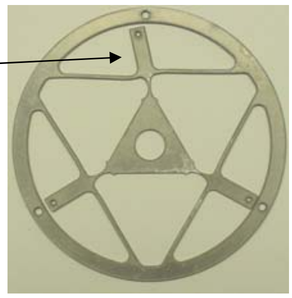
Video





Permanent set due to plastic deformation

Have option to rely on elastic deformation



z , θ_x , θ_y

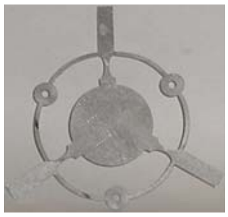
x and θ_z

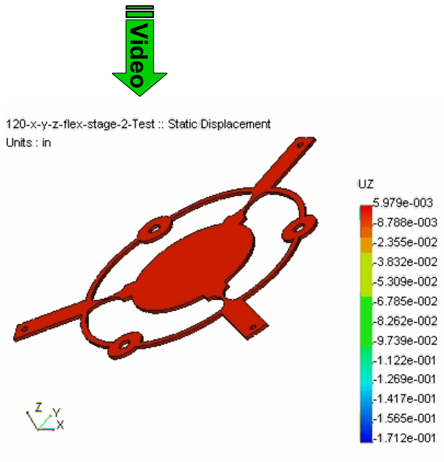
Patent Pending

Constraint based compliant mechanisms cont.

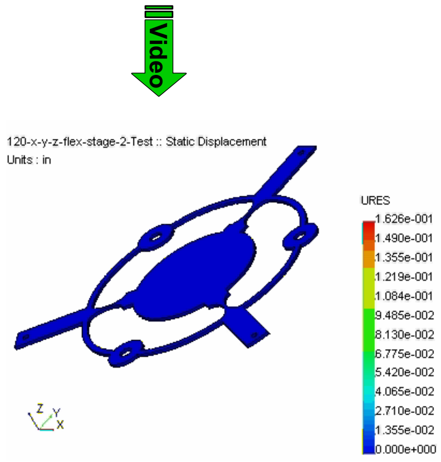
Example 6 DOF alignment capability

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

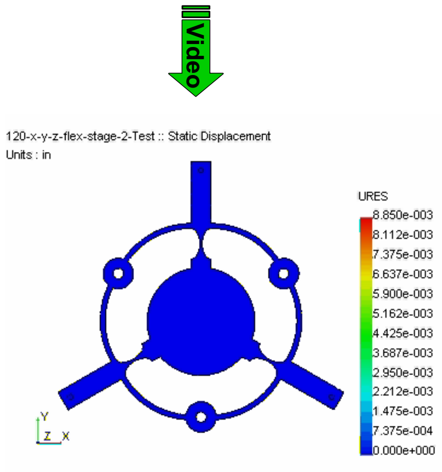




z , θ_x , θ_y



z



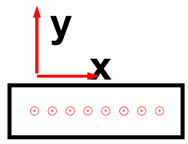
θ_z

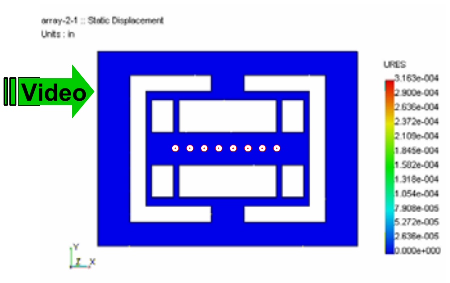
Patent Pending

Constraint based compliant mechanisms cont.

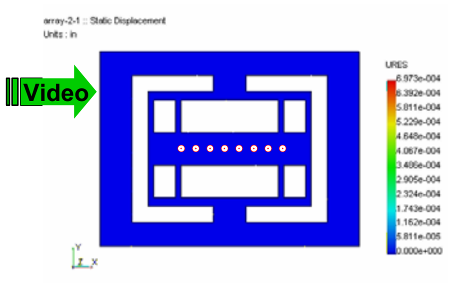
3 DOF active alignment [x, y, z]   &   2 DOF passive alignment [z, θ_y]

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

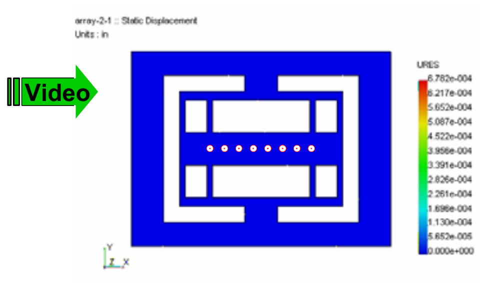




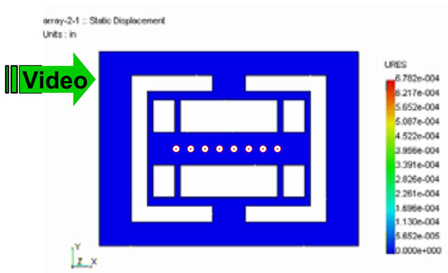
Δx



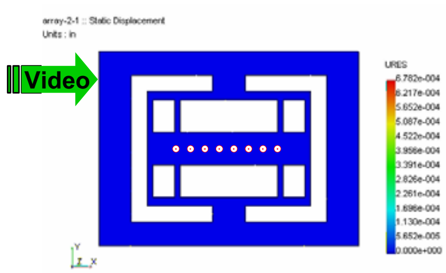
Δy



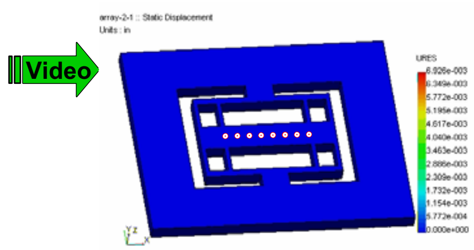
$\Delta\theta_z$



Δx & $\Delta\theta_z$



Δx & Δy

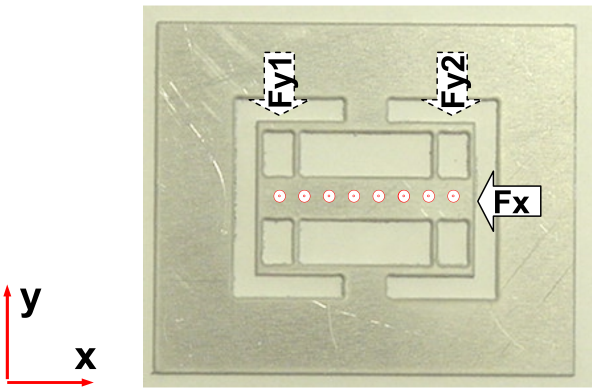


$\Delta\theta_y$

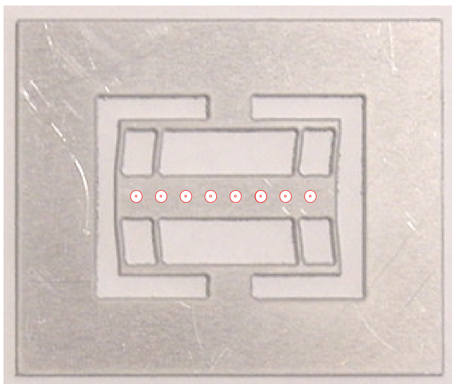
Constraint based compliant mechanisms cont.

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