FUNdaMENTALS of Design

Topic 9

Structural Connections & Interfaces

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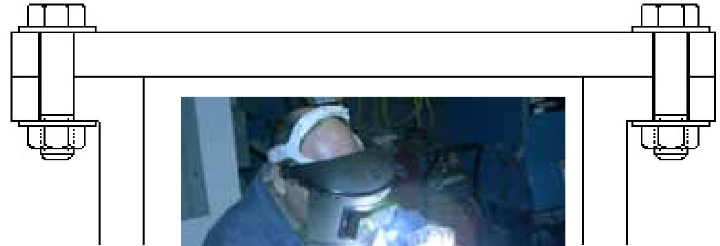
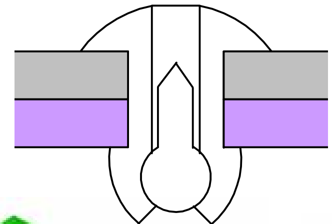
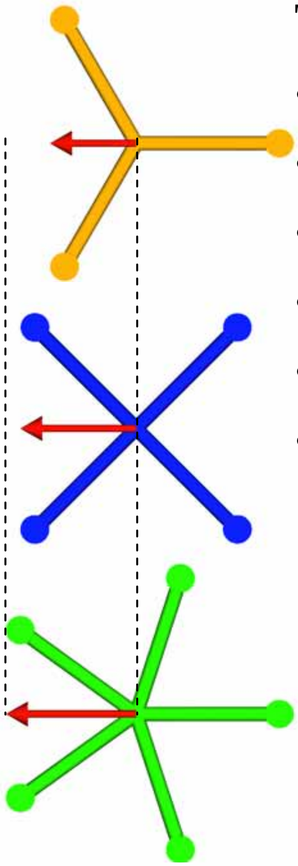
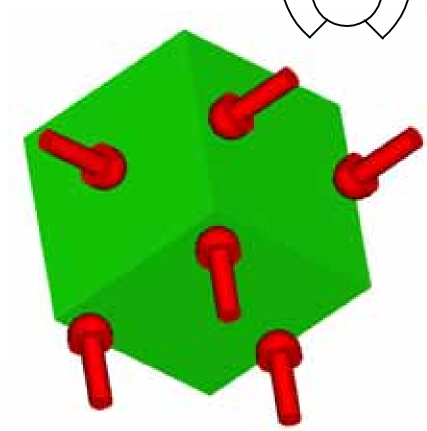
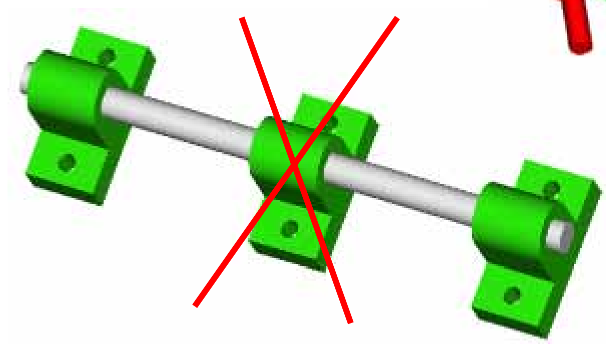


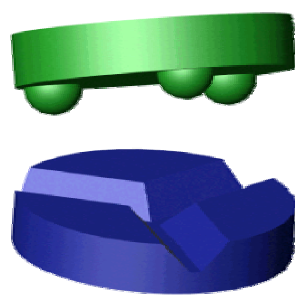


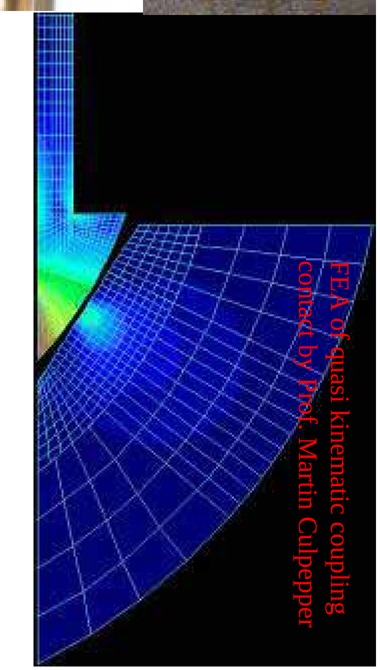
Topics
- Connections
- Structural Joints
- Structural Interfaces
- Hertz Contact
- Kinematic Couplings
- Elastic Averaging

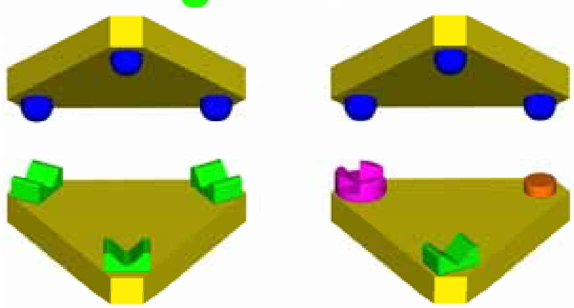




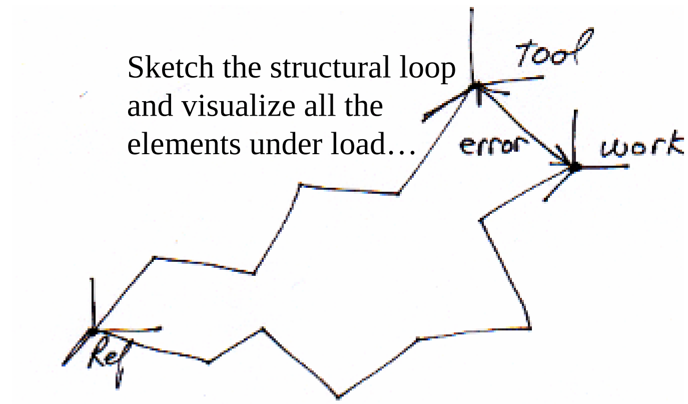




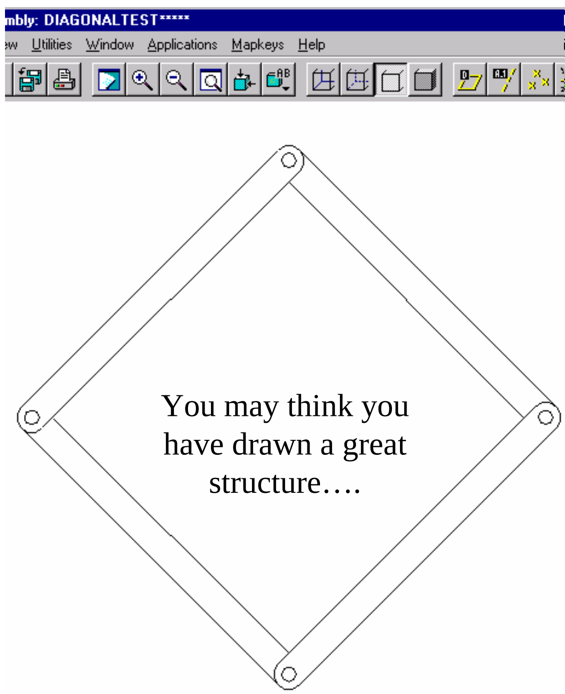


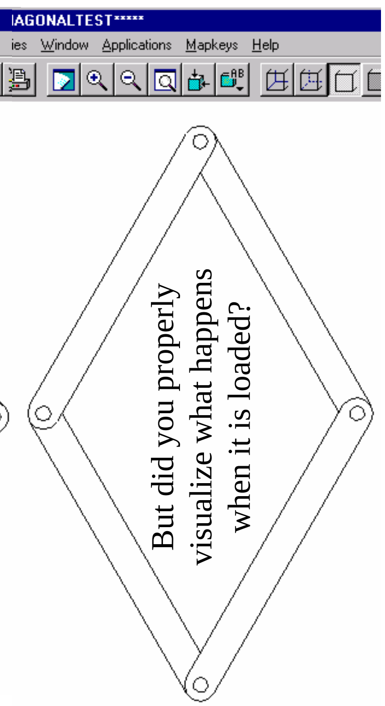


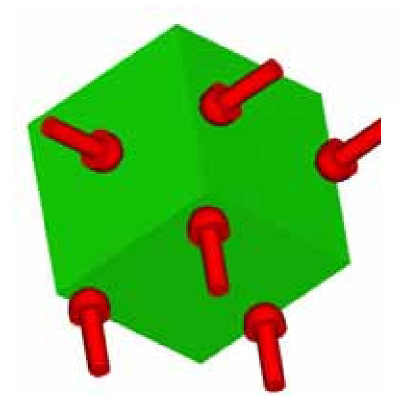
Connections: *Visualization*



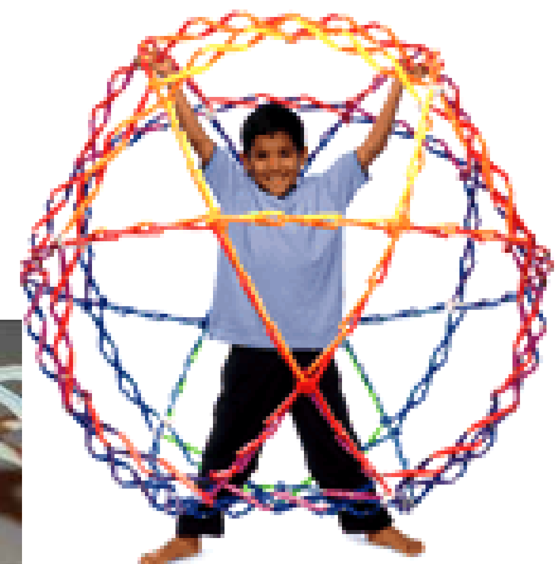
- As a visualization tool for a joint of which are unsure:
  - Make a cardboard model of the joint
    - If the model is stable, there is a good chance that the real parts will also be stable!
- What happens to the performance of your structure if you assume the joints are just pinned?



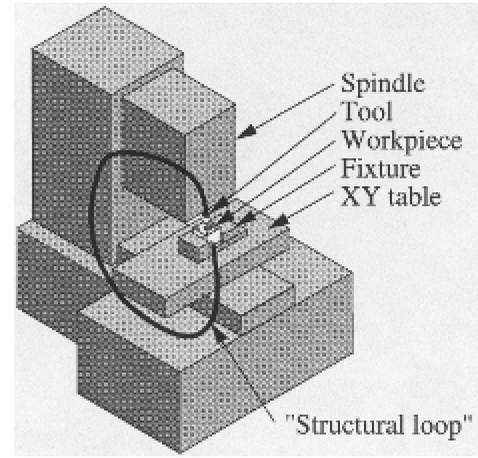




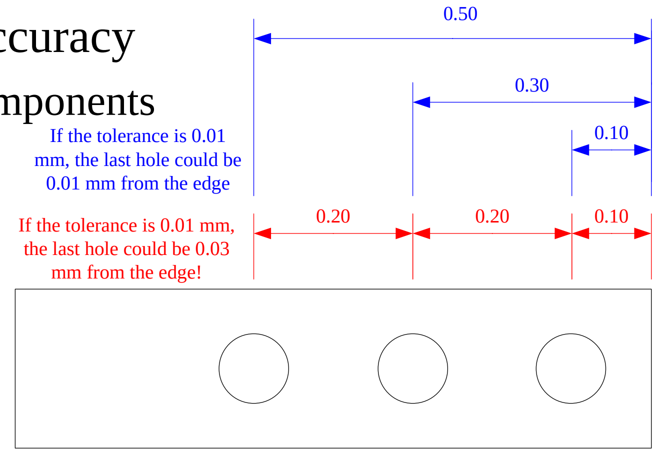
www.hoberman.com

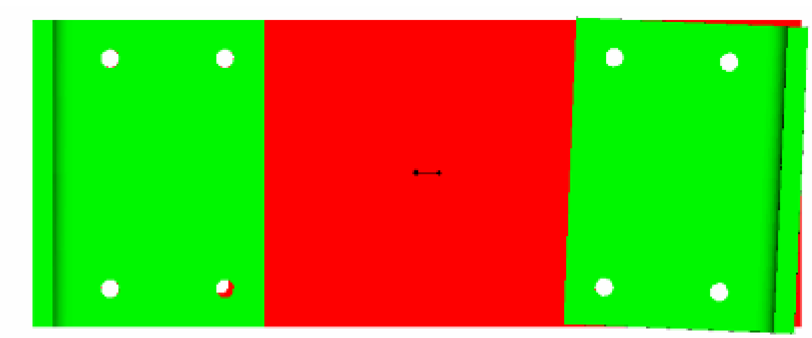


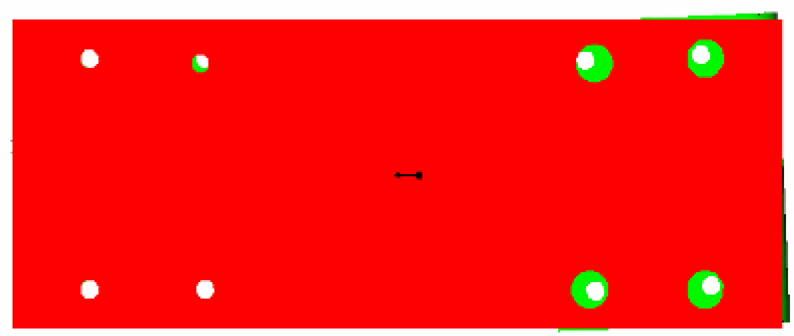
Connections: *Accuracy*



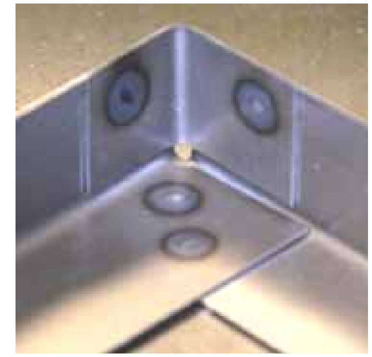
- Interfaces must enable parts to fit together with the desired accuracy
  - You cannot create two sets of exactly matching holes in two components
    - You can oversize the holes
    - The clearance between the bolts and holes means that the components do not have a unique assembly position
- "Error budgets" keep track of interferences & misalignments
  - These methods often assume "worst case tolerance"
  - For complex assemblies, advanced statistical methods are required
  - Deterministic designs are created using *financial*, *time*, and ***error*** budgets



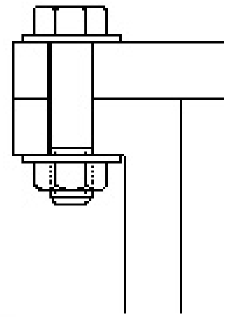


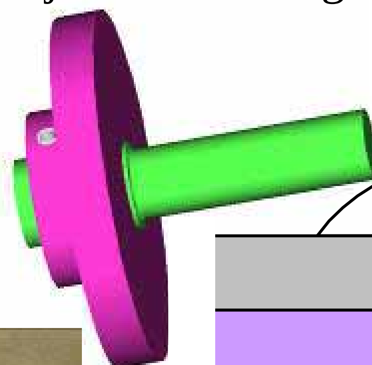


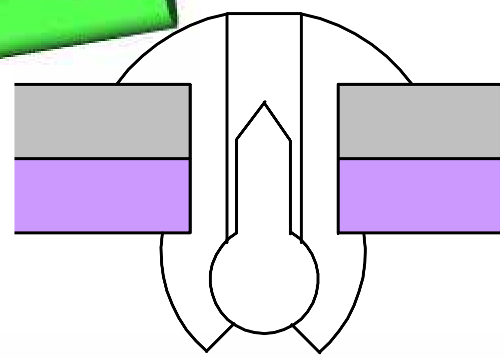
Structural Joints



- *Structural joints* (non moving) transfer loads between members, and are a necessary part of almost all structures
  - They can take up space and add cost
  - They can provide damping and design flexibility
- There are many different types of joints including:
  - Welded
  - Adhesive
  - Bolted
  - Pinned & Riveted
  - Press-fit (see page 5-25)

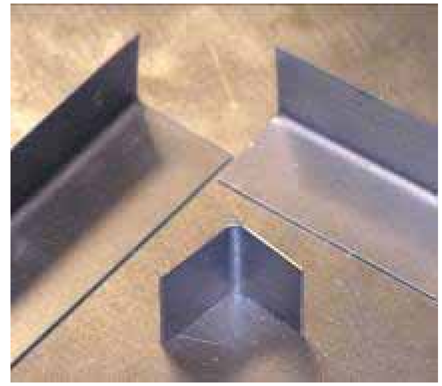




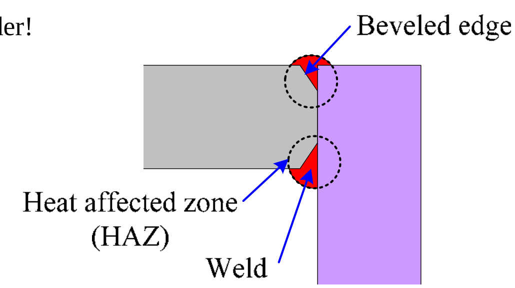


Structural Joints: *Welded*

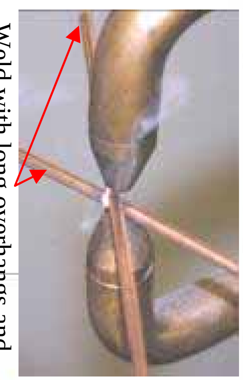
- A good weld is as strong as the base metal
  - Surface preparation is VERY IMPORTANT
    - Cleanliness
    - On thicker parts, bevel edges to be welded
- Shop personnel will help you with your welding needs
  - Consult with them during the **concept** stage about options
  - Spot welding is used for sheet metal and thin rods
  - Arc welding is typically used for heavier sections
  - TIG (*Tungsten Inert Gas*) is used for welding aluminum, or for very precise welds on steels and special alloys

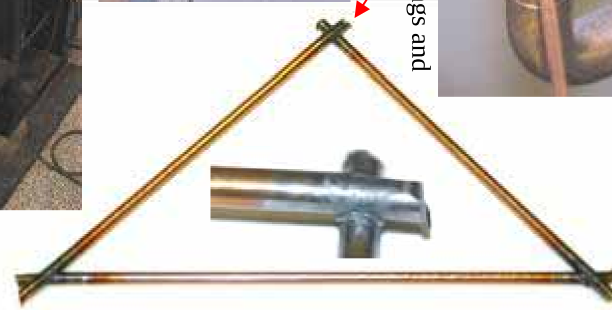


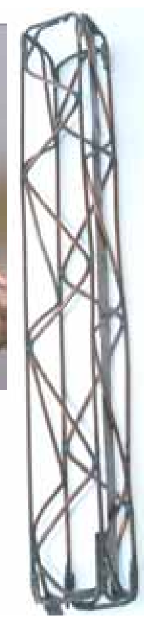
Steve Haberek, master welder!



Weld with long overhangs and then trim

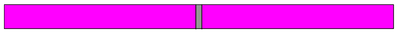






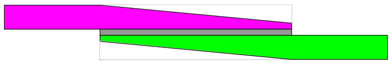
Structural Joints: *Adhesive*

Butt Joint: OK



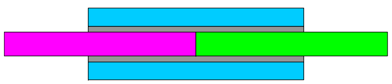
Lap Joint: Good

Tapered Lap Joint: Very Good

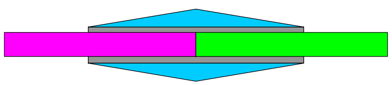


Stepped Lap Joint: Very Good

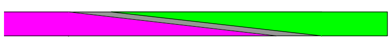
Double Strap Joint: Very Good



Tapered Double Strap Joint: Excellent

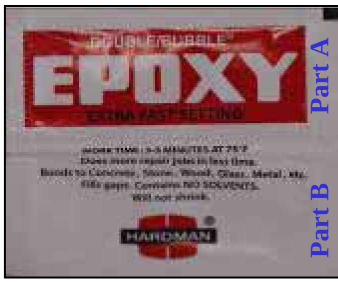


Scarf Joint: Excellent

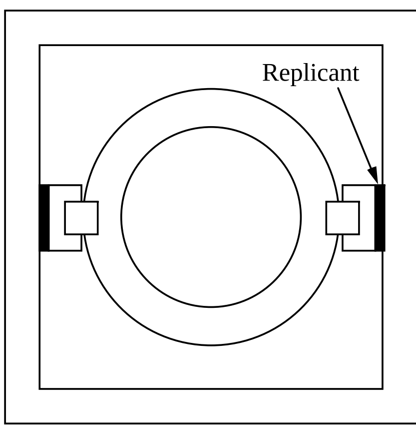


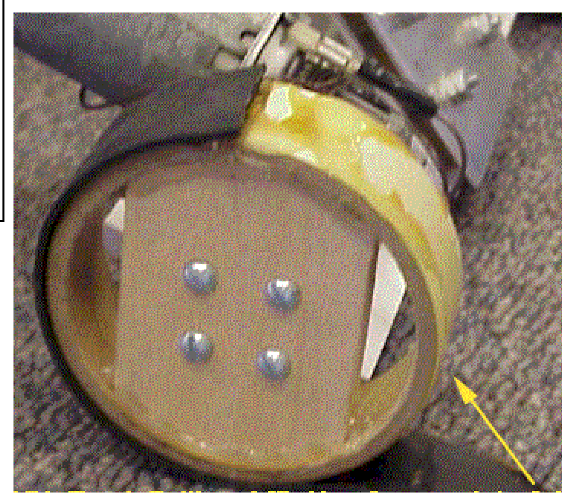
Single strap joint used to bond toothed belt with Super glue

- Adhesives are often used to bond large surface areas
  - Epoxy are often used for making laminates
  - Adhesive joints are usually not meant to be moment connections
  - Thread locking agents are used to keep screw threads from coming undone
  - ***CLEANLINESS IS OF UTMOST IMPORTANCE***
  - Check out binding reccomendations: http://www.thistothat.com/
- Strengths vary greatly with the type of adhesive, but the lap shear strength is typically are on the order of 15 MPa at 80 ^{o}F
  - K. Lewis, “Bonds That Take a Beating”, *Machine Design*, Aug. 8, 2002 pp 69-72



“Double Bubble” two-part epoxy. Make sure to squeeze out all the material from BOTH packets



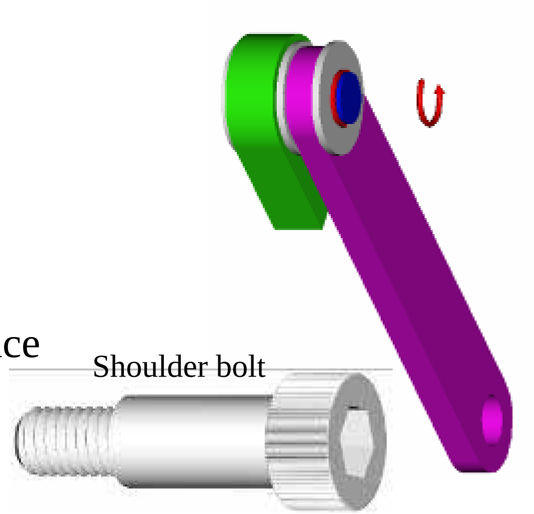


Improper surface preparation (rubber should be clean and rough), and the rubber should have been scarf joined

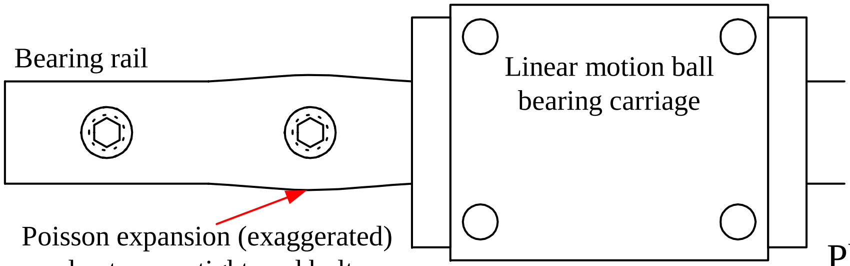
Aluminum epoxied to both sides of plywood which acts as a core

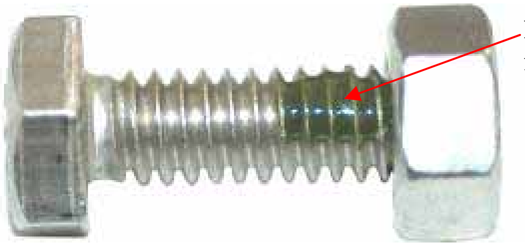
Aluminum epoxied to one side of plywood

Structural Joints: *Bolted*



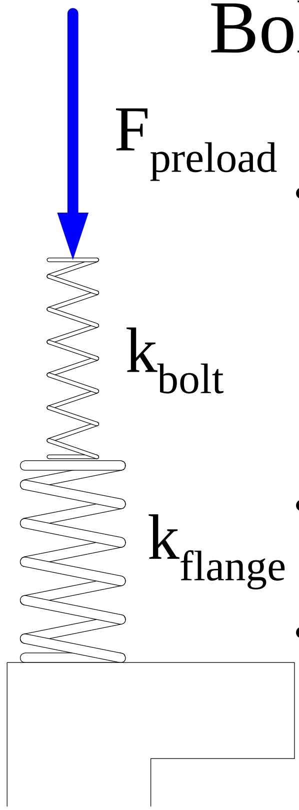
- Bolts and screws ONLY clamp one element to another!
  - Friction and the clamping force are what hold the joint together
  - Washers are used to keep hex-nut edges from chewing up the surface
- Bolts and screws DO NOT themselves take shear loads
  - Unless you use a shoulder bolt
- A shoulder bolt can act as a shaft or element of a linkage (pin):
  - When a bolt is to be used to support a bearing, or act as an axle (pin) in a linkage:
    - One end of the bolt must be firmly anchored so it is preloaded and rigid
    - The cantilevered end ideally has a precision ground shoulder that acts as an axle

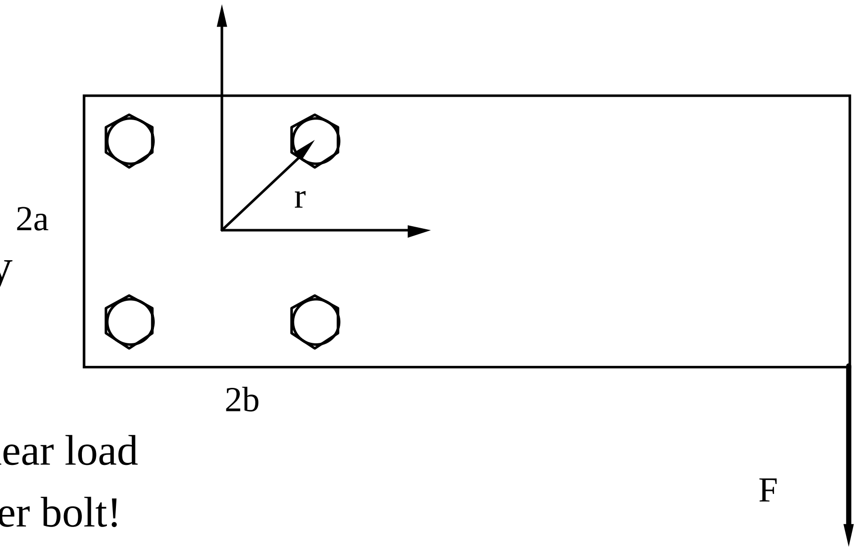




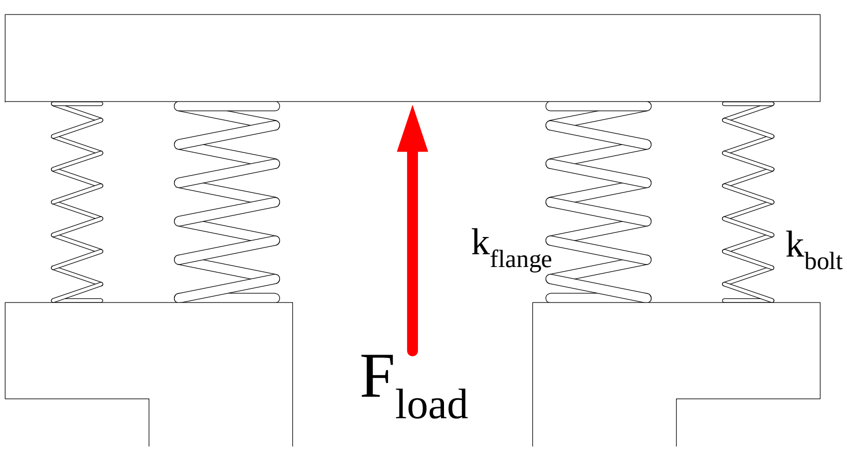
JointCompliance.xls	
Spreadsheet to estimate bevel gear tooth strength	
Production gears must be designed using AGMA standards	
Last modified 9/8/2003 by Alex Slocum	
Enter numbers in **BOLD,** results in **RED**	
Be consistant with units! (in, lb or N, m or N, mm)	
Angle of cone of influence, theta (degrees)	**45**
Diamter of bolt head, Done	**20**
Diameter of affected zone, Dtwo	32
Thickness of material, t	**6**
Diameter of bore for bolt, Dbore	**12**
Modulus of elasticity, E	**200000**
Compliance, C	3.24941E-07
Stiffness, k	3.08E+06
From FEA	
Load (4x load for quadrant)	**1000**
displacement	**4.55E-04**
resulting stiffness, k	2.20E+06
Ktheory/Kfea	1.40
With shear stiffness term (assumes theta = 45 degrees)	
Poisson ratio, n	**0.29**
Shear stiffness	4216149
Total stiffness (1/(1/compression + 1/shear))	1.78E+06
Ktheory/Kfea	0.81

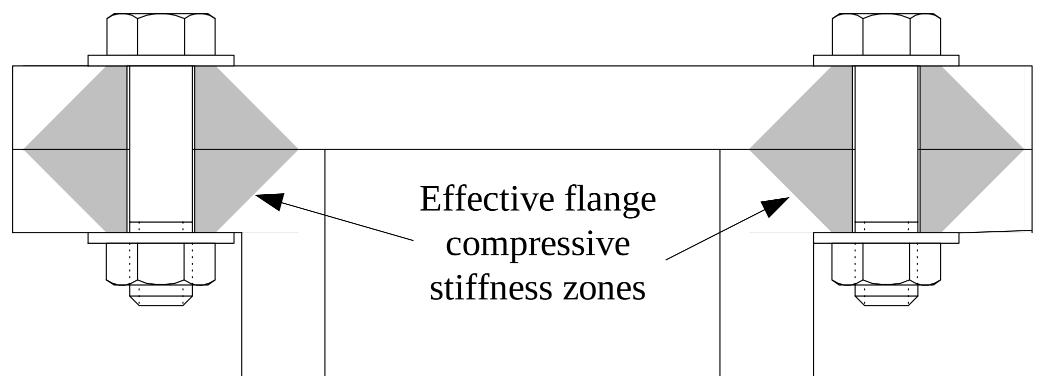
Bolted Joints: *Mechanics*





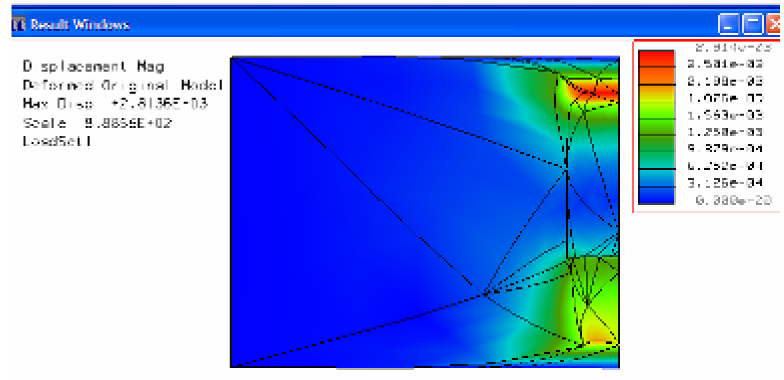
- Bolted joints resist shear ONLY by clamping action and friction!
  - Lubrication of threads is critical
  - NEVER rely on a bolt to take a shear load
    - UNLESS the bolt is a shoulder bolt!
- Shear and moment capacity: Find the center of the bolt pattern, and compute the moments about it
- Bolts act in parallel with the stiffness of the joint
  - By tightening them to create a preload higher than the applied load, the effects of alternating stresses created by a load are reduced



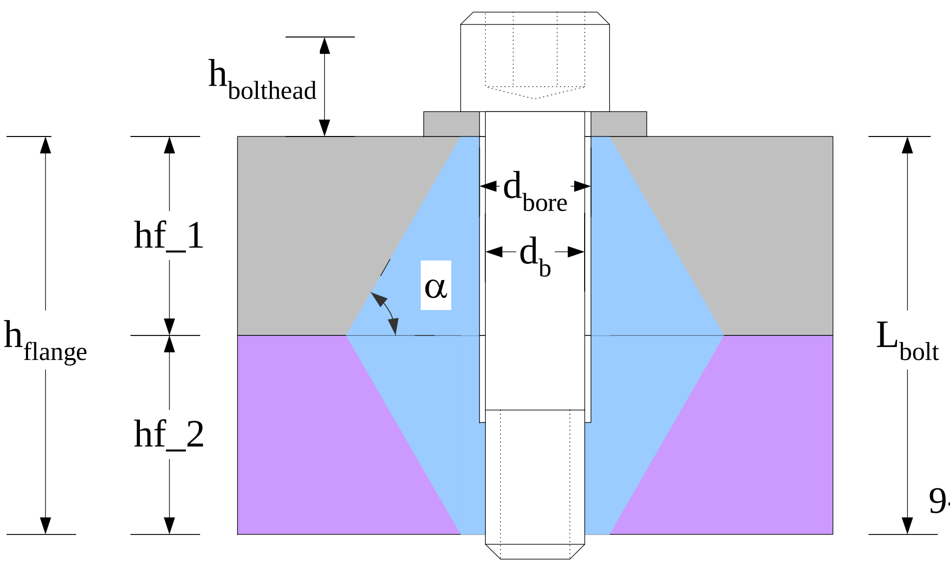


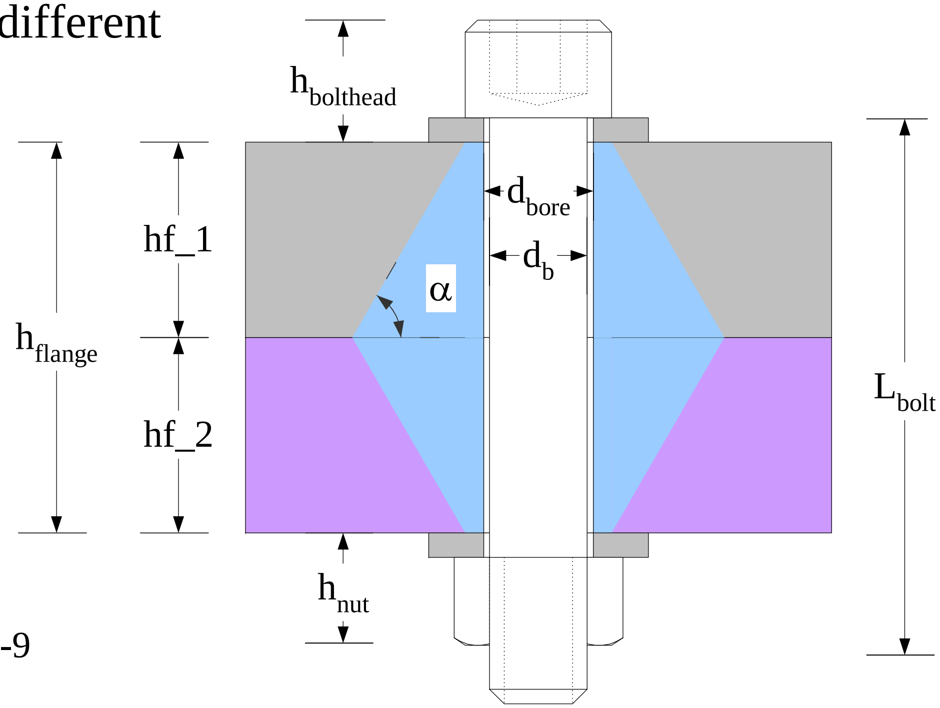
Bolted Joints: *Stiffness*

- As bolts are tightened (preloaded), their stiffness acts in series with the flange stiffness
- As external loads are applied to the joint, bolts' stiffness acts in parallel with flange stiffness
- Preloading bolts allows large loads to be applied to a joint while minimally affecting the bolt stress
- A joint can be designed so it "leaks" before a bolt breaks
  - Make the stress cones overlap!
- *Bolt_preload.xls* lets you experiment with different dimensions!

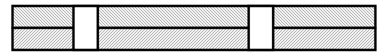


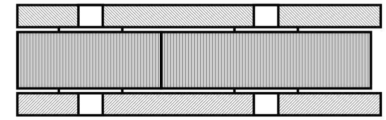
FEA results compared to analysis (60^{o} stress cones)	
Applied load (N)	**4000**
deflection under bolt head (mm)	**0.002810**
deflection from threaded region (mm)	**0.002100**
Total deflection (mm)	**0.004910**
Stiffness (N/mm)	**814664**
FEA/Analytical	**0.86**



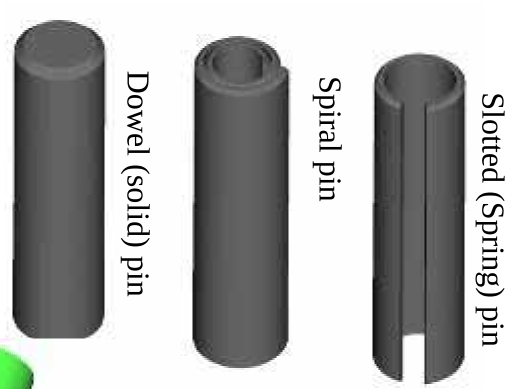


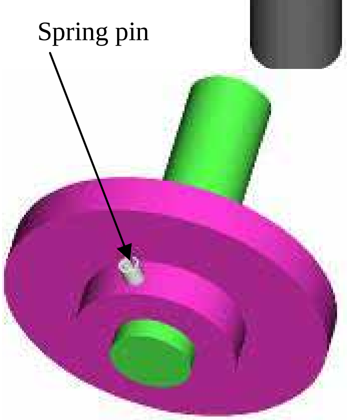
Structural Joints: *Pinned & Riveted*

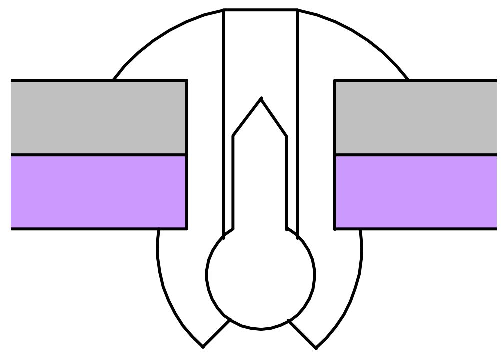


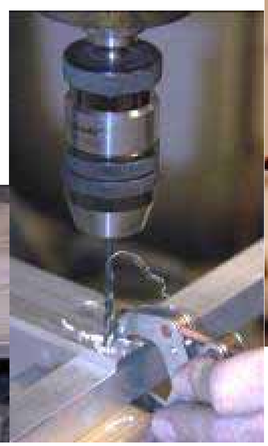


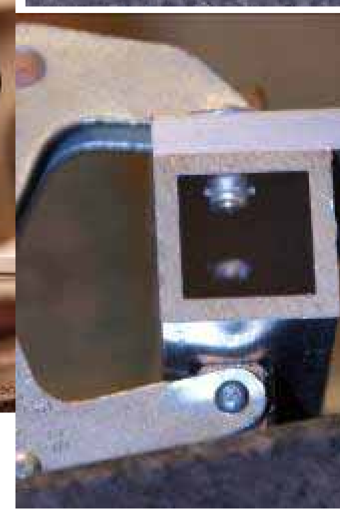
- Pinned joints use pins pressed into holes to transmit forces (or torque) ***<u>(see page 5-25)</u>***
- Pinning parts together can help during alignment during manufacturing or assembly
  - Line-bore holes for shafts and bearings by pinning plates together and drilling all the holes at once!
- A riveted joint uses expanded members to transmit shear forces and resist peeling forces
  - The expanding nature of the rivet allows many holes to be drilled in parts to be fastened together



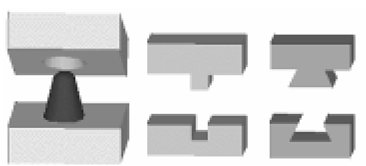






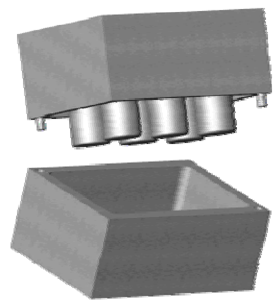


Structural Interfaces



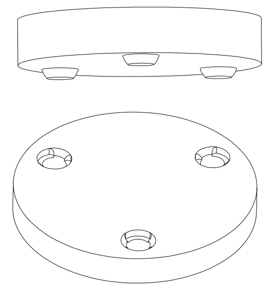
Keys

Over Constrained



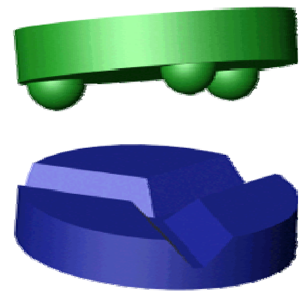
Pinned Joints

Often over Constrained



Quasi-Kinematic Couplings

Near Kinematic Constraint



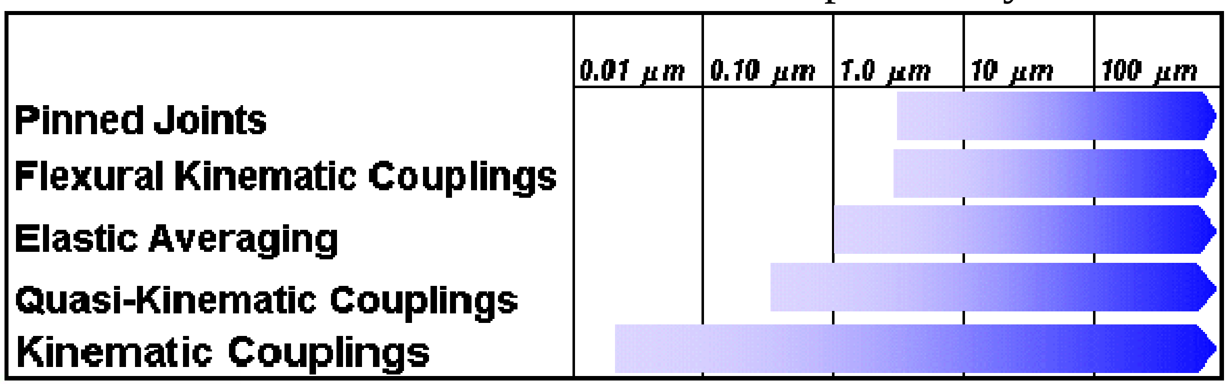
Kinematic Couplings

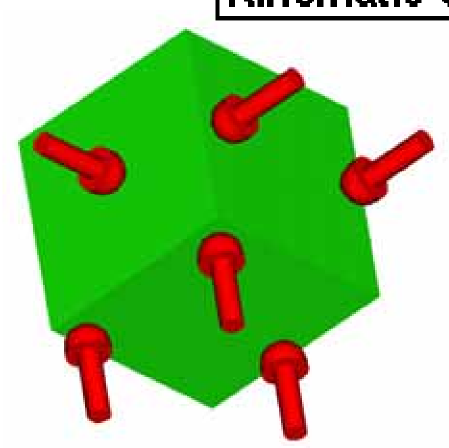
Exact Constraint

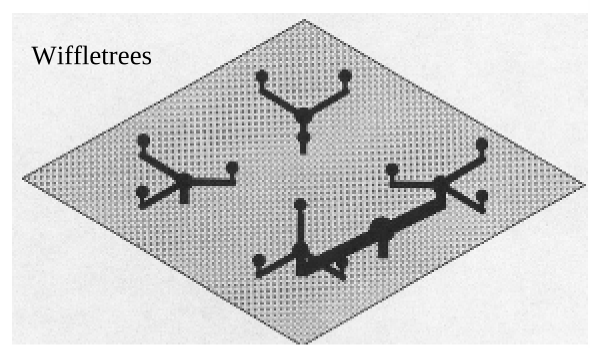
Flexural Kin. Couplings

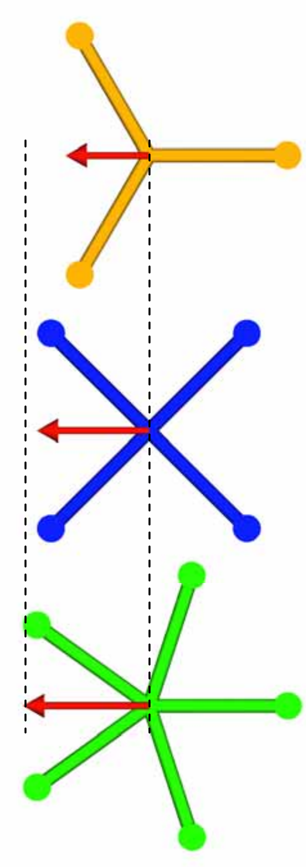
Exact Constraint

Repeatability







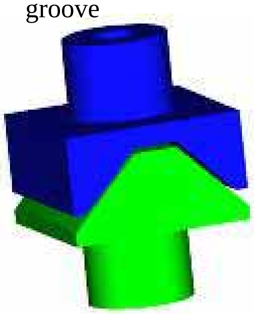


Hertz Contact

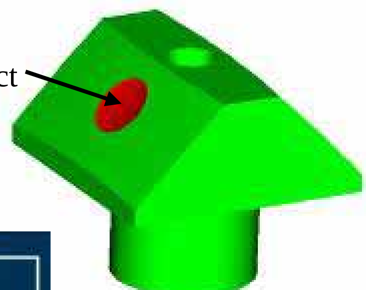
Crowned cone

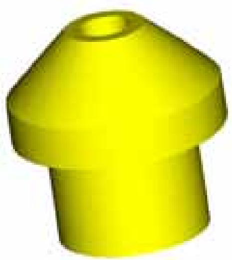
- A most important aspect of interface design are the stresses at the contact points
- In the 1800's, railroad ***wheels*** were damaging tracks, and rolling element bearing designs were very limited
  - Heinrich Hertz, the mathematician famous for his work in the frequency domain, created the first analytical solution for determining the stress between two bodies in point contact
  - *Hertz_Contact_point.xls, Hertz_Contact_line.xls, Kinematic_Coupling_3Groove_Design.xls*

Canoe ball and vee groove

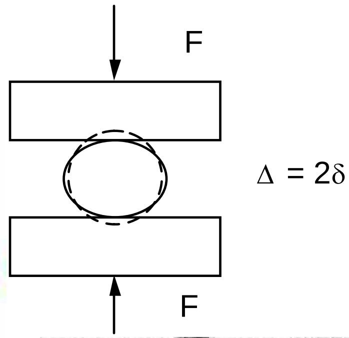


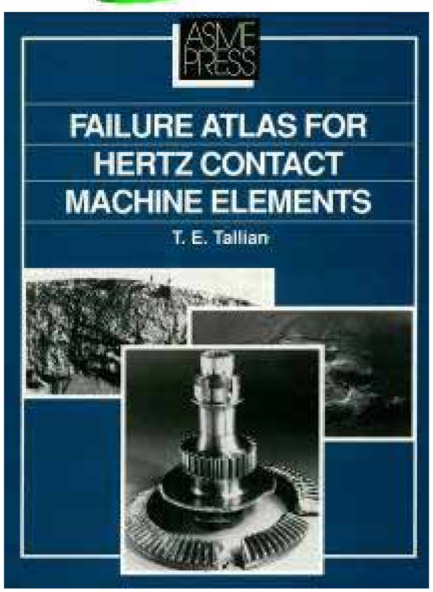
Hertz contact zone





Crowned cone

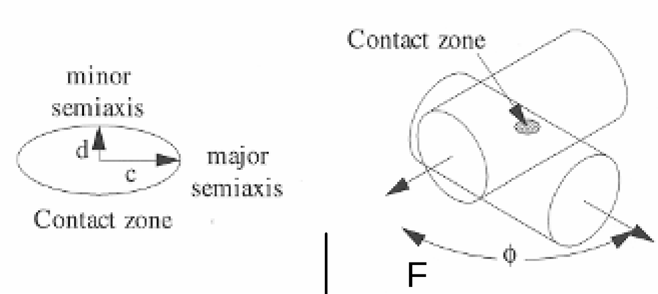




Heinrich Hertz 1857-1894

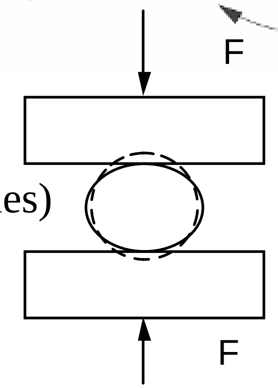
HertzContact.xls	
To determine Hertz contact stress between bodies	
By Alex Slocum, Last modified 1/17/2004 by Alex Slocum	
Last modified 12/28/03 by Alex Slocum	
Enters numbers in **BOLD,** Results in **RED**	
Be consistent with units!!	
Ronemaj	**1.00E+06**
Ronemin	**1.00E+06**
Rtwomaj	**0.500**
Rtwomin	**0.500**
Applied load F	**4,358**
Phi (degrees)	**0**
Ultimate tensile stress	**3.45E+08**
Elastic modulus Eone	**1.93E+11**
Elastic modulus Etwo	**1.93E+11**
Poisson's ratio vone	**0.29**
Poisson's ratio vtwo	**0.29**
Equivelent modulus Ee	**1.05E+11**
Equivelent radius Re	**0.2500**
ellipse c	**2.50E-03**
ellipse d	**2.50E-03**
Contact pressure, q	**3.33E+08**
Stress ratio (must be less than 1)	**0.97**
Deflection at the one contact interface	
Deflection (μunits)	**12.4**
Stiffness (load/μunits)	**350.8**
for circular contact a = c, a	**2.50E-03**
Depth at maximum shear stress/a	**0.634**
Max shear stress/ultimate tensile	**0.324**

Hertz Contact: *Point Contact*



- Equivalent radius R_e and modulus E_e (ν is Poisson ratio):

$$R_e = \frac{1}{\dfrac{1}{R_{1\,\mathrm{major}}} + \dfrac{1}{R_{1\,\mathrm{minor}}} + \dfrac{1}{R_{2\,\mathrm{major}}} + \dfrac{1}{R_{2\,\mathrm{minor}}}} \qquad E_e = \frac{1}{\dfrac{1-\nu_1^2}{E_1} + \dfrac{1-\nu_2^2}{E_2}}$$



$\Delta = 2\delta$

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

$$\cos\theta = R_e\sqrt{\left(\frac{1}{\mathrm{R}_{1\,\mathrm{major}}} - \frac{1}{\mathrm{R}_{1\,\mathrm{minor}}}\right)^2 + \left(\frac{1}{\mathrm{R}_{2\,\mathrm{major}}} - \frac{1}{\mathrm{R}_{2\,\mathrm{minor}}}\right)^2 + 2\left(\frac{1}{\mathrm{R}_{1\,\mathrm{major}}} - \frac{1}{\mathrm{R}_{1\,\mathrm{minor}}}\right)\left(\frac{1}{\mathrm{R}_{2\,\mathrm{major}}} - \frac{1}{\mathrm{R}_{2\,\mathrm{minor}}}\right)\cos 2\phi}$$

- Elliptic Integrals

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

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

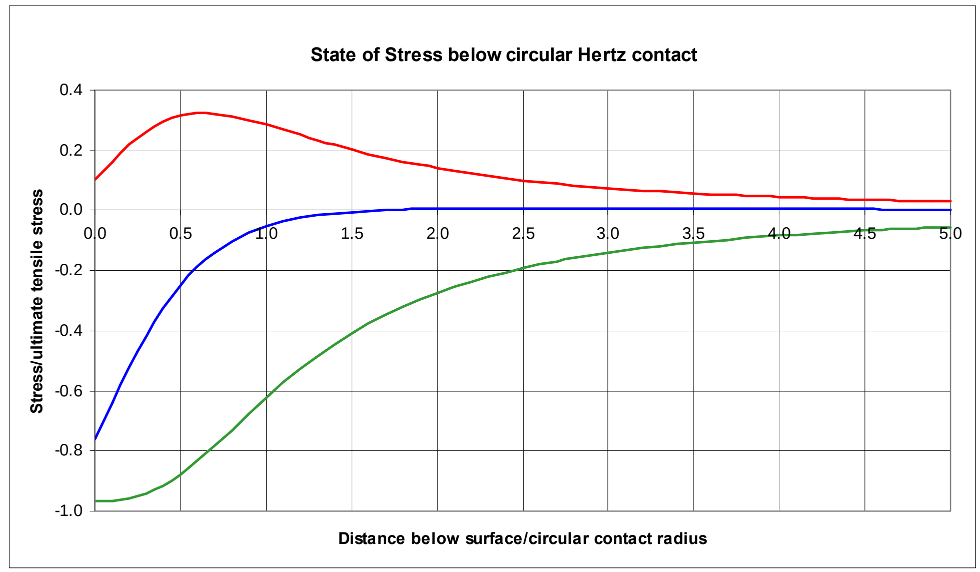
$$\lambda = -0.214e^{-4.95\theta} - 0.179\theta^2 + 0.555\theta + 0.319$$

- Major and minor contact area elliptical semi-axes

$$c = \alpha\left(\frac{3FR_e}{2E_e}\right)^{1/3} \qquad d = \beta\left(\frac{3FR_e}{2E_e}\right)^{1/3}$$

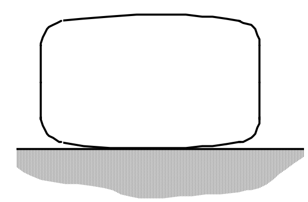
- Contact pressure & Deflection

$$q = \frac{3F}{2\pi cd} \le 1.5\sigma_{\text{for metals, ultimate tensile strength}} \qquad \delta = \lambda\left(\frac{2F^2}{3R_eE_e^2}\right)^{1/3}$$

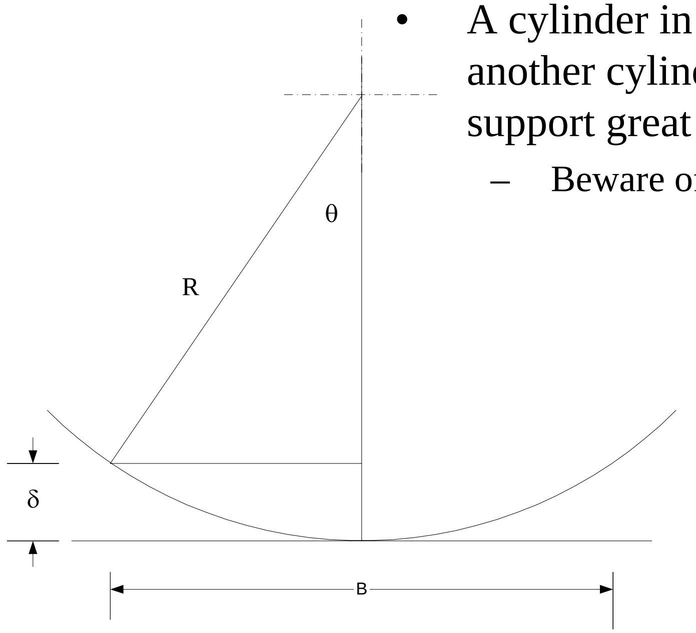


- State of stress for circular contact of radius a as a function of depth z below the surface

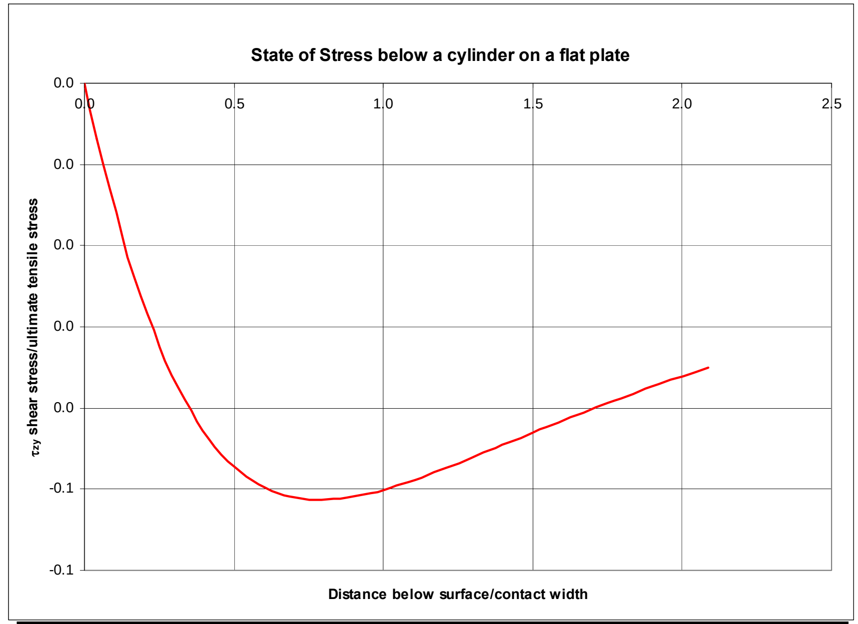
$$\sigma_z(z) = q\left(-1 + \frac{z^3}{\left(a^2+z^2\right)^{1.5}}\right) \qquad \sigma_r(z) = \sigma_\theta(z) = \frac{q}{2}\left(-(1+2\nu) + \frac{2(1+\nu)z}{\sqrt{a^2+z^2}} - \frac{z^3}{\left(a^2+z^2\right)^{1.5}}\right) \qquad \tau(z) = \frac{\sigma_\theta(z) - \sigma_z(z)}{2}$$



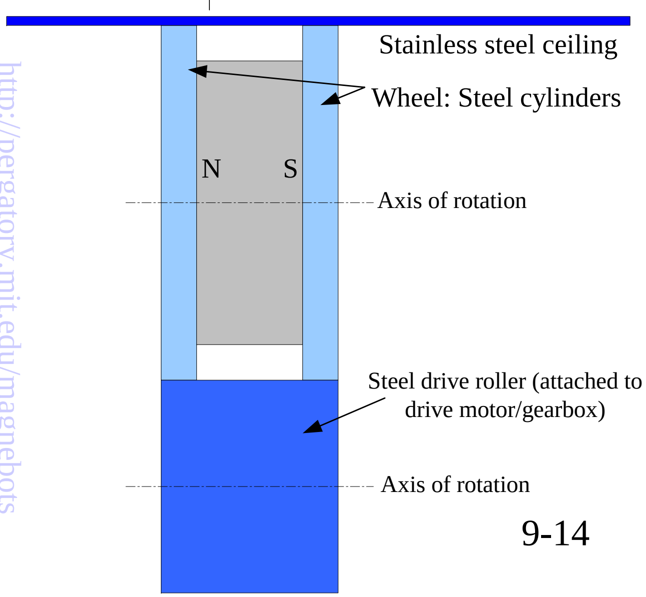
Hertz Contact: *Line Contact*



- A cylinder in contact with another cylinder or a plate can support great forces
  - Beware of edge loading!

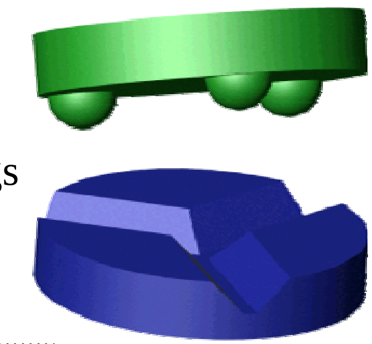


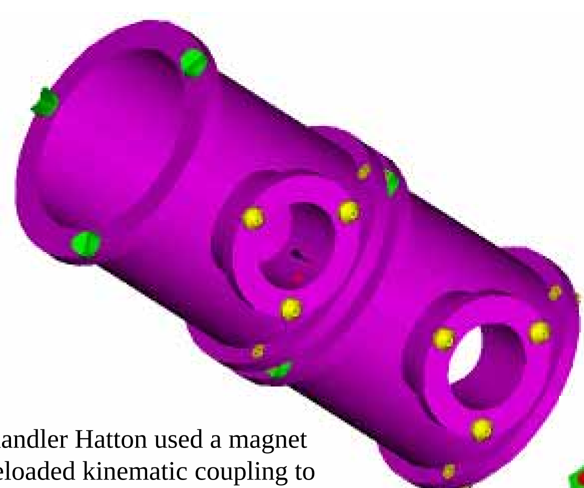
Hertz_contact_line.xls	
To determine Hertz contact stress between two cylinders	
By Alex Slocum, last modified 2/10/2004 by Alex Slocum	
Enters numbers in **BOLD**, Results in **RED**	
Smaller cylinder 1 diameter, d_1 (mm)	**10**
Larger cylinder 2 (or flat plane) diameter, d_2 (mm)	**100**
Length, L (mm)	**10**
Applied load, F (N)	**8,184**
Elastic modulus Eone (N/mm^2)	**2.00E+05**
Elastic modulus Etwo (N/mm^2)	**2.00E+05**
Poisson's ratio vone	**0.29**
Poisson's ratio vtwo	**0.29**
Ultimate tensile stress, sigult (N/mm^2)	**1500**
Depth below contact surface for evaluating deflection, do	**300**
Rectangular contact zone width, 2b (mm)	0.42
Contact pressure, qcyl (N/mm^2)	2502
Deflection motion of d_1 center, defl_1 (mm)	0.0104
Deflection motion of d_2 center, defl_2 (mm)	0.013159
Total relative displacement of the cylinder's centers, dcyls (mm)	0.0236
Stress factor: Must be less than 1	
Maximum shear stress/(ultimate tensile/2)	1.00
Manufacturing issues	
Surface roughness, Ra (mm)	**0.005**
Potential induced contact width, Bra (mm)	0.4



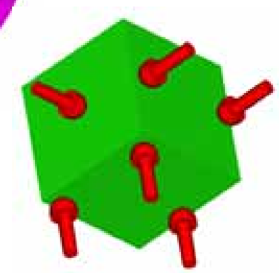
Kinematic Couplings

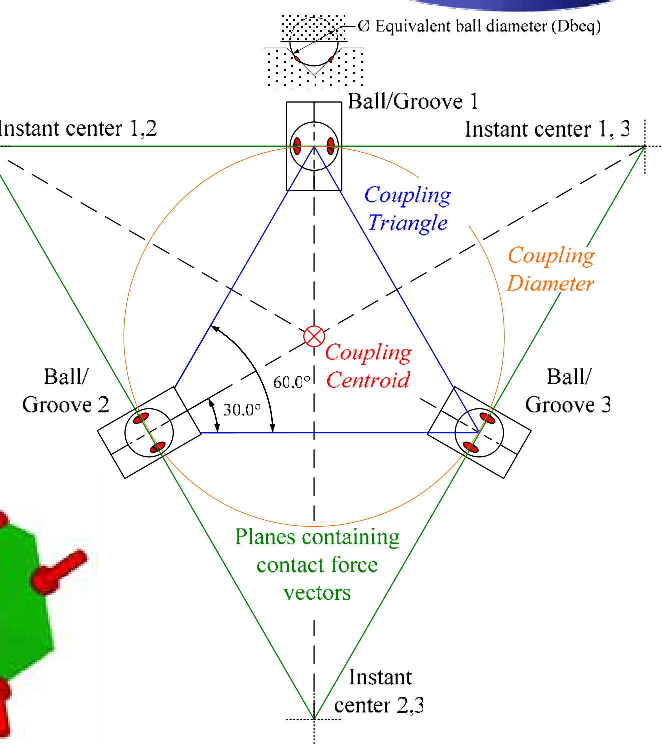
- When a component is constrained by a number of points equal to the number of degrees of freedom, it is said to be *exactly constrained*
  - Kinematics is the study of motion, assuming bodies are rigid, so when a design is "kinematic" it means it is exactly constrained, and geometric equations can be written to describe its motion
- Kinematic Couplings are couplings that exactly constrain components
  - They are not stable unless ALL six contact points are engaged
  - There are no intermediate stability configurations like those in 3-2-1 couplings
  - They can provide repeatability on the order of parts' surface finish
  - ¼ micron repeatability is common
- Managing the Hertz contact stresses!

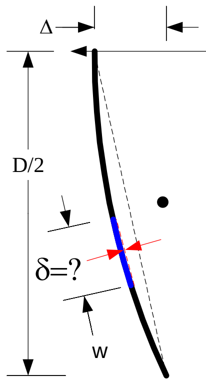




Chandler Hatton used a magnet preloaded kinematic coupling to enable her machine's module to be easily flipped depending which side of the table on which she had to setup

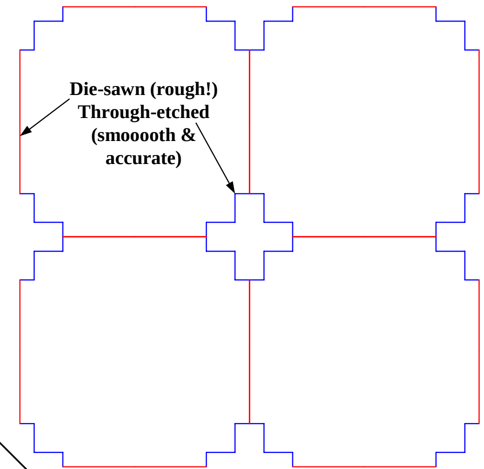


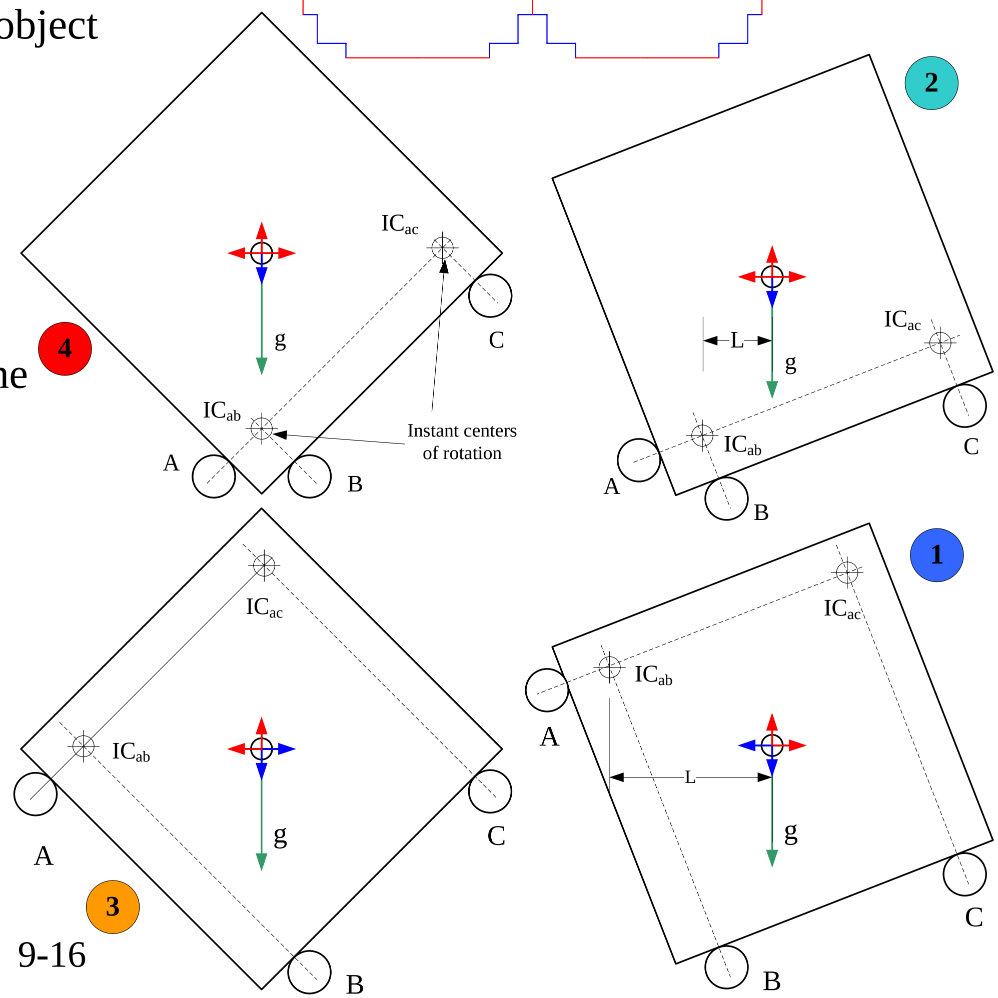


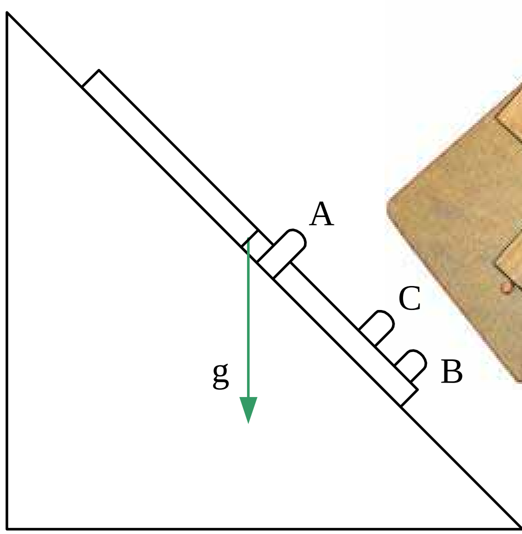


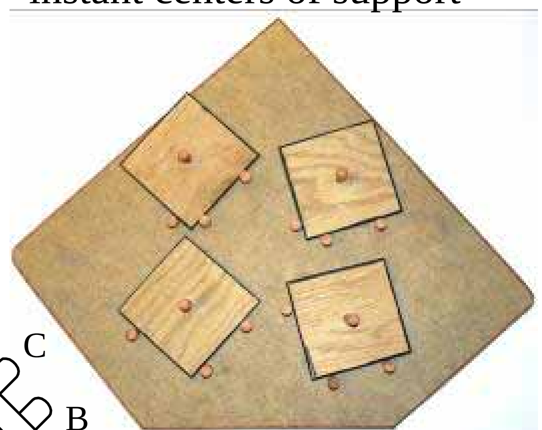
Kinematic Couplings: *2D*

- How to fixture a 2D object, such as a silicon MEMS chip, so several could be stacked upon each other for bonding?
  - 3 DOF (translation, pitch & roll) are defined by the plane on which the object rests
  - 3 DOF (2 translations and yaw) must be established
    - 3 contact points are needed
    - Gravity provides preload
    - Align the gravity vector wrt the instant centers of support







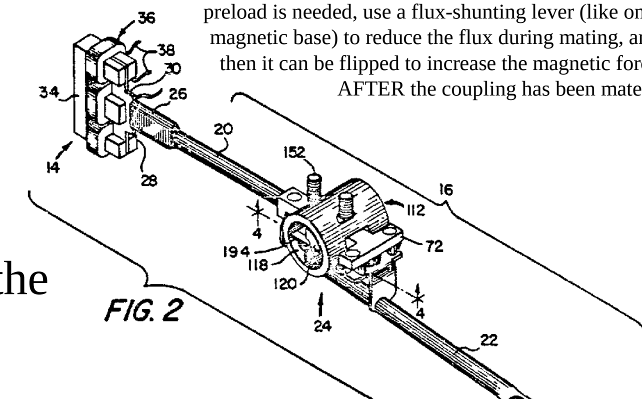


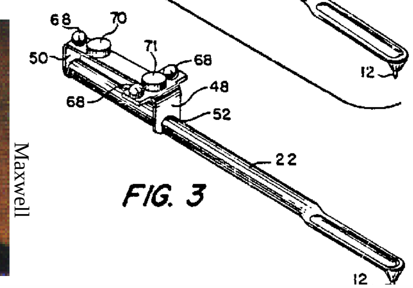
Wooden bench level experiment

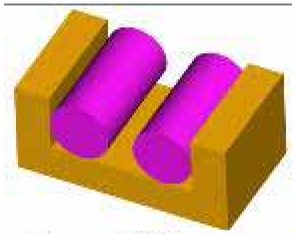
See US patent 4,574,625. NOTE magnet preload needs to be applied gently else the sudden THWAP (impact) of contacts drawn together can cause subsurface failure or surface indentation (*Brinelling*); hence if a LOT of preload is needed, use a flux-shunting lever (like on a magnetic base) to reduce the flux during mating, and then it can be flipped to increase the magnetic force AFTER the coupling has been mated!

Kinematic Couplings: *3D*

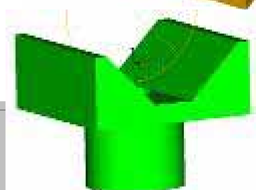
- James Clerk Maxwell (1831-1879) liked the three-grooves
  - Symmetry good for manufacture, dynamic stability
  - Easy to obtain very high load capacity
- William Thomson (later Lord Kelvin) (1824 - 1907) liked the ball-groove-tetrahedron
  - More intuitive, and applicable to non-planar designs

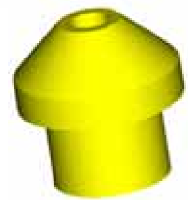


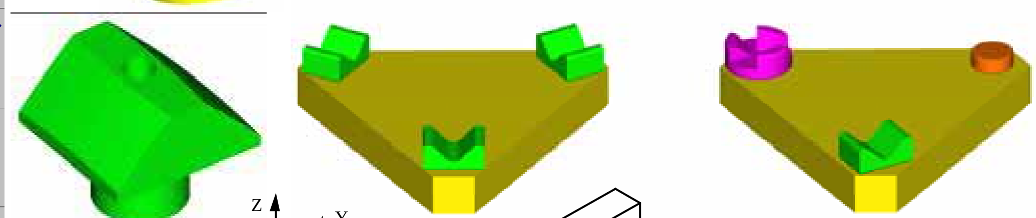


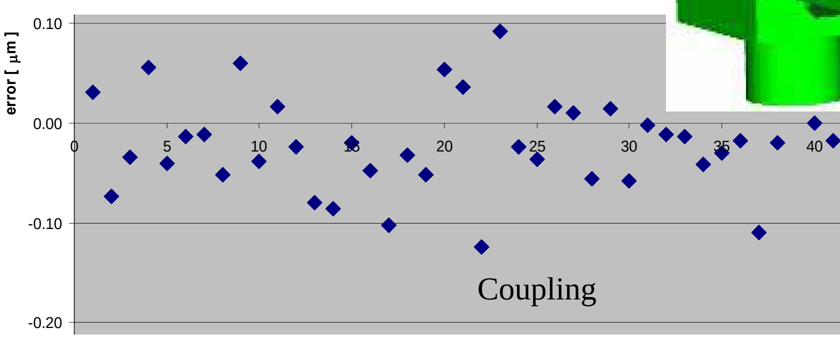


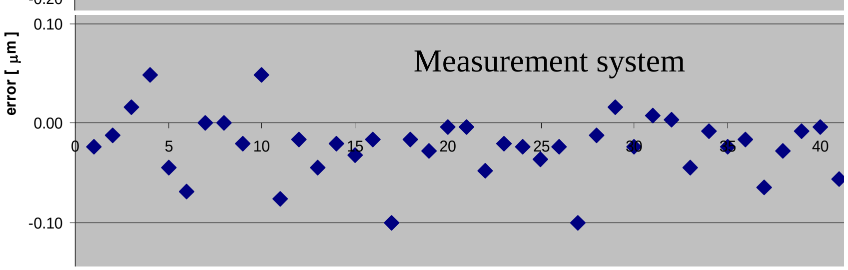
Maxwell

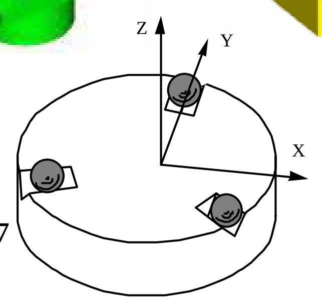


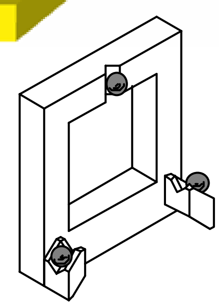


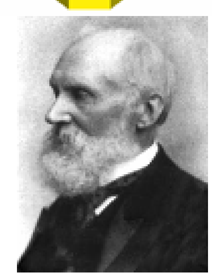








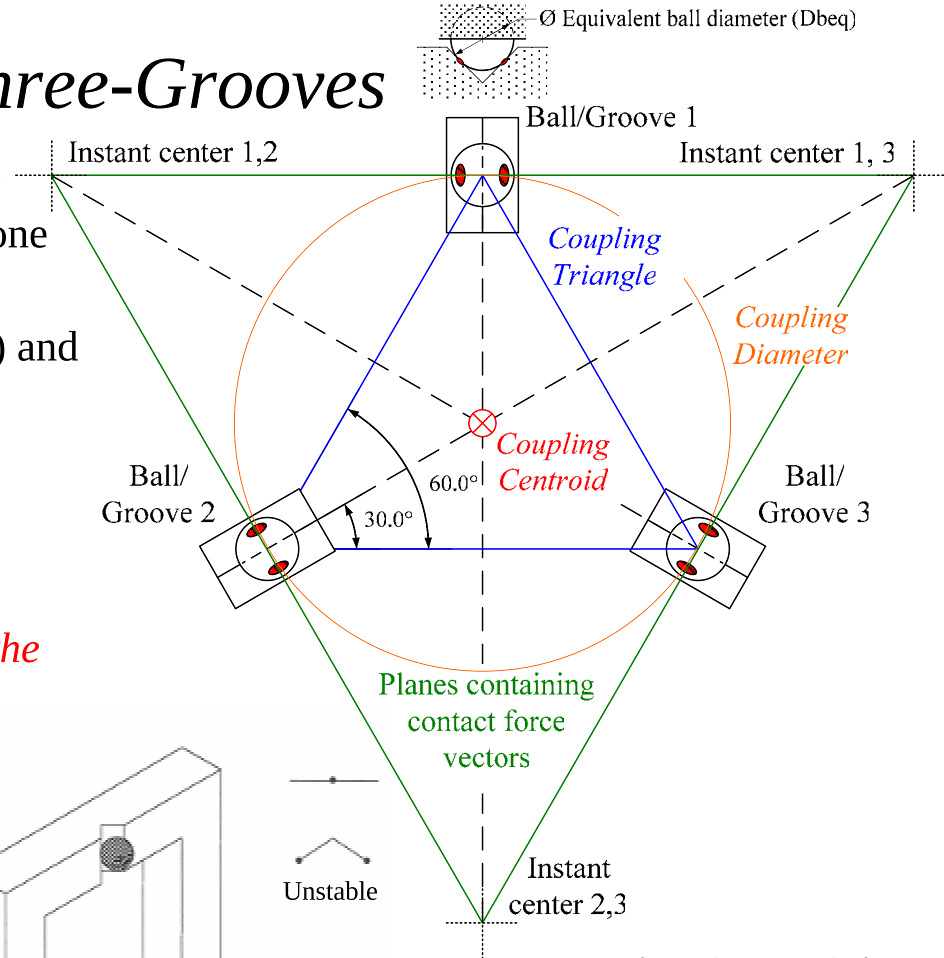


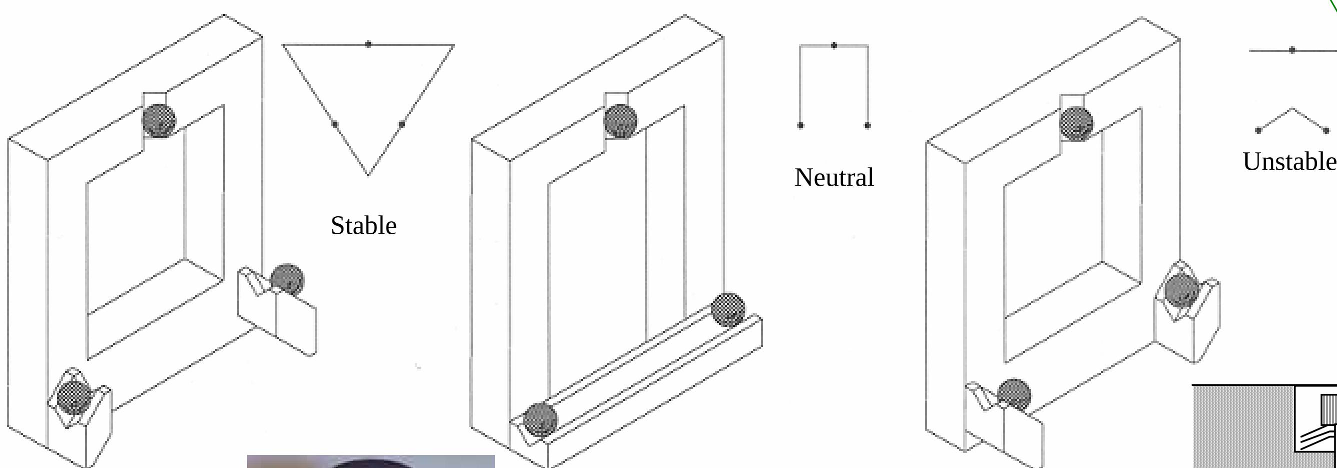


Kelvin

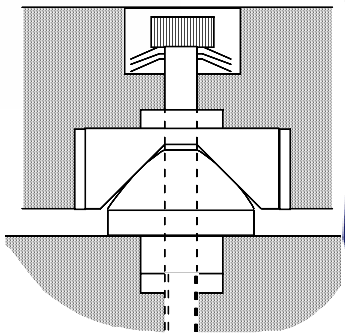
Kinematic Couplings: *Three-Grooves*

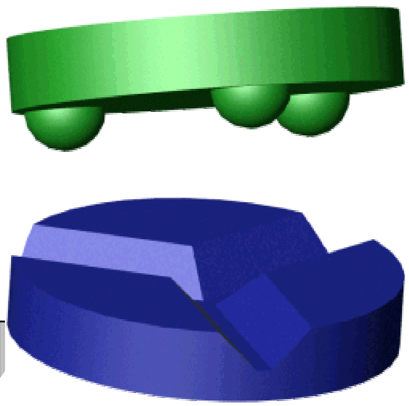
- For long life, Hertz contact pressure $q < \sigma_{yield}$
  - Contact area center should not be closer than one contact ellipse diameter from groove edge
  - Materials must be non-galling (no AL on AL!) and non-fretting
  - Preload to keep coupling from tipping
- *Ideally align the grooves with the coupling triangle's angle bisectors*
  - *The coupling centroid will NOT always be at the coupling triangle centroid!*





Don't forget the potential of using magnets for light load applications!





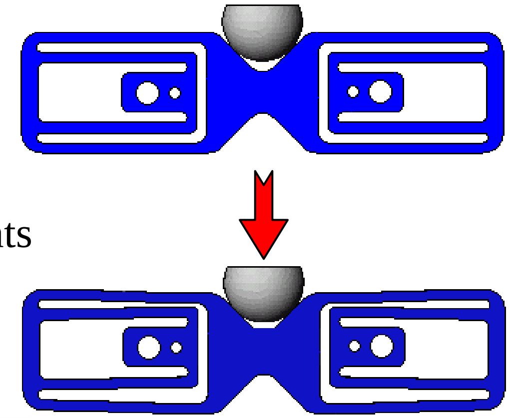
See www.kinematiccouplings.org for spreadsheets, articles, and suppliers

Kinematic Couplings: *Three-Grooves*

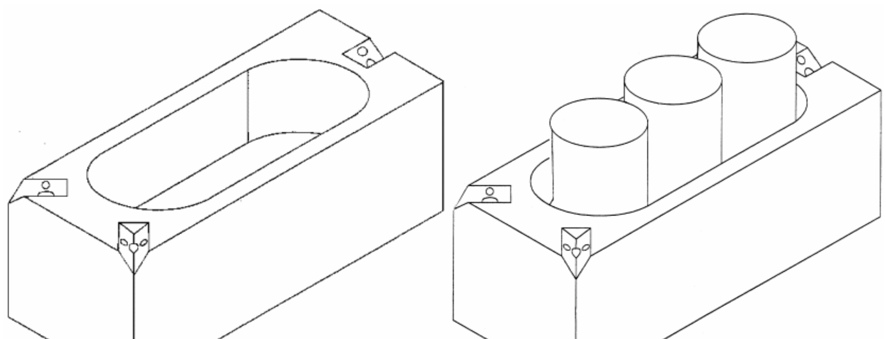
Kinematic_Coupling_3Groove_Design.xls						
To design three groove kinematic couplings						
Written by Alex Slocum. Last modified 10/27/2004 by Alex Slocum						
Metric units only! Enters numbers in **BOLD**, Results in **RED**					**Material properties**	
Standard 120 degree equal size groove coupling? (contact forces are inclined at 45 to the XY					*User defined material*	***aluminum***
plane. For non standard designs, enter geometry after results section)				**TRUE**	*Yield stress*	
System geometry (XY plane is assumed to contain the ball centers)					plastic	**3.45E+07**
Dbeq (mm) =	**5**	Equivalent diameter ball to contact the groove at the same points			RC 62 Steel	**1.72E+09**
Rbminor (mm) =	**2.5**	"Ball" minor radius			CARBIDE	**2.76E+09**
Rbmajor (mm) =	**2.5**	"Ball" major radius			user defined	**2.76E+08**
Rgroove (mm) =	**1.00E+06**	Groove radius (negative for a trough)			*Elastic modulus*	
Costheta =	**TRUE**	Is ball major radius along groove axis?			plastic	**2.07E+09**
Dcoupling (mm) =	**150**	Coupling diameter			RC 62 Steel	**2.04E+11**
Fpreload (N) =	**-100**	Preload force over each ball			CARBIDE	**3.10E+11**
Xerr (mm) =	**0.0**	X location of error reporting			user defined	**6.80E+10**
Yerr (mm) =	**0.0**	Y location of error reporting			*Poisson ratio*	
Zerr (mm) =	**0.0**	Z location of error reporting			plastic	**0.20**
Auto select material values (enter *other_4* to the right)					RC 62 Steel	**0.29**
Matlabball =	**1**	Enter 1 for plastic, 2 for steel, 3 for carbide, 4 for user defined, 5			CARBIDE	**0.30**
Matlabgroove =	**4**	where each ball and groove is defined individually			user defined	**0.29**
Min. yield strength (Pa, psi)		**3.45E+07**	**5,000**			
Largest contact ellipse major diameter (mm)		**0.831**				
Largest contact ellipse major diameter (mm)		**0.829**				
Largest contact stress ratio		**3.826**	Max Hertz shear stress/Material's max shear stress (tensile yield/2)			
RMS applied force F (N)	**17.32**					
RMS deflection at F (micron)	**2.238**					
RMS stiffness (N/micron)	**7.74**					
Applied force's Z,Y,Z values and coordinates				**Coupling centroid location**		
FLx (N) =	**10.00**	XL (mm) =	**0**	xc (mm)	**0.000**	
FLy (N) =	**10.00**	YL (mm) =	**0**	yc (mm)	**0.000**	
FLz (N) =	**10.00**	ZL (mm) =	**100**	zc (mm)	**0.000**	

Kinematic Couplings: *Compliant Mounts*

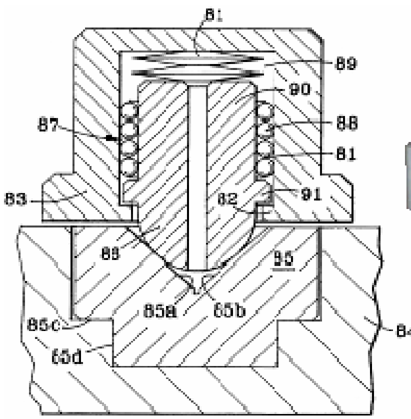
- Allows a component to be kinematically located
  - Application of the preload force deflects the kinematic components until surface-to-surface contact occurs to resist tipping loads
- Many forms from simple sheet metal to flexure-based linkages
- Deformation can be elastic, or permanent
  - Even sand cores can be aligned



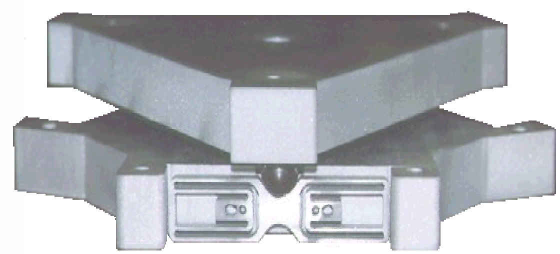
U.S. Patent 5, 678, 944

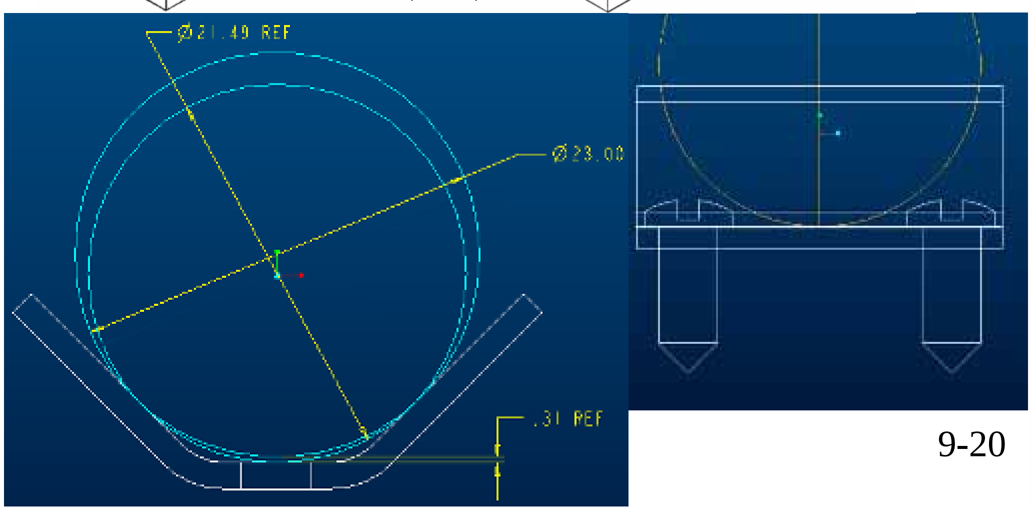


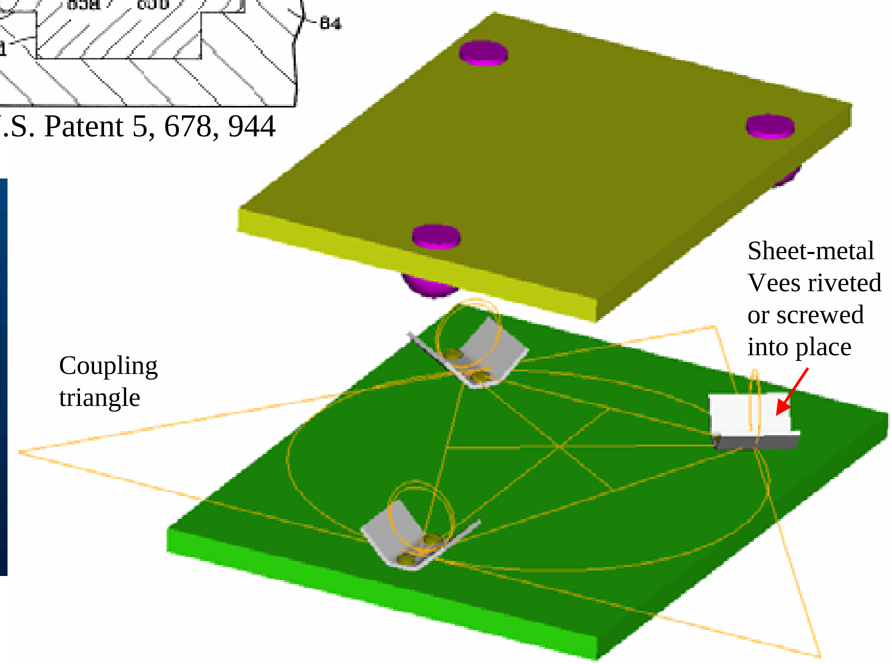
U.S. Patent 5, 769, 554



U.S. Patent 5, 678, 944

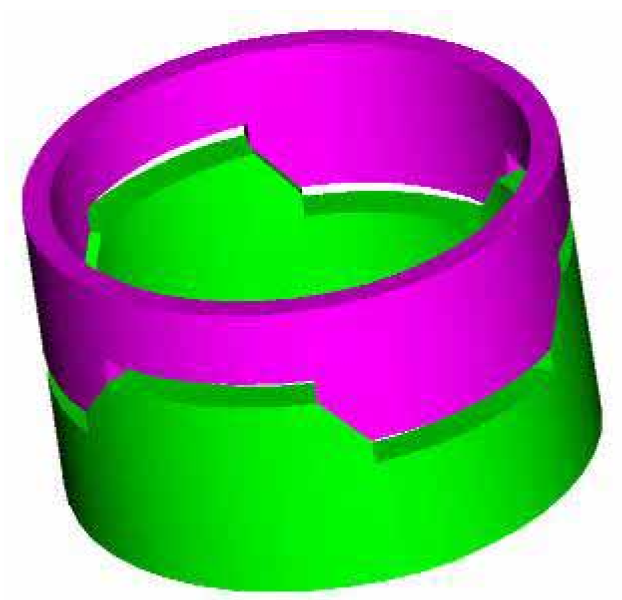






Kinematic Couplings: *Three-Tooth*

- A semi-kinematic effect can be achieved by having three teeth each on two coupling halves mate at six points
  - 3-5 micron repeatability can be obtained with this simple design
- Layton Hale at LLNL put crowns on one set of the teeth to create a nearly true kinematic three tooth coupling:
  - 1 micron repeatability can be obtained with this simple design



United States Patent [19]
Hale

[11] **Patent Number: 6,065,898**
[45] **Date of Patent: *May 23, 2000**

[54] **THREE TOOTH KINEMATIC COUPLING**

[75] Inventor: **Layton C. Hale**, Livermore, Calif.

[73] Assignee: **The Regents of the University of California**, Oakland, Calif.

[*] Notice: Under 35 U.S.C. 154(b), the term of this patent shall be extended for 634 days.

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[51] **Int. Cl.**[7] **F16D 1/00**
[52] **U.S. Cl.** **403/364**; 403/190; 403/340; 403/381; 464/157; 192/69.83
[58] **Field of Search** .... 403/190, 291, 403/364, 311, 340, 381; 192/114 T, 69.81, 69.82, 69.83; 464/149, 157

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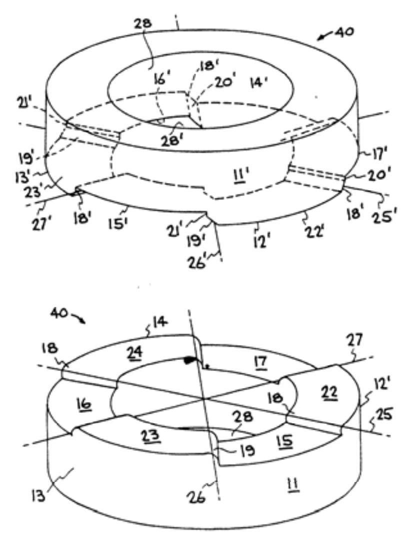
[57] **ABSTRACT**

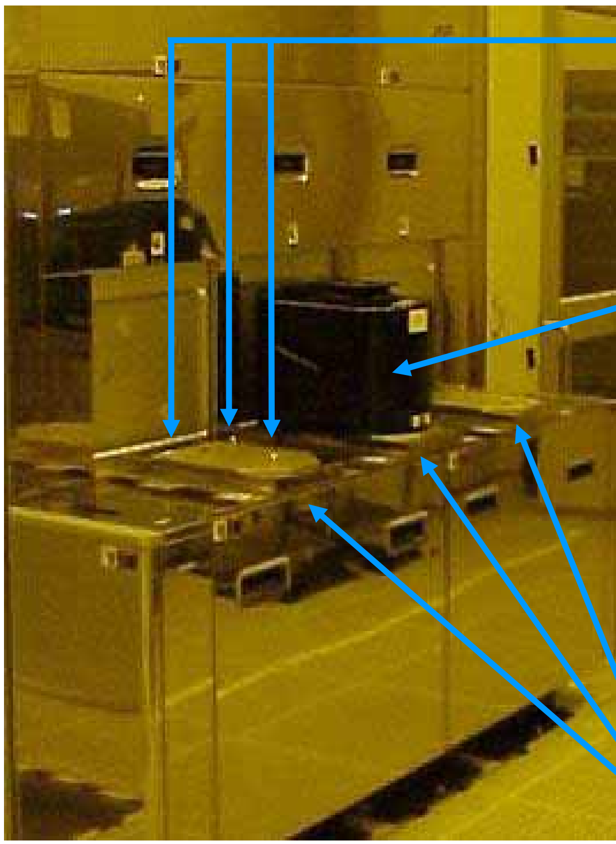
A three tooth kinematic coupling based on having three theoretical line contacts formed by mating teeth rather than six theoretical point contacts. The geometry requires one coupling half to have curved teeth and the other coupling half to have flat teeth. Each coupling half has a relieved center portion which does not effect the kinematics, but in the limit as the face width approaches zero, three line contacts become six point contacts. As a result of having line contact, a three tooth coupling has greater load capacity and stiffness. The kinematic coupling has application for use in precision fixturing for tools or workpieces, and as a registration device for a work or tool changer or for optics in various products.

15 Claims, 2 Drawing Sheets



Kinematic Couplings: *300mm Wafer Transport*

- How to precisely locate a plastic wafer carrying structure (FOUP) on a tool, so a robot can precisely load/unload wafers?
  - Exactly constrain it of course with an interface that contacts the FOUP at 6 unique points!
  - BUT success requires careful management of contact stresses, and development of standards upon which manufacturers can agree

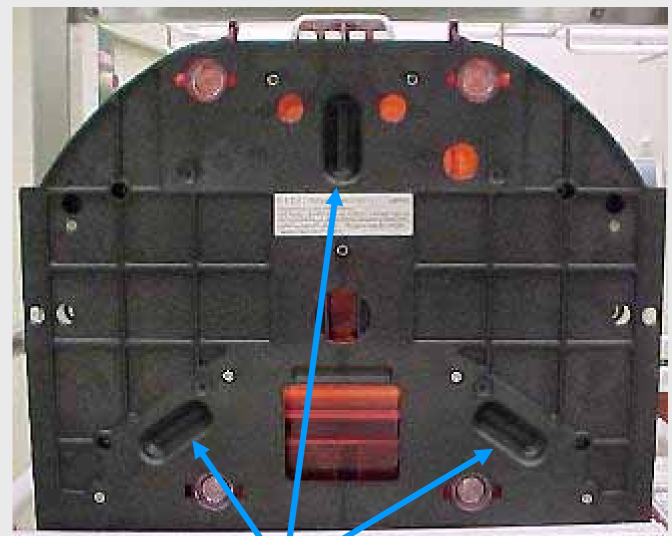


Kinematic coupling pins on loadport based on SEMI E57 standard

300mm Wafer carrier (FOUP) precisely positioned on kinematic coupling pins on loadport

Production equipment loadports based on SEMI E15.1 standard

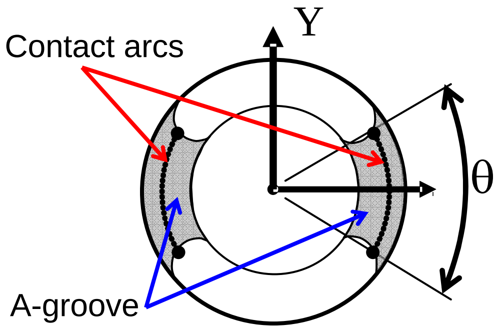
Base of the FOUP

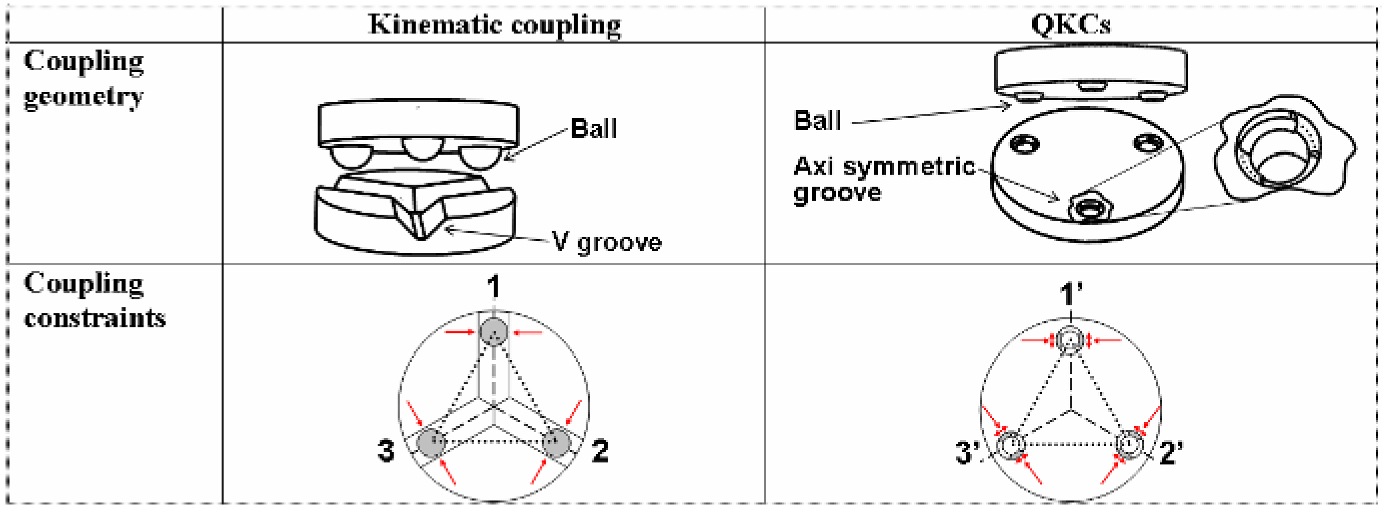


Mating kinematic coupling grooves on the FOUP, permitting precise alignment on load ports, so robots can precisely access 300 mm wafers

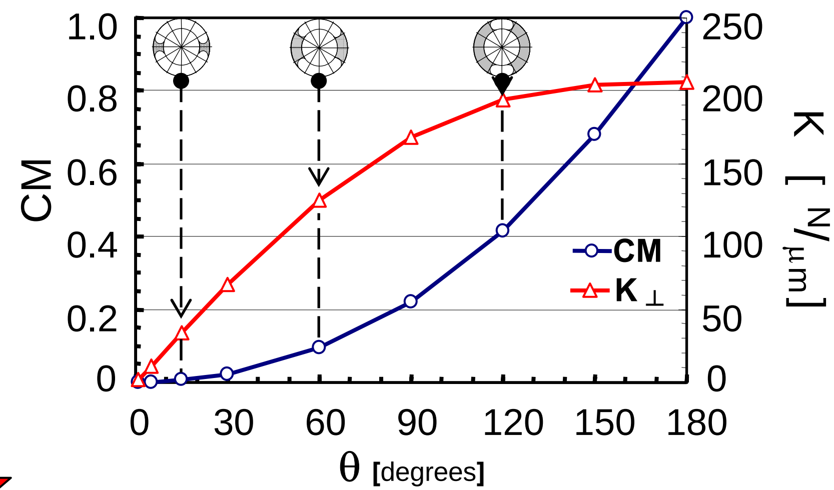
Thanks to Devadas Pillai at Intel for preparing this slide

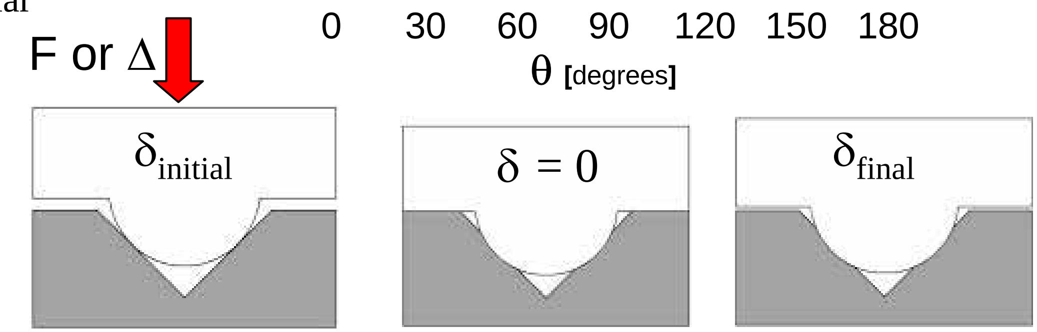
Quasi-Kinematic Couplings





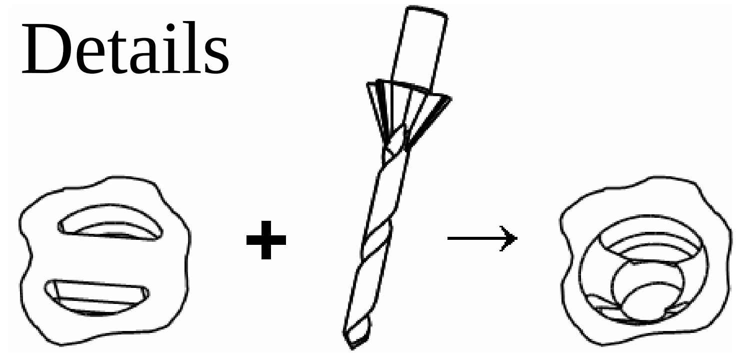
- **KC geometry**
  - 3 balls (hemispheres)
  - 3 *v-grooves*
  - 6 "point" contacts
- **QKC geometry**
  - 3 balls (hemispheres)
  - 3 *A-grooves* (surface of revolution)
  - Circular arc contact
  - Remove groove material to form arc contacts
- **Ideal constraint:**
  - Desire all constraints to be perpendicular to bisectors of triangle angles
  - Desire no constraint parallel to bisectors of triangle vertices
  - Constraint metric (CM) = <u>constraint parallel</u>
constraint perpendicular
  - Ideal CM = 0
- **QKCs are weakly over constrained**
  - Contact angle q defines arc geometry
  - Larger θ = stiffer joint but more over constraint
  - Choose design which delivers adequate stiffness, K, and minimizes CM
  - θ < 60 degrees typically emulate a true kinematic coupling

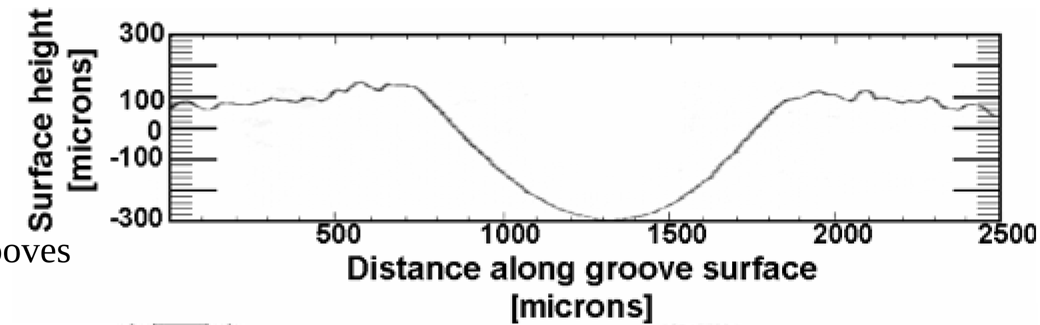


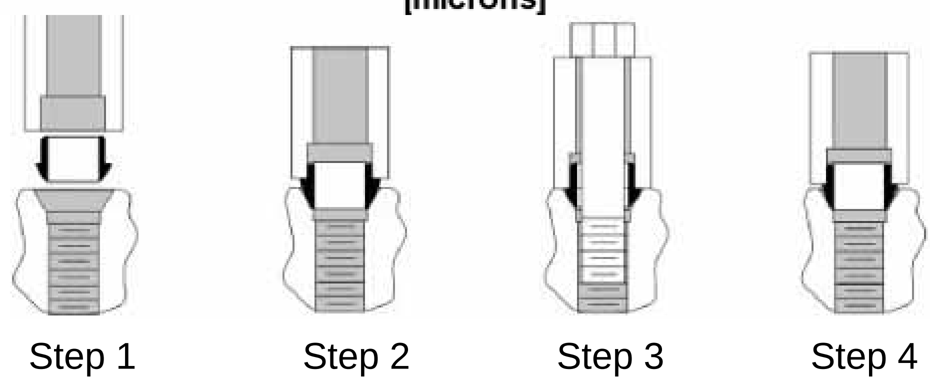


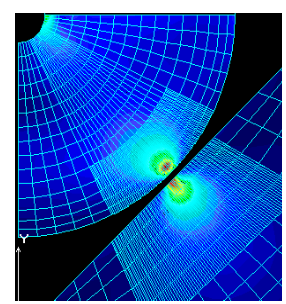
Quasi-Kinematic Couplings: Details

- **Fabricating QKC geometry**
  - Pre-cast or machine reliefs
  - Form tool machines axisymmetric *A-grooves*
  - Balls can be ball bearings or may be ground
- **QKC mating cycle**
  - Step 1: Balls are assembled into top part
  - Step 2: Balls mate A-grooves; finite gap between components
  - Step 3: Balls are preloaded into A-grooves
    - Gap is closed allowing interface to seal
  - Step 4: When preload is released, balls and grooves elastically recover
- **Ball and groove deformation**
  - During Step 3, grooves plastically deform
  - Plastic deformation reduces mismatch between ball and groove patterns
  - Balls and grooves elastically recover in Step 4
  - Recovery restores gap between parts
- **Surface finish**
  - Repeatability of coupling scales as 1/3 RA
  - Rough finish = poor repeatability
  - Grinding and/or polishing are expensive
  - Press hard, fine-surfaced ball surface into rough groove surface



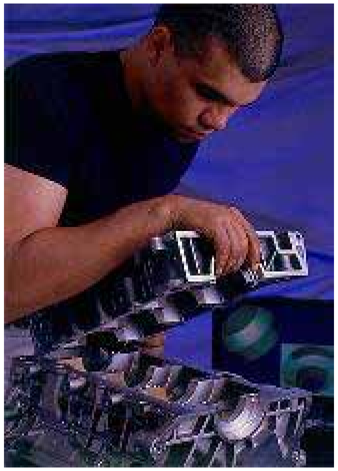




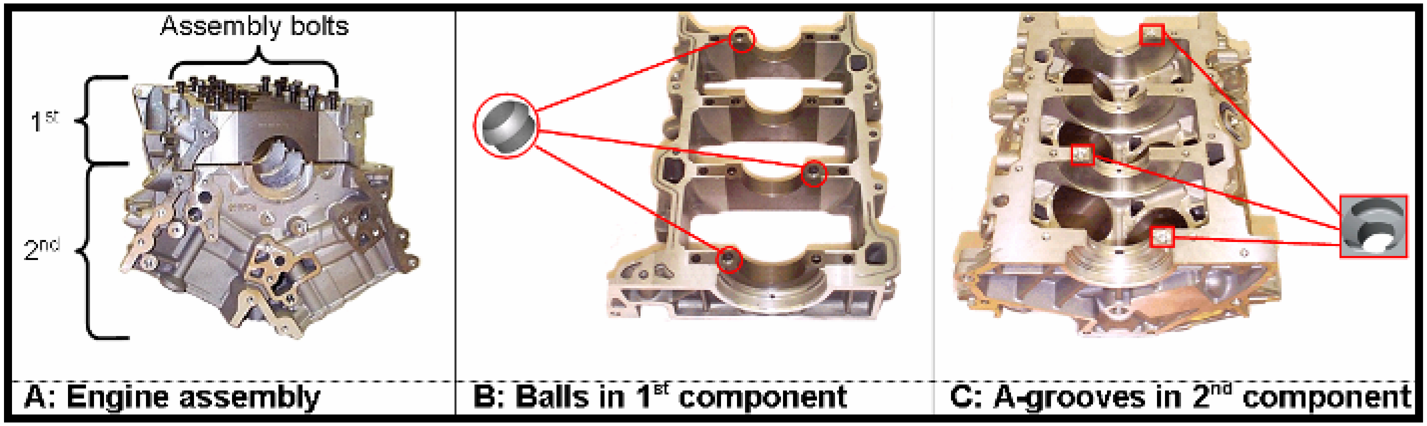


Quasi-Kinematic Couplings: Automotive Example

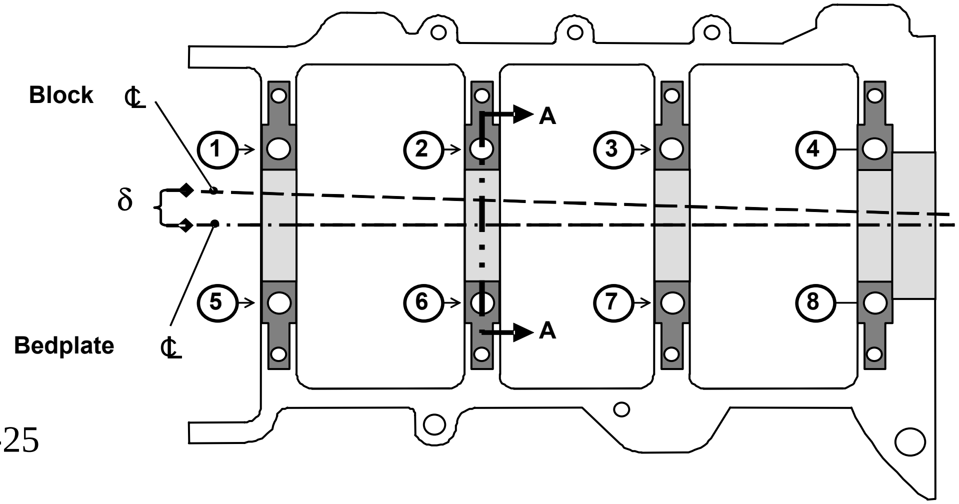
- Original alignment design
  - Components were aligned with 8 pin-hole joints
  - This design is very over constrained
  - Pin-hole patterns requires tight tolerances
  - 8 precision ground dowels required
  - 16 precision holes are bored
- QKC design
  - 8 pins => 3 balls
  - 16 holes =>
    - 3 holes
    - 3 A-grooves



Prof. Martin Culpepper with his Ph.D. thesis, the QKC

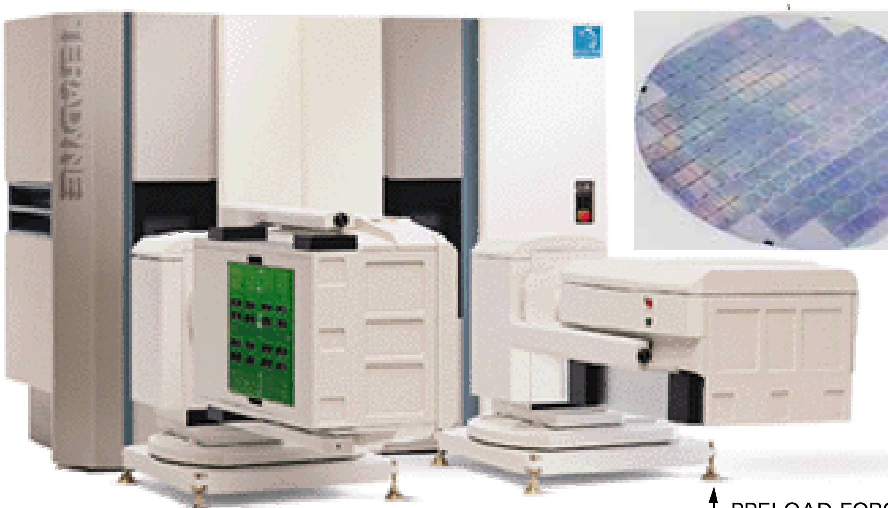


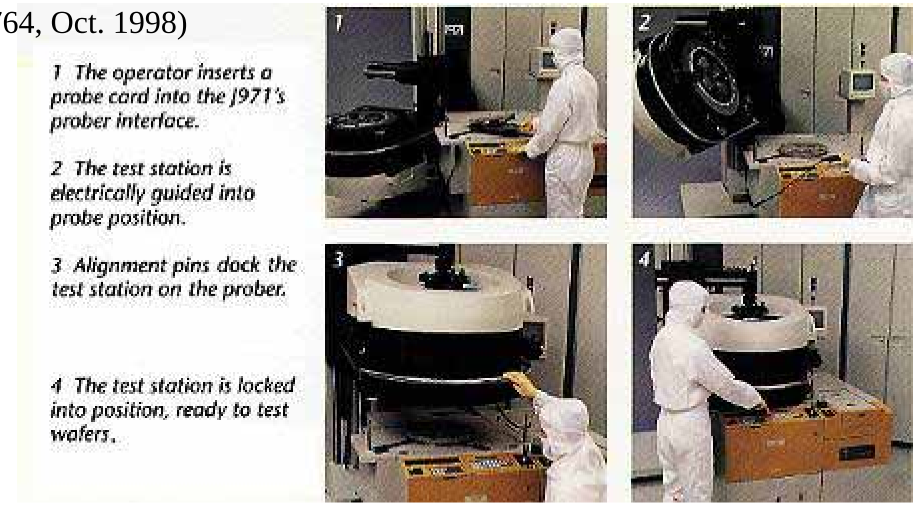
Engine QKC	8 dowels	QKC
Precision pieces	**8**	**3**
Precision features	**16**	**6**
Tolerance [microns]	**40**	**80**
Repeatability [microns]	**5**	**1.5**
Cost reduction/engine	**N/A**	**$1**

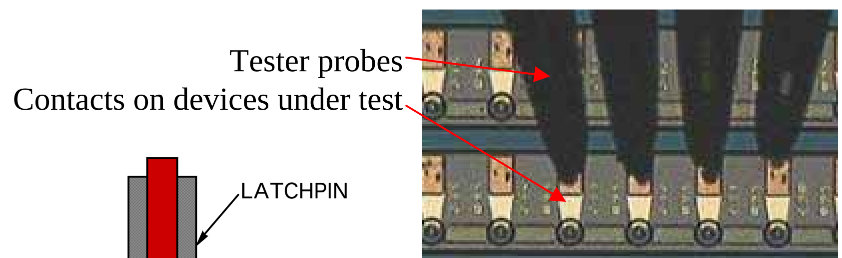


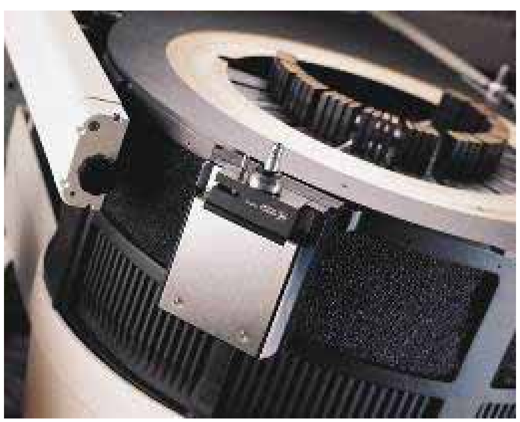
Kinematic Couplings: *Servo-Controlled*

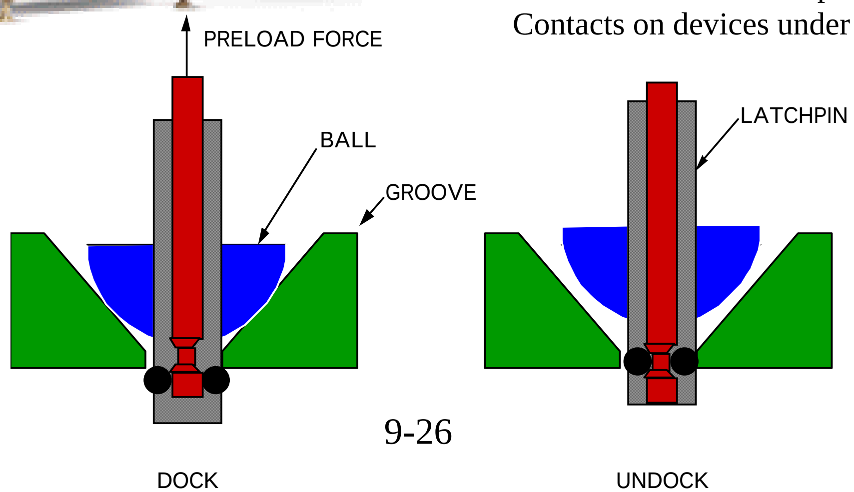
- Automatic Test Equipment (ATE) is used to test computer chips during their manufacture
  - Testing wafers requires a very high precision interface between the tester and wafer
- Sevro-controlled kinematic couplings automatically level ATE test heads to wafer plane
  - Michael Chiu's Doctoral Thesis (US Patent #5,821,764, Oct. 1998)
- Teradyne has shipped over 500 systems





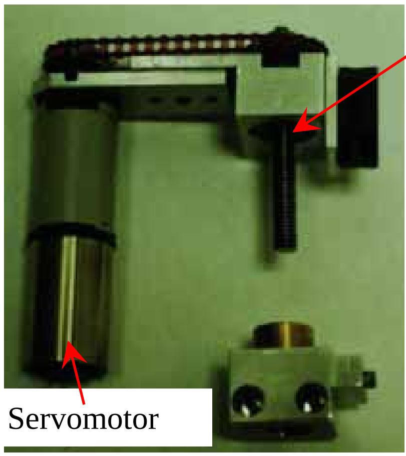


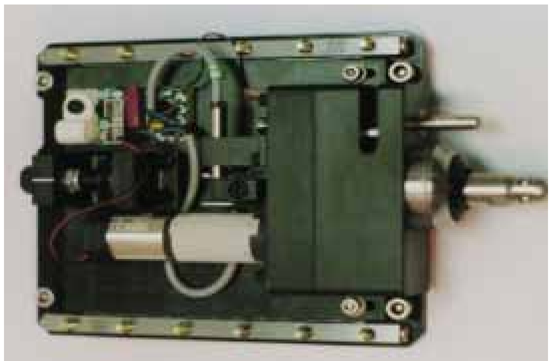


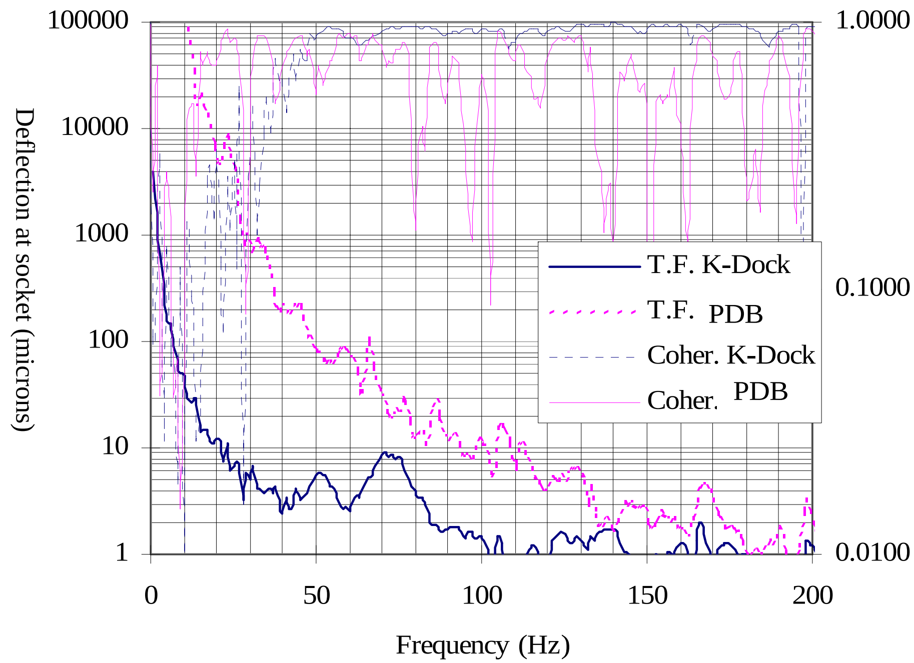


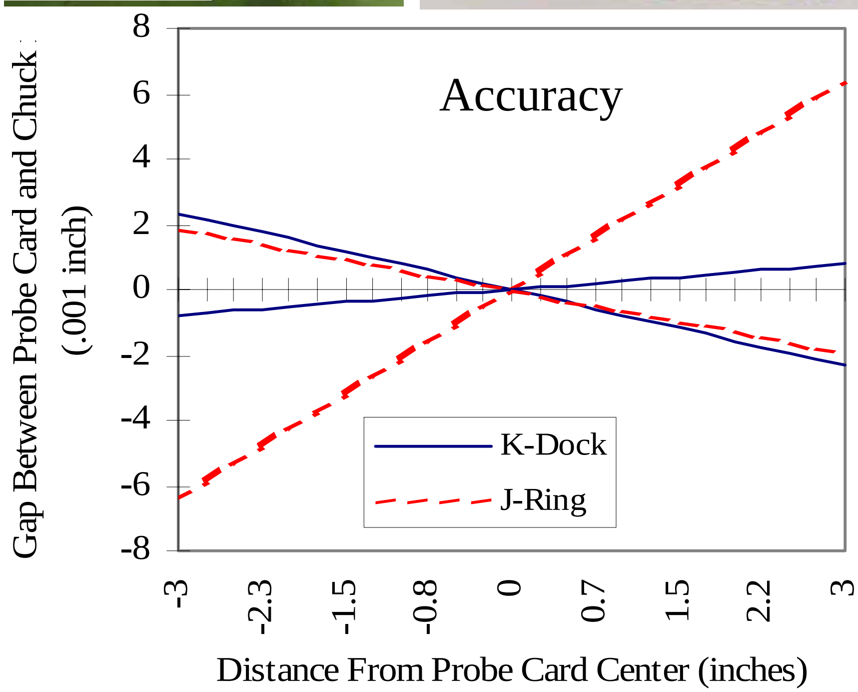
SCKC: *Details*

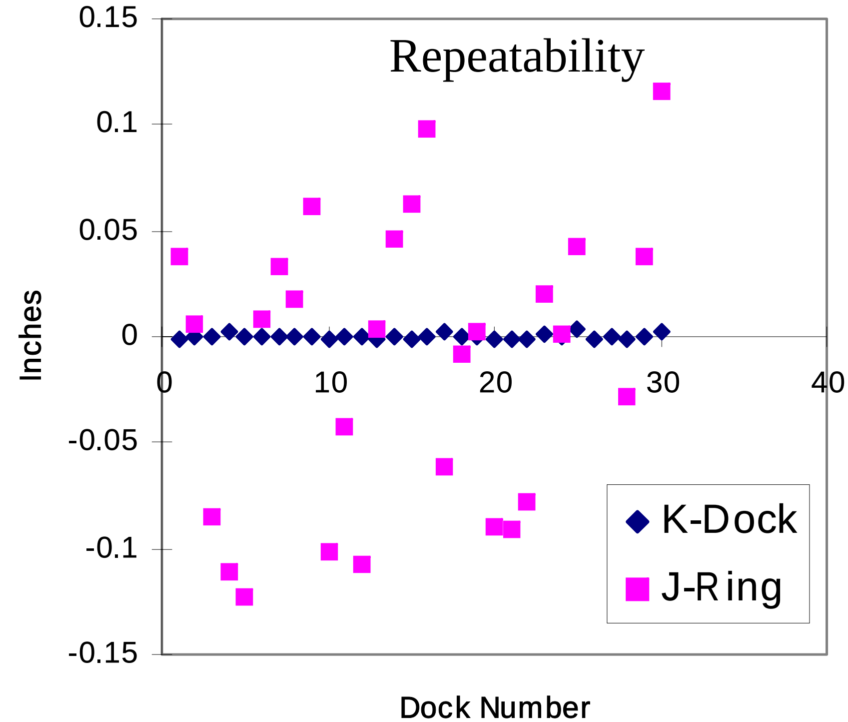
- Improved repeatability, accuracy, dynamic stiffness







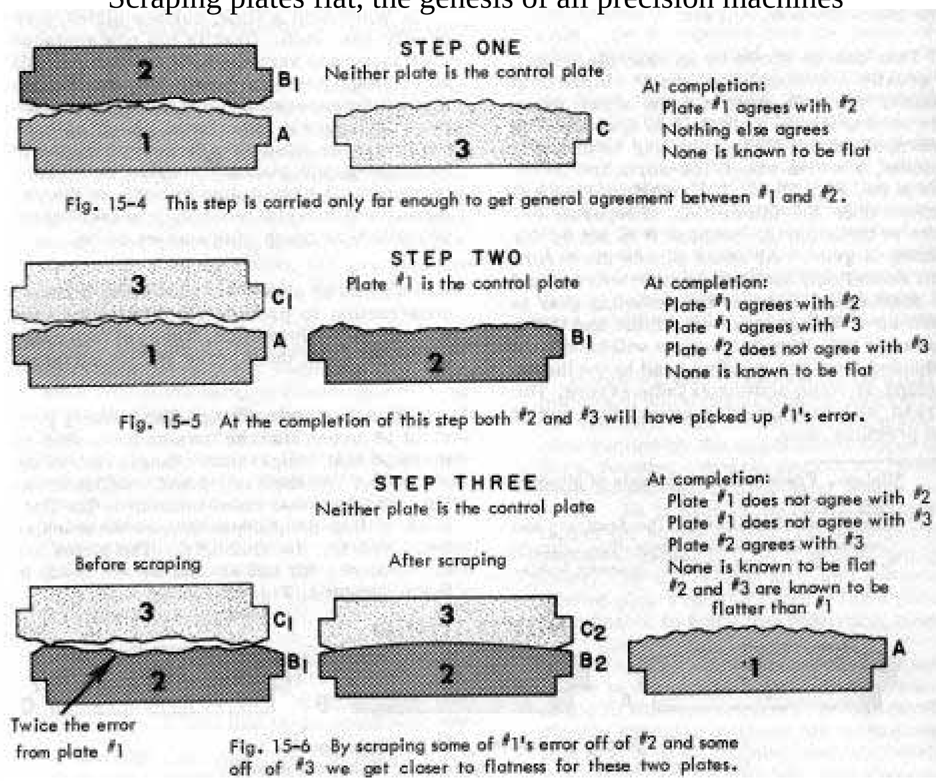




Elastic Averaging

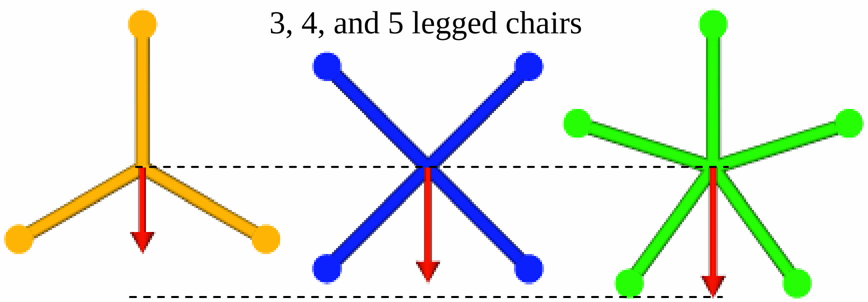
- Any one error can be averaged out by having many similar features
  - As in gathering data with random errors, the accuracy of the reading is proportional to the square root of the number of samples taken
- Local errors are accommodated by elastically deforming the members
  - Overall high stiffness is obtained by the sum of many compliant members

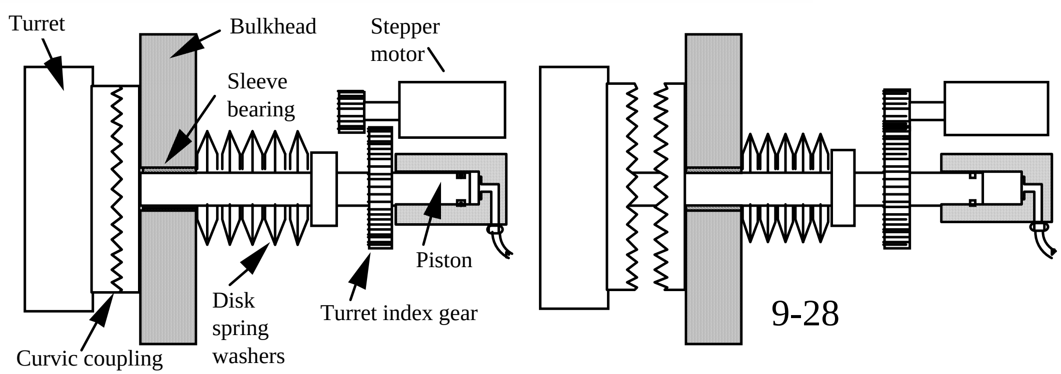
Scraping plates flat, the genesis of all precision machines



From T. Busch, Fundamentals of Dimensional Metrology, Delmar Publishers, Albany, NY, 1964

3, 4, and 5 legged chairs

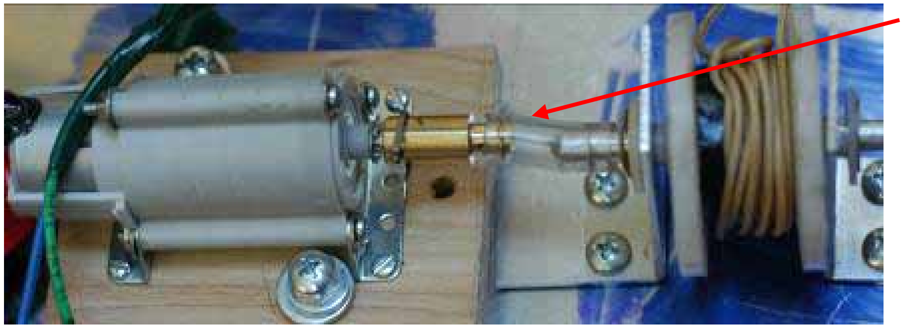




Curvic coupling mechanism for indexing precision machine tools

Elastic Averaging: *Overconstraint?*

- Over-constraint is NOT Elastic Averaging
  - Example: One component (a carriage) wants to move along one path and another (ballscrew nut) along another, but they are attached to each other
    - They will resist each other, and high forces can result which accelerates wear
    - Either more accurate components and assembly are required, or compliance, or clearance (pin in oversized hole) must be provided between the parts
  - Designers should always be thinking of not just an instant along motion path, but along the entire motion path



The coupling may be elastic, but to get it to bend, means large forces are placed on the delicate motor shaft!