Kinematic Couplings: A Review of Design Principles and Applications

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Abstract

From the humble three legged milking stool to a SEMI standard for wafer pod location to numerous sub micron fixturing applications in instruments and machines, exactly constrained mechanisms provide precision, robustness, and certainty of location and design. Kinematic couplings exactly constrain six degrees of freedom between two parts and hence closed-form equations can be written to describe the structural performance of the coupling. Hertz contact theory can also be used to design the contact interface so very high stiffness and load capacity can also be achieved. Potential applications such as mechanical/electrical couplings for batteries could enable electric vehicles to rapidly exchange battery packs.

Keywords

Kinematic coupling, exact constraint design, elastic averaging, repeatability

Introduction

This paper focuses on the design of kinematic couplings and how they have been used in the past and how they can be used in the future. In addition to specific "how to" knowledge and examples for kinematic coupling design, the methodology of exact constraint design will be considered as a catalyst for mechanical design innovation.

Before precision manufacturing equipment and solid modeling became commonplace, design engineers often had to think very carefully how they would attain precision in products they were designing. This often led to the use of exact constraint design principles or careful application of elastic averaging to ensure that components could be assembled without causing undue stresses. However, as the quest for greater precision, reliability and lower cost become ever more apparent in a world of shrinking economies and global resources, design methodologies from the past could be a great asset for the future.

All mechanical things have a structure, and the structure is often made up of parts. Structural connections keep the parts permanently attached to each other. Structural interfaces allow parts to be easily attached and detached. Both cases require the design engineer to think in terms of springs and degrees of freedom. Two critical steps in the analysis of a design are to identify the structural loop and to assess the compliance of elements along it. Next, the stresses on elements along the structural loop are evaluated to ensure, for example, that bearings do not become overstressed when parts are bolted together.

The design of connections and interfaces can be bracketed by kinematic design (exact constraint design) and elastically averaged design [1,2,3]. As shown in Figures 1a and 1b, consider a three legged chair, and its interface with the ground. For a three legged chair, leg length and compliance are nominally not critical. Three legs will always contact the ground. However, such a chair is more prone to tipping as the load must be applied within the bounds of a triangle. On the other hand, consider a five legged chair where each leg has modest compliance, such that when a person sits on it, all the legs deform a little bit and so all legs make contact with the ground. The chair is more expensive to design and manufacture, but loads can generally be applied anywhere within the polygon that bounds the contact points.

Exact Constraint Design

A structural interface is considered to be a repeatable mechanical connection capable of withstanding structural loads, and it can be routinely taken apart and put back together. This is in contrast to structural joints which are not intended to be routinely taken apart. A structural interface must therefore provide constraints to control all the intended degrees of freedom. According to the principle of *Exact Constraint Design* (ECD): *The number of points of constraint should be equal to the number of degrees of freedom to be constrained*. This is the minimum, although some interfaces may utilize more constraints in order to achieve higher load capacity, repeatability, and accuracy using the principle of *Elastic Averaging*.

Figure 2 shows the relative repeatability of different types of connections, and the goal of the designer is to pick the lowest cost method for the desired performance. Often it is good enough to use low cost keyways or pinned connections. Because they would typically be over constrained if an attempt were made to create an exact fit, tolerances are set so there is always some room between components. This loose fit ensures that parts can be assembled, but the accuracy and repeatability that can be obtained is limited by the toleranced gaps. The alternative is to use spring pins or numerous elastic elements that accommodate misalignment and tolerances by elastic deformation. However, designing a system that is over constrained takes exceptional care to ensure that deformations do not occur that may overload sensitive components such as bearings. Thus if possible, a good strategy is to try and create an exactly constrained design.

ECD often concentrates loads at single points which can lead to a smaller region of stability (e.g., 3 legged chair); however, ECD does not always mean that systems have to be designed like three-legged chairs. Ponder the following: Can you support a plate at multiple points yet not get the "four legged chair with one short leg" syndrome? How do windshield wiper blades distribute the point force applied by the arm uniformly across the blade? The answer to both questions is to use a wiffle tree as can be seen on any windshield wiper blade. Exact constraint design can sometimes be visualized by imagining how support points need to be applied to uniquely define the position of a cube using a 3-2-1 fixturing philosophy illustrated in Figure 3. Each of the support points can in fact be the center of stiffness of an array of points on a wiffle tree arm; however, eventually at the connection to ground, 3-2-1 points are established:

1. One side is placed on three support points
2. A second side is pushed up against two support points
3. The first side slides across the three support points
4. A third side is pushed against one support point

5. The first and second side slide across their support points

With the above, ideally, exact constraint is theoretically achieved, and for all practical purposes, it is when the loads are very light; however, when heavier loads are applied, which may be due to the weight of the object itself, point loads cause local deformations that act like additional orthogonal constraints. These point contact deformations and friction at them fully constrain the cube when it is first placed. As the cube is pushed against the other constraints, the point contact deformations and friction reduce the repeatability of the system. Hence the 3-2-1 fixturing method practically has repeatability on the order of 3-5 microns.

Both types of kinematic couplings are exact constraint designs to a point: the contact forces at each of the six points are high and micro indentations, although elastic typically, occur which serve to add in effect micro over constraints. The instant center of the couplings is known, and thus if there is uniform thermal expansion of the coupling elements, the instant center also is the thermal center. If the expansion is non-uniform, then hysteresis in coupling position will depend on if the thermal expansion forces cause micro slip at the contacts, or the static friction forces combined with the micro over constraints act to pin the preloaded coupling components together. This can be modeled to some degree using conventional methods or FEA. For a critical application, once engineered, an experiment should be done before committing to production.

What countermeasure can be used to address the risk of point contact deformations essentially exactly constraining the system when the first three contact points are established? The fundamental principles of stability, or rather that of an unstable system, and self-help can provide insight: If initial contact points can be established such that the weight of the system creates a moment that requires more contact points to be established in order to create force equilibrium, then "exact" constraint can be achieved that is far less sensitive to point contact deformations. This type of coupling is called a *kinematic coupling*. The primary point contact issue becomes: what is the maximum point load that can be applied at any support point without causing permanent deformation?

Hertz Contact

Railroads were a critical transformative industry for the human race. Imagine a railroad locomotive that is heavily loaded so it has the tractive effort to pull a longer train. If the contact stresses between its wheels and the rails are too high, the rails could prematurely fail, resulting in the need for replacement of hundreds of miles of tracks and massive disruption of the transportation system. Such was the motivation to create a formula for determining how much load a train could carry. Fortunately Heinrich Hertz, like many mathematicians, saw no real difference between different types of problems[1], only different boundary conditions to be applied to differential equations. As a result, he developed a theory for predicting the contact stress between bodies and the term "Hertz contact" symbolizes the high stresses that arise between bodies in point or line contact. Indeed the contact stress analysis methods pioneered by Hertz allowed the industrial

[1] Hertz also made major contributions to the field of electrical engineering, and the symbol for frequency "Hertz" comes from his name.

revolution to roll forward. Although his life was short, Hertz made a lasting impression that will be remembered as long as time rolls on [4].

Because Hertz stresses can locally be so high, they often act as initiation sites for spalling, crack growth and other failure mechanisms. An understanding of Hertz contact stresses is thus of fundamental importance in machine design, and intense volumes have been written about Hertz-induced failures [5]. The resulting equations for Hertz contact are developed from analysis methods that require the evaluation of elliptical integrals, which are not simple to evaluate. The results are often shown as plots, from which designers must interpolate the values. Approximate polynomials can be used, which when incorporated into a spreadsheet make it easy to evaluate Hertz contact stresses [6]. The equations are well known and incorporated into many different analysis tools such as spreadsheets, but it is important for designers to remember the overall relations for Hertz contacts:

Stress is proportional to:
- Force to the 1/3rd power
- Curvature of bodies in contact to the –2/3rd power
- Modulus to the 2/3rd power

Deflection (stiffness) is proportional to:
- Force to the 2/3rd power
- Curvature of bodies in contact to the –1/3rd power
- Modulus to the –2/3rd power

Contact ellipse diameter is proportional to:
- Force to the 1/3rd power
- Curvature of bodies in contact to the 1/3rd power
- Modulus to the –1/3rd power

Kinematic Couplings

Kinematic couplings are exact constraint design couplings because ideally they use six known contact points to locate one component with respect to another. They have long been known to provide an economical and dependable method for attaining high repeatability in fixtures [7,8]. Figure 4 shows the two main types of kinematic couplings. When a spherical surface on one part rests in a concave tetrahedron on another part, another rests in V-groove pointing towards the tetrahedron, and a third rests on a flat plate, the coupling is referred to as a *Kelvin Coupling* (or Kelvin Clamp) after William Thompson (Lord Kelvin) who favored this design. The origin of the two designs appears, however, to be in the early 1800's but the "inventor" is apparently not specifically known [9]. The primary advantage of this non-symmetric design is that its instant center of rotation is always located at the center of the tetrahedron's contact points. However, it suffers from the 3-2-1 Hertz contact problem. James Clerk Maxwell, on the other hand, preferred the symmetry and ease of manufacture of *three-groove couplings*: three V grooves on one part are oriented towards the center, and three mating curved surfaces on the other part.

Kinematic couplings are deterministic because they only make contact at a number of points equal to the number of degrees of freedom that are to be restrained.

This makes performance predictable [10,11] and helps to reduce design and manufacturing costs.   On the other hand, contact stresses are often very high and no lubrication layer remains between the elements that are in point contact. For high-cycle applications it is advantageous to have the contact surfaces made from corrosion-resistant materials (e.g., stainless steels, carbides, or ceramic materials). When non-stainless steel components are used, one must be wary of fretting at the contact interfaces, so steel couplings should only be used for low-cycle applications.

Historically kinematic couplings were typically used for precision instruments and not for heavily loaded situations. In the mid 1980's, Dr. Robert Hocken at NBS (now NIST) asked the author to figure out how to use kinematic couplings for heavily loaded applications. The author applied Hertz contact theory and kinematic analysis to predict load capacity and accuracy under load of kinematic couplings. Tests on a heavily loaded (80% of allowable contact stress) steel ball/steel groove system have shown that sub-micron repeatability can be attained;. However, with every cycle of use, the repeatability worsened until an overall repeatability on the order of ten microns was reached after several hundred cycles. At this point, fret marks were observed at the contact points. Tests on a heavily loaded (80% of allowable contact stress) silicon nitride/steel groove system have shown that 50 nm repeatability could be attained over a range of a few dozen cycles, and that with continued use the overall repeatability asymptotically approached the surface finish of the grooves. An examination of the contact points showed burnishing effects, but once the coupling had worn in, submicron and better repeatability was obtained.   The tests also showed that with the use of polished corrosion-resistant (preferably ceramic) surfaces, a heavily loaded kinematic coupling can easily achieve submicron repeatability with little or no wear-in required [13,14]. Accuracy of kinematic couplings depends on the tolerancing of the components and their assembly, and in general some averaging effect is achieved [15]. In practice, it is not unusual for the accuracy of the coupling to be two to three times better than the accuracy of the components used.

Indeed controlling deformation and friction at the contact interface are keys to achieving a high level of repeatability. Hard ground steel surfaces work well to the micron to sub-micron level. Coating the hard steel, for example with TiN, can help prevent corrosion, but the difference in elastic modulus can lead to failure of the coating with repeated contacts at high stress levels [15]. Hard polished ceramic or tungsten carbide surfaces are preferred when possible.

For even greater repeatability, the tangential frictional constraint at the contact surface needs to be controlled [16]. For modestly loaded couplings, flexures can be wire cut in the Vee groove structures so the flats or the Vee grooves are supported by flexural elements that are reasonably stiff in a normal direction but tangentially compliant [17]; however the flexures can buckle under high loads. For higher loads, a more complex wire cut can be used so the flexural elements are in tension. Damping elements can also be added to the flexural elements [18]. For the ultimate elimination of friction at the interface, air bearings can even be used [19]. In the end, the literature provides detailed design considerations for the different types and expected performance levels, and the designer gets what they pay for.

Spatial Kinematic Couplings

A spatial kinematic coupling is one that has six contact points that define the position and orientation of one component with respect to another. Three-groove kinematic couplings are particularly easy to create because the grooves and spherical contact elements, typically hemispheres, can be made all at once and then simply bolted in place. With three grooves, the question naturally arises as to what is the best orientation for the grooves. Mathematically, to guarantee that the coupling will be stable, James Clerk Maxwell stated the following:

When an instrument is intended to stand in a definite position on a fixed base it must have six bearings, so arranged that if one of the bearings were removed the direction in which the corresponding point of the instrument that would be left free to move by the other bearings must be as nearly as possible normal to the tangent plane at the bearing. This condition implies that, of the normals to the tangent planes at the bearings, no two coincide; no three are in one plane, and either meet in a point or are parallel; no four are in one plane, or meet in a point, or are parallel, or, more generally, belong to the same system of generators of an hyperboloid of one sheet. The conditions for five normals and for six are more complicated.

With respect to practical implementation of the theoretical requirement for stability, for precision three-groove kinematic couplings, stability, and good overall stiffness will be obtained if the normals to the plane of the contact force vectors bisect the angles of the triangle formed by lines joining the centers of the hemispheres (e.g., balls) that lie in the grooves. This is illustrated in Figure 5. For balanced stiffness in all directions, the contact force vectors should intersect the plane of coupling action at an angle of 45 degrees. Note that the angle bisectors intersect at a point that is also the center of the circle that can be inscribed in the *coupling triangle*. This point is referred to as the *coupling centroid* and it is only coincident with the coupling triangle's centroid when the coupling triangle is an equilateral triangle. Fortunately designers of precision kinematic couplings are not faced with the generic grasp-a-potato problem faced by researchers in robotics. Indeed, any three-groove kinematic coupling's stability can be quickly assessed by examining the intersections of the planes that contain the contact force vectors (two per ball/groove interface). For stability, the planes must form a triangle as illustrated. In terms of mechanism design, this means that the instant center for each ball/groove interface must lie outside the coupling triangle.

Three-groove kinematic couplings are commonly used to create repeatable interfaces for machines and instruments. Modular components are available off-the-shelf[2], but can also be made custom when needed. The design equations for three groove kinematic couplings are well established, and are incorporated into a readily available spreadsheet such as *Kinematic_Coupling_3Groove_Design.xls*[3]. Figure 5 shows the primary geometric design parameters which include the effective ball radii, the diameter of the circle on which the ball centers lie (the coupling diameter), and the groove orientations. The performance under load is also affected by the coupling materials and the preload force. The sensitivity to these parameters can be studied using the spreadsheet. Figures 6a and 6b show input and output sections from the spreadsheet.

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[2] See www.kinematiccouplings.org which provides many references and design ideas. Components are commercially available, for example from Bal-Tec, Inc., 1550 E. Slauson Avenue, Los Angeles, CA 90011, (800) 322-5832, www.precisionballs.com/

[3] www.kinematiccouplings.org

Dbeq (mm) is the diameter in millimeters of a ball that could fit in the groove. This is important because the plane through the virtual ball's centers is assumed to be the plane of the coupling. As discussed before, a canoe ball or a crowned cone could be used to increase the load capacity by entering in the actual radii of curvature at the contact points.

Rbminor (mm) is the minor radius of contact of the "ball"

Rbmajor (mm) is the major radius of contact of the "ball"

Rgroove (mm) is the radius of curvature of the groove at the contact point. For a Vee groove, enter in a huge number, like 109 to indicate a flat plane. If a Gothic arch groove is used, it should be no more conformal than about 1.5 times the radius of the ball, else edge loading is likely to occur. Remember, for a conformal groove, the radius of curvature is negative. Hence one would enter, for example, Rgroove = -1.5Rbmajor.

Costheta is the angle between the major diameters of the contact surfaces. This value has always been TRUE.

Dcoupling (m) is the diameter of the coupling circle on which the centers of the balls' centers lie. The bigger the coupling diameter, the greater the moment load capacity of the coupling.

Fpreload (N) is the preload applied over each ball. Because the Z direction is pointing upwards from the coupling plane, the preload force is negative. Even if the preload is centrally applied, enter the preload force over each ball. it is assumed that all balls have the same initial preload.

Xerr (mm) is the X location at which the deflections of the coupling are to be determined with respect to the coupling coordinate system located at the center of the coupling circle.

Yerr (mm) is the Y location at which the deflections of the coupling are to be determined.

Zerr (mm) is the Z location at which the deflections of the coupling are to be determined.

Matlabball is the material label for the ball. Select a standard material, or enter in your own values (to the right).

Matlabgroove is the material label for the ball. Select a standard material, or enter in your own values (to the right).

Min. yield strength reminds the designer what stress on which the stress ratio will be based.

Largest contact ellipse major diameter (mm) reports the contact ellipse dimension at each contact point. This is important to ensure that the contact zone does not extend to the edge of the Vee groove. It should be one characteristic dimension away.

Largest contact ellipse minor diameter (mm) is similar to major diameter.

Max shear stress/(ult. tensile/2) shows if the subsurface shear stress due to Hertz contact > material's maximum allowable shear (syield/2).

RMS applied force (N) shows the root mean square of the contact forces.

RMS stiffness (N/micron) shows the root mean square of the dx, dy, dz deflections at Xerr, Yerr, Zerr.

FLx (N) = the applied X direction force. All the force components are applied at XL, YL, ZL.
FLy (N) = the applied Y direction force at XL, YL, ZL.
FLz (N) = the applied Z direction force at XL, YL, ZL.

The spreadsheet is written to enable a designer to assume a standard common shape for kinematic couplings, where the grooves are spaced 120 degrees apart on a circle which is referred to as the *coupling diameter*. It is also assumed that the " balls" that engage the grooves make contact at a 45 degree angle. If the design engineer needs to create a kinematic coupling where the grooves are not 120 degrees apart, then the direction cosines through the contact points need to be defined. In this case, the designer can modify the spreadsheet accordingly for each groove.

Figure 7a shows a high load capacity and stiffness kinematic coupling with a 300 mm coupling diameter that was designed for a precision grinding fixturing application [20]. "Canoe balls", shown in Figure 7b[4], enable the Hertz contact equivalent of a 250 mm diameter ball in a groove. The resulting coupling had an axial stiffness of over 200 N/micron and a repeatability of 50 nm. The canoe ball merely takes the contact zones of the large ball and brings them onto the surface of a much smaller "ball" (*Dbeq*) so there are large contact radii at the contact points. This is a very important design philosophy that gives great versatility to the application of kinematic couplings.

The load capacity is a direct function of the Hertz contact stress, and the contact stress is a very strong function of the shape of the contact interface. Consider the following types of ball-groove contact shape options based on maximum contact pressure of 1.3 GPa, and both components having a modulus of elasticity of 193 GPa:

- 25 mm diameter stainless steel half-sphere on 25 mm diameter cylinders
  - F_{max} = 111 N
  - Vertical deflection = 3.2 µm
  - Contact ellipse major diameter = 0.425 mm, minor diameter = 0.269 mm
- 25 mm diameter stainless steel half-sphere in a Vee
  - F_{max} = 229 N
  - Vertical deflection = 4.7 µm
  - Contact ellipse major diameter = 0.488 mm, minor diameter = 0.488 mm
- 25 mm contact diameter x 125 mm radius crowned cone in a Vee
  - F_{max} = 1106 N
  - Vertical deflection = 11 µm
  - Contact ellipse major diameter = 2.695 mm, minor diameter = 0.603 mm
- 250 mm diameter stainless steel half-sphere in a Vee
  - F_{max} = 16160 N
  - Vertical deflection = 47 µm
  - Contact ellipse major diameter = 4.878 mm, minor diameter = 4.878 mm

[4] Canoe balls are made using a punch grinder: a CNC cylindrical grinding machine that can grind non-round parts. A source for such grinding machines and/or canoe balls is Weldon Machine Tool: http://www.weldonmachinetool.com

Preload is the force applied to the coupling to hold it together, and it is one of the most important parameters that affect repeatability of a kinematic coupling. Preload establishes the initial stiffness, given that the Hertzian contact stiffness is nonlinear: Hertzian deflection is proportional to applied force to the 2/3rds power. To get good stiffness, the preload must be high, repeatable, and must NOT deform the rest of the structure. For heavy duty applications, such as fixtures for machine tools, this can be accomplished by preloading through the center of the kinematic elements with bolts that compress springs as shown in Figure 8. For lightly loaded systems, such as instruments, magnets can be used as preload elements [21].

Materials also play a major role in the performance of the coupling. They not only dictate the maximum stress, and hence the loads that can be applied, they must also have low relative friction between each other, and they must not be subject to corrosion. For super precision high load applications, silicon nitride, silicon carbide, or tungsten carbide are the best materials to provide high load capacity and to minimize the coefficient of friction and to prevent corrosion at the Hertz contact interface.

Two forms of three-groove couplings are shown in Figure 9. *Horizontal couplings* are often found in metrology applications. They can also be used in the manufacture of precision parts. For example, a planar three-groove coupling can be used to hold a grinding fixture on a profile grinder. A matching three-groove plate on a CMM allows the grinding fixture to be transferred to the CMM with the part. The part can be measured and then placed back onto the grinder so the errors can be corrected.

To minimize Abbe errors in some applications, *vertical couplings* can be designed where the preload is obtained with a clamping mechanism or by gravity acting on a mass held by a cantilevered arm. An industrial machinery example would be in a machine where a module must be precisely located with respect to the rest of the machine, yet for maintenance purposes, it has to be easily disconnected and rolled away to gain access to the interior of the machine. Once the repair is complete, the faster the module can be reinstalled, the sooner the machine can become productive.

Compliant Kinematic Couplings

Kinematic couplings have an inherent problem in that unless sufficient preload is applied, externally applied loads can cause them to become unstable and tip. A countermeasure to this risk is to use a high preload, but this can create the need for expensive kinematic contact elements. Furthermore, for applications such as locating forming dies, where the process forces on the dies are huge, or locating mold components, where the surfaces have to seal against each other tightly, conventional kinematic couplings are not practical. In these cases there are two functional requirements. The first functional requirement is for locating. The resulting design parameter is then a kinematic coupling. The second functional requirement is for having the two part surfaces to touch. The resulting design parameter is to mount the kinematic coupling elements in such a way that they have a degree of freedom normal to the interface surfaces [22].

Only one set of kinematic coupling elements needs to be mounted upon movable members. The motion needs to have minimal parasitic error motions (unwanted motions) so as to not lose precision as the mating surfaces come into contact. This means that in addition to not having any backlash (clearance) in the mechanism, the mechanism must have high lateral stiffness in the sensitive directions. Perhaps the simplest way to accomplish these functions with minimal cost is to use a compliant mechanism as shown in Figure 10. The compliant mechanism (flexures) can support rigid elements, such as the vees, or the function of the Vees can be incorporated into the flexural elements themselves.

Figure 11 shows how the vee-grooves can be made from formed sheet metal or spring steel (for heavy duty applications). The sheet metal Vees are easily made, and they are attached to the base plate by sheet metal screws or rivets. Before a vertical load is applied in the Z direction, the balls contact the sides of the Vees and this locates one part with respect to the other in X, Y, and yaw. After the system is preloaded, the balls spread the Vees out and flat surfaces between the parts contact to establish Z and pitch and roll. The bottoms of the balls can even serve as the flat surfaces which then contact the bases of the Vees. How much vertical travel can the sheet metal vees tolerate? Is it reasonable to install them, use the coupling, and just allow them to plastically deform? If they are bent beyond their elastic limit, when the load is released, some recovery will occur due to elastic spring back. However, vertically compliant sheet-metal vees will not have high lateral stiffness and because of the sliding action between the balls and the Vees, repeatability on the order of tens of microns is to be expected which is actually very good for a low cost coupling.

As shown in Figure 12a, wavy spring washers can even have mating “male” (ball) and “female” (groove) features that allow the washers to be precisely located with respect to one part, and then to precisely locate another part to both the first part and the washer stack. When preloaded, significant torque could be transmitted, or the spring stack could act as a slip clutch, as shown in Figure 12b. For mating parts, highly repeatable component assembly without loss of stiffness can be achieved at minimal cost and effort. This is a great advancement over the use of dowel pins: Simply counter bore the two mating faces at the fastener positions and insert a kinematic spring in each pocket. The inserted springs lightly clinch the bore ODs and grip tighter under pressure, a third coupling spring then ensures repeatable alignment between the two parts and can average out bore pitch errors over several fastener positions. Tightening the bolts achieves normal face-to-face stiffness. The author and his good friend Peter Bailey created this design for use in a new machine concept which did not come to life, but the kinematic spring washer concept still has great potential.

Flexural bearings can take up a lot of space, and as an alternative design, Figure 13 shows a compact system for enabling a kinematic coupling element to be constrained to move axially by a die-set bushing and spring washer stack. A very low cost method is to use ball spring plungers. In either case, error motions, including backlash, of the support bearings must be considered, but is easily controlled with good design and manufacturing techniques.

Figure 14 shows another variation on the compliant coupling theme: Sand cores for casting can be aligned with respect to each other using hard balls which engage sand Vees [23]. The balls can be located in tetrahedron vees, and then a sand vee mates with them. This can give mm location. Alternatively, metal vees bonded to sand vees in one sand part can mate with metal balls bonded into hemispherical depressions in another sand part. Even wooden vees and balls can be used. After the casting is poured, the metal balls and vees can be recovered for reuse.

MEMS Kinematic Couplings

Speaking of sand, compliant kinematic couplings can be used to enable silicon wafers to be stacked upon each other to precisely locate them with respect to each other prior to the stack being compressed for bonding [24]. Figure 15 shows a silicon wafer with an array of silicon cantilevers around its perimeter, where each cantilever has a pyramid on its tip. The cantilevers and pyramids are made by low cost KOH etching. The pyramids mate with KOH etched grooves in the mating wafer. The project started out assuming many units would be needed to achieve accuracy via elastic averaging, but in the end, the accuracy of the etching process showed only 4 elements (in a + pattern across the wafer perimeter) were needed. The coupling is not strictly kinematic nor strictly elastically averaging, yet it is sufficiently deterministic and hence can be referred to as a *pseudo kinematic coupling*.

Zia Couplings

Another type of *pseudo kinematic coupling* based on a + pattern coupling with mating pyramids and grooves is the *Zia* coupling [25, 26] created by Sandia National Laboratories for manufacturing high precision aspheric lenslet arrays. A Zia coupling is shown in Figures 16a (the faceplate of a spindle) and 16b (the parts coupled) and it was used to enable a lenslet array to be accurately fixtured and indexed in two orthogonal directions parallel to a spindle face. Hence each lenslet could be machined on-center while maintaining lenslet center to center positioning of 2μm and form error of λ/10. One half of the Zia coupling is made integral to the part and the mate utilizes scribed orthogonal sets of grooves that locate the part on the chuck. The averaging of the grooves increases the repeatability of the system. The part can then be moved an integral number of grooves across the chuck to index the lenslet array for machining the individual lenslets. The pattern of features that make the coupling resembles the Zia pattern created by American Indians and hence the name for the coupling.

Servo Controlled Kinematic Couplings

A particularly useful variation on the compliant kinematic coupling theme is to make the moving elements' positions servo controllable to enable an instrument to be precisely positioned and adjusted on top of a device that delivers parts to be tested. Three identical servo modules are used. Figure 17a shows a module where a leadscrew controls the position of a slide supported by crossed-roller bearings, to which is attached the ball. A preloading post passes through the center of the ball, and at its end is a ball-lock device much like that which is found on quick-connect pneumatic couplings. A separate actuator controls the action of the preloading mechanism. This design is called the *K-Dock*™ [27, 28]. The K-Dock precisely locates and then servo controls the height, pitch, and roll location of Teradyne Corp.'s automated semiconductor testheads with respect to a handler or prober for testing semiconductor devices as shown in Figure 17b. Figure 17c compares the repeatability of the servo controlled kinematic coupling, the K-Dock, and a standard interface known as a J-Ring. Hundreds of these units have been sold and are in continual use with a very high degree of reliability. They reduce the undock/dock/align time from tens of minutes to a few minutes which means a great deal on the factory floor, as the process is typically done several times per shift and the machines cost millions each.

Servo controlled kinematic couplings can also use servo controlled actuators to adjust the groove width and position or the relative ball position to control the position of the coupling with three [29] or even six degrees of freedom [30,31,32,33]. The groove contact surfaces are supported by flexures which are then positioned by actuators. Tens of nanometer controllability is obtainable, and thus with appropriate metrology, system accuracy can approach repeatability.

Three Tooth Kinematic Couplings

To prevent thermal expansion, it is often desirable to make precision instrument components from Super Invar, which is an iron alloy with 36% nickel. However, Invar cannot be hardened, and hence it cannot support significant point-contact loads. Three-tooth couplings were invented by Dr. Layton Hale of the Lawrence Livermore National Laboratory to overcome this limitation [34]. The three-tooth coupling, shown in Figure 18, forms three theoretical lines of contact between the cylindrical teeth on one component and flat teeth on the other component. Practically, both members can be made identical with cylindrical teeth, which yields the best repeatability. Good performance for low cost can also be obtained if the members are made with flat teeth, which yields the greatest load capacity, least cost, and good repeatability. Each line of contact across two teeth represents a two-degree-of-freedom constraint, thereby giving a total of six constraints. Ideally manufactured with three identical cuts directly into each member, the teeth must be straight along the lines of contact but other tolerances may be relatively loose.

A three tooth kinematic coupling can also be made to have flat-on-flat tooth contacts which is easier and cheaper but will have worse repeatability. Flat teeth are easily molded into plastic to enable very repeatable coupling and orientation of cylindrical members, and the flat-on-flat contact will not creep with time. This can be a good design for some lens assemblies. With added snap-fits, a very precise assembly can be obtained.

Using Hertz contact theory, the load capacity of a single Super Invar tooth-to-tooth line contact is actually greater than the load capacity of a hardened 20 mm diameter ball on a hardened flat steel contact. However, one must take care to make sure the contact area is not larger than the thickness of the mating teeth else the contact zone has probably saturated and in effect, the teeth have flattened and face-to-face contact has occurred.

Planar Kinematic Couplings

A common problem faced in many applications is how to repeatedly fixture (locate) a 2D object. An example is stacking many layers of a printed circuit board into a book which is then to be pressed and heated to bond the layers together [35,36]. Because there are three degrees of freedom to define (x, y, θ), at least three contact points must be established. Four candidate locations are shown in Figure 19a. Dotted lines drawn normal to the contact points and their intersections represent the instant centers of rotation for the body. In configuration 1, the preload force is applied between and below the instant centers. Configuration 2 has the preload force applied between but above the instant centers. Configuration 3 has the preload applied below the instant centers, but it is in line with one of them, which makes this design therefore marginally stable in one direction. Configuration 4 has the preload force above one of the instant centers and also directly in line with one of them. To *feel* what is more stable, it is easy to make a bench level experiment, such as shown in Figure 19b, and play with it! It is also straightforward to model it analytically to optimize the locations.[5]

Split Groove Kinematic Couplings

In multi-station production systems, the product being manufactured is held in a fixture attached to a pallet, and the pallet is conveyed between workstations. In high-precision systems, variation in the position of the pallet is one of the largest sources of error which can reduce quality and yield. Conventional alignment methods for pallets utilize a round pin to engage a bushing for X-Y location and a diamond shaped pin to engage a second bushing which thus acts as a rotational constraint

Split groove kinematic couplings split one of the grooves and separate it so at first glance the coupling looks like it uses four spherical surfaces mating with four grooves. The design concept and schematic of the groove layout is shown in Figure 20a [37]. This design allows for greater tipping stability and also facilitates coupling rectangular pallets in transfer line machines often used in automated assembly systems. Split-groove kinematic couplings can reduce variation in pallet location by an order of magnitude over conventional pin-in-bushing systems; in addition, they never jam and they get better with wear (when they wear at all).

A particularly interesting application is the display of archival documents in environmental cases. Encasements that hold and protect historic documents for display must balance the preservation demands of conservators, the aesthetic requirements of curators, and the technical constraints imposed by physical phenomenon such as diffusion. The Massachusetts Archives needed five encasements for permanent public display of the Massachusetts Constitution, original copies of the Declaration of Independence and Bill of Rights, and two 17th century charters. Collectively they are known as the *Documents of Freedom* and are priceless as they simply cannot be replaced and they are foundations of US democracy. The encasements are filled with a humidified argon and helium mixture to prevent oxidation of the documents. As soon as the encasements are filled with inert gas, oxygen will inevitably begin creeping in. The case design is intended to maintain the proper environment for 25 years without intervention. Inside the case the documents are held on to a perforated platen by special plastic clips. The platen allows the humidified encasement gas access to the backside of the document to maintain the utmost in stability. Once the document was removed from its vault and placed on a platen, it had to be rapidly put into its case and then sealed up. A split groove kinematic coupling enabled the platen to be snapped into place in the case in the exact desired position without any chance of it shifting during encasement handling. Figure 20b shows the modular split groove elements and Figure 20c shows a platen with a document being snapped into place in an encasement.[6]

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[5] Both activities are fun to do with students (even one's own kids) as the bench level experiment devices can be used as kinematic paper holders and then given away as gifts!

[6] This encasement design project is the topic of Keith Durand's doctoral thesis with Prof. Slocum (in progress at this time).

Quasi Kinematic Couplings

Quasi-kinematic couplings (QKCs) utilize elastic/plastic deformation to emulate the performance of kinematic couplings whose grooves are mounted on flexures. Figure 21a shows how in a QKC the first instance of coupling is used to force hardened convexly curved surfaces into softer concave grooves to plastically form mating surfaces which upon releases of the load and the mechanism of elastic spring back form kinematic point contacts for subsequent couplings. One simple way QKCs can be made is to mate three hemispheres attached to a first component with three split cones in a second component. The hemispheres can look like mushrooms that are made from hard steel on a lathe and then pressed into bored holes. The split cones are made by casting or milling slots aligned with the coupling triangle's angle bisectors and then using a countersink tool to form the conical surfaces. The result is referred to as an *A-groove* as shown in Figure 21b. The mating of these surfaces of revolution defines a circular line contact. As a result, quasi-kinematic couplings use ball and groove geometries which are symmetric, making them easier to manufacture for less cost. The trade-off between KCs and QKCs is cost vs. constraint. The lower-cost geometries depart from the constraint characteristics of ideal kinematic couplings, yielding some degree of over constraint. However, with careful design, quasi-kinematic couplings can be made to emulate the performance of kinematic couplings [38,39,40,41].

KC and QKC coupling constraints are contrasted in Figure 21c. QKC's arc contacts differentiate the constraint characteristics of a quasi-kinematic coupling from a kinematic coupling. In an ideal, perfectly constrained kinematic coupling, the constraint forces are perpendicular to the bisectors of the coupling triangle's angles and permit unobstructed freedom of motion parallel to the bisectors. Any constraint parallel to the bisectors (e.g. along the grooves) over constrains the coupling. Ideally the ratio between the parallel and perpendicular constraint, called the *Constraint Metric* (CM) should be zero. The key to designing a good quasi-kinematic coupling is to minimize over constraint by minimizing over constraint parallel to the bisectors. In a QKC there is a balance between stiffness and over constraint for a given contact angle θ. Designs with contact angles less than 60 degrees (CM < 0.1) are typically safe choices. However larger contact angles may be used when one is willing to trade over constraint for higher stiffness.

Stiffness in a QKC is desired so when the coupling first occurs and the planar faces have not yet been forced into contact by the elastic compression of the balls into the grooves, off axis preload forces and torques (e.g., bolt torques) do not cause in plane deformations. In plane deformations would lead to a loss in accuracy and repeatability. The initial application for QKCs was to replace dowel pins in engine block assembly to achieve greater precision in boring the main bearing journals for the crankshaft as shown in Figure 21d. The repeatability of top-to-bottom half bearing alignment was increased from 5 microns using a doweled connection to 1.5 microns using QKCs. The cost was also reduced from several dollars per engine block to less than a dollar. QKCs are thus quite useful in applications where two parts are to be held together for machining, and then must be separated for assembling other parts into the system. The QKC then allows the machined parts to be re-mated with micron level precision and no chance for jamming as is common with dowel pinned assemblies.

Figure 21e shows an interesting variation on the QKC theme where bent metal tabs on gaskets to precisely align a gasket to one part by having bent-down metal tabs engage countersunk holes. A second part to be mated to the first part and sealed by the gasket between the two can then have countersunk holes that engage bent-up metal tabs in the gasket. Bolts can pass through the holes and be used to compress the entire assembly together.

Fluid and Electrical Interconnect Kinematic Couplings

Coupling modules together often involves distinct structural, fluid, and electrical connections; however, the chance for misalignment between the different coupling elements increases with the number of elements. If instead of distinct coupling elements for each type of connection, a single element could be used, robustness might increases: As shown in Figure 22, one coupling design parameter could satisfy several coupling functional requirements via the use of the Hertz contact zone. The Hertz contact zone is not a point, but rather a circle (in the case of a spherical contact on a flat groove), and if the center of the circle is removed, the contact becomes an annulus. The annular contact rings at each of the six contacts of a kinematic coupling thus serve as mechanical constraints. The high contact stresses around the annuli can also effectively seal high pressure fluids that could pass through the centers in the same manner that fluid quick connects work. In addition, the high contact stresses around the annuli create contact areas that can allow high electrical currents to pass [42].

Kinematic couplings could prove to be particularly useful for electric cars where it is proposed that battery modules are swapped at battery stations so drivers will not have to wait for a charge. A kinematic coupling could provide the robust structural interface needed to precisely hold a heavy battery in position, even under bumpy ride conditions, while maintaining electrical and even fluid connections. In fact, the initial microsliding that occurs between the contact regions during coupling would act as a scrub to help maintain good electrical contacts.

Kinematic Coupling Standards

There are many applications for kinematic couplings, and is often the case with a mechanical element, the need for common use by many often leads to a standard. In the case of pods to hold semiconductor wafers during manufacturing, the author found himself struggling to design a wafer transfer robot for 300 mm wafers that could operate within the allocated error of a tight system error budget. Previously this was not a problem for 150 mm wafers, but for 300 mm wafers the pitch of the wafers in the pod was the same but the larger diameter wafer meant that tilt errors are amplified more at the wafers' edges and that could lead to a crash between the wafer and the slot in the pod. In such crashes, the robot wins, and the wafer and system designer lose. The author had done error budgets for 150 mm systems and knew that the fat-rabbit error was the "H-Bar" in the base of the pod. The H-Bar was an H-shaped protrusion in the base of the pod that mated with an H-shaped female in the tool which received the pod.

The author and his student Michael Chiu made a bench level prototype pod using a conventional pod retrofitted with three small plastic hemispheres that mated with wooden grooves on a pierce of plywood. The repeatability was an order of magnitude better than a production H-Bar coupling could achieve. Working with Rick Scott and

Ron Billings at SEMATECH, we were able to in the short time of a few months convince the entire semiconductor industry that the transition to 300 mm wafers could only be reliably accomplished if wafer carrying pods (FOUPs) mated to tools using a standardized kinematic coupling interface. This is illustrated in Figure 23 and is represented by SEMI E57-1296, a kinematic coupling standard for wafer transport pods.

Kinematic couplings have been shown to be a viable option for rapid placement of factory tools, such that robots, for example, can be calibrated off line to a standard kinematic mount and then rapidly swapped on the factory floor without the need for manual recalibration. Standards could readily be developed for such an interface to be used widely by industry [43,44].

Conclusions

Kinematic couplings have been known for many many years and have had many many applications. Much has been published on their design and use, and many patents have been issued for unique embodiments[7]. Typically they have been designed for specific purposes, but as in the case of wafer pod carriers (FOUPs), a generic use led to the creation of a very useful standard. There are many other applications for kinematic couplings, modular elements, and their variations that may also benefit from standards which could help make them more easily used. For applications ranging from tooling interfaces to quick-connect batteries for automobiles, the creation of kinematic coupling standards could help more designers take advantage of the unique properties of these robust versatile deterministic machine elements.

Acknowledgements

Over the decades many people have contributed to the body of knowledge that makes kinematic couplings (and their mechanical relatives) possible, and I sincerely apologize if I have missed referencing other important sources. I was introduced to kinematic couplings while I was working at NBS (now NIST) by my late boss Donald Blomquist and Dr. Robert Hocken. They challenged me to write a spreadsheet for the generalized design of KCs and the rest is history. Since then, many of my former students and other researchers, have helped develop the technology, and there is still much to be done! For preparation of this article, I would especially like to thank R. Peter Bailey, Danny Braunstein, Michael Chiu, Martin Culpepper, Keith Durand, David Gill, Layton Hale, Ryan Vallance, and Alexis Weber.

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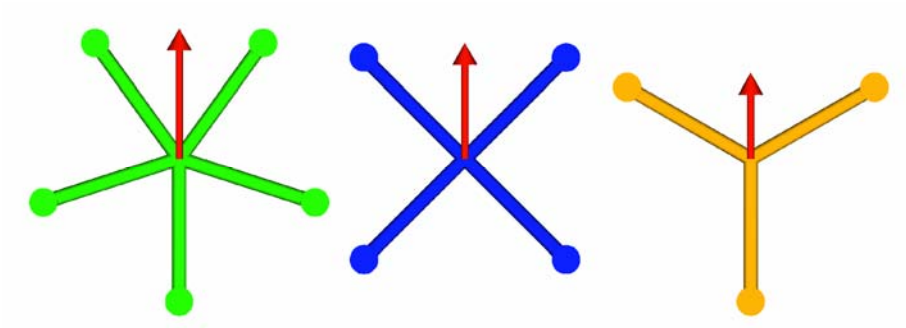


Figure 1a Support legs and stability arrows for chair legs ranging from elastically averaged to kinematic

Figure 1b Kinematic design principles applied to create a collapsible camping stool as a modern variation of the classic three legged milking stool ☺

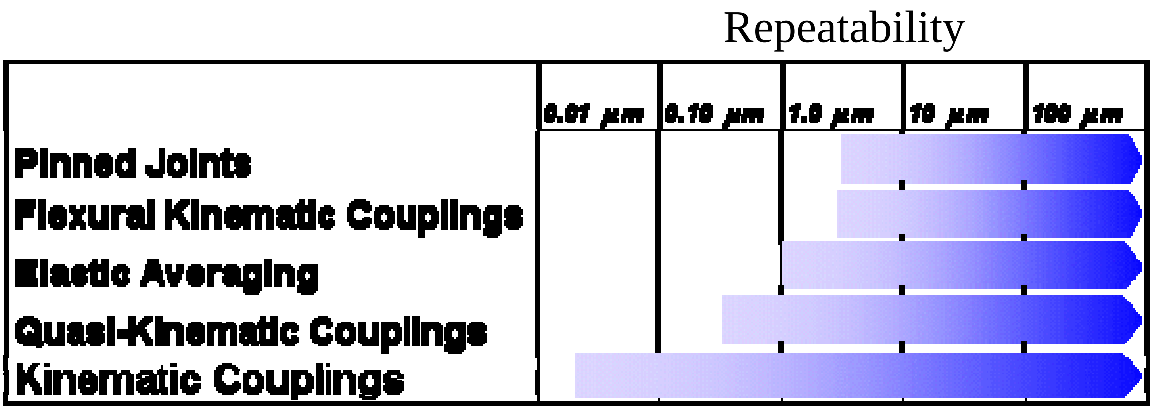


Figure 2 Relative repeatability of different types of connections. (Courtesy M. Culpepper)

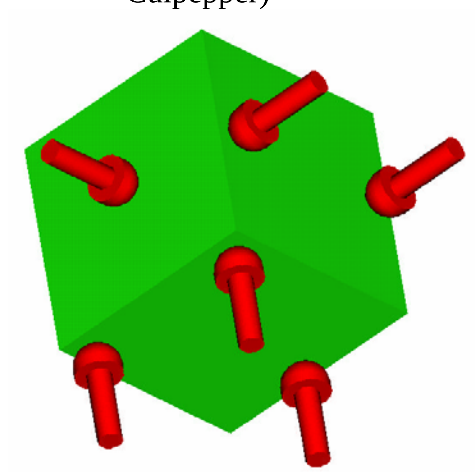


Figure 3 3-2-1 Fixturing principle applied to a cube

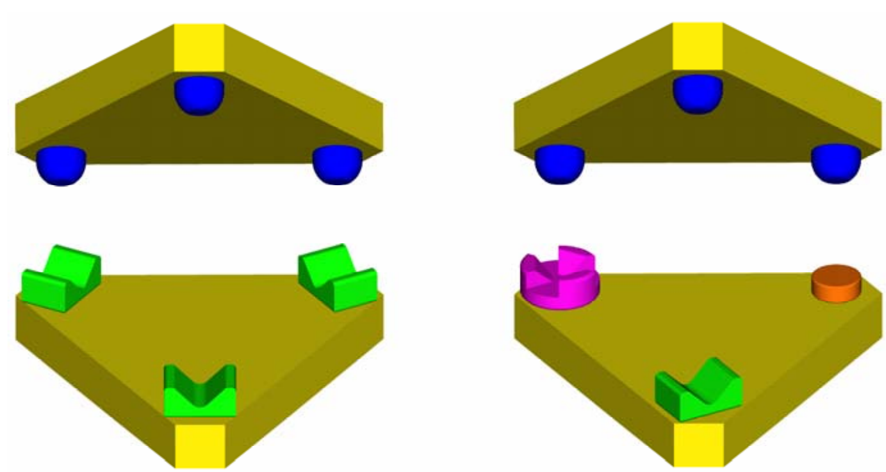


Figure 4 *Three Groove* (l) and *Kelvin Clamp* (r) kinematic couplings.

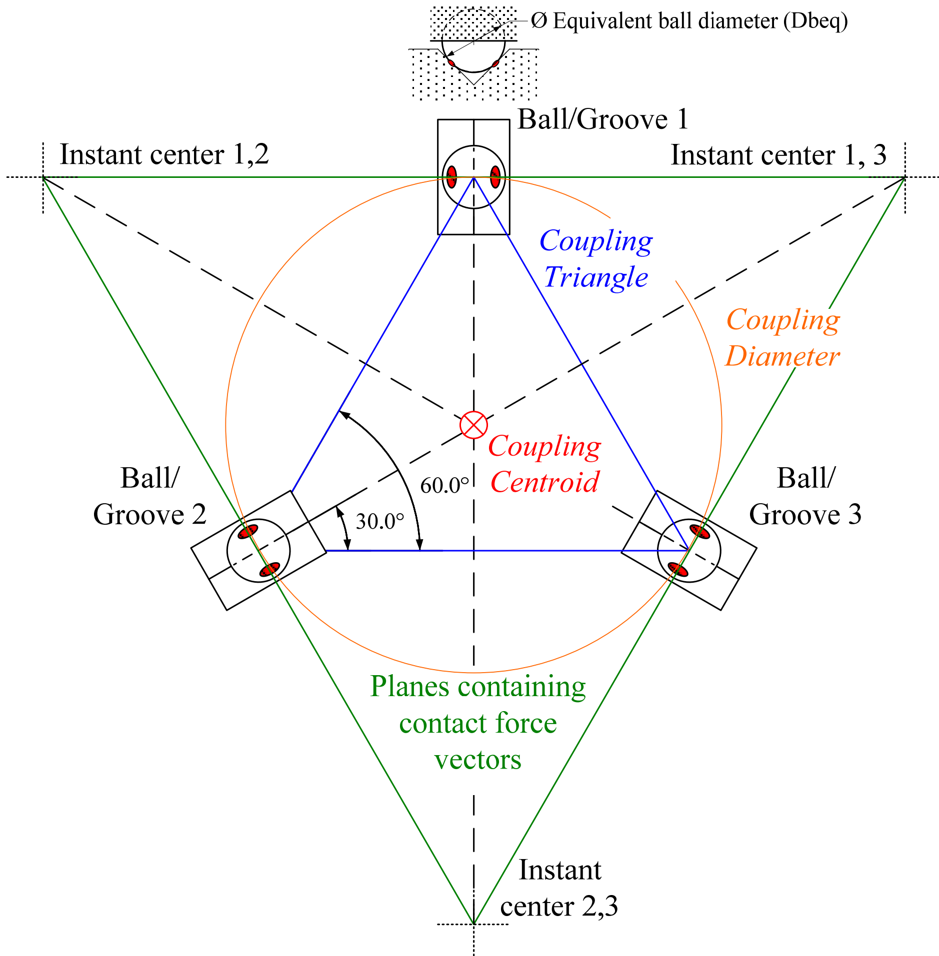


Figure 5 Geometric parameters for a three groove kinematic coupling.

System geometry (XY plane is assumed to contain the ball centers)				
Dbeq (mm) =	**20**	Equivalent diameter ball to contact the groove at the		
Rbminor (mm) =	**500**	"Ball" minor radius		
Rbmajor (mm) =	**500**	"Ball" major radius		
Rgroove (mm) =	**1.00E+06**	Groove radius (- for a trough)		
Costheta =	**TRUE**	Ball major radius along groove?		
Dcoupling (mm) =	**250**	Coupling diameter		
Fpreload (N) =	**-2000**	Preload force over each ball		
Xerr (mm) =	**0.0**	X location of error reporting		
Yerr (mm) =	**0.0**	Y location of error reporting		
Zerr (mm) =	**0.0**	Z location of error reporting		
Auto select material values (enter *other_4* to the right)				
Matlabball =	**2**	Enter 1 for plastic, 2 for steel, 3 for carbide, 4 for user		
Matlabgroove =	**2**	defined, 5 for each ball and groove to be defined		
Min. yield strength (Pa, psi)		1.72E+09	249,855	
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**

Figure 6a Inputs to kinematic_coupling_3Groove_design.xls

Results: Hertz stresses and deformations								
Error displacements at the point of interest (micron)								
DeltaX	-5.63E-09	DeltaY	-5.63E-09	DeltaZ	-5.64E-09		Max	5.64E-09
Largest contact ellipse major diameter (mm)			3.380					
Largest contact ellipse major diameter (mm)			3.373					
Max shear stress/(ult. tensile/2)			0.083					
RMS applied force F (N)			17.32					
RMS deflection at F (micron)			0.010					
RMS stiffness (N/micron)			1,774.36					
				Max shear/(ult. tensile/2)	*Deflection (+into ball) (m)*		*Contact ellipse*	
Groove normal forces (N)		*Contact stress (Pa)*					*Rmajor (m)*	*Rminor (m)*
Fbnone	*1.42E+03*	sigone	*2.38E+08*	*0.083*	delone	*1.63E-08*	*1.69E-03*	*1.69E-03*
Fbntwo	*1.41E+03*	sigtwo	*2.37E+08*	*0.083*	deltwo	*-8.80E-09*	*1.69E-03*	*1.68E-03*
Ball-Groove 2							*Contact ellipse*	
Groove normal forces (N)		*Contact stress (Pa)*			*Deflection (m)*		*Rmajor (m)*	*Rminor (m)*
Fbnthree	*1.41E+03*	sigthree	*2.37E+08*	*0.083*	delthree	*-1.54E-08*	*1.69E-03*	*1.68E-03*
Fbnfour	*1.40E+03*	sigfour	*2.37E+08*	*0.083*	delfour	*-2.46E-08*	*1.68E-03*	*1.68E-03*
Ball-Groove 3							*Contact ellipse*	
Groove normal forces (N)		*Contact stress (Pa)*			*Deflection (m)*		*Rmajor (m)*	*Rminor (m)*
Fbnfive	*1.41E+03*	sigfive	*2.37E+08*	*0.083*	delfive	*-1.98E-08*	*1.68E-03*	*1.68E-03*
Fbnsix	*1.42E+03*	sigsix	*2.38E+08*	*0.083*	delsix	*1.45E-08*	*1.69E-03*	*1.69E-03*
Results: Error motions								
Error motions at X,Y,Z coordinates:		*0.000*	*0.000*	*0.000*				
deltaX (m)	-5.63E-09	RMS Δ (m)	9.76E-09					
deltaY (m)	-5.63E-09	*Homogenous Transformation Matrix:*						
deltaZ (m)	-5.64E-09	1.00E+00	-2.70E-14	-5.69E-08	-5.63E-09			
EpsX (radian)	5.69E-08	2.70E-14	1.00E+00	-5.69E-08	-5.63E-09			
EpsY (radian)	-5.69E-08	5.69E-08	5.69E-08	1.00E+00	-5.64E-09			
EpsZ (radian)	2.70E-14	0.00E+00	0.00E+00	0.00E+00	1.00E+00			

Figure 6b Outputs from kinematic_coupling_3Groove_design.xls

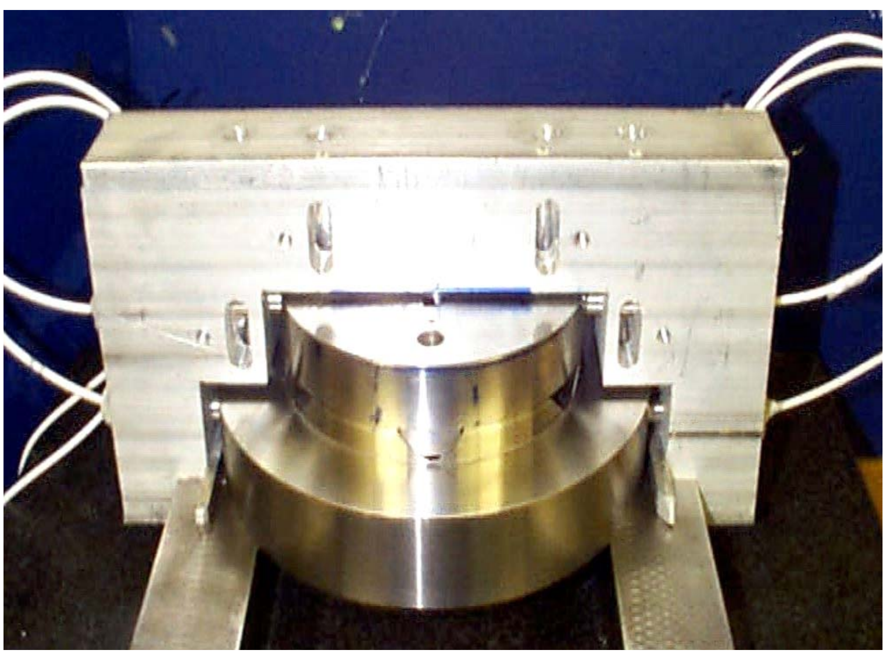


Figure 7a Three Groove kinematic coupling with canoe balls. (Courtesy B. Mullenheld)

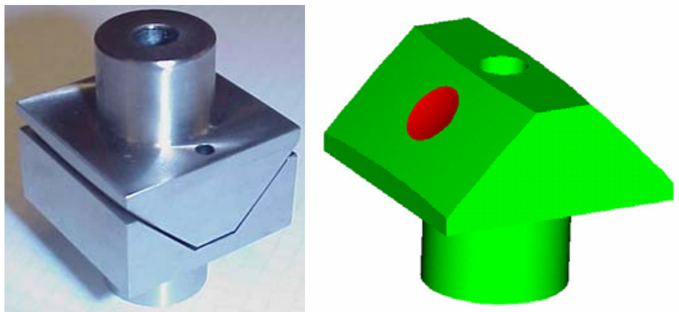


Figure 7b Modular "Canoe ball" and vee (l) and solid model showing Hertz contact zone (r) used in the kinematic coupling of Figure 7a.

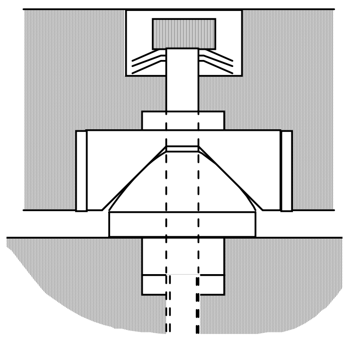


Figure 8 Preloading a kinematic coupling through the coupling elements to avoid deforming the rest of the structure.

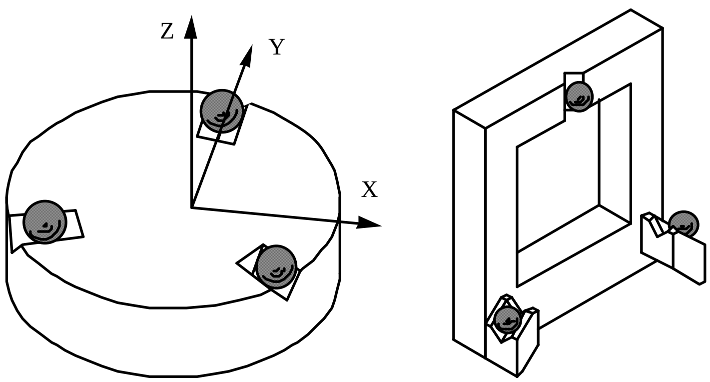


Figure 9 Horizontal (l) and vertical (r) kinematic couplings.

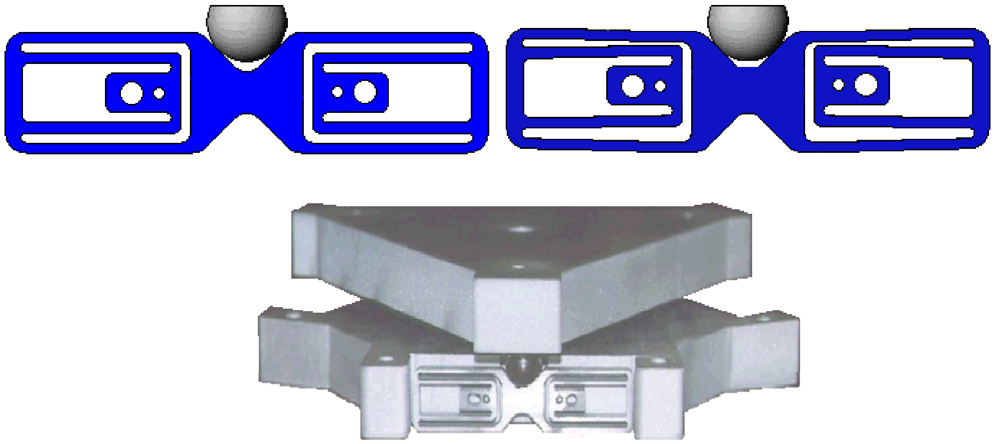


Figure 10 Kinematic coupling element supported by flexural bearings. (Courtesy D. Braunstein)

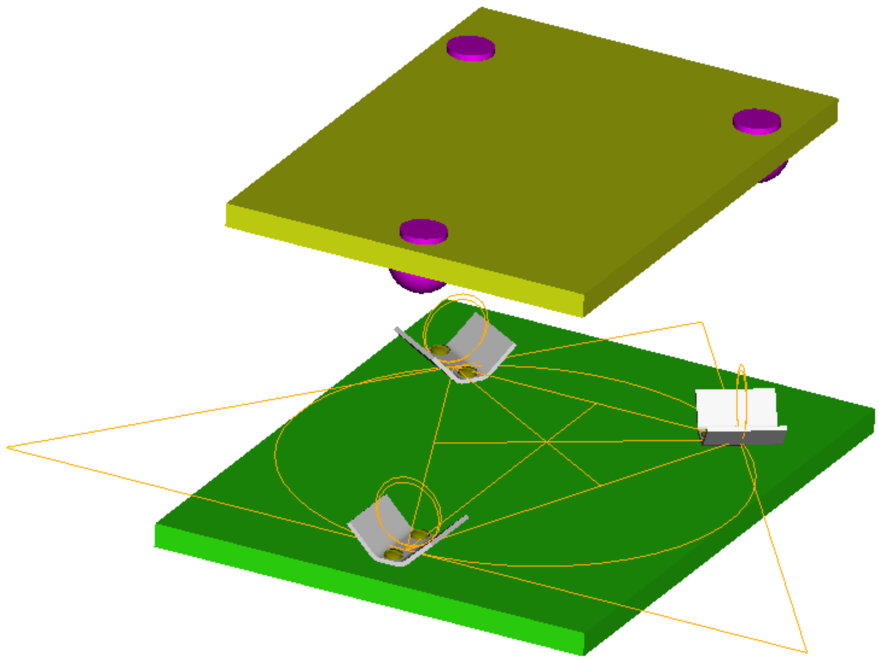


Figure 11 Low cost kinematic coupling elements supported by flexural vees

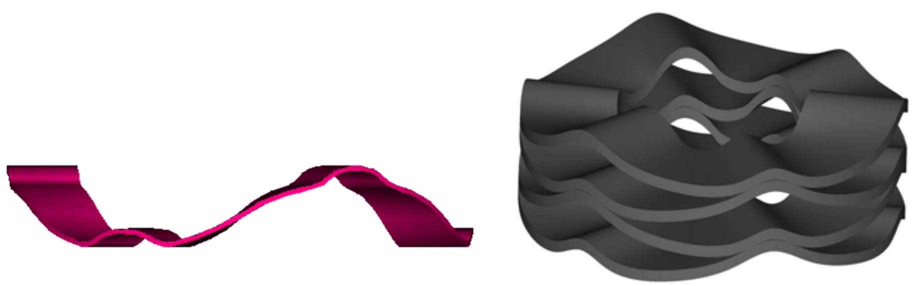


Figure 12a Kinematic spring washer concept. (Courtesy R. Bailey)

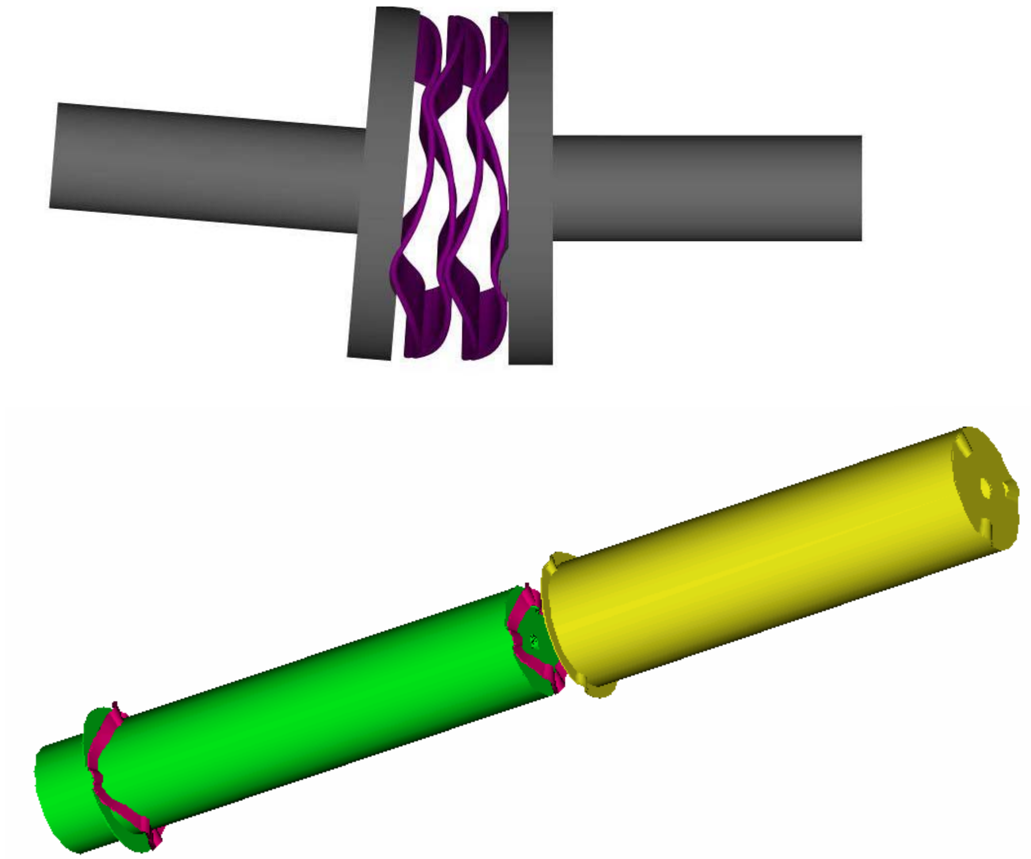


Figure 12b Kinematic spring washer concept used to couple and align shafts while also acting as a slip clutch. (Courtesy R. Bailey)

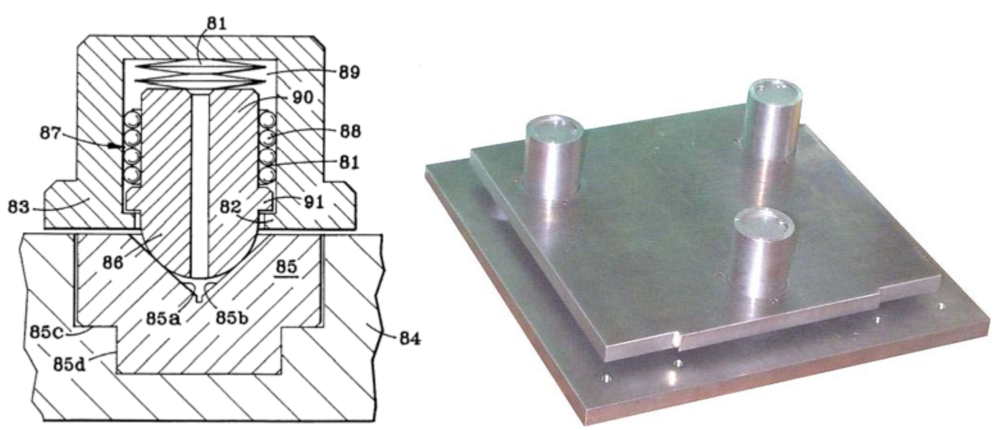


Figure 13 Kinematic coupling element constrained to move axially by a die-set bushing and spring washer stack. (Courtesy D. Braunstein)

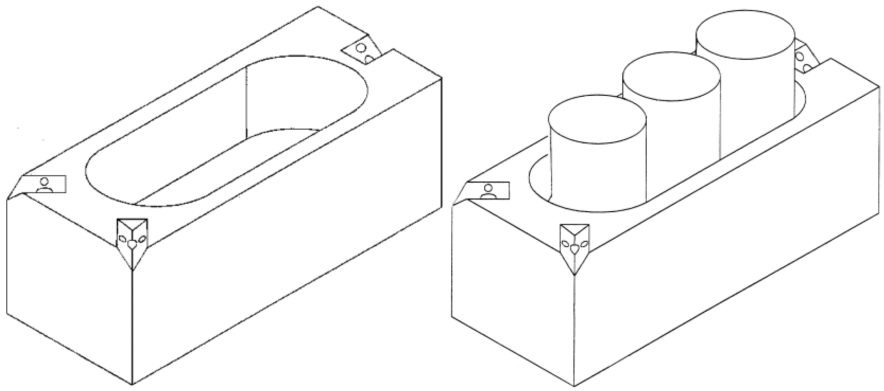


Figure 14 Kinematic coupling elements can be used to precisely locate sand cores to each other for precision casting..

Figure 15 Silicon compliant pseudo kinematic coupling elements for locating wafers with respect to each other prior to bonding: wafers stacked (l), KOH etched "males" on silicon cantilevers (m) and KOH etched "females" (vees) (r). (Courtesy A. Weber)

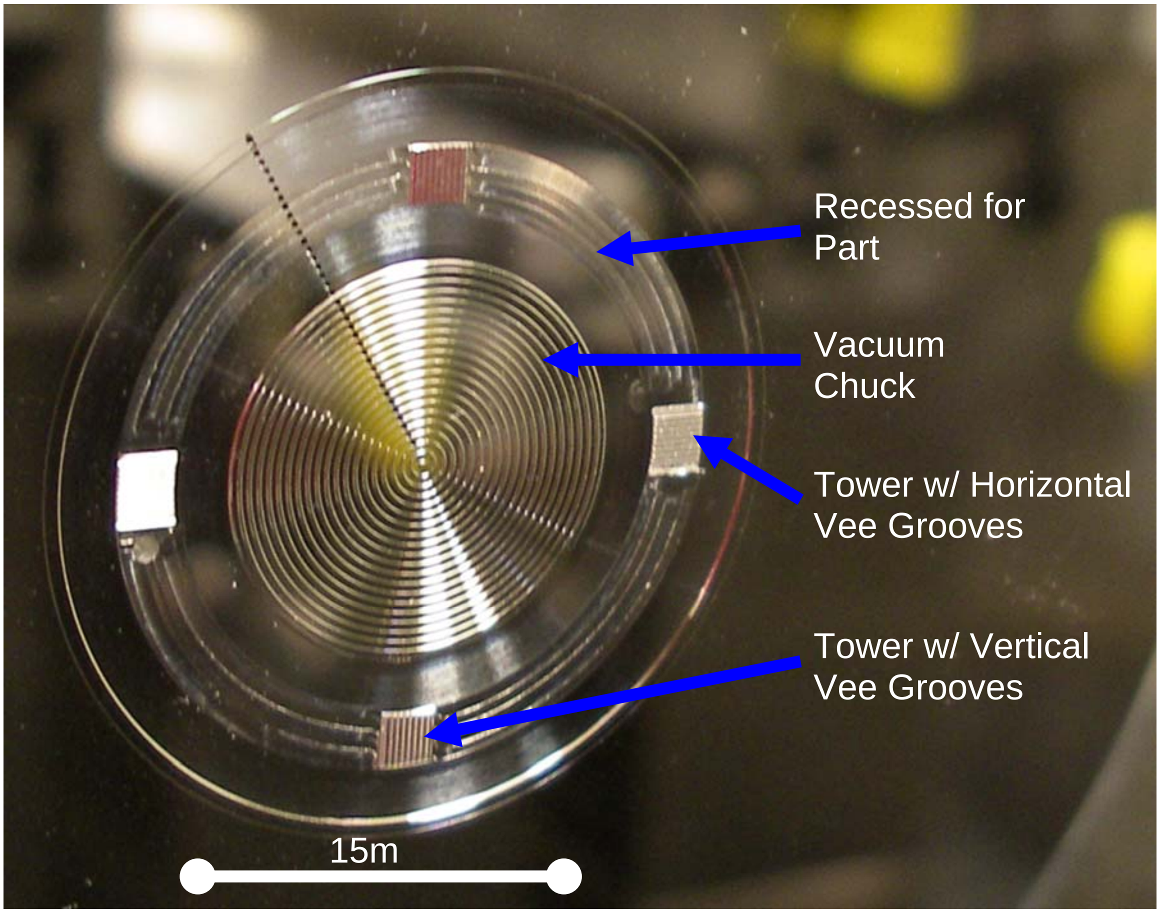


Figure 16a Zia pseudo kinematic coupling for fixturing and indexing. (Courtesy D. Gill)

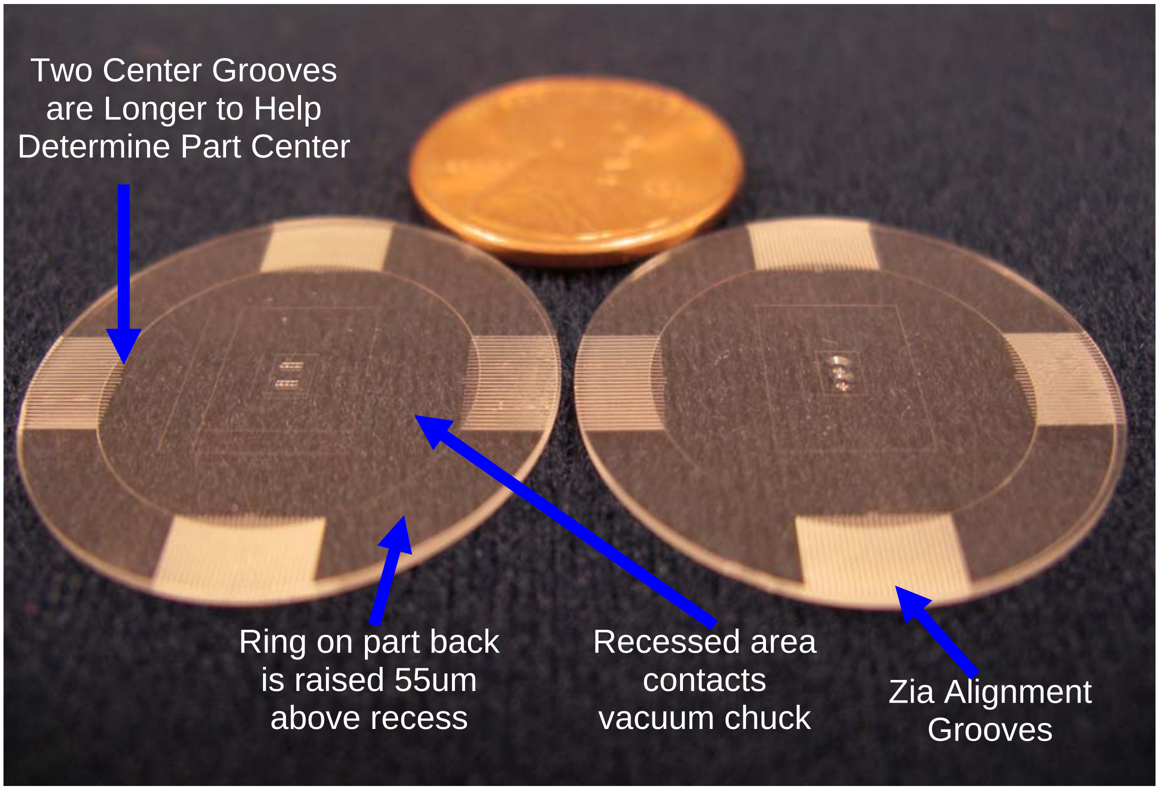


Figure 16b Zia coupling on parts to be fixtured. (Courtesy D. Gill)

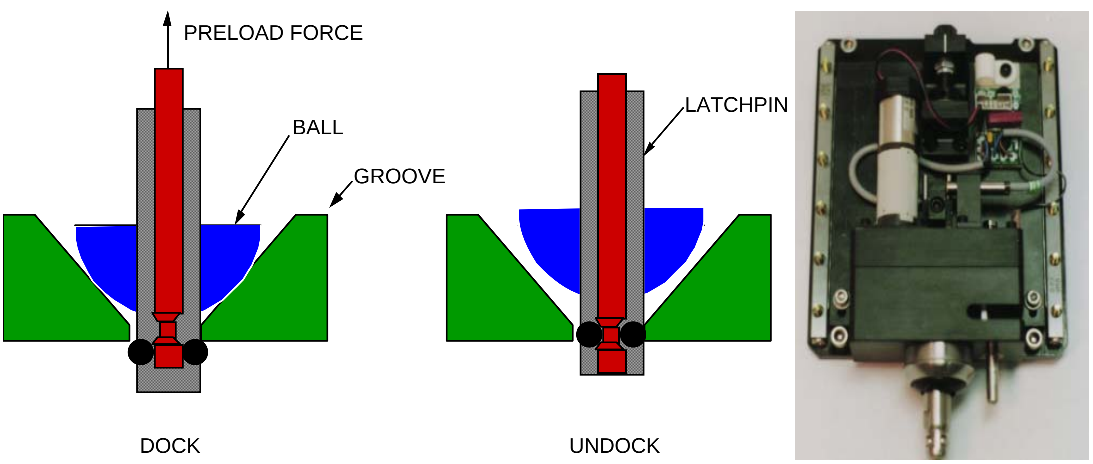


Figure 17a K-Dock servo controllable kinematic coupling. Operating principle (l) and unit module (r). (Courtesy M. Chiu)

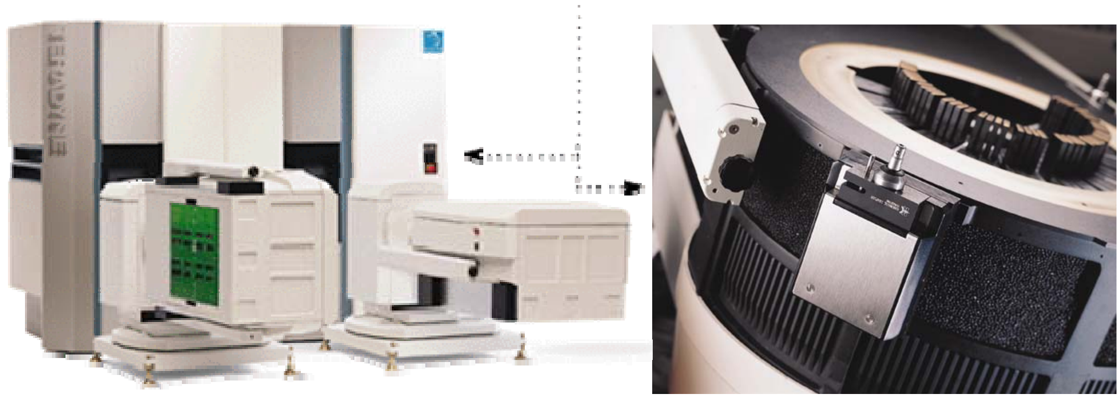


Figure 17b K-Docks used in precision alignment application for semiconductor test equipment. (Courtesy M. Chiu)

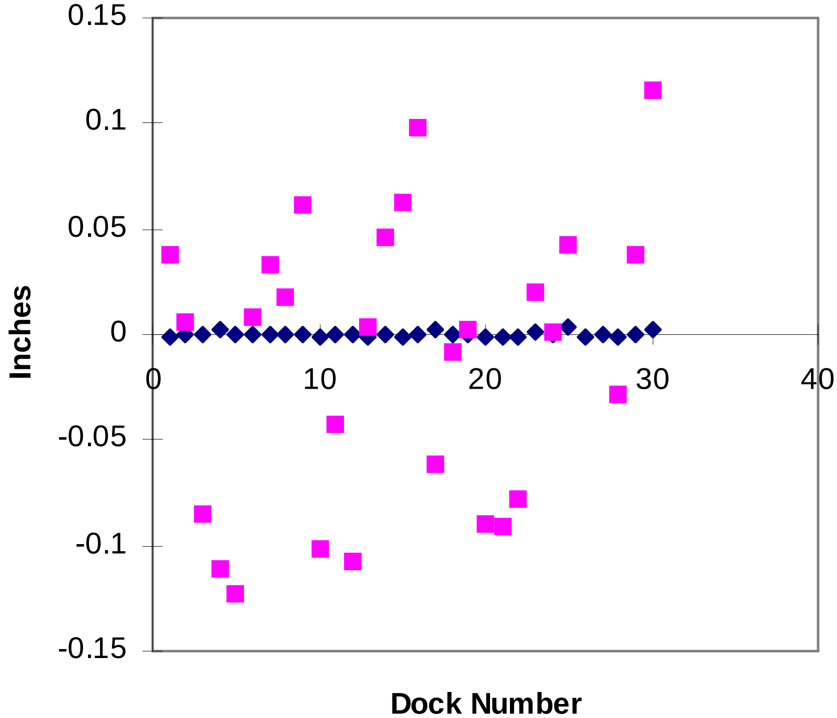


Figure 17c Repeatability of K-Dock couplings compared to conventional J-ring overconstrained coupling. (Courtesy M. Chiu)

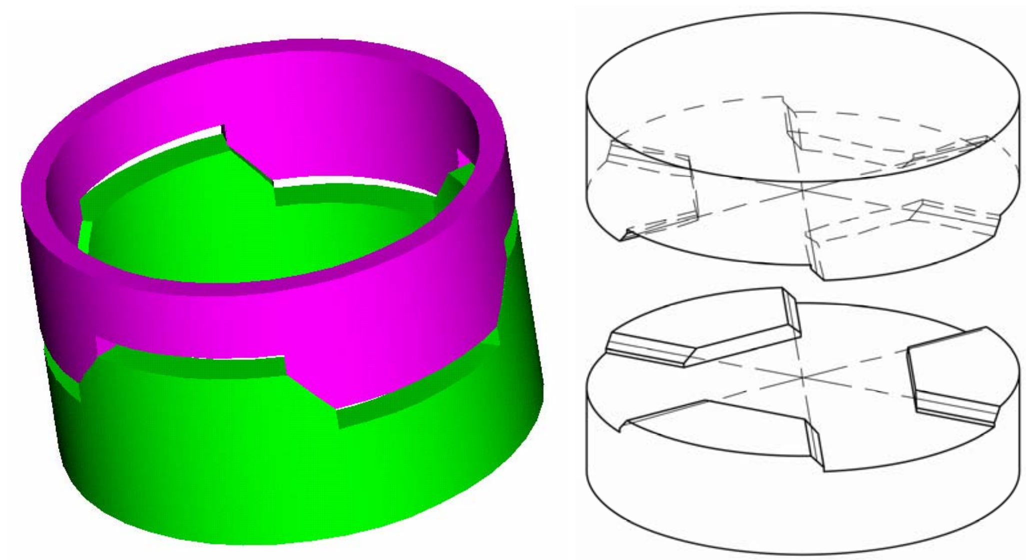


Figure 18 Three tooth kinematic coupling: solid model of low cost planar face contacts (l) and ideal line contact model (r). (Courtesy L. Hale)

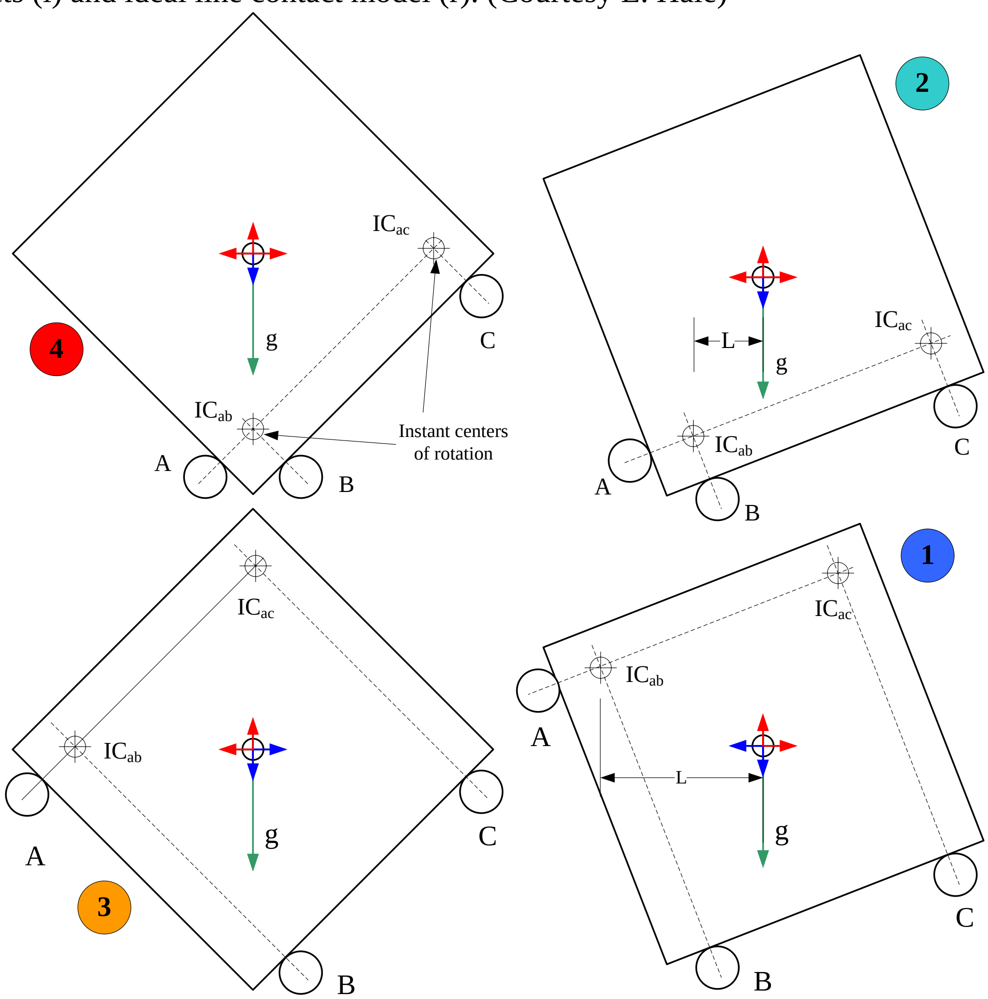


Figure 19a Planar kinematic coupling scenarios

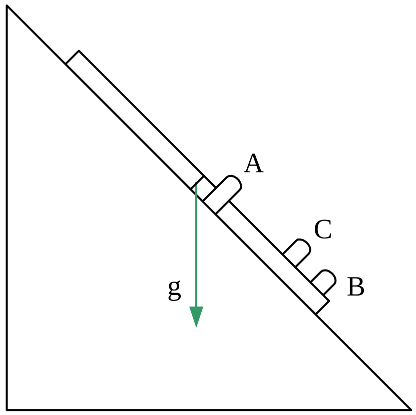
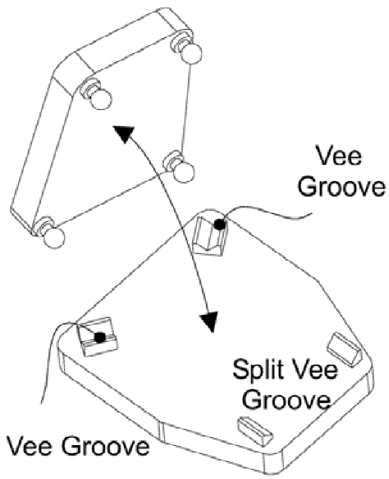


Figure 19b Bench Level Experiment for planar kinematic coupling scenarios



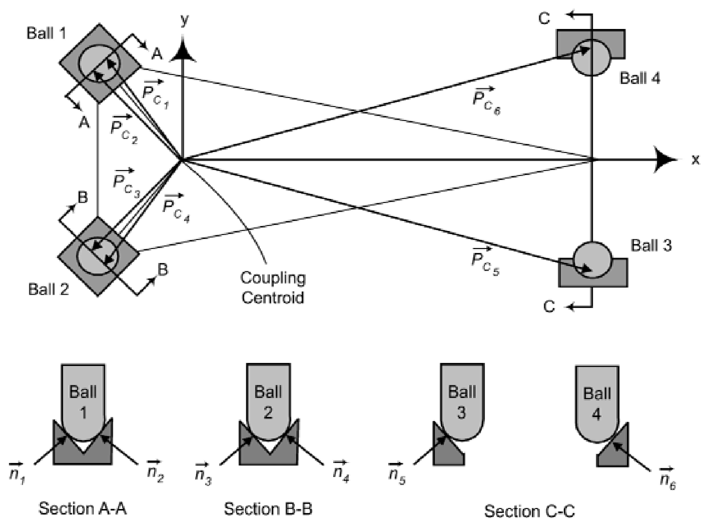


Figure 20a Split groove kinematic coupling concept (l) and schematic (r). (Courtesy R. Vallance)

Figure 20b Split groove kinematic coupling elements made from a conical male (l) and mating grooves made from slots milled with a V cutter (r). A ball-spring plunger provides preload and manually removable snap-fit locking (Courtesy Massachusetts State Archives)

Figure 20c A platen holding a priceless document is held in position using split groove kinematic couplings in a special environmental display case (Courtesy Massachusetts State Archives)

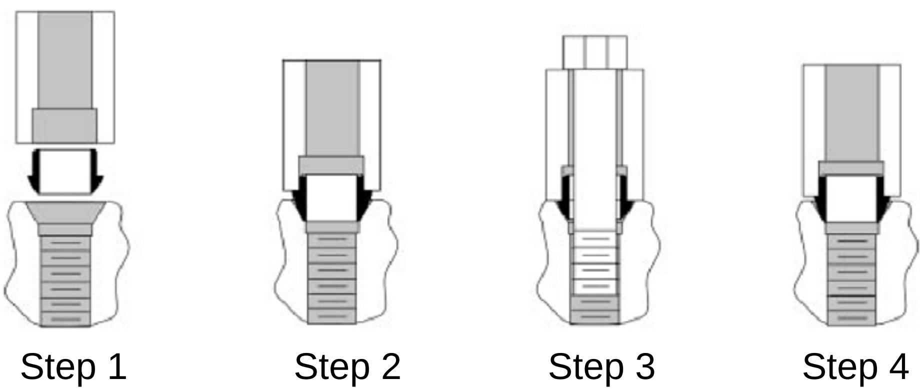


Figure 21a Quasi kinematic coupling contact steps. (Courtesy M. Culpepper)

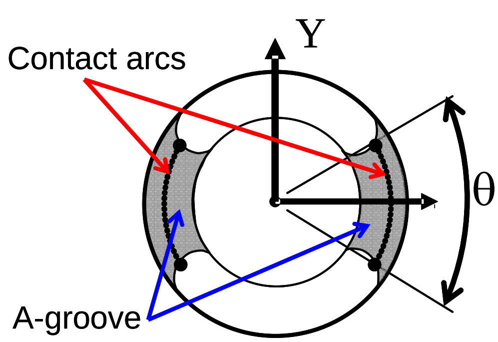


Figure 21b Quasi kinematic coupling contact geometry. (Courtesy M. Culpepper)

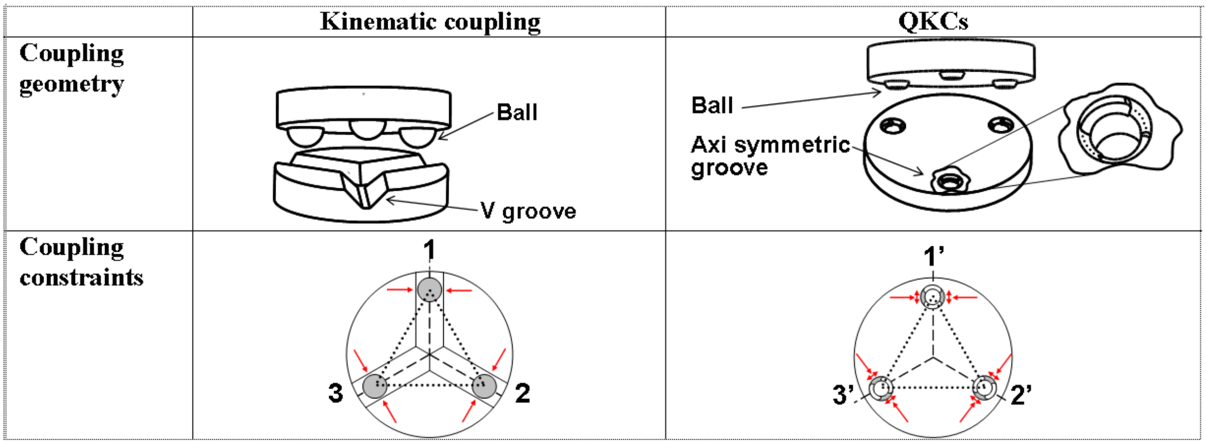


Figure 21c Comparison of kinematic and quasi kinematic coupling contact geometries. (Courtesy M. Culpepper)

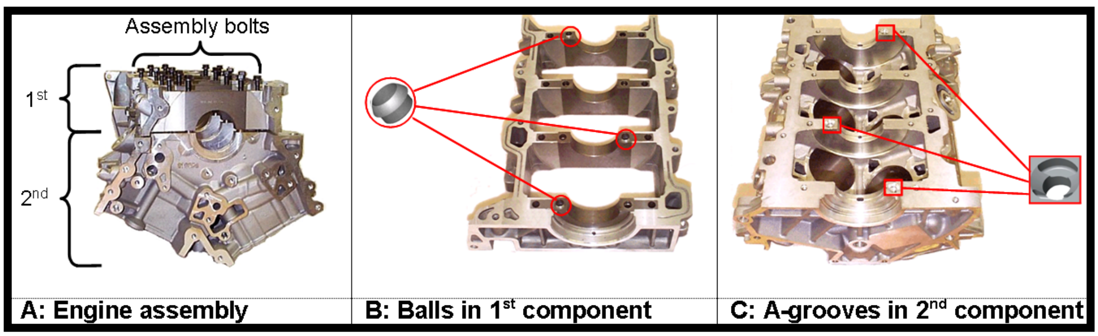


Figure 21d Application of kinematic and quasi kinematic couplings to engine assembly.(Courtesy M. Culpepper)

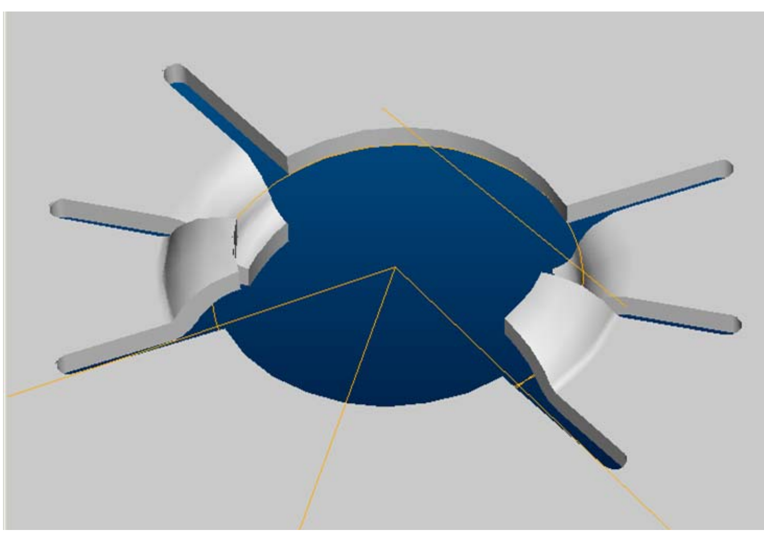


Figure 21e Sheet metal components of a gasket can be formed to locate the gasket in a conical hole in one part, and then locate a second part with similar conical holes to the gasket to form a quasi kinematic assembly.

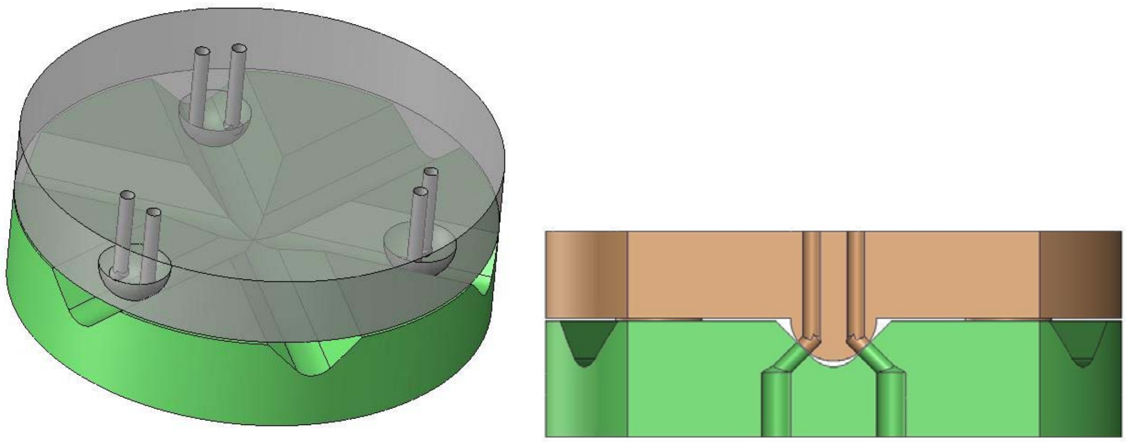


Figure 22 Kinematic fluid coupling concept

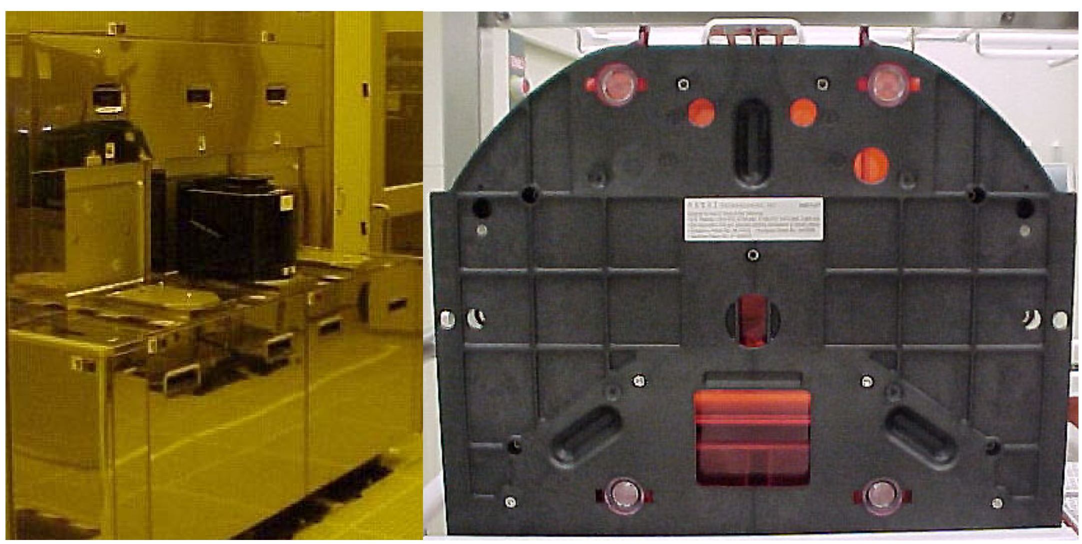


Figure 23 Mating kinematic coupling grooves on the wafer carrying pod (FOUP) enable precise alignment on the load ports of semiconductor processing equipment, so wafer handling robots can precisely access 300 mm silicon wafers. (Courtesy SEMATECH)