QUASI-KINEMATIC COUPLINGS FOR PRECISION AUTOMOTIVE ASSEMBLIES

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ABSTRACT

There are an innumerable number of precision automotive assemblies which use pinned joints for alignment. Many of these suffer from poor performance and cost/quality issues because pinned joints are inherently over-constrained. This paper introduces the Quasi-Kinematic Coupling (QKC) as a solution for precision (better than 10 microns) automotive assemblies. For these applications, QKCs require fewer parts, have a lower total cost, and provide sub-micron repeatability.

This paper covers design and manufacturing considerations for automotive QKCs. Proof of concept is demonstrated via a case study on the block-main bearing assembly of a Ford six cylinder engine. Repeatability tests yield average QKC bearing center line repeatability of 0.65 microns at the outer journal bearings as compared to 4.85 microns for pinned joints. This QKC uses 60% fewer precision features, costs 36% less, and allows feature size and placement tolerance ranges which are twice as wide as those of the previous pinned joints.

NOMENCLATURE

G_{max}	Maximum gap between block and bedplate
G_{min}	Minimum gap between block and bedplate
K_a	Stiffness in axial direction
K_c	Stiffness perpendicular to bearing center line
R_a	Repeatability in axial direction
R_c	Repeatability perpendicular to bearing center line
δ_a	Alignment error in axial direction
δ_c	Alignment error perpendicular to bearing center line
δ_{GMS}	Gap margin of safety
δ_v	Controllable gap variation

1. INTRODUCTION - THE NEED FOR REPEATABILITY

For automakers to maintain a competitive advantage, the ability to repeatably locate and position parts is crucial. As a result, better precision at lower cost is a driving force in automotive manufacturing. Many applications (i.e. block to main bearing assemblies) require better than 10 micron repeatability. The absence of a low-cost means of satisfying this requirement has motivated the first application of a Quasi-Kinematic Coupling (QKC) to an automotive application.[1]

2. COUPLING REQUIREMENTS AND STANDARD METHODS

2.1 Kinematic Constraint and Kinematic Couplings

Precision alignment requires the kinematic constraint of six degrees of freedom. This can be accomplished by establishing six points of contact, as done in a kinematic coupling (see Fig. 1) via the contact of three balls and corresponding V grooves (Slocum, 1988). However, these couplings are not suited to high volume applications which require low cost or sealing between components. This has been addressed for some applications by flexural kinematic couplings (Culpepper et. al., 1998) and (Slocum et. al., 1992) but, due to the cost

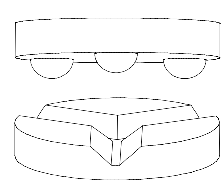

Figure 1 Kinematic Coupling

1. US patent pending by Culpepper, Slocum, Shaikh, Vrsek

of the integrated flexures, they are not suited for mass production.

2.2 Pinned Joints

Traditionally, manufacturers have used pinned joints which rely on elastic averaging instead of kinematic principles. As a result, the practical repeatability of two close-fit pin-hole joints is limited to about 10 microns. Better performance is costly as the additional elastic averaging of more joints (5 microns requires about eight joints) is required (Slocum, 1998). However, with no alternatives, manufacturers have been left with little choice but to suffer their use.

2.3 Quasi-Kinematic Couplings

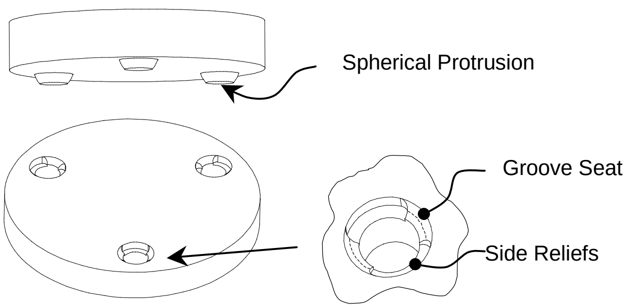

Figure 2 QKC With Detail of Conical Groove

In general, QKCs[2] are well-suited for high volume precision assembly, particularly where integral locating features and/or sealing contact are required. QKCs consist of three spherical protrusions and corresponding *conical grooves*. Mating these components results in six arced lines of contact (dotted lines in Fig. 2) not six points as in a kinematic coupling. The result is a weakly over-constrained coupling whose behavior approaches that of a kinematic coupling as the contact angle shown in Fig. 3 decreases (Culpepper, 1999).

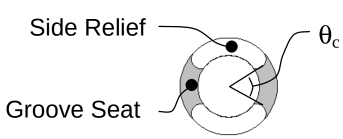

Figure 3 Contact Angle of QKC Groove

3. QUASI-KINEMATIC FUNCTION AND CHARACTERISTICS

Due to friction and surface irregularities (surface finish affects (Slocum, 1992)), when the coupling is first mated as in Fig.4-A, the components will not occupy the most stable equilibrium. In force driven coupling proper seating can be induced by a preload that overcomes the contact friction and causes the spherical elements to brinell out surface irregularities at the contacts.

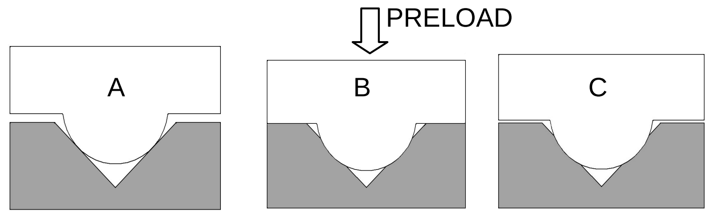

Figure 4 Mating Cycle of QKCs

If mating of opposed faces is desired (i.e. for sealing or stiffness) the gap between components and the compliance of the kinematic elements can be chosen such that the preload will close the gap (Fig. 4-B). This is termed displacement driven coupling. On removal of the load, all or part of the gap is restored through elastic recovery of the kinematic elements (Fig. 4-C), thereby preserving the kinematic nature of the joint for subsequent mates. If the initial deformation is elastic, the whole gap will be restored. If elastic and plastic, only a portion of the gap will be recovered.

With the gap closed, high stiffness can be achieved. This is because the coupling stiffness will depend on interactions at the contacting surfaces, not the moderately stiff quasi-kinematic interfaces. As such, the stiffness in the direction perpendicular to the plane of the mated surfaces depends primarily on the stiffness of the clamping method, and the stiffness in directions contained in the plane of the mated surfaces depends on the contact friction between the components (Culpepper et. al., 1998).

4.METHODS FOR MANUFACTURING QKC ELEMENTS

4.1 Manufacture of Conical Grooves

Figure 5 illustrates an inexpensive method for making QKC groove seats, where cast-in reliefs are machined with a form tool. This process has the benefit that grooves can be formed simultaneously with drilled holes. As shown in section 5.3, this makes the switch to QKCs possible with minor changes in the manufacturing process.

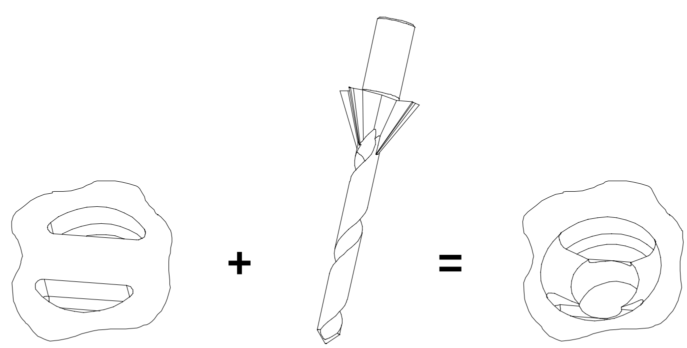

Figure 5 Features and Tool Used to Make QKC Grooves

2 US patent pending by Culpepper, Slocum

4.2 Manufacture of Spherical Members

Instead of machining spherical features, crowned pegs as in Fig. 6 can be pressed into one component. Note the peg is equipped with a lip to establish position when pressed in. The axisymmetric geometry of this part and high volumes (3 per engine required) make it well suited for production on screw machines. When purchased in volume, these pegs can be attained for roughly the same cost as hollow dowels.

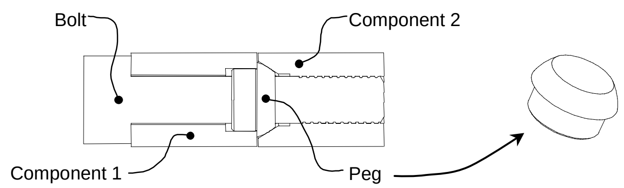

Figure 6 Cross Section of QKC Joint

With respect to material, the properties of the pegs are important. If the peg plastically deforms during the initial mate, the resulting indentions on the peg's surface will catch at random locations on the edges of the conical grooves, thereby decreasing repeatability in subsequent mates (Culpepper, 1999). Making the peg of harder material than the grooves can help prevent this.

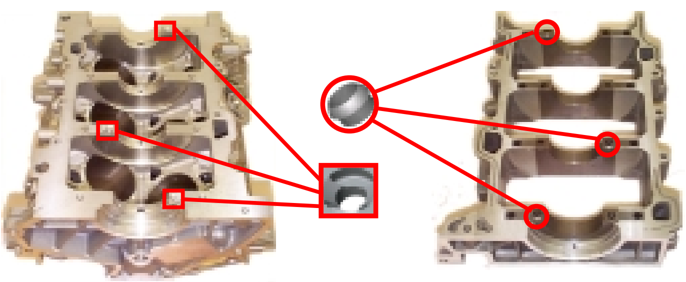

Figure 7 Block and Bedplate With QKC Elements

4.3 Placement of QKC Joints

The choice of QKC joint locations should be made such that the joints form the largest and most near equilateral triangle to maximize resistance to errors from assembly loads. For the components in Fig. 7, the ideal location would be coaxial with bolt holes on the periphery of the engine. However, the only space in which the pegs would not interfere with tooling on the transfer line was on the inner bolt holes directly adjacent the crank bore.

Tolerances on the location of QKC joints are less sensitive to misalignment than pinned joints, as the pegs can easily fit into a conical hole which is slightly misaligned; then force it to conform during the initial deformation (Culpepper 1999). Pinned joints are incapable of eliminating initial misalignment. Depending upon the application, the tolerance range for QKC placement can be a factor of two to three times wider.

5. DESIGN METHOD AND ISSUES FOR QKCS

Following is a synopsis of the design method for sub-micron QKCs. Thorough coverage can be found in reference 2.

5.1 STEP I Constraints and Functional Requirements

Before designing the coupling, one should list the constraints (i.e. cost and space) and functional requirements (i.e. repeatability, sealing) and the interactions between them. This forces consideration of all aspects of the design and prepares the designer for possible trade-offs between competing constraints, requirements, and other factors. A more rigorous method for structuring the design is Axiomatic Design (Suh, 1990). Although this method is very useful, it is just becoming widely used in industry, and is therefore only mentioned as an alternative for those familiar.

5.2 STEP II Coupling Geometry

Generally, one first chooses the location of the joints and the orientation of the conical grooves (see section 4.3 for example) which maximize the stiffness of the coupling. Then other geometric values, such as the dimensions of the kinematic elements, the cone angle of the groove, and other quantities are chosen to maximize the contact stiffness.

5.3 STEP III Force or Gap Choice

For force induced coupling, one chooses a preload force which is sufficient to overcome the friction at the peg-cone contact lines and ensures that the surface irregularities are brinelled smooth. In displacement driven coupling, an analysis is done to determine the range of variation of the gap, δ_V, as a function of the variation in kinematic element geometry, the location of the elements, and the depth of the conical grooves. The gap is directly dependent upon the depth of the conical grooves, but less so upon other factors (via dependence on the grooves cone angle). As such, first efforts should be concentrated on groove depth control. Next, to ensure the existence of a gap, one specifies the minimum gap, G_{min}, to be greater than the controllable gap variation, δ_V, by some finite margin δ_{GMS}. From Eqs. (1) and (2), the maximum gap, G_{max} is found.

$$G_{min} = \delta_V + \delta_{GMS} \tag{1}$$

$$G_{max} = \delta_V + G_{min} \quad = \quad 2 \times \delta_V + \delta_{GMS} \tag{2}$$

5.4 STEP IV Plastic Deformation

It is necessary to ensure that the given geometries and material properties of the kinematic elements do not result in plastic deformation of the peg surfaces. In displacement driven coupling, one must either lower the maximum gap or make one or more of the following changes:

1. Increase the radii of the grooves and/or spheres

2. Increase the yield stress of the spherical member
3. Decrease the modulus of the peg material

In force driven coupling, avoiding plastic deformation can be accomplished by increasing the contact angle, θ_c (See Fig. 3). If the necessary angle is greater than 120 degrees (contact angles less than 120 degrees yield sub-micron repeatability (Culpepper, 1999)), it is recommended that the contact angle be set at 120 degrees and one (or more as needed) of the three solutions for displacement induced coupling be implemented.

5.5 STEP V Testing / Proof of Performance

Repeatability testing (section 6.4) and thermal fatigue (in thermally cycled joints) should be performed to verify performance.

6. CASE STUDY ON BLOCK TO BEDPLATE MATING

6.1 Review of Current Alignment Method

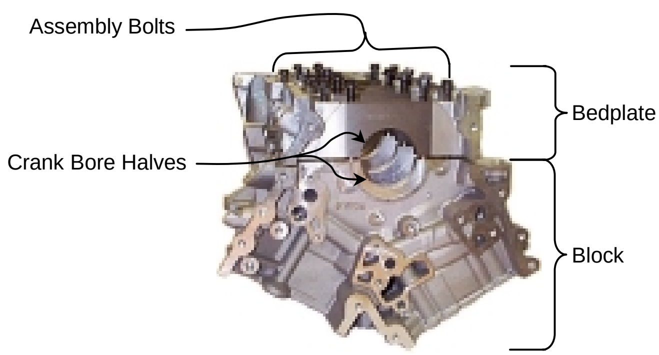

Figure 8 Block and Bedplate Assembly

Figure 8 shows the block and bedplate (monolithic part containing main bearing caps) of the engine. Maintaining the specified 5 micron alignment (Vrsek, 1997) requires eight pinned joints (8 dowels can be seen in Fig. 9). The corresponding 16 holes require feature placement tolerance of +/- 0.04 mm and feature size tolerance of +/- 0.02 mm, making this design difficult and costly to manufacture (Heck, 1997).

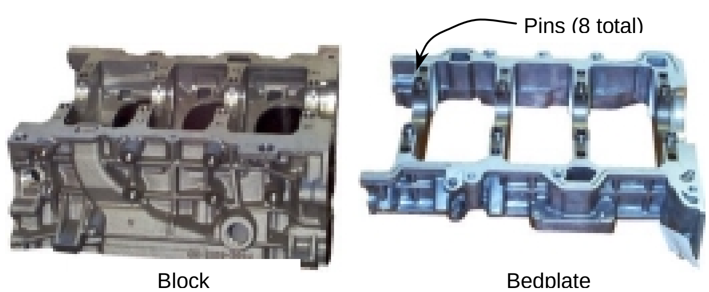

Figure 9 Block and Bedplate With Pinned Joints

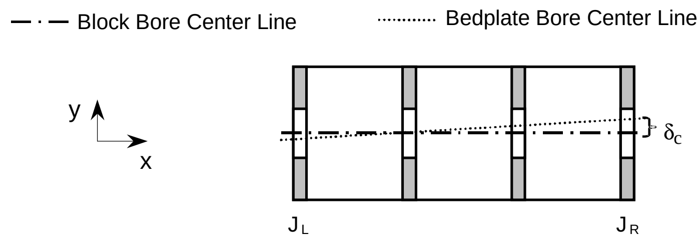

Figure 10 Center Line Error Between Block and Bedplate

With regard to the need for repeatability in this application, consider that in manufacturing this engine type, the components are bolted together and the crank bore is simultaneously machined into each component (a half bore in each). Afterwards, the two components are disassembled, the main bearings and crank shaft are installed, and the block and bedplate are reassembled. Maintaining the same alignment of the block and bedplate half bores before and after assembly is critical as the error, δ_c, between the bearing center lines (see Fig. 10) will result in bearing misalignment. This in turn can adversely affect the load capacity, running temperature, and friction coefficient of the bearing. Accelerated wear or bearing seizure can also result (Shigley and Mischke., 1989). Note the maximum error occurs at either the left most (J_L) or the right most (J_R) journals.

6.2 Application of the Quasi-Kinematic Coupling

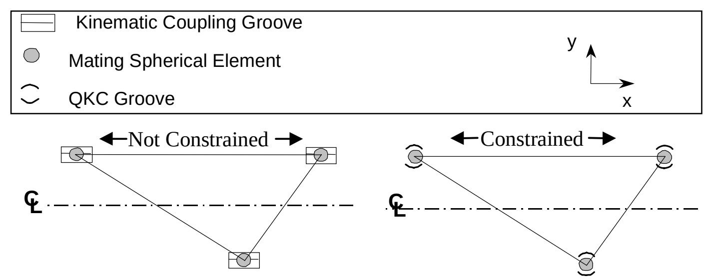

Figure 11 Mated QKC and Kinematic Couplings

To maximize the centering ability and stiffness of the coupling in the y direction (sensitive direction), the QKC grooves were oriented as shown in Fig. 11. This is not possible with a traditional kinematic coupling as the spherical elements could slide in the x direction. With a QKC the curvature of the groove seats helps constrain movement in the x direction.

The crowned pegs and conical grooves were manufactured and placed as discussed in sections 4.1 - 4.3. The peg and groove were designed such that a nominal gap of 0.13mm existed between the mating surfaces of the block and bedplate. This gap was closed by clamping the components together with the assembly bolts shown in Fig. 8. As this happened, the grooves deformed plastically while the pegs deformed elastically. The machining of the crank bore proceeded as in the pinned design. When disassembled for main bearing and crank shaft installation, part of the gap between the mating faces was restored by the elastic recovery of the pegs and grooves, making possible another quasi-kinematic mate during reassembly.

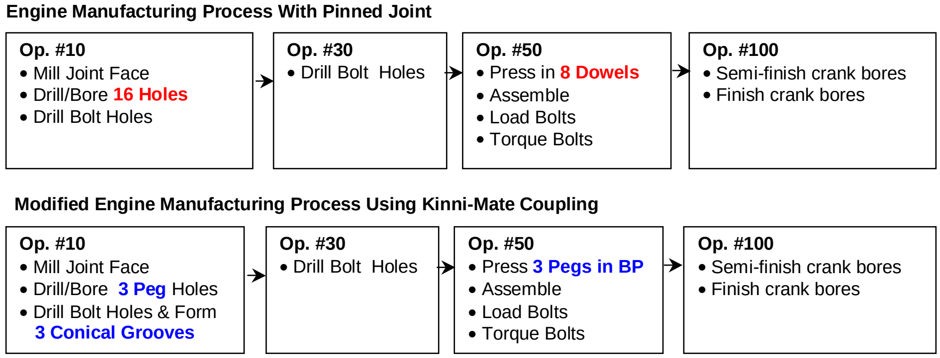

Figure 12 Manufacturing Processes of Alignment Methods

6.3 Comparison of Pinned and QKC Manufacturing

Figure 12 shows the basic operations of interest in the machining of the locating members in the pinned and QKC methods. The only change needed to switch between the two designs is a tooling change from drill to the form tool (fits in the same holder as the replaced drill) and the elimination of 13 of the 16 bored dowel pin holes (three were needed to accommodate the pegs).

6.4 Performance of QKCs

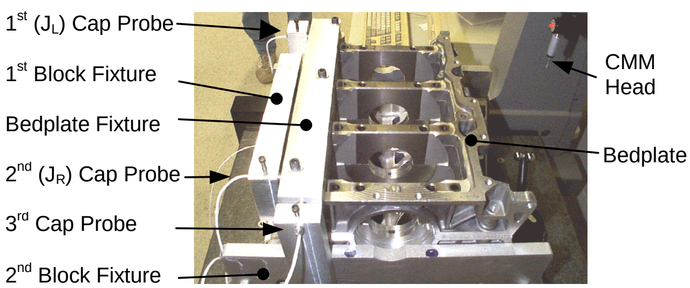

Figure 13 Engine Repeatability Test Setup

Figure 13 shows the block and bedplate assembled in the test stand used to take repeatability data. The bedplate and block fixtures were rigidly attached to their respective components. Relative movement between the block and bedplate was determined by measuring the movement between the fixtures with three capacitance probes attached to the block fixtures. The entire assembly was mounted on a coordinate measuring machine which was used to measure the pre-mate and post-mate gap between the bedplate and block by measuring the height of the bedplate's top surface. For each data point the procedure involved bolting the components together, taking position readings (resolution 0.05 microns), disassembling the components, reassembling them, then taking the final reading. Test results are shown in Figs. 14 and 15. Refer to Figs. 10 and 11 for the sensitive (y) and axial (x) directions.

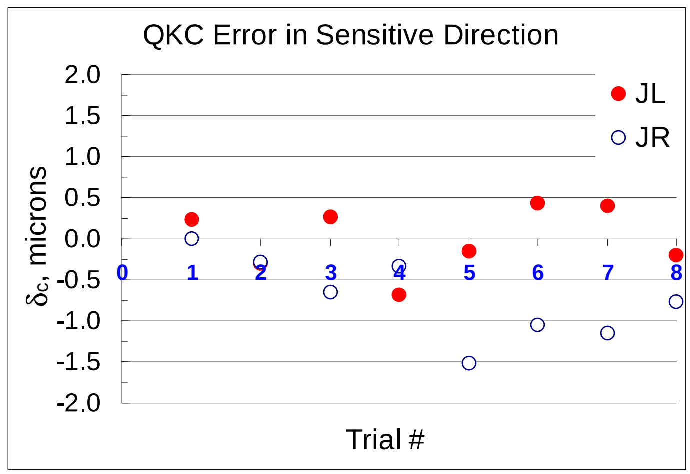

Figure 14 QKC Repeatability in Sensitive Direction

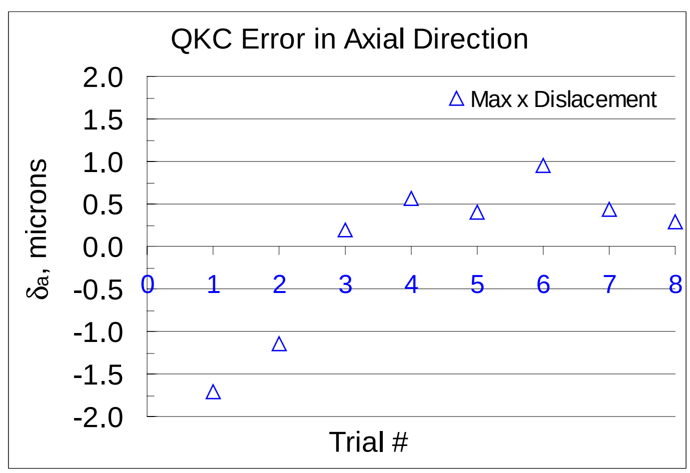

Figure 15 QKC Repeatability in Axial Direction

Table 1 Comparison of Six Cylinder Couplings		
Item	**QKC**	**Pinned Joint**
Fewer precision pieces	3 Pegs	8 Dowel Pins
Fewer precision features	6 (3 Grooves and 3 Bored Peg Holes)	16 (8 Bored Dowel Holes In Each Component)
Better average center line repeatability	0.65 microns	4.85 microns
Wider feature placement tolerances	+/- 0.08mm	+/- 0.04 mm
Lower cost per engine (data is normalized)	0.64	1.00

Repeatability of the couplings is calculated by dropping the high and low readings, then dividing the maximum difference (range) between the remaining data points by two. The repeatability in the sensitive direction at the J_L and J_R journals is 0.55 and 0.75 microns respectively, giving an average repeatability of 0.65 microns. The axial repeatability is calculated as 1.35 microns. The difference in performance in the two directions is expected as the coupling uses a 120 degree groove seat which when orientated as shown in Fig. 11 is roughly 1.9 times more stiff in the sensitive direction than in the axial direction (Culpepper, 1999). Considering the deformation of the kinematic elements was linear elastic during all but the first mate, and ignoring the effects of surface finish, friction, and initial misalignment between kinematic elements, the ratio of axial to center line repeatability can be estimated with Eq. (3).

$$\frac{R_c}{R_a} = \frac{K_c}{K_a} \qquad (3)$$

The repeatability ratio (left hand side of Eq. (3)) of 2.1 is in moderately close agreement with the theoretical stiffness ratio of 1.9 (right hand side).

6.5 Comparison of QKC and Pinned Joint Methods

A comparison of important characteristics and the performance of the pinned and QKC characteristics and performance is provided in Table 1. In all areas, the QKC out performs or has more desirable characteristics. The authors found no important characteristics in which the pinned joint outperformed the QKC.

7. CONCLUSION

In this paper we have introduced the application of QKCs to automotive applications with discussion of high level design and manufacturing issues. The financial, manufacturing, and performance benefits of the QKC versus pinned joints have been covered showing substantial increase in performance at lower cost. Though only one application was discussed, the QKC coupling can be used to align tooling, fixtures, heads to blocks, and applications with close fit tolerances or bearing clusters.

8. ACKNOWLEDGEMENTS

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9. REFERENCES

1. Culpepper, M. L., Slocum, A. H., Shaikh, F. Z., 1998, "Compliant Kinematic Couplings For Use In Manufacturing And Assembly," Proceedings of the 1998 ASME International Mechanical Engineering Congress and Exposition.

2. Culpepper, M. L., 1999, "Design and Application of Compliant Quasi-Kinematic Couplings," Ph.D. Thesis, Massachusetts Institute of Technology, Cambridge, MA.

3. Heck, K., December 1997, Conversations with Kevin Heck, Manufacturing Engineer, Ford Engine Plant #2, Cleveland, OH.

4. Shigley, J. E., Mischke, C. R., 1989, "Mechanical Engineering Design," McGraw-Hill, Inc., New York, New York, pp 508 - 509

5. Slocum, A. H., April 1988, "Kinematic Couplings for Precision Fixturing -Part I: Formulation of Design Parameters," Precision Engineering, Vol. 10, No. 2, pp. 85-91.

6. Slocum, A. H., 1992, "Precision Machine Design, " Prentice-Hall, Inc., Englewood Cliffs, New Jersey , pp. 401-412 and pp. 228-235.

7. Slocum, A. H., Mueller, L., Braunstein, D., 1997, "Flexural Mount Kinematic Couplings and Method", United States Patent No. 5,678,944, pp. 1 - 10.

8. Slocum, A. H., May 1998, Conversations with Prof. A. H. Slocum of MIT, Cambridge, MA.

9. Suh, N. P., 1990, "The Principles of Design", Oxford University Press, New York, pp. 1 - 30.

10. Vrsek, Gary, December 1997, Conversations with Gary Vrsek, Powertrain Research Manager, Ford Scientific Laboratory, Dearborn, MI.