

**Mechanism Design Principle for Optical-Precision,
Deployable Instruments**

Mark S. Lake
*NASA Langley Research Center
Hampton, Virginia
757-864-3114*

M. Roman Hachkowski
*Raytheon Systems Company
Danbury, Connecticut
203-797-5609*

Presented at the 41st AIAA/ASME/ASCE/AHS/ASC
Structures, Structural Dynamics, and Materials Conference
AIAA Paper No. 2000-1409

Atlanta, Georgia
April 3-6, 2000

Mechanism Design Principles for Optical-Precision, Deployable Instruments

Mark S. Lake*

NASA Langley Research Center, Hampton, Virginia, 23681-2199

M. Roman Hachkowski†

Raytheon Systems Company, Danbury, Connecticut, 06810

Abstract

The present paper is intended to be a guide for the design of “microdynamically quiet” deployment mechanisms for optical-precision structures, such as deployable telescope mirrors and optical benches. Many of the guidelines included herein come directly from the field of optomechanical engineering, and are neither newly developed guidelines nor are they uniquely applicable to high-precision deployment mechanisms. However, the application of these guidelines to the design of deployment mechanisms is a rather new practice, so efforts are made herein to illustrate the process through the discussion of specific examples. The present paper summarizes a more extensive set of design guidelines for optical-precision mechanisms that are under development.

Introduction

Despite growing interest in deployable optical instruments within both NASA and the Department of Defense¹, no such instruments have been developed and flown to date within the unclassified community^{2,3}. Consequently, there is currently no comprehensive and accepted set of guidelines for the development of high-precision deployment mechanisms for optical instruments⁴. Developing such guidelines requires the consideration of past experiences in the design of both conventional deployment mechanisms and optical-precision mechanisms for non-deployable applications.

Organizations currently engaged in the development of deployable optical instruments include both

optomechanical design groups and aerospace-mechanical design groups. In general, optomechanical design groups have expertise in the design of positioning devices and kinematic mounts for optical system components^{5,6,7}, but they have little or no expertise in the design of hinges and latches for large deployable structures. Conversely, the aerospace-mechanical design groups have expertise in the design of deployment mechanisms for lower-precision applications (e.g., solar arrays and radio frequency antennas^{8,9}), but they have little or no expertise in the design of optical-precision mechanisms. It is reasonable to suggest that future success in the design of deployment mechanisms for optical instruments demands the consideration of both optomechanical design principles and aerospace mechanical design principles.

This paper presents a summary of the relevant practices in the design of conventional deployment mechanisms and optomechanical devices, and provides specific recommendations for the design of high-precision deployment mechanisms. These recommendations are derived from recent studies of the effects, on dimensional stability, of interface mechanics within precision deployment mechanisms¹⁰.

Conventional Deployment Mechanisms

The current state of the art in design of conventional (i.e., non-optical-precision) deployment mechanisms is captured in refs. 4 and 11. Reference 11 is the military specification most commonly applied to the development and qualification of deployment mechanisms. Although ref. 11 was cancelled by the Air Force in 1996, it includes a complete set of formal guidelines and criteria that are still commonly applied to the design of deployment mechanisms, and its continued use in that capacity is strongly encouraged. Chapter 16 of ref. 4 presents a good summary and interpretation of these guidelines along with a good reference list documenting specific examples of deployment mechanisms currently in use throughout the industry.

*Senior Research Engineer, Mechanics and Durability Branch, Associate Fellow AIAA.

† Senior Mechanical Engineer, Member AIAA.

Copyright © 2000 by the American Institute of Aeronautics and Astronautics, Inc. No copyright is asserted in the United States under Title 17, U.S. Code. The U.S. Government has a royalty-free license to exercise all rights under the copyright claimed herein for Governmental Purposes. All other rights are reserved by the copyright owner.

Features to Maximize Stiffness and Strength

In accordance with ref. 11, conventional deployment mechanisms are typically simple in design with a minimum number of parts to minimize fabrication cost and complexity, and to maximize deployment reliability. In order to maximize stiffness and strength, the internal load paths within conventional deployment mechanisms are often highly redundant and interfaces between internal, load-bearing components are usually conforming (i.e., interfacing surfaces match over a relatively large area) with relatively high interface preloads.

For example, a simple hinge joint consists of a tang that rotates around a clearance-fit pin embedded in a clevis (ref. 12). As shown in Fig. 1, this typical hinge design is symmetric about two perpendicular planes passing through the center of the joint. This symmetry insures that the joint will not bend laterally as axial loads are applied. The symmetry also places the pin in a state of double shear, giving the joint higher stiffness and strength than a joint in single shear. In addition to incorporating load paths designed for high stiffness and strength, conventional hinges typically incorporate some form of mechanical preload to minimize nonlinear load-displacement response.

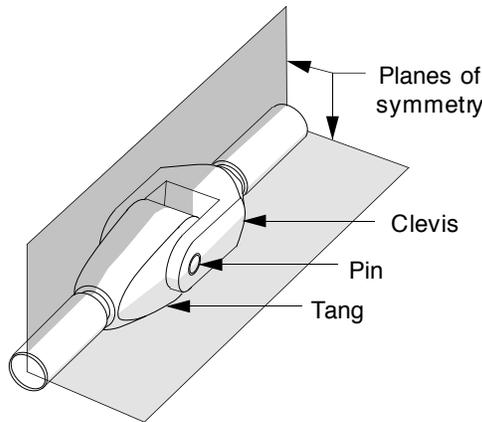


Fig. 1. Simple pin-clevis hinge joint.

Preload to Reduce Nonlinear Response

Under tension-compression load cycling, a mechanical joint can exhibit the three types of nonlinear load-displacement response illustrated in Fig. 2 (ref. 13). Freeplay is typical in mechanisms that include clearances between components to allow articulation (e.g., the pin and tang in the case of the simple pin-clevis joint). Nonlinear elasticity occurs due to different internal load paths in tension and compression and due to increasing regions of contact at mechanical interfaces

with increasing load. Finally, hysteresis arises from friction-induced slippage between contacting components within the joint.

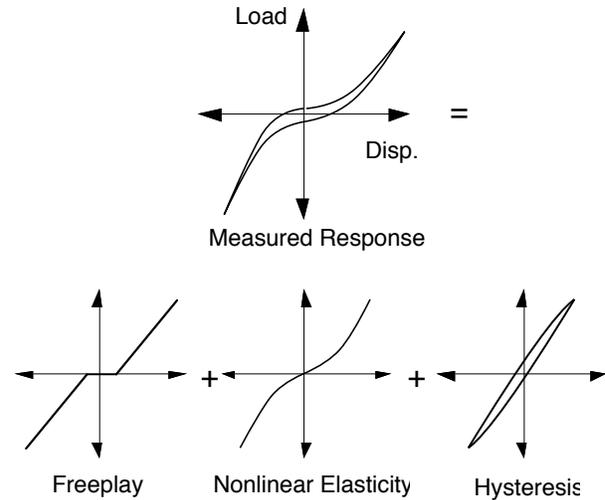


Fig. 2. Nonlinear load-displacement response of mechanical joints.

In conventional hinge and latch designs, a mechanical preload device is often incorporated to reduce all forms of load-displacement nonlinearity to less than a few percent of full-scale displacement. Preload devices are usually designed to maintain intimate contact across load-transferring interfaces, and eliminate any gross nonlinear effects like freeplay and gross slippage. These preload devices usually apply a uniaxial compression load across the mechanism in the primary direction the mechanism is loaded under operational conditions. The magnitude of the preload is usually designed to be substantially larger than the expected disturbance loads. Experience has shown that conventionally designed deployment mechanisms, preloaded in this fashion, provide adequate dimensional stability for non-optical-precision applications (ref. 4).

However, recent research has also shown that simply increasing mechanical preload *may not* provide adequate dimensional stability for optical-precision applications (ref. 10). In some mechanism designs, an increase in preload can actually increase hysteresis if operating loads are carried through the preload device, or if the design of the preloaded interface is such that a substantial portion of the operating load is carried through traction forces. Therefore, in reviewing existing mechanism designs, it is important to consider what effect preload has on hysteresis, and select only designs in which preload reduces hysteresis.

Optomechanical Design Principles

The current state of the art in the design of optomechanical devices is captured in refs. 14 through 16. As mentioned previously, few examples of optical-precision hinges and latches can be found in the literature (e.g., ref. 17). The vast majority of literature on optomechanical design deals with the design of positioning devices and optical-component mounts. However, the following principles employed in the design of positioning devices and mounts are equally applicable to the design of hinges and latches.

Determinate vs. Indeterminate Load Paths

In optomechanical design, great emphasis is placed on the use of statically determinate (i.e. kinematic) load paths between interfacing components. Between large optical-system components, a determinate load path is often established by the use of six truss members arranged as a Stewart platform¹⁶ or the more compact Kelvin mount¹⁴ which includes three mounts arranged as shown in Fig. 3. Between small optical-system components (e.g., lenses and lens mounts), a determinate load path is often established using a convenient arrangement of flexures.

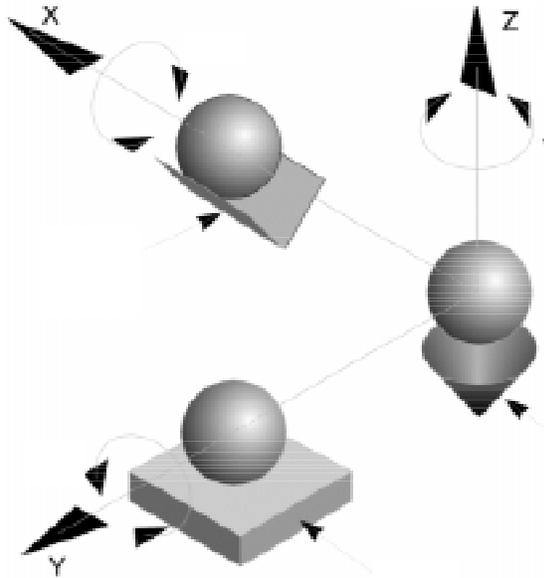


Fig. 3. Three interface geometries used in kinematic “Kelvin” mount.

The reason for using kinematic load paths between optical-system components is that they preclude the development of unwanted assembly preloads (i.e., loads between assembled components due to manufacturing tolerances and differential, thermo-mechanical response of the components). It is commonly recognized that

unwanted assembly preloads can degrade optical system performance because of the mechanical distortions that they induce in the optical components¹⁶. However, it is not commonly recognized that unwanted assembly preloads can increase hysteresis and hysteresis-induced instabilities. Unwanted assembly preloads can force local stresses in the region of mechanical interfaces to become excessively high, and if these stresses include traction components (i.e., tangential components involving friction at interfaces), the likelihood of friction-induced slippage, and hence hysteresis, increases. Therefore in the interest of minimizing hysteresis and hysteresis-induced instabilities:

Optomechanical Design Principle 1: When practicable, design determinate load paths between components, and design the interfaces along these load paths to carry little or no load through friction.

Non-Conforming vs. Conforming Interfaces

To make interfaces between mechanical components very stable, non-conforming (i.e., point or line) contacts are preferred over conforming (i.e., areal) contacts whenever practicable. Of course this recommendation is only applied to moving mechanical interfaces or interfaces that must be assembled and disassembled. Fixed interfaces that never require disassembly are often bonded or welded to eliminate all possibility of friction-induced slippage. Figure 3 illustrates three non-conforming interface concepts using a spherical contacting surface. It is also common to use cylindrical surfaces against flat surfaces as a means of establishing a non-conforming interface.

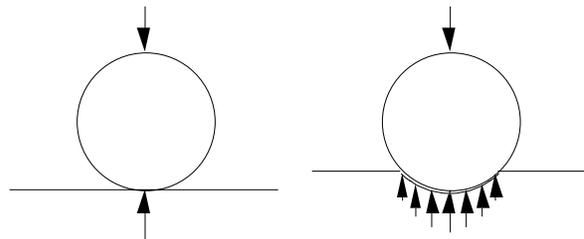


Fig. 4. Stress distributions at non-conforming and conforming interfaces.

In a sense, the reason for using non-conforming geometries at load-bearing interfaces is the same as the reason for using determinate load paths between assembled components. At the local level, a conforming interface is a highly redundant load path since local elasticity determines the interface stress distribution (see Fig. 4). For example, two conforming

surfaces that are not perfectly matched in shape will exhibit significant variations in the interface stress distribution and a high likelihood of localized slippage under load cycling as the interface continually seeks to “re-seat” itself as the applied load changes. Conversely, the use of non-conforming interfaces virtually guarantees that the interface stress distribution will be accurately known, and be independent of localized imperfections in the mating surfaces. Consequently:

Optomechanical Design Principle 2: When practicable, use non-conforming geometries at mechanical interfaces.

Compliant vs. Non-Compliant Preload Devices

Before discussing preload devices, it is prudent to define clearly the distinction between preload and operational load within the context of loading across a mechanical interface. Operational loads are defined to be loads that must pass across the mechanical interface due to the operation of the optical system (i.e., loads generated in response to global disturbances). Conversely, preload is defined to be the load that is intentionally applied across the interface, by means of some secondary device (i.e., preload device), in order to maintain intimate and stable contact at the interface. Preload is commonly applied to all mechanical interfaces between optical-system components¹⁶.

In general, a compliant (i.e., low-stiffness) linkage is used between a preload device and the mechanical interface being preloaded. This is a good means of maintaining relatively constant preload across the interface despite manufacturing tolerances and localized thermo-mechanical deformations. In addition, a compliant linkage also tends to prevent the transmission of operational loads through the preload device, a condition that is undesirable since preload devices are not typically designed to be precision mechanisms. Therefore:

Optomechanical Design Principle 3: All preload mechanisms should be designed not to participate directly in the transfer of operational loads across the interface (e.g., through the use of a compliant linkage between the preload device and the point of application of preload).

Design Recommendations for Optical-Precision Deployment Mechanisms

Before considering specific design recommendations for high-precision deployment mechanisms, it is important to convey several general guidelines related to

the geometric layout of a deployable structure and the location of the deployment mechanisms within that structure¹⁸. Specifically:

- it is highly desirable to locate deployment mechanisms in line with the primary load-carrying members of the structure such that no significant offsets occur that can amplify the loads induced across the mechanisms.

If mechanisms are located in this fashion, the mechanical loads induced across mechanisms will be kept to a minimum, and hysteresis-induced instabilities will also be kept to a minimum. Also:

- it is highly desirable to select deployable structure geometries with sufficient depth such that the percent uncertainty in the optical-alignment degrees of freedom in the structure will be on the order of the percent uncertainty in the displacement response of an individual deployment mechanism.

As a general rule of thumb, this result can be achieved by avoiding geometries in which hinges and/or latches are spaced substantially more closely in one direction than in other directions (e.g., see Fig. 5).

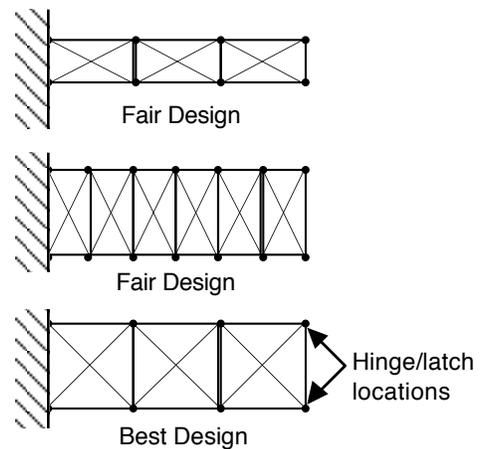


Fig. 5. Illustration of recommended mechanism spacing in a deployable structure.

Use of Existing Mechanism Designs

As mentioned previously, designers are encouraged to continue to apply the design principles of conventional deployment mechanisms (refs. 4 and 11) to the design of high-precision deployment mechanisms. Although not summarized herein, these guidelines and the past four decades of industry

experience in developing deployment mechanisms for non-optical-precision applications, represent an invaluable experience base that should not be ignored. Specifically:

- designers are encouraged to use existing, flight-proven, hinge and latch mechanisms (e.g., ref. 17) for optical-precision deployable structures, when such application can be shown to satisfy dimensional-stability requirements.

In cases where existing designs might prove inadequate:

- designers are encouraged to consider the modification of existing designs or the synthesis of new designs using the practices recommended herein.

Designers who are inexperienced in the design of optomechanical systems are encouraged to consult with optomechanical designers to ensure proper application of the recommended practices and consideration of additional optomechanical design principles not interpreted herein.

Mechanism Stiffness Considerations

Since the deployment mechanism and the structural member it is embedded within act as springs in series, it is desirable for the stiffness of the mechanism to be comparable to the stiffness of the structural member. In general, mechanisms exhibit relatively low stiffnesses due to the use of non-conforming interfaces and circuitous internal load paths. The stiffness of a mechanism can be increased by using high-modulus materials (e.g., composites, invar, or titanium) in the non-mechanical components of the mechanism, and decreasing the effective length of the mechanism (i.e., the distance the mechanism spans between connected structural members). Hence, it can be generally stated that:

- it is desirable to minimize the effective length and maximize the elastic stiffness of the deployment mechanism.

Dropping Hinges Out of the Load Path

In most applications, the deployable structure does not have to exhibit high dimensional stability during deployment. It is only required to exhibit high dimensional stability post-deployment. Therefore, it is not surprising that one approach often considered for reducing the overall challenge of designing for precision

deployment is to use “sloppy” hinges to affect deployment and high-precision latches to maintain dimensional stability post-deployment. In most cases, this approach is considered largely because there is a general misconception that it is easier to design a dimensionally stable latch than a dimensionally stable hinge.

Indeed, a few examples of flight-qualified, optical-system latches exist in the literature (e.g., ref. 17), whereas there are no known examples of *flight-qualified* optical-system hinges. However, recent design experience has shown that high-precision hinges are just as easy (if not more easy) to design than high-precision latches (refs. 13 and 19). Hence, the practice of intentionally dropping hinges out of the load path post deployment will not, necessarily, lead to a more microdynamically stable deployable structure. This coupled with the fact that additional latches (and hence additional mass and complexity) are required to eliminate the hinges from the load path, leads one to the general conclusion that:

- it is not inherently advantageous to drop hinges out of the load path.

Use of Distributed Preload Systems

Typically in the design of optomechanical systems, all mechanisms (and their load-bearing interfaces) are individually preloaded via “local” preload devices. These preload devices are classified as “local” devices because they only provide preload to one hinge or latch mechanism, and are usually an integral part of the hinge or latch mechanism. By contrast, many conventional (i.e., non-optical-precision) deployable structures utilize a network of tension cables or other means to provide “distributed” preload to a large number of hinges and/or latches (e.g., refs. 8 and 9).

Although such a distributed preload system might be applied with success in the design of a deployable optical instrument, most conventional distributed preload system concepts violate optomechanical Design Principle 3 (see previous section), because they carry substantial operational loads in addition to applying preload. In other words, it is difficult to design a distributed preload system that loads all deployment mechanisms but does not provide an alternative load path through the structure. Specifically:

- distributed preload systems should be designed: 1) not to involve any primary load-bearing components of the structure, and 2) not to change preload despite thermo-mechanical loading of the structure.

Design of Fixed Interfaces

Fixed interfaces are hereby defined to be load-bearing interfaces between mechanical components that do not move (in the case of a hinge) or do not require mating and de-mating (in the case of a latch). Examples of fixed interfaces are: bonded; welded; press-fit; and bolted interfaces. In general, bonded and welded interfaces exhibit no measurable friction-induced hysteresis, and hence, they function very well for high-precision applications. However, experience has shown that even highly-preloaded, press-fit and bolted interfaces can exhibit measurable friction-induced hysteresis²⁰. Hence:

- whenever practicable, it is recommended to bond or weld fixed interfaces.

If it is necessary to use a bolted or press-fit fixed interface (e.g., for assembly/disassembly purposes):

- it is recommended that the load capacity (i.e., stick-slip load) of bolted or press-fit interfaces be designed to be AT LEAST a factor of 10 greater than the maximum expected operating load of the mechanism.

Usually, this can be achieved by using large pins or bolts assembled with high press-fits and torque values.

Design of Non-Fixed Interfaces

Non-fixed interfaces are hereby defined to be interfaces between components that must move relative to one another (e.g., two halves of a hinge), or must be mated and de-mated (e.g., two halves of a latch). As discussed previously:

- non-fixed interfaces should, whenever practicable, incorporate non-conforming (i.e., point or line) contacts instead of conforming (i.e., large-area) contacts.

In the case of a hinge, essentially the only way to allow rotation across the non-fixed interface, while incorporating non-conforming contact, is to use a pre-loaded rolling-element bearing as the non-fixed interface. In the case of a latch, there are a number of design options for establishing non-conforming contact at the non-fixed (i.e., mate/de-mate) interface. For example, the Kelvin mount illustrated in Fig. 3 includes three latches that incorporate point contact between a sphere and a plane, and line contact between a sphere and a cone. These three interface geometries are the simplest non-conforming geometries for latches that exhibit one-, two-, and three-degrees of restraint, respectively.

Minimizing Traction at Non-Fixed Interfaces

All of the interface geometries depicted in Fig. 3 are commonly used in the design of mate/de-mate interfaces because they are simple to manufacture, and they tend to exhibit good repeatability between mate/de-mate cycles. However, all three interface geometries can allow the development of traction (i.e., friction) forces, and hence friction-induced hysteresis under load cycling, under certain loading conditions. Hence:

- to reduce friction-induced slippage across non-conforming interfaces, it is desirable to minimize the tangential stiffness at the interface.

Figure 6 depicts a very simple method of eliminating tangential stiffness at a single-point-contact, non-conforming interface. Instead of establishing the single-point contact between two bodies that might be loaded tangential to the contact surface, it is better to “trap” a rolling element (i.e., ball or needle bearing) between the two contacting bodies that cannot transmit traction forces.

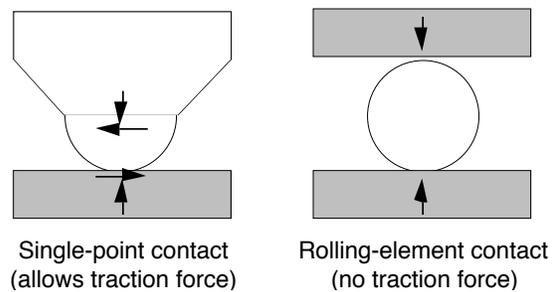


Fig. 6. Method to eliminate traction forces at a non-conforming contact.

Figure 7 depicts a method of eliminating tangential stiffness at a two-point-contact using a simple V-groove. Under the area of contact within the V-groove, it is better to relieve the tangential stiffness by cutting slots as shown in the right-hand sketch of Fig. 7. This detail effectively eliminates potential traction forces from developing and hence, reduces the potential for friction-induced hysteresis under load cycling.

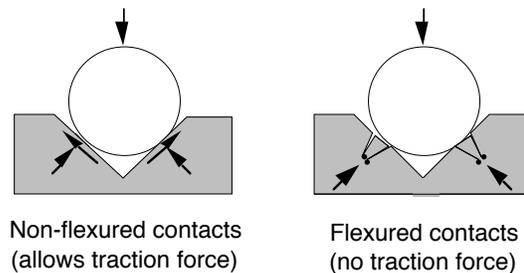


Fig. 7. Use of flexures to reduce traction.

Increasing the Stiffness of Non-Conforming, Non-Fixed Interfaces

A disadvantage to using non-conforming contact geometries at non-fixed interfaces is that they tend to exhibit a low stiffness due to the high localized stresses and deformations under loading. In order to mitigate this effect it is possible to design the interface to include multiple (i.e., redundant) non-conforming contacts. Of course, it is realized that this approach eliminates static determinacy across the interface, but in general:

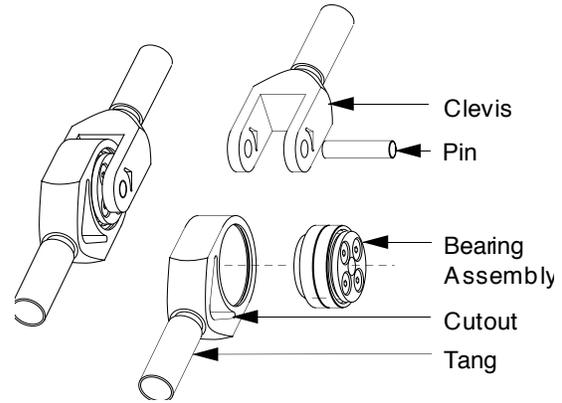
- in the interest of increasing the stiffness of non-fixed interfaces, it is desirable to design the interfaces with multiple non-conforming contacts.

Rolling-element bearings are particularly well-suited for application to the design of non-fixed interfaces in high-precision deployment mechanisms because all interfaces in rolling-element bearings are non-conforming, and each bearing is capable of transmitting very little load through friction (as shown in Fig. 6). Hence, despite the inherent load-path redundancy within a rolling-element bearing, it can still provide a highly stable, non-fixed interface.

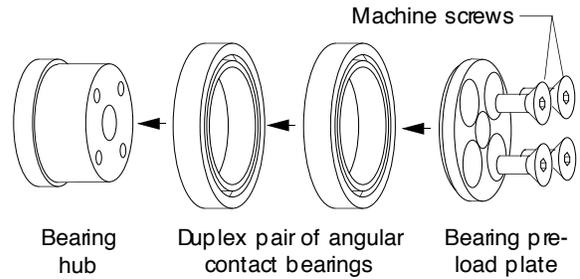
Figure 8 includes sketches of a high-precision hinge developed for a deployable telescope mirror¹. Reference 19 describes this hinge in addition to a high-precision latch that uses a pre-loaded, rolling-element bearing as its mate/de-mate interface. The pair of angular-contact bearings used in this hinge include over 40 balls that are preloaded within the bearing races. Each ball within the bearing pair is held between its inner and outer races by opposing point contacts that are capable of transmitting load in one direction as shown in Fig. 9 (i.e., along the line of action established by the two contact points). Because of its freedom to roll, each ball transmits very little load through friction. Hence, the assembly of balls, by definition, transmits very little load through friction¹⁰.

The assembly of bearing balls provides restraint against load in five directions as shown in Fig. 9 (three orthogonal directions of force and two directions of moment). Therefore the bearing assembly represents a highly redundant load path (i.e., there are far more than 5 interfaces between the balls and the races). Since the load transferred across the bearing is shared between a fairly large number of balls, the localized load at each of the non-conforming contacts is only a small part of the total load. This distribution of load reduces substantially the elastic deformations in the vicinity of the individual contacts, and leads to substantially higher

overall stiffnesses than typically achievable with statically determinate arrangements of non-conforming contacts.



(a) Main sub-assemblies of hinge.



(b) Bearing assembly.

Fig. 8. High-precision hinge with bearings.

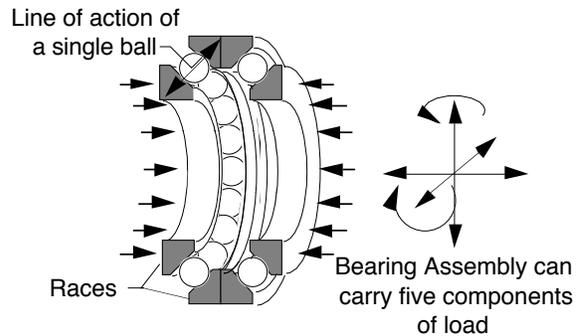


Fig. 9. Angular contact bearing pair.

Use of Axisymmetry for Athermalization

Although thermal mis-match of materials has little direct impact on hysteretic response in high-precision deployment mechanisms, it is an important issue that can influence the selection of components and indirectly affect hysteretic response. Therefore, a few comments relating to athermalization are provided here.

First, it is recognized that low-CTE materials, such as laminated composites, will be used in members spanning between hinges and latches in an effort to minimize the net CTE of the deployable structure. Second, it is recognized that many components of hinges and latches must be made from metal due to the emphasis on the use of non-conforming contacts with high localized stresses. Hence it is desirable to identify effective means of incorporating metal hinge and latch components in athermal designs for which the high CTE of the metal components has minimal effect on the overall CTE of the structure.

One traditional approach to athermalization of metal joints is to attach the metal joints in series with other members having a low or negative CTE (usually composite members) such that the net CTE of the assembly is near zero (Fig. 10). This approach involves balancing the product of the length and CTE of the metal joint and the product of the length and CTE of the composite member. For composite members with negative CTEs, this balance can be achieved by connecting the metal joint and composite member in series with one another (upper sketch of Fig. 10). For composite members (or metal members) with low positive CTEs, this athermalization balance can be achieved by connecting the metal joint and composite member in parallel (lower sketch of Fig. 10).

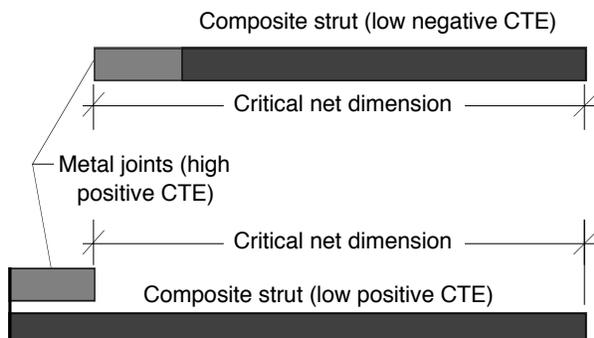


Fig. 10. Traditional approach to athermalization of metal joint components.

Another method by which athermalization can be achieved is to embed the metal joint components in a near-zero-CTE composite member in such a way that thermal mis-match only causes local deformations and not net length changes. An illustration of this approach to athermalization is given in Fig. 11. In this example, an axisymmetric (i.e., circular) metal joint mechanism is embedded in a quasi-isotropic, flat-laminated composite member in such a way that thermal growth of the metal joint only results in uniform radial expansion of the hinge within the composite member.

A cutout is incorporated in the member adjacent to the hinge to ensure athermal expansion of the hinge by making the tension and compression load paths through the hinge of equal stiffness¹³.

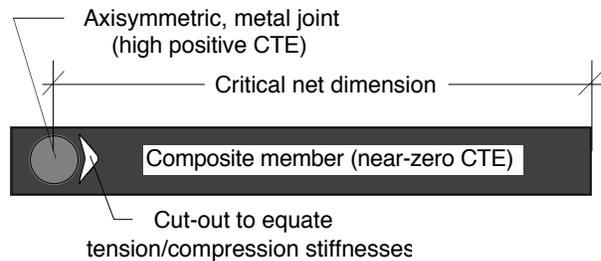


Figure 11. Approach to athermalization of axisymmetric metal joints.

This example is shown to illustrate another advantage to the use of rolling-element bearings in the design of non-fixed interfaces in hinges and latches. Due to the fact that bearings are axisymmetric, it is possible to integrate them into composite structural members in an athermal fashion.

Concluding Remarks

This paper is intended to facilitate the development of deployable, optical instruments by providing recommendations on the design of high-precision deployment mechanisms. Many of the guidelines included herein come directly from the field of optomechanical engineering, and are therefore neither newly developed nor uniquely applicable to the design of high-precision deployment mechanisms. However, the application of these guidelines to the design of deployment mechanisms is a rather new practice, so efforts are made herein to illustrate the process through the discussion of specific examples.

Designers are encouraged to continue to apply the design principles of conventional deployment mechanisms^{4,10} to the design of high-precision deployment mechanisms. Although not summarized herein, these guidelines and the past four decades of industry experience in developing deployment mechanisms for non-optical-precision applications, represent an invaluable experience base that should not be ignored. Designers who are inexperienced in the design of optomechanical systems are encouraged to consult with optomechanical designers to ensure proper application of the recommended practices and consideration of additional optomechanical design principles not interpreted herein.

Finally, this paper is to be regarded as a guide to design and not a set of NASA requirements, except as

may be defined in formal project specifications. Furthermore, due to the rapid pace of advancement in the field of precision deployment, this paper should be regarded as a preliminary set of guidelines.

References

¹Lake, Mark S., et al., "A Deployable Primary Mirror for Space Telescopes," presented at the 1999 SPIE International Symposium on Optical Science, Engineering, and Instrumentation, Denver, Colorado, July 18-23, 1999, SPIE Paper No. 3785-02.

²Hachkowski, M. Roman, and Peterson, Lee D.: "A Comparative History of the Precision of Deployable Spacecraft Structures," University of Colorado Pub. No CU-CAS-95-22, December 1995.

³Lake, Mark S., et al., "Research on the Problem of High-Precision Deployment for Large-Aperture, Space-Based Science Instruments," presented at the 1998 Space Technology & Applications International Forum, Albuquerque, New Mexico, January 25-29, 1998.

⁴Conley, Peter L., (Editor), Space Vehicle Mechanisms: Elements of Successful Design, John Wiley & Sons, Inc. 1998.

⁵Chadwick, Curt H., "A High Resolution, Adjustable, Lockable laser Mirror Mount," NASA TM 33-777, pp. 116-123, April 1976.

⁶Sawyer, Kevin A., et al, "Launch Rated Kinematic Mirror Mount with Six Degree of Freedom Adjustments," presented at the 1999 SPIE International Symposium on Optical Science, Engineering, and Instrumentation, Denver, Colorado, July 18-23, 1999.

⁷Pollard, Lloyd W., "Design of a Flexure Mount for Optics in Dynamic and Cryogenic Environments," NASA CR-177495, February 1989.

⁸Thomson, Mark W., "The Astromesh Deployable Reflector," proceedings of the Fifth International Satellite Conference, Pasadena, California, June 16-17, 1997.

⁹Takamatsu, K.A., and Onoda, J., "The New Deployable Truss Concepts for large Antenna Structures or Solar Concentrators," Presented at the 30th AIAA/ASME/AHS/ ASC Structures, Structural Dynamics, and Materials Conference, Mobile Alabama, April 3-5, 1989.

¹⁰Hachkowski, M. Roman, "Reduction of Hysteresis in the Load-Displacement Response of Precision Deployment Mechanisms Through Load-Path Management," Ph.D. Dissertation, University of Colorado, 1998.

¹¹Annon., "Assemblies, Moving Mechanical, for Space and Launch Vehicles, General Specification for," MIL-A-83577B (USAF), February, 1988.

¹²Rhodes, M. D., "Design Considerations for Joints in Deployable Space Truss Structures," presented at the First NASA/DOD CSI Technology Conference, Norfolk, Virginia, November 18-21, 1986.

¹³Lake, Mark S., Warren, Peter A., and Peterson, Lee D., "A Revolute Joint With Linear Load-Displacement Response for Precision Deployable Structures," presented at the 37th AIAA/ASME/ASCE/AHS/ASC Structures, Structural Dynamics, and Materials Conference, Salt Lake City, Utah, April 15-17, 1996.

¹⁴Jacobs, D. H., Fundamentals of Optical Engineering, McGraw-Hill Book Co., 1943.

¹⁵Yoder, P. R., Jr., Optomechanical Systems Design, 2nd edition, Marcel Dekker, Inc., 1993.

¹⁶Vukobratovich, Daniel, "Optomechanical System Design," in Electro-Optical Systems Design, Analysis, and Testing, Vol. 4, The Infrared & Electro-Optical Systems Handbook, M. C. Dudzik, ed, ERIM and SPIE Optical Engineering Press, 1993.

¹⁷Dozier, Jan D., and Kaelber, Everett, "Latch Fittings for the Scientific Instruments on the Space Telescope," NASA CP-2273, pp. 253-265, May, 1983.

¹⁸Hedgepeth, John M., "Critical Requirements for the Design of Large Space Structures," NASA CR-3483, 1981.

¹⁹Phelps, James E., "Fabrication and Assembly of High-Precision Hinge and Latch Joints for Deployable Optical Instruments," NASA CR-1999-209117, March 1999.

²⁰Lake, Mark S., et al., "Experimental Characterization of Hysteresis in a Revolute Joint for Precision Deployable Structures," presented at the 38th AIAA/ASME/ ASCE/AHS/ASC Structures, Structural Dynamics, and Materials Conference, Orlando, Florida, April 7-10, 1997.