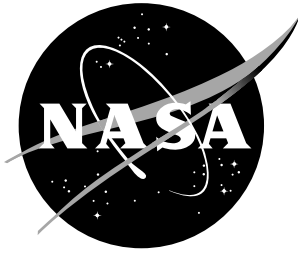


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Design of Mechanisms for Deployable, Optical Instruments: Guidelines for Reducing Hysteresis

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March 2000

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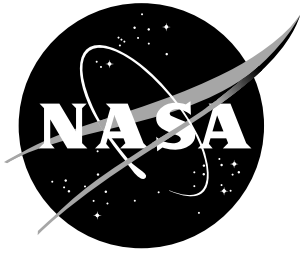
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Foreword

The research, upon which the present paper is based, has evolved from in-depth study of the following basic question: *How does the structural response behavior of a mechanically deployable optical bench or metering structure differ from that of a non-deployable (i.e., monolithic) optical bench or metering structure?*

The general answer to this question is that mechanically deployable structures might exhibit low-level, nonlinear structural response behavior (commonly referred to as microdynamics) that can affect the dimensional precision and stability of the structure. Unfortunately many aspects of microdynamics (e.g., temporal-frequency content and propagation/attenuation characteristics) are still poorly understood and the subject of ongoing research. However, our understanding of the origin of microdynamics is much more complete. Specifically, it is now commonly accepted that microdynamics are caused largely by instabilities in the mechanical joints of a structure arising from friction and friction-induced slippage between mechanical components. Hence, it is clear that reducing or eliminating microdynamics requires the reduction or elimination of friction-induced slippage within the joints.

Therefore, this paper is an effort to answer specifically the following derived question: *How does one design a mechanically deployable optical bench or metering structure such that its structural response behavior is as close to that of a non-deployable optical bench or metering structure as possible?*

In a general sense, the present paper was developed to facilitate advancement of a new sub-discipline of space-vehicle design: the design of precision deployable structures for optical instruments. This paper is intended for use in the development of any deployable optical instrument, but it was written in response to the immediate needs of the Space Interferometry Mission (SIM), currently under development at the Jet Propulsion Laboratory (JPL). Specifically, the present paper is intended to provide the SIM development team with a methodology for designing, testing, and comparing concepts for precision deployment mechanisms.

For the broader deployable optical instrument design community, the present paper is intended to be a guide for the design of deployment mechanisms that exhibit minimal friction-induced slippage. The paper represents what might be considered the most important step (and certainly the first step) in the development of a deployable optical instrument – the design of a deployable structure that exhibits a high degree of passive dimensional stability. Clearly the passive stability of the structure directly affects the complexity of the active control problem. Therefore, provided that reasonable design alternatives exist to improve passive stability, prudence demands that these alternatives be considered.

The outline of this paper follows roughly the outline of a series of space vehicle Design Criteria Monographs developed during the 1960's and published as NASA

Special Publications: SP-8001 through SP-8015. Like these previous design documents, the present document is to be regarded as a guide to design and not a set of NASA requirements, except as may be specified in formal project specifications. It is hoped, however, that this paper, revised as experience may indicate to be desirable, eventually will form the basis for uniform design requirements for high-precision deployment mechanisms on future NASA space-based science instruments.

This paper was prepared by Mark S. Lake, of the Langley Research Center, and M. Roman Hachkowski, of Raytheon Systems Company, at the request of Marie B. Levine of the Jet Propulsion Laboratory. A number of other individuals assisted in developing the material and reviewing the drafts of the paper. In particular, the significant contributions made by Lee D. Peterson, Jason D. Hinkle and Lisa M. Hardaway of the University of Colorado, Robert J. Calvet of the Jet Propulsion Laboratory, and Peter A. Warren of Foster-Miller, Assoc. are hereby acknowledged and greatly appreciated.

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Symbols

δ	-	deflection degree of freedom in simplified hysteresis model
δ_{hys}	-	maximum width of the hysteresis loop for a mechanism
δ_{ten}	-	maximum displacement across a mechanism in tension
δ_{com}	-	maximum displacement across a mechanism in compression
$\delta_{\text{structure}}$	-	displacement of a structure under applied load
η	-	normalized hysteresis
$\eta_{\text{operating}}$	-	normalized hysteresis of a mechanism at its operating load
$\eta_{\text{mechanism}}$	-	normalized hysteresis of a mechanism
η_{peak}	-	peak normalized hysteresis of a mechanism
$\eta_{\text{structure}}$	-	normalized hysteresis of a structure
k_1, k_2, k_3	-	equivalent stiffness parameters in simplified hysteresis model
$(EA/L)_{\text{mechanism}}$	-	effective linear stiffness of a mechanism
$(EA/L)_{\text{strut}}$	-	effective linear stiffness of a strut in a structure
$L_{\text{mechanism}}$	-	length a mechanism
L_{strut}	-	length a strut in a structure
μN	-	stick-slip load in simplified hysteresis model
P_{ten}	-	maximum load applied to a mechanism in tension
P_{com}	-	maximum load applied to a mechanism in compression
$P_{\text{operating}}$	-	maximum expected operating load of a mechanism
P_{peak}	-	load-cycle magnitude at peak hysteretic loss
$P_{\text{structure}}$	-	maximum expected operating load applied to a structure
U_{hys}	-	hysteretic energy loss within a mechanism
U_{ten}	-	maximum strain energy of a mechanism in tension
U_{com}	-	maximum strain energy of a mechanism in compression

Design of Mechanisms for Deployable, Optical Instruments:

Guidelines for Reducing Hysteresis

“The rational design of all structures must start with a definition of the task or function of the structure. . . For each application, a large number of detailed requirements exists, which, if taken collectively, expresses the means by which the proper performance of the structure’s task can be met. Usually a small subset of these requirements dominates the design and is hence termed ‘primary.’” - J. M. Hedgepeth (ref. 1)

Introduction

In 1981, Hedgepeth (ref. 1) astutely predicted that future large space structures would “be designed to deal with phenomena as primary criteria which have been considered as only secondary in the past.” Indeed, recent research on deployable optical instrument structures has shown that a class of response phenomena prevalent in these structures is low-level, nonlinear dynamic response commonly referred to as microdynamic response (ref. 2). Microdynamic response can be important in deployable optical instruments because it can drive requirements for the active alignment-control systems. For example, the temporal-frequency content and propagation/attenuation characteristics of microdynamic events can drive controller bandwidths and stroke requirements.

Unfortunately our understanding of many aspects of microdynamic response that might affect the design of active-control systems is incomplete. Luckily though, our understanding of the origin of microdynamics is much more complete. Specifically, it is now commonly accepted that microdynamics are dominated by instabilities in the mechanical joints of a structure arising from friction and friction-induced slippage between mechanical components.¹ Hence, it is clear that reducing or eliminating microdynamics requires the reduction or elimination of friction-induced slippage within the joints. *Herein, it is asserted that microdynamics in a deployable structure are related to hysteretic response within the deployment mechanisms. Furthermore, it is asserted that a good mechanical design for a high-precision deployment mechanism is identically one that exhibits low-hysteresis response to load cycling, in addition to the traditionally accepted high-stiffness and high-strength response.*

This paper is intended to facilitate the development of deployable, optical instruments by providing a rational approach for the design, testing, and qualification of high-precision (i.e., low-hysteresis) deployment mechanisms for these instruments. Many of

¹ Hysteresis can result from a number of different material and/or structural response effects including viscoelasticity, plasticity, and friction-induced slippage. The only source of hysteresis considered in the present paper is friction-induced slippage at mechanical interfaces within the mechanism. That is not to say that other forms of hysteresis are unimportant in optical-precision deployment mechanisms, but rather, the dominant source of hysteresis is friction. Furthermore, the design recommendations included herein are specifically tailored to reduce friction-induced hysteresis.

the guidelines included herein come directly from the field of optomechanical engineering, and are, therefore, neither newly developed guidelines, nor are they 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.

The organization of the paper is as follows. Section 2 discusses the aspects of hysteretic response that relate to microdynamic instabilities and lead to general mechanical design principles. Section 3 gives an interpretation of the present state of the art in mechanism design from the viewpoint of how current mechanism designs comply with these general design principles. Section 4 provides specific criteria for quantifying and qualifying the hysteretic response of a deployment mechanism. Finally, section 5 provides specific guidelines for synthesizing new mechanism designs that should exhibit very low levels of hysteretic response. Although the sections are interrelated, they may be considered separately by readers with more narrow interests (e.g., mechanical designers might find section 5 to be most helpful.) Finally, attempts have been made to cite the significant literature throughout, and readers with broader interests are encouraged to refer to this body of work for more detailed information.

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. However, it is expected that this paper, with revisions as experience may indicate to be desirable, might eventually form the basis for a set of uniform design requirements for high-precision deployment mechanisms on future NASA space-based science instruments.

Hysteresis in Deployment Mechanisms

Hysteresis in the response of a structure to loading and unloading is commonly associated with energy loss under load cycling and hence damping within the structure. In addition, hysteresis is an indication of the existence, within the structure, of multiple equilibrium shapes in reaction to a particular load condition (ref. 3). Under quasi-static loading, hysteresis implies a dependency on load history in the response of the structure. Quite literally, hysteresis indicates that the history of loading and unloading, not just the final load condition, determines which of the multiple equilibrium shapes will be attained by a structure. Under dynamic loading, hysteresis can cause dynamic instabilities and nonlinear modal response (e.g., changing frequencies and mode shapes with loading level). In general, the magnitude of these nonlinear response effects relative to the total dynamic response of the structure is expected to be small since hysteresis is generally small in comparison to total elastic response. Hence, nonlinear dynamics arising from hysteresis are commonly referred to as *microdynamic* response.

If microdynamic response is of concern, why focus attention on hysteresis?

- First, microdynamic response is inherently a system-response effect that is easiest to observe during testing of complete deployable structures. In other words, it is difficult to experimentally determine the microdynamic stability of a single

deployment mechanism, but it is possible to determine the hysteretic response of a mechanism and relate that response to the microdynamic stability of a deployable structure incorporating the mechanism.

- Second, research to date has resulted in numerous analytical models of hysteretic response that can relate the response to physical design variables in deployment mechanisms. This ability to predict hysteretic response implies the ability to affect (i.e., *reduce*) hysteretic response through implementation of good design practices.

The first reason given above provides the motivation behind section 4 of the present paper – establishment of criteria for hysteretic response in high-precision deployment mechanisms. The second reason given above provides the motivation behind section 5 of the present paper – articulation of a concise set of design principles to reduce hysteresis. Following are detailed discussions of the relationship between hysteretic response and microdynamic response, and the relationship between hysteresis and mechanical design.

2.1 Relationship between Hysteretic Response and Microdynamic Response

Hysteretic systems are nonlinear and their response, even to low-frequency time-varying loading, can involve high-frequency components (ref. 4). Precision deployable structures with intentionally low levels of hysteresis can still exhibit nonlinear dynamic response phenomena with magnitudes at or below the microstrain level (i.e., 10^{-6} times a characteristic dimension of the structure). These microdynamic responses can include: changes in static structural shape (i.e., microlurch, ref. 2) and spontaneous, high-frequency dynamics (i.e., “snapping”), as well as more traditional nonlinear-dynamics such as harmonic distortion and viscoelastic effects (ref. 4). The microdynamic behavior of hysteretic systems can not be completely modeled by linear dynamic modeling methods (e.g., modal techniques). Instead, adequate characterization of microdynamics probably requires the application of non-causal, perturbation methods (ref. 5).

An exact relationship between hysteretic response and microdynamic response is difficult to derive. First, as will be discussed in the next section, the magnitude of hysteresis in a high-precision deployment mechanism varies with load-cycle magnitude. Therefore, additional issues must be considered in order to quantify “critical” values of hysteresis for a given design. Second, accurate predictions of microdynamic response arising from hysteresis require accurate modeling of the entire time-history of loading as well as accurate characterization of the initial conditions of the hysteretic elements (i.e., initial stress states).

Nevertheless, a “practical” relationship between hysteretic response and microdynamic response (i.e., one that can be applied in the development and qualification of designs) can be suggested:²

- the magnitude of microdynamic response expected in a system is equivalent to the magnitude of hysteresis in the system.

² This relationship between hysteresis and microdynamics is logical, but has yet to be proven rigorously. The reader is therefore encouraged to apply it only in the preliminary qualification of mechanism designs, and rely on microdynamic testing of complete deployable structures for final qualification.

For example, a system that exhibits no more than 1% hysteresis under quasi-static load cycling, should exhibit microdynamic response of *NO MORE THAN 1%* in magnitude relative to the linear-response magnitude of the system. (Note: this relationship is with total system hysteresis and not individual component hysteresis. The distinction between system and component hysteresis will be discussed in detail in section 4.1.1.) In other words, the order of magnitude of expected microdynamic instabilities is equal to the order of magnitude of the hysteretic response. This relationship will be expanded in section 2.2.1 in the discussion of microslip versus gross stick-slip.

2.2 Relationship between Hysteresis and Mechanism Design

In a deployable optical instrument, a significant source of hysteresis is inelastic (i.e., frictional) interface mechanics within deployment mechanisms (refs. 6 and 7). Specifically, hysteresis arises from frictional load transfer within the deployment mechanisms. Measuring the magnitude of hysteresis exhibited by a deployment mechanism is a way of quantifying (in a relative sense) the amount of load being transferred through friction at the interfaces between internal components of the mechanism. Unfortunately, due to the geometric complexity of most deployment mechanisms, it is difficult or impossible to predict precisely the magnitude of hysteresis. However, studies have shown that substantial insight into mechanical design can be gained by applying fairly simple models to interpret hysteretic-response data.

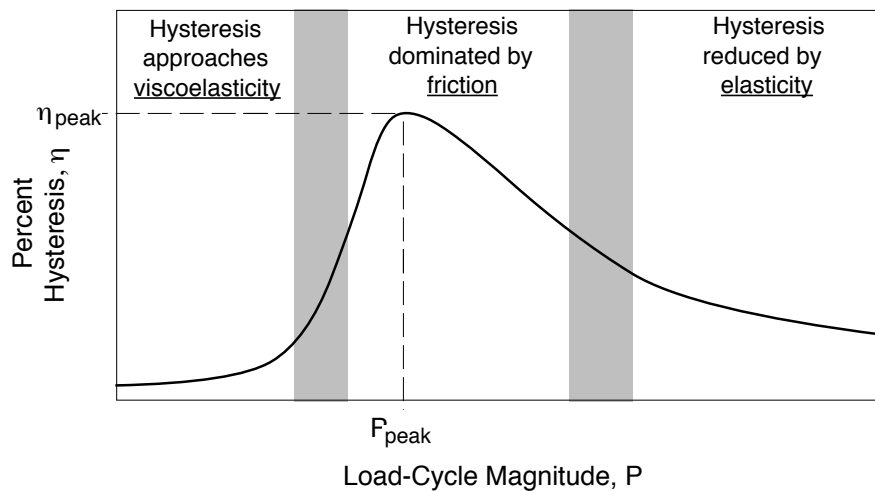


Figure 1. Hysteretic response of a high-precision deployment mechanism (ref. 7).

The normalized (i.e., percent) hysteresis³ exhibited by a high-precision deployment mechanism is generally expected to vary with the load-cycle magnitude as sketched in Fig. 1 (ref. 7). For low load-cycle magnitudes, relatively little friction-induced slippage occurs at interfaces within the mechanism and the percent hysteresis approaches material (i.e., viscoelastic) hysteresis. At higher load-cycle magnitudes, the percent hysteresis increases dramatically, reaching a peak value substantially greater than material hysteresis. The percent hysteresis is high in this region because motions due to friction-

³ Note: Section A.3 of the Appendix defines percent hysteresis and outlines methods for calculating percent hysteresis from load-displacement response data.

induced slippage are large relative to the elastic deformations. Finally, at even higher load-cycle magnitudes, the *absolute* hysteresis might become nearly constant if the amount of stick-slip is limited, causing the *percent* hysteresis to decrease substantially (as elastic deformations become large relative to the limited deformations due to slippage).

2.2.1 Microslip versus Gross Stick-Slip. Substantial insight can be gained into the general hysteretic-response behavior illustrated in Fig. 1, by considering the simplified model shown in Fig. 2. The left-hand sketch illustrates how loads applied in the vicinity of a mechanical interface result in normal pressure and shear stress at the interface. The right-hand sketch shows a simplified model of this effect using two parallel load paths: one that is purely elastic described by the spring k_3 ; and the other that is inelastic involving the springs k_1 and k_2 and the friction element μN . The elastic load path represents load that is transferred across the interface through normal pressure and the inelastic load path represents load that is transferred through shear.

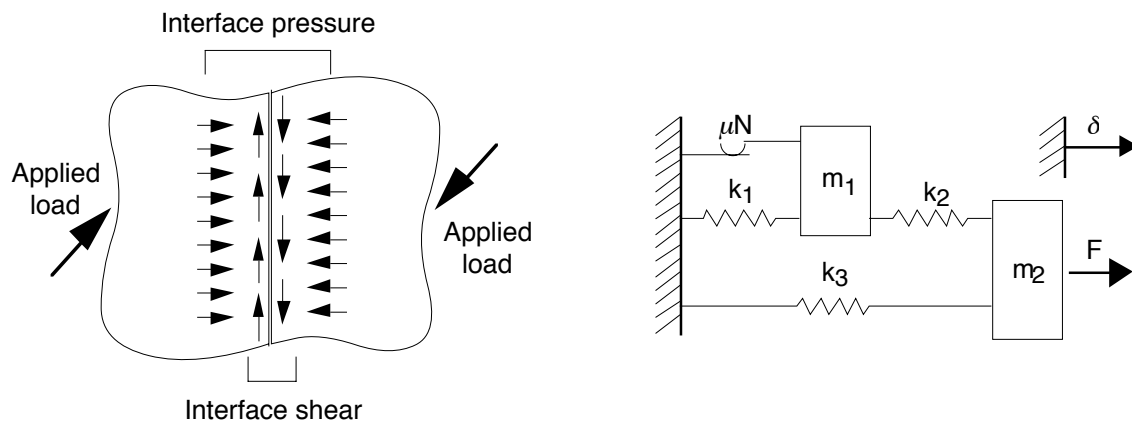


Figure 2. Simplified model of load transfer across a mechanical interface (ref. 7).

Figure 3 illustrates the percent hysteresis as a function of load-cycle magnitude predicted using the simplified model from Fig. 2 and assuming that the friction element represents Coulomb friction (i.e., pure stick at loads below μN and pure slip at loads above μN). This simplified model predicts results similar to those from more complex analyses (e.g., Fig. 1). Specifically, the simplified model predicts that mechanical interfaces exhibit a peak in hysteretic loss (η_{peak}) under load cycling at some critical load-cycle magnitude (P_{peak}), and substantially lower percent hysteresis at other load-cycle magnitudes. However, unlike the more complex models (and data from most deployment mechanisms) the simplified model predicts that hysteresis vanishes for load-cycle magnitudes below one-half that of the peak-hysteresis load-cycle magnitude (i.e., $P_{\text{peak}}/2$).

This “collapse” in hysteresis is an artifact of the Coulomb model of friction. The Coulomb model assumes gross stick-slip behavior in which the entire mechanical interface is loaded uniformly in both normal and tangential directions, and slippage occurs throughout the interface at one critical value of tangential load. However, most mechanical interfaces do not behave in this idealized way because the distribution of loading across the interface varies greatly. In most cases, slippage occurs over localized regions of the interface, and the amount of slippage is limited by local elasticity in the vicinity of the interface. This so-called microslippage behavior is complex and difficult

to predict precisely due to the sensitivity of the results to small variations in interface conditions and model parameters (ref. 8). However, qualitative trends, like those presented in Fig. 1, can be predicted using simplified models such as the Todd-Johnson model of microslip (ref. 7).

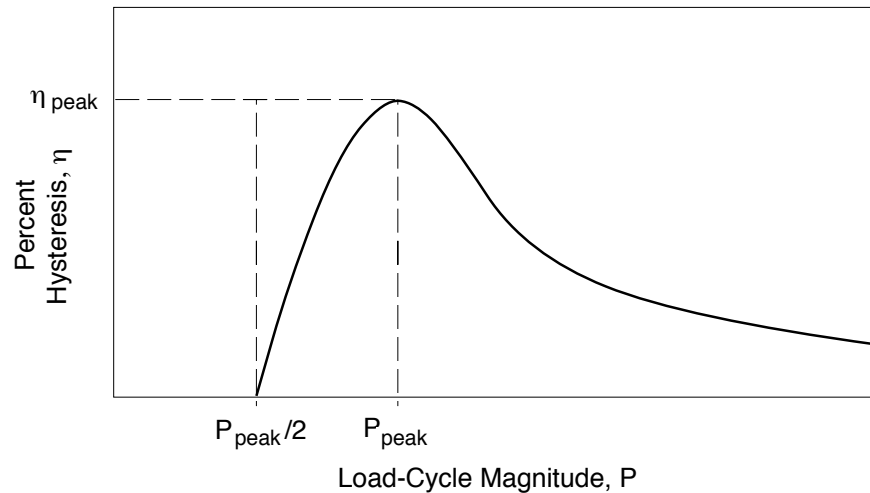


Figure 3. Hysteretic response of simplified model (ref. 7).

The significant difference between microslip and gross stick-slip has to do with their effect on dimensional stability and microdynamic behavior. Numerous models have been developed of dynamic systems with gross stick-slip (e.g., ref. 3 and ref. 7). These models predict both chaotic response and various types of microdynamic instabilities that are generally believed to be of concern to designers of optical systems. Although the exact nature and spectral content of these instabilities is only partially understood today, it is generally agreed that gross stick-slip response should be avoided in any precision deployment mechanism design. On the other hand, insufficient evidence exists from which to conclude that microslip behavior leads to high-frequency microdynamic instabilities (ref. 8).⁴

First of all, microslip implies very small relative motion between the contacting bodies because a substantial portion of the contact region does not slip. Second, as dynamic disturbances are dissipated in a system with “progressive” microslip (regions of microslip progressively decreasing at decreasing load levels), the magnitude of local slip gradually decreases and there is no sudden transition from a slip condition to a stick condition that might trigger high-frequency microdynamic instabilities. Microslip can cause nonlinear modal response, such as changes in frequencies with load magnitude, but such effects are probably of little concern if the variations are small. Therefore:

- the presence of microslip should have a rather benign effect on system dynamics.

⁴ Few models have been developed of dynamic systems with variable-friction elements (e.g., ref. 9). To date, no models have been found that accurately represent microslip behavior in dynamic systems with friction. Although it would be desirable to develop such models in the future and investigate the microdynamic implications of microslip, at the present time, it is generally believed that microslip should not trigger significant high-frequency microdynamic instabilities.

2.2.2 Superimposition of Multiple Interfaces. The discussion in the last section was related to the behavior of a single mechanical interface. Virtually any practical mechanism design will have multiple interfaces acting in parallel or series to transfer load, and any hysteretic-response data will, implicitly, include the effects of microslippage or gross slippage at all of these interfaces. Therefore it is important to consider how hysteretic response due to multiple interfaces will superimpose and be reflected in the global behavior. For this discussion, slippage at an interface can be considered to be a form of compliance in an otherwise elastic system. Hence, the superimposition of slippage from multiple interfaces can be viewed as the superimposition of the compliance at each interface.

From elasticity, it is known that the total compliance of a system of discrete elements acting *in series* is the sum of the individual compliances of the elements. Conversely, the total stiffness of the system of elements acting *in parallel* is the sum of the stiffnesses of the elements. Since hysteretic response is a type of compliance, the total hysteretic response of an assembly of mechanical interfaces acting in series is simply the sum of the hysteretic response of each of the interfaces. One implication of this result is that a mechanism with multiple interfaces that are subject to gross stick-slip should exhibit multiple peaks in its hysteretic response as a function of load-cycle magnitude.

The superimposition of hysteretic response for mechanisms with multiple interfaces acting in parallel is not as easy to visualize since it is the stiffnesses and not the compliances of these interfaces that add. In this case, the combined hysteretic response is expected to appear like the response of a single interface possibly with less sharp features than those that might be seen for a single interface (e.g., flattened region of maximum hysteretic response). The net result though, is that in interpreting the hysteretic-response data from a joint with multiple interfaces in parallel, the same qualitative trends developed for a joint with a single interface can be applied.

2.2.3 Interpreting Hysteretic-Response Data. Substantial insight can be gained into the nature of the frictional interface mechanics within a precision deployment mechanism by measuring its hysteretic response as a function of load-cycle magnitude (see the Appendix for recommendations on hysteretic-response testing and data reduction.) Figure 4 depicts hysteretic-response trends that might be expected from three different mechanism designs. These trends are derived from simplified models of frictional load transfer across interfaces (e.g., Fig. 2).

As discussed in Fig. 3, a mechanism that exhibits gross stick-slip at its mechanical interfaces would be expected to exhibit essentially no hysteresis at load-cycle magnitudes below a stick-slip threshold, and a peak in the hysteresis at a load-cycle magnitude equal to twice the stick-slip threshold load. A mechanism that exhibits only microslip at its interfaces would be expected to exhibit monotonically increasing percent hysteresis with a monotonically increasing slope (dashed curve in Fig. 4). Finally, mechanisms that exhibit microslip that develops into gross stick-slip would exhibit monotonically increasing percent hysteresis at low load-cycle magnitudes, a peak in the percent hysteresis, and monotonically decreasing hysteresis at high load-cycle magnitudes (solid curve in Fig. 4).

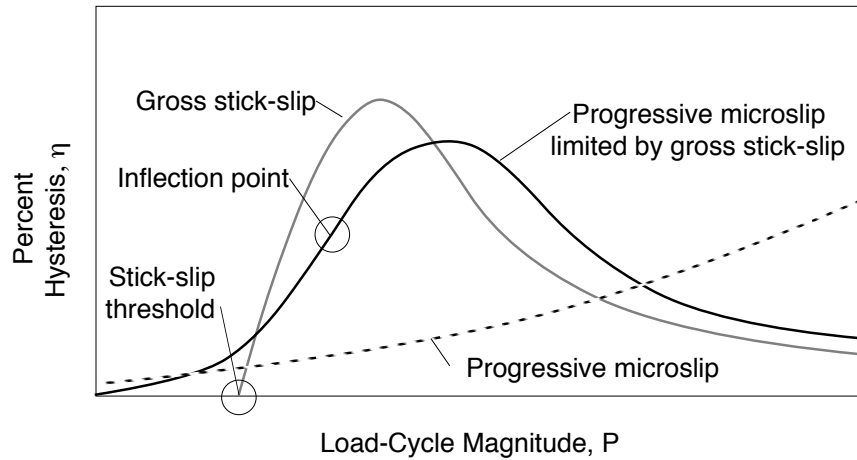


Figure 4. Trends in hysteretic-response data.

An important artifact in the hysteretic-response data of a mechanism with limited microslip would be an inflection point in the percent hysteresis versus load-cycle magnitude curve (ref. 8). This inflection point occurs at a load-cycle magnitude approximately equal to the stick-slip limit. In other words, the load-cycle magnitude associated with this inflection point is equivalent to the stick-slip threshold load of a simple Coulomb element. These hysteretic-response trends will be discussed in more detail in section 4 in the context of establishing criteria on the hysteretic response of precision deployment mechanisms.

For illustration, figure 5 shows a photograph and hysteretic-response data from two versions of a high-precision hinge (one made of composite material and the other made of aluminum) developed for a deployable telescope mirror (ref. 10). The design of this hinge will be discussed in section 5.2.5 of the present paper. The data in Fig. 5 indicate that the two designs exhibit slightly different microslip behavior. In particular, the composite hinge exhibits a maximum percent hysteresis at a load-cycle magnitude of about 800 N (180 lb_f) indicating the presence of limited, microslip. And the inflection in the curve at roughly 600 N (135 lb_f) indicates a gross slip threshold of about 600 N (135 lb_f). Whereas the aluminum hinge exhibits no maximum hysteresis (within the range of load-cycle magnitudes considered), indicating no limit to the progression of microslip.

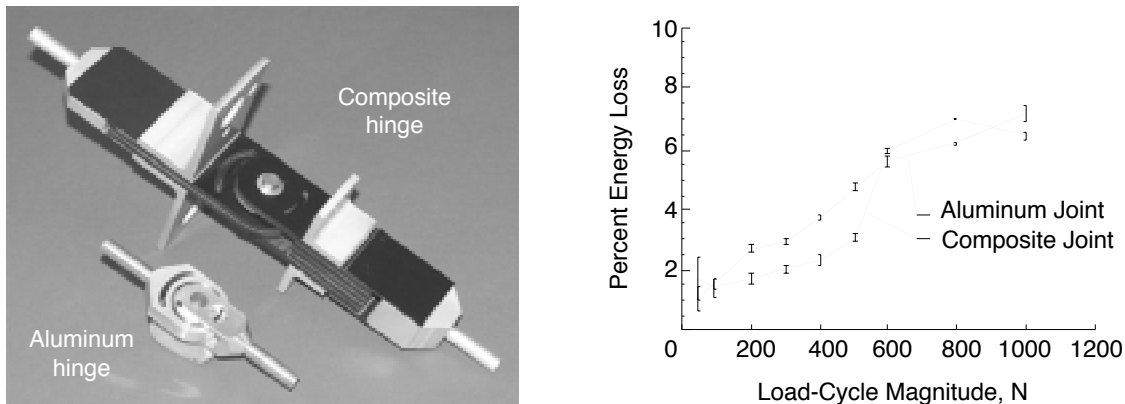


Figure 5. Aluminum and composite prototype hinges.

2.2.4 Implications for Mechanism Design. The results presented in the last section can be summarized and interpreted as follows relative to the design of high-precision deployment mechanism:

- Microslip is not (at present) believed to cause high-frequency microdynamic instabilities, but it is expected to cause nonlinear modal response (i.e., frequency variations with response magnitude). *Therefore, as long as disturbance forces across a precision deployment mechanism remain within its range of microslip, the induced microdynamic effects are expected to be benign (i.e., not a significant problem for active optical systems).*
- Hysteresis in precision deployment mechanisms at low load-cycle magnitudes is dominated by microslip (i.e., pre-sliding friction), as opposed to gross stick-slip (i.e., Coulomb friction).
- The presence of a peak value for the percent hysteresis indicates gross-slip behavior within the mechanism (i.e., the presence of a Coulomb, stick-slip threshold). This gross-slip threshold load should be considered to be an upper bound on operating load-cycle magnitude, and is approximately equal to the load at which the hysteretic-response curve exhibits an inflection point.
- Regardless of the nature of the frictional response (i.e., microslip versus gross slip), the presence of hysteresis in general indicates that load is being transferred through traction forces at mechanical interfaces, and reducing the hysteretic response requires design modifications that reduce frictional load transfer.

The second and third conclusions will be used in section 4.1.2 as the basis of a rationale for establishing criteria on hysteretic response. The fourth conclusion forms the basis of the following general principle for the design of deployment mechanisms that exhibit low hysteresis in response to load cycling:

General Design Principle: When practicable, design the load-carrying components of a high-precision deployment mechanism such that minimal load is transferred through friction at the mechanical interfaces.

The next section of this paper provides an interpretation of the present state of the art in mechanism design from the viewpoint of how current mechanisms comply with this general design principle. Section 5 of the present paper provides specific guidelines for synthesizing new mechanism designs that follow this general principle closely.

State of the Art in Mechanism Design

Despite growing interest in deployable optical instruments within both NASA and the Department of Defense (e.g., ref. 10), no such instruments have been developed and flown to date within the unclassified community (e.g., refs. 11 and 12). Consequently, there is currently no comprehensive and accepted set of guidelines for the development of high-precision deployment mechanisms for optical instruments (ref. 13). This does not imply that there is no current “state of the art” in the design of high-precision deployment mechanisms. Rather, the current state of the art is not explicit, and must be derived from

past experiences in the design of conventional deployment mechanisms and optical-precision mechanisms for non-deployable applications. This “split history” in the present state of the art is reflected in the fact that organizations currently engaged in the study of deployable optical instruments include both optomechanical design groups and aerospace-mechanical design groups with experience in deployable system design.

In general, optomechanical design groups have expertise in the design of positioning devices and kinematic mounts for optical system components (e.g., refs. 14 through 16), but they have little or no expertise in the design of hinges and latches for large deployable structures. Conversely, the aerospace-mechanical design groups tend to have expertise in the design of deployment mechanisms for lower-precision applications like solar arrays and RF antennas (e.g., refs. 17 and 18), but 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 section is intended to present a summary of the relevant state of the art in both deployment-mechanism and optomechanical design from which can be derived specific recommendations for the design of high-precision deployment mechanisms.

3.1 Conventional Deployment Mechanisms

The current state of the art in design of conventional (i.e., non-optical-precision) deployment mechanisms is captured in refs. 13 and 19. Reference 19 is the military specification most commonly applied in the development and qualification of deployment mechanisms. Although ref. 19 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. 13 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.

3.1.1 Design Features to Maximize Stiffness and Strength. In accordance with ref. 19, conventional deployment mechanisms for non-optical-precision structures 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 minimize nonlinear load-displacement response behavior, 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. 20). As shown in Fig. 6, 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 stiffness and strength, conventional hinges typically incorporate some form of mechanical preload to minimize nonlinear load-displacement response (see section 3.1.2)

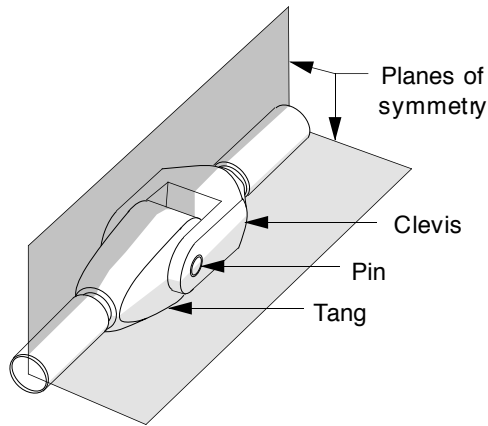


Figure 6. Simple pin-clevis hinge joint.

An example of a latch mechanism designed for assembly of RF antenna structures is shown in Fig. 7 (ref. 21). This latch includes tapered, tongue-and-groove interfaces that contact over a large area for high stiffness and strength. The interfaces are clamped together and preloaded via an internal locking mechanism that is driven by an external collar after the latch halves are mated.

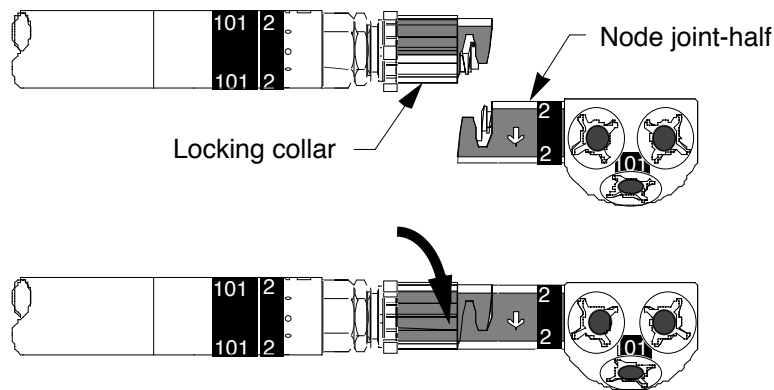


Figure 7. Latch joint developed for erectable construction of RF antenna structures.

3.1.2 Application of Preload to Reduce Nonlinear Response. Under tension-compression load cycling, any mechanical joint can exhibit the three types of nonlinear load-displacement response illustrated in Fig. 8 (ref. 22). 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 increasing regions of contact at mechanical interfaces with increasing load. Finally, hysteresis arises from friction-induced slippage between contacting components within the joint.

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. These 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 (e.g., ref. 13).

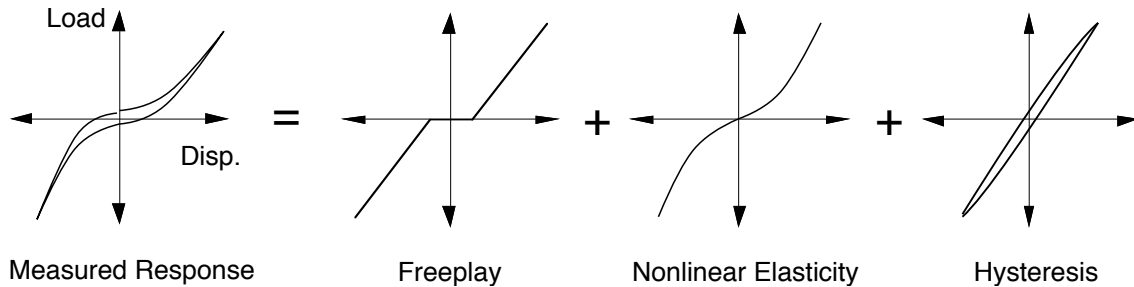


Figure 8. Nonlinear load-displacement response of mechanical joints.

However, recent research has also shown that simply increasing mechanical preload *may not* provide adequate dimensional stability for optical-precision applications (ref. 7). In some mechanism designs, increases in preload can actually increase hysteresis if operating loads are carried through the preload device (see section 3.2.3), 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.

3.2 Optomechanical Devices

The current state of the art in design of optomechanical devices is captured in refs. 14 through 16 and 23 through 33. As mentioned previously, few examples of optical-precision hinges and latches can be found in the literature (e.g., ref. 23 and 24). The vast majority of literature on optomechanical design deals with the design of positioning devices and optical-components mounts rather than hinges and latches. However, the principles employed in the design of positioning devices and mounts are equally applicable to the design of hinges and latches.

Also virtually absent from the optomechanical design literature are references to hysteresis. That is not to say that hysteresis is unimportant. Rather, hysteresis is seldom *explicitly* characterized in the standard process of design and validation of optomechanical devices and systems. However, within the broadly accepted principles of optomechanical design (e.g., ref. 28) there are guidelines for the design of load paths, interfaces, and preload, which *implicitly* lead to designs that exhibit minimal pre-gross-sliding hysteresis, and hence, improved dimensional stability. In the following sections, these salient optomechanical design principles are reviewed and their implications for reducing hysteresis are explained.

3.2.1 Determinate versus Indeterminate Load Paths. Within the optomechanical design community, 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 platform (ref. 28) or the more compact arrangement of three, semi-kinematic mounts known as a Kelvin clamp (Fig. 9, ref. 32). Between small optical-system components (e.g., lenses and lens mounts), a determinate load path is typically established using a convenient arrangement of flexure or ball-bearing mounts.

Figure 10 shows an example of a three-point, Kelvin clamp used to mount the CERES instrument package onto the Earth Observing System (EOS) spacecraft. This three-point mount includes three receptacles that are located on the spacecraft at the vertices of an equilateral triangle. These receptacles include flat, v-grooved, and conical surfaces, respectively, that interface to three spherical fittings mounted to the instrument package. Illustrations of the three interface geometries are shown in Fig. 9 with an indication of how all six rigid-body degrees of freedom are restrained by the three-point mount.

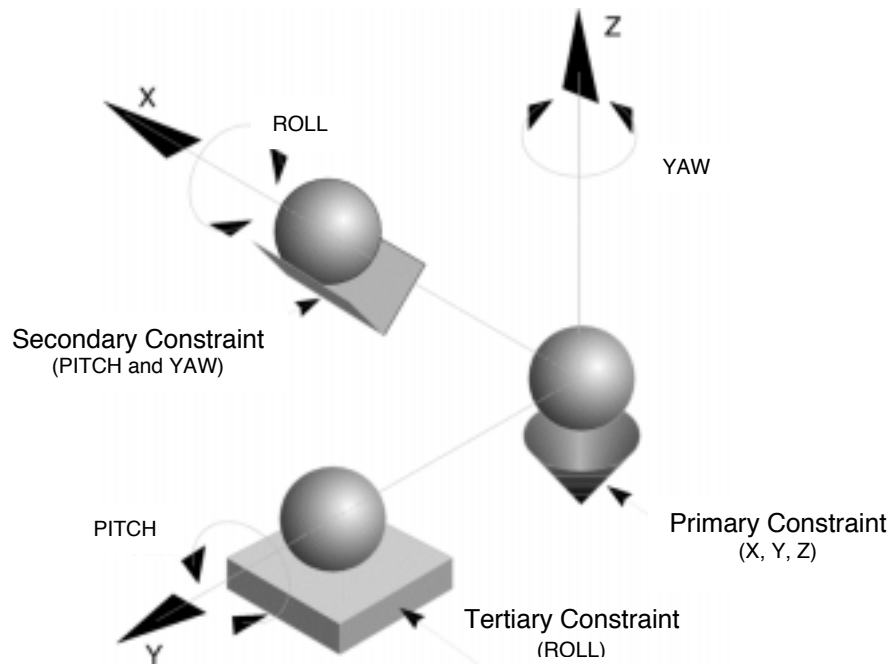


Figure 9. Three interface geometries used in three-point (Kelvin) mount.

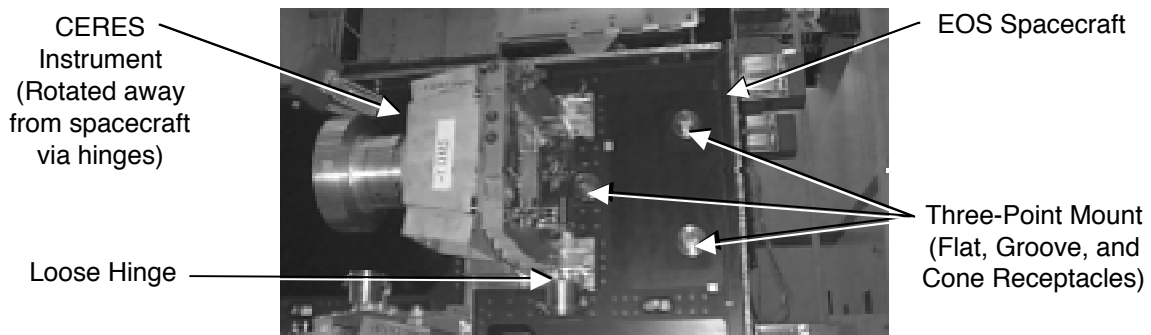


Figure 10. Three-point Kelvin mount between CERES instrument and EOS spacecraft.

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 components (e.g., ref. 28). 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.

3.2.2 Non-Conforming Versus Conforming Interfaces. To make interfaces between 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 (see Section 5.2). Figure 9 shows illustrations of typical 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.

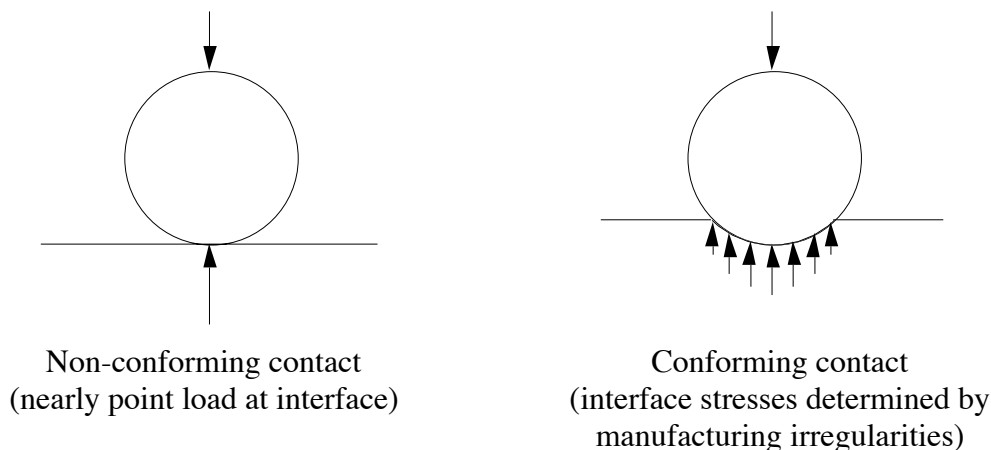


Figure 11. Non-conforming versus conforming interfaces.

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. 11). For example, if the two conforming surfaces are not perfectly matched in shape, then there will be 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 applied load change. Conversely, the use of non-conforming interfaces virtually guarantees that the interface stress distribution will be fairly accurately known, and independent of localized imperfections in the mating surfaces. Consequently:

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

3.2.3 Compliant versus Non-compliant Application of Preload. 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 components (e.g., ref. 29).

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).

Suggested Criteria

Section 2 discussed the relationships between hysteresis in deployment mechanisms and microdynamic instabilities and presented background information for interpreting hysteretic-response data. The Appendix presents a rational methodology by which to conduct hysteresis-response testing for preliminary design qualification of candidate high-precision deployment mechanisms. This section presents a suggested set of specific criteria for qualifying the hysteretic response of a design. These criteria have been derived from thoughtful consideration of the aspects of hysteretic response that can have a significant effect on the dimensional stability of a deployable optical instrument. *However, the criteria suggested herein have not been exhaustively tested and proven, and as such they are to be regarded as a guide to design and not a set of NASA requirements, except as may be defined in formal project specifications.*

4.1 Stiffness Criterion

It is impossible to suggest any rational criteria on the hysteretic response of a single deployment mechanism without first considering the relationship between the response of

the mechanism and the response of the structure into which the mechanism might be installed. As will be shown, this relationship is determined largely by the relative stiffness of the deployment mechanism and the structural members it interconnects. In addition, the geometric arrangement of a structure and the location of the mechanism within that structure determine how dimensional instabilities, due to hysteresis in the mechanism, affect the critical, optical-alignment dimensions of the structure (e.g., locations of hard points for mounting of optical components). Furthermore, it should be apparent that by designing a structure with reasonable depth (e.g., ref. 1) and judiciously locating mechanisms within the structure, it is possible to avoid significant “amplification” of mechanism-induced dimensional instabilities (see section 5.1.1).

For a given structural design, a requirement on deployment mechanism stiffness can be derived from the relationship between the hysteretic response of the mechanism and the hysteretic response of the structure. Figure 12 presents a simple example of a uniform deployable truss beam that can be used for the purpose of illustrating this process. Assume that each longeron strut of the truss is identical with identical deployment mechanisms at each of its ends (as shown in the inset in Fig. 12). Also assume that all batten and diagonal struts are rigid, the structural loading condition of interest is a lateral tip load ($P_{\text{structure}}$), and the critical optical-alignment degree of freedom ($\delta_{\text{structure}}$) is measured at the point of application of the load. If all deployment mechanisms exhibit the same hysteretic response, it can be shown that the percent hysteresis in the optical-alignment degree of freedom ($\eta_{\text{structure}}$) is related to the percent hysteresis in each of the deployment mechanisms ($\eta_{\text{mechanism}}$) by:

$$\eta_{\text{structure}} = \frac{\eta_{\text{mechanism}}}{1 + \frac{(EA/L)_{\text{mechanism}}}{(EA/L)_{\text{strut}}}} \quad (1)$$

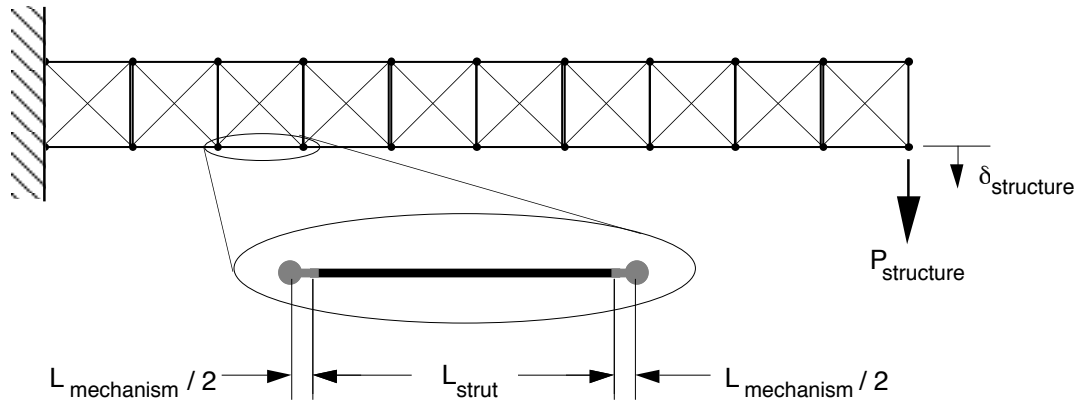


Figure 12. Illustration of a simple deployable truss beam.

Equation (1) can be rearranged to give the following criterion on the stiffness of the mechanism, $(EA/L)_{\text{mechanism}}$, in terms of the stiffness of the longeron strut, $(EA/L)_{\text{strut}}$, and the ratio of the percent hysteresis in the mechanism to that in the structure:

$$(EA/L)_{\text{mechanism}} \geq (EA/L)_{\text{strut}} \left(\frac{\eta_{\text{mechanism}}}{\eta_{\text{structure}}} - 1 \right) \quad (2)$$

Equation (2) applies only to the uniform truss beam illustrated in Fig. 12, but equations similar to Eq. (2) can be derived for any deployable structure geometry and mechanism location. Although it might be difficult in some cases to derive an exact relationship for Eq. (2), it should be possible to derive a reasonable approximation that is explicit (like Eq. (1)) and useful for establishing mechanism performance requirements. Also, it should be recognized that Eq. (2) (or a similar equation for another structure) requires an estimate for the percent hysteresis of the mechanism. This estimate will be derived in the next section.

4.2 Maximum-Load Criterion

The variations in hysteretic response exhibited by a high-precision deployment mechanism under load cycling were discussed in some detail in section 2 of the present paper. Figure 13 illustrates (once again) typical hysteretic response trends that are indicative of load transfer across a deployment mechanism comprised of multiple mechanical interfaces. A maximum-load criterion can be derived for the operating load of the mechanism by requiring that the mechanism be load-cycled within its microslip response regime. As illustrated in Fig. 1, (and explained in section 2.2.3) this maximum load is approximately the load at which the curve of percent hysteresis versus load-cycle magnitude reaches an inflection.

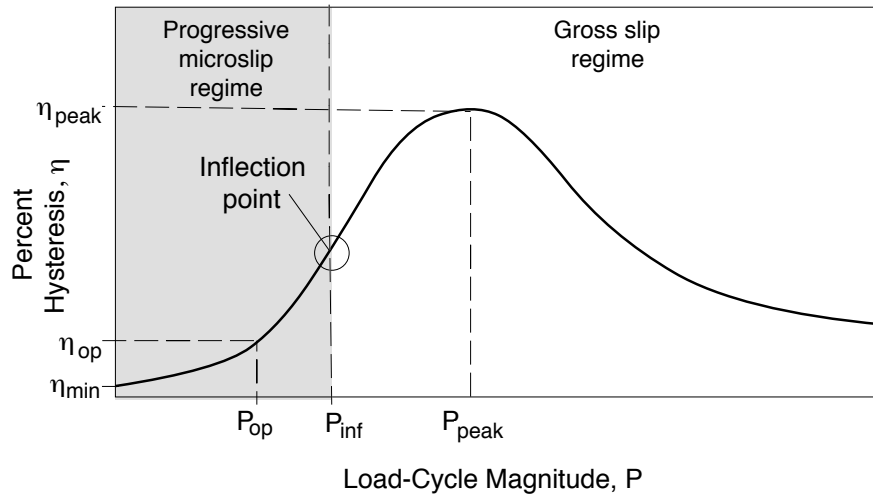


Figure 13. Hysteretic response of a high-precision deployment mechanism.

4.3 Hysteresis Criterion

For deployable instruments with quasi-static re-alignment capability between observational windows the operational load level of interest for qualifying hysteretic response in the mechanism is the nominal disturbance load during the observation window. For deployable instruments without active re-alignment, the operational load level of interest is the worst-case disturbance load seen on-orbit.

As discussed in section 2.2.4, as long as the operating load of the mechanism is kept within its microslip regime (i.e., below the point of inflection of the percent hysteresis versus load-cycle magnitude curve), no significant microdynamics are expected to occur. However, this conjecture is based on the assumption that the percent hysteresis of the

mechanism essentially vanishes (or approaches viscoelastic limits) as the load-cycle magnitude vanishes. If the percent hysteresis approaches a finite value as load-cycle magnitude vanishes (η_{\min} in Fig. 13), this limiting value must be used in Eqs. (1) or (2) (or equivalent equations for the structure of interest) to determine if adequate stability is ensured.

If the operating load of the mechanism exceeds the load limit of microslip, the value of percent hysteresis that should be used in Eqs. (1) or (2) (or the equivalent), is the value of percent hysteresis at the maximum load-cycle magnitude attained.

4.4 Guidelines for Compliance

In order to minimize restrictions on hysteretic response and maximize the range of applications for a specific mechanism design, it is prudent to make every effort to apply the design principles outlined in the next section before beginning testing and qualification of a mechanism design. Once a well-conceived design has been developed, it can either be qualified for a specific application following the criteria of the previous sections, or its hysteretic response can be quantified over a range of load-cycle magnitudes for use in developing stability requirements for active control systems.

Recommended Design Practices

With the criteria on hysteretic response suggested in the last section as a means of evaluating precision mechanism designs, and the salient optomechanical design principles identified in section 3.2 that affect hysteretic response, it is now important to consider specific design practices that can reduce hysteretic response in deployment mechanisms. It is assumed herein that the reader is familiar with the process of designing deployment mechanisms for space, and specifically, the non-structural-performance issues normally considered in the design process such as material selection, lubrication, manufacturing constraints, etc. Although these important issues are not explicitly addressed herein, they must be considered in addition to the issues addressed herein. For readers not familiar with these important issues, it is recommended that they review refs. 13, 19, 34, and/or any other comprehensive mechanical design guide for aerospace applications.

5.1 General Deployable System Design Recommendations

Before considering specific design recommendations for high-precision deployment mechanisms, it is important to consider several general issues that can affect the selection of mechanism and/or the hysteretic-response requirements placed on the mechanism.

5.1.1 Deployable Structure Design. As described in Section 4, it is impossible to place requirements on the hysteretic response of a single deployment mechanism without first considering the relationship between the response of the mechanism and the response of the structure into which the mechanism might be installed. Clearly, the geometric arrangement of a structure and the location of the mechanism within that structure determine how dimensional instabilities, due to hysteresis in the mechanism, affect the critical, optical-alignment dimensions of the structure (e.g., locations of hard points for mounting of optical components). Furthermore, it should be apparent that by

designing a structure with reasonable depth (e.g., ref. 1) and judiciously locating mechanisms within the structure, it is possible to avoid significant “amplification” of mechanism-induced dimensional instabilities. 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. 14). A corollary to this rule is :

- *develop designs for the deployable structure with a maximum structural depth, and a minimum total number of deployment mechanisms.*

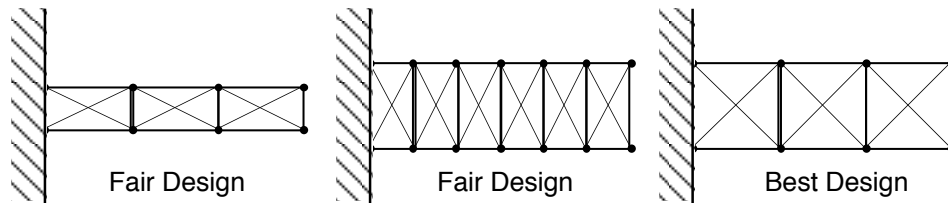


Figure 14. Illustration of “ideal” mechanism spacing in deployable structure.

5.1.2 Use of Existing Mechanism Designs. As mentioned previously, designers are encouraged to continue to apply the design principles of conventional deployment mechanisms (refs. 13 and 19) 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:

- *when practical, use existing, flight-proven, hinge and latch mechanisms (e.g., ref. 23) for optical-precision deployable structures, when such application can be shown to satisfy dimensional-stability requirements.*

In cases where existing designs might prove inadequate:

- *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.

5.1.3 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 (e.g., ref. 35). In most cases, this approach is considered largely because there is a general feeling 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. 23), whereas there are no known examples of *flight-proven* 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. 22 and 24). 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.*

5.1.4 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 as discussed in section 3.2.3. 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. 17 and 18).

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 (section 3.2.3), because they carry substantial operational loads in addition to applying preload. In other words, it is difficult to design a distributed preload system that effectively loads all deployment mechanisms but does not provide an alternative load path through the structure. This issue makes distributed preload systems less desirable for application to optical-precision, deployable structures. Especially for small to moderately sized deployable structures that contain a relatively small number of hinges and latches.

For deployable optical structures with a large number of hinges and latches and for which distributed preload systems might prove attractive, it is important to adhere to optomechanical Design Principle 3 (section 3.2.3) to the extent possible in developing

such distributed preload systems. Specifically, for optical-precision deployable structures:

- 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 .

5.2 High-Precision Mechanism Design Recommendations

As mentioned in section 3, designers are encouraged to continue to apply the design principles of conventional deployment mechanisms (refs. 13 and 19) 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. The following specific recommendations, derived from experience in the design of optomechanical systems, are intended to complement those encompassed by refs. 13 and 19.

5.2.1 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.

5.2.2 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 hysteresis (e.g., ref. 36). Hence:

- when practical, 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 the interface be designed to be much greater than the maximum expected operating load of the mechanism (e.g., a factor of 10 greater).

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

5.2.3 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 in section 3.2.2:

- 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 (see section 5.2.5). 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 clamp illustrated in Fig. 9 and photographed in Fig. 10 includes three latches that incorporate point contact between a sphere and a plane, and line contact between a sphere and a cone (Fig. 15). These three interface geometries are the simplest non-conforming geometries for latches that exhibit one-, two-, and three-degrees of restraint, respectively.

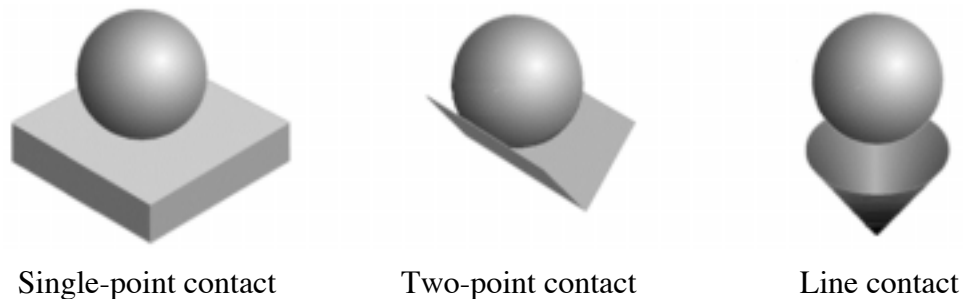


Figure 15. Spherical, non-conforming, latch-interface geometries.

In addition to being simple, the latch-interface geometries presented in Fig. 15 are kinematic (or semi-kinematic in the case of the sphere-in-a-cone interface). Hence, they lend themselves to the design of a latch that has determinate internal load paths (see section 3.2.1). However, as pointed out previously, simple (i.e., one-, or two-point-contact) latch-interface geometries like this might exhibit unacceptably low stiffnesses, so it might be necessary in some applications to sacrifice determinacy in the load path for stiffness across the interface by using multiple non-conforming contacts (see sections 5.2.5 and 5.2.6).

5.2.4 Minimizing Friction Forces at Non-Fixed Interfaces. All of the interface geometries depicted in Fig. 15 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 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 16 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 friction forces.

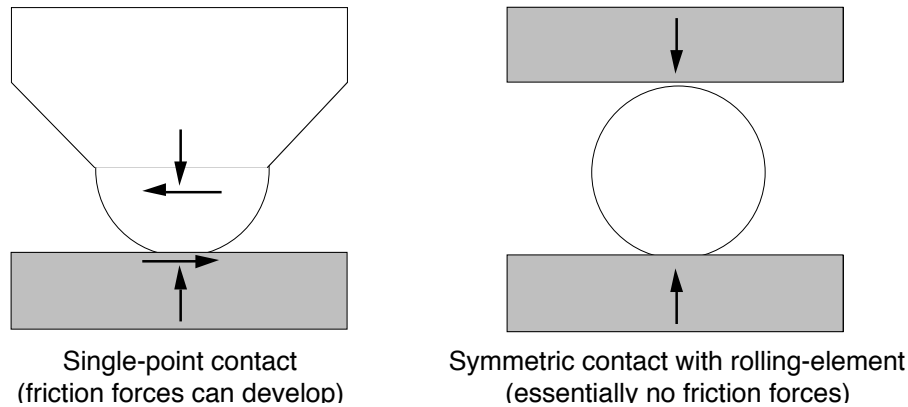


Figure 16. The use of symmetric, non-conforming contacts to eliminate friction forces.

Figure 17 depicts a method of eliminating tangential stiffness at a two-point contact using a V-groove (as depicted in Fig. 15). Under the area of the 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. 17. This detail effectively precludes friction forces from developing and, hence, reduces the potential for friction-induced hysteresis under load cycling.

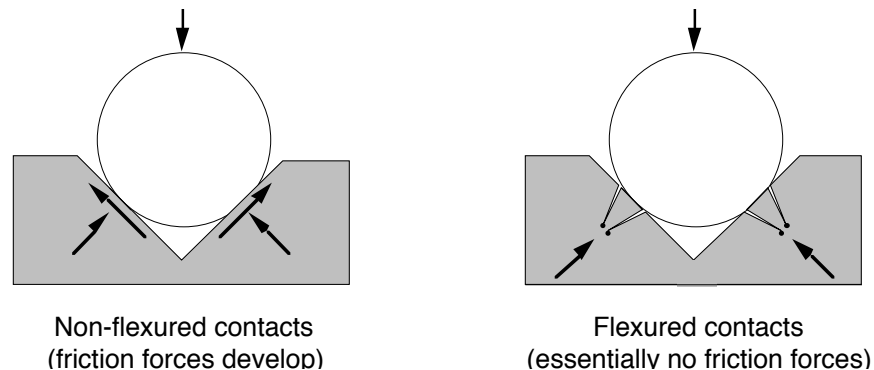


Figure 17. The use of flexures to eliminate friction forces.

5.2.5 Increasing the Stiffness of Non-Conforming, Non-Fixed Interfaces: Use of Rolling-Element Bearings in Hinges and Latches. 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 use in non-fixed interfaces 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 18 includes sketches of a high-precision hinge developed for a deployable telescope mirror (ref. 10). Reference 24 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 includes over 40 balls that are preloaded within the bearing races. Each ball contacts the inner and outer races at points and is capable of transmitting load only along the line of action established by the two contact points as shown in Fig. 19. 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 friction (ref. 7).

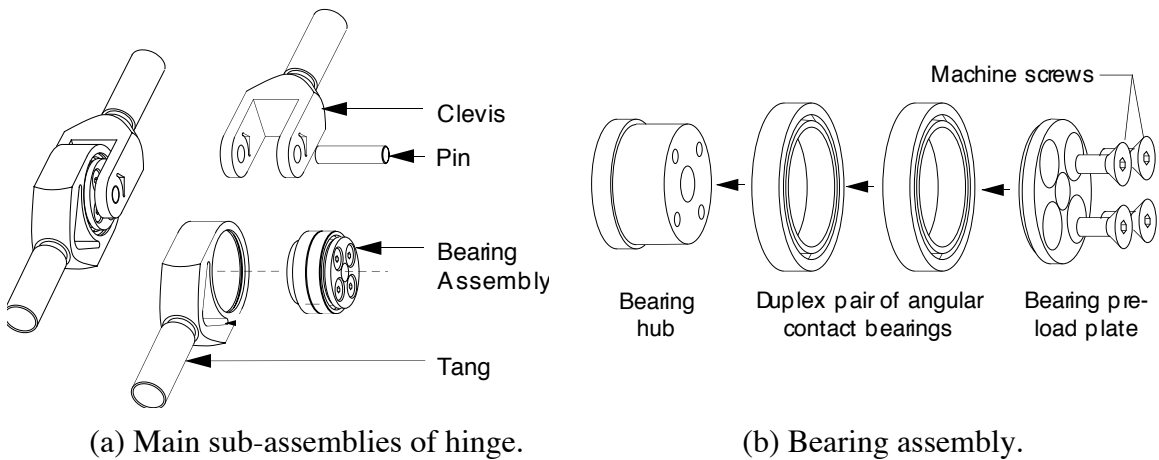


Figure 18. High-precision hinge incorporating pre-loaded, rolling-element bearings.

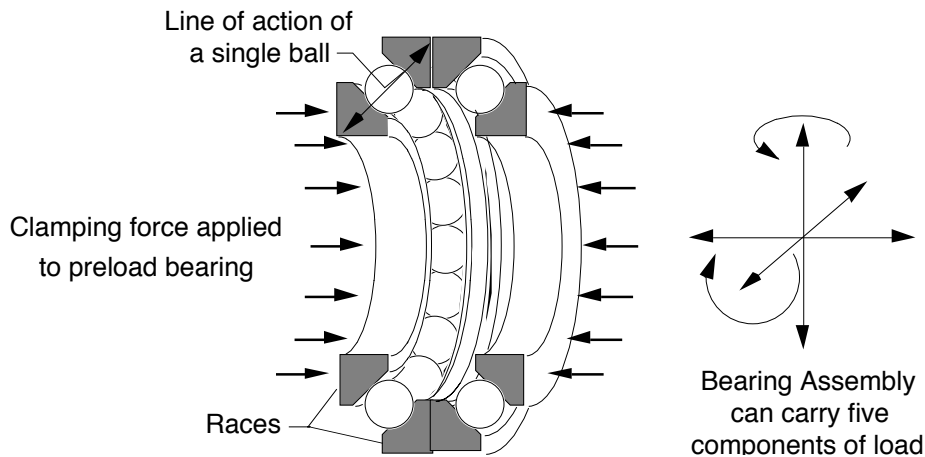


Figure 19. Angular contact bearing pair.

The assembly of bearing balls provides restraint against load in five directions as shown in Fig. 19 (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.

5.2.6 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 in the design of optomechanical devices that can influence the selection of components and affect, indirectly, hysteretic response. Therefore, a few comments relating to athermalization are provided here.

First, it is recognized that low-CTE materials like 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 of dramatically different CTE (usually composite members) such that the different CTEs and lengths of the components cancel out (Fig. 20). This approach amounts to achieving a balance between 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. 20). 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. 20).

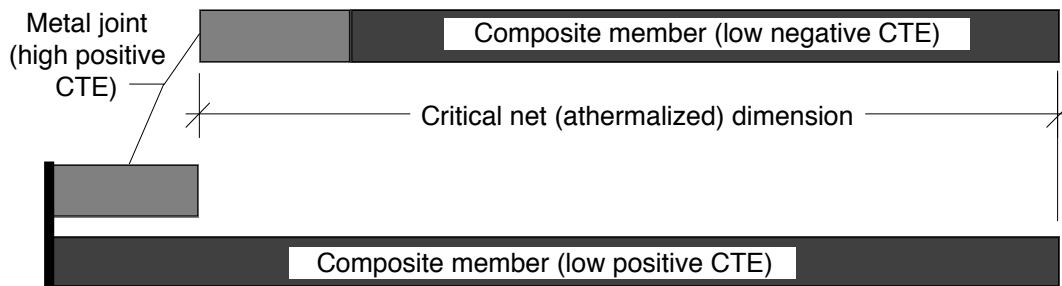


Figure 20. Traditional approach to athermalization of metal joint components.

Another method by which athermalization that 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. 21, and an example of this approach applied to the design of an athermalized hinge is shown in Fig. 5 (ref. 24). In the example shown in Fig. 5, a nearly axisymmetric hinge mechanism is embedded in a quasi-isotropic, flat-laminated composite member in such a way that thermal growth of the hinge only results in uniform radial expansion of the hinge within the composite member. A cut-out 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 stiffness (ref. 22).

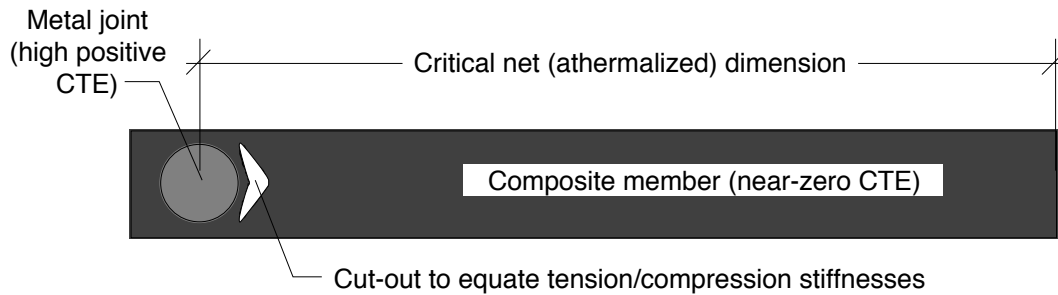


Figure 21. Alternative approach to athermalization of metal joint components.

Hence, another advantage to the use of rolling-element bearings in the design of non-fixed interfaces in hinges and latches is that axisymmetric geometries are easy to devise with rolling-element bearings, and these geometries lend themselves to athermal integration into composite structural members as shown in Fig. 21.

Summary

This paper is intended to facilitate the development of deployable, optical instruments by providing a rational approach for the design, testing, and *preliminary* qualification of precision (i.e., low-hysteresis) deployment mechanisms for these instruments. It is tacitly assumed that *final* qualification of any deployment mechanism requires system-level testing in a significant portion (or all) of the deployable, optical-instrument structure. Suggestions regarding such system-level tests are beyond the scope of the present paper except that the following assertion regarding the relationship between system-level microdynamic response and hysteretic response is assumed.

- the magnitude of microdynamic response expected in a system is equivalent to the magnitude of hysteresis in the system.⁵

The Appendix of the present paper includes recommended standards for conducting preliminary qualification tests on a candidate precision deployment mechanism. Criteria are suggested in Section 4 of the present paper for preliminary qualification of precision deployment mechanisms. These criteria follow from consideration of some basic principles that have become commonly accepted by the precision-deployment research

⁵ This relationship between hysteresis and microdynamics is logical, but has yet to be proven rigorously. The reader is therefore encouraged to apply it only in the preliminary qualification of mechanism designs, and rely on microdynamic testing of complete deployable structures for final qualification.

community. Most notably, the criteria suggested herein are derived from the general belief that:

- the presence of microslip in precision deployment mechanisms should have a rather benign effect on system dynamics. Therefore, as long as loads across a precision deployment mechanism remain within its range of microslip, the induced microdynamic effects are expected to be benign (i.e., not a significant problem for active optical systems).⁶

In the process of developing and qualifying designs for precision deployment mechanisms, designers are encouraged to:

- use existing, flight-proven, hinge and latch mechanisms 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 in Section 5 of the present paper.

The design guidelines suggested herein are motivated by the following basic assertion regarding load-cycle response behavior in precision mechanisms:

- the presence of hysteresis in the load-cycle response of a precision deployment mechanism indicates that load is being transferred through traction forces at mechanical interfaces, and reducing hysteresis requires design modifications that reduce frictional load transfer.

Many of the design guidelines included herein come directly from the field of optomechanical engineering, and are, therefore, neither newly developed guidelines, nor are they 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.

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. However, it is expected that this paper, with revisions as experience may indicate to be desirable, might eventually form the basis for a set of uniform design requirements for high-precision deployment mechanisms on future NASA space-based science instruments.

⁶Although it is accepted that microslip can cause nonlinear modal response, such as changes in frequencies with load magnitude, such effects are probably of little concern if the variations are small. Also, at the present time, it is generally believed that microslip should not trigger significant high-frequency microdynamic instabilities.

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Appendix: Recommended Test Procedures

A critical step in establishing a methodology for qualifying structural designs is to first establish a rational set of test procedures to adequately characterize the response behavior of interest. As discussed in Section 2 of the present paper, experience has shown that the magnitude of hysteresis exhibited by a given high-precision deployment mechanism varies dramatically with test condition (e.g., load-cycle magnitude, direction of load application, and rate of load application, refs. 3, 6, 7, 12, 22, and 36). Also, it has been found that no single value of hysteresis fully characterizes the behavior of a given mechanism, and hence there is a need to perform a series of test instead of a single test during the process of qualifying a design. However, it is possible to define a reasonably concise set of tests that adequately characterizes hysteretic response for the purpose of preliminary qualification of a design and in order to gain substantial insight that can lead to thoughtful design revisions. This section defines a set of tests that can be used for such purposes.

Ultimately, it should be recognized that final qualification of any mechanism design requires system-level testing of the deployable instrument structure (or a substantial subassembly thereof) with the mechanism of interest installed and subjected to global loads representative of those expected under operational conditions.

A.1 Load Conditions

One of the most important considerations in defining test requirements is that of the load condition to be applied during testing. Ideally, for the purpose of qualifying a design, one would prefer to apply precisely the same load condition to the deployment mechanism as it would see under operational loading of the structure. However, two issues make it difficult to precisely define such a loading condition. First, the operational loads within the structure are typically not well known, and second, many typical mechanism installations include unavoidable load-path offsets and asymmetries that result in hard-to-predict, elastically coupled loads being applied to the mechanism (e.g., combined tension/compression and bending loads).

Despite uncertainties in precise operational loading, most deployment mechanisms are designed for a single primary load direction (e.g., uniaxial tension/compression). In other words, most deployment mechanisms are installed within a deployable structure at a location and in an orientation for which the dominant component of the operational loads is expected to be in one direction. Therefore:

Load Condition Recommendation 1: For the purpose of preliminary qualification testing apply load to the mechanism in only the primary loading direction considered during the design of the mechanism (e.g., pure uniaxial tension/compression, or pure bending, etc.)

In order to quantify hysteretic response, it is necessary to apply the primary loading in a cyclic fashion. Furthermore, in order to characterize adequately variations in the hysteretic response, it is necessary to conduct load-cycle tests over a range of load-cycle magnitudes. Although operational loads are dynamic loads (i.e., time-varying cyclic

loads), it has been found that quasi-static load cycling provides sufficient insight into hysteretic response for the purpose of preliminary design qualification. Therefore:

Load Condition Recommendation 2: For the purpose of quantifying variations in the hysteretic response for preliminary design qualification, conduct quasi-static, load-cycle testing of the candidate mechanism over a range of primary load magnitudes at least up to the maximum load expected under reasonable operational conditions.

The minimum load-cycle magnitude for a series of tests is usually determined by instrumentation limitations that will be addressed in section A.3. In cases where response under the application of secondary loads and/or coupled loads might be a concern, a broader test matrix can be constructed by extension of the above criteria.

A.2 Test Setup

By definition, high-precision deployment mechanisms exhibit *very low levels of hysteresis* under load cycling (i.e., less than 1% of full-scale displacement). Hence, to ensure accurate characterization of hysteretic response, great care must be taken to apply the load cleanly to the specimen, provide adequate instrumentation to characterize the response, and minimize noise and hysteresis in the instrumentation. Otherwise, computed values of hysteresis can easily be corrupted by the mechanical test set up.

To ensure the load is applied cleanly to the specimen (i.e., no unwanted secondary loads arise during application of the primary load), it is advisable to incorporate compliant linkages (e.g., flexures or hinges) as appropriate between the specimen and the test apparatus (e.g., load frame) to accommodate specimen misalignment. A less desirable alternative, is to provide adequate load instrumentation to determine the existence and magnitude of any undesirable load coupling, and to attempt to eliminate such coupling by adjusting (e.g., shimming) the specimen within the test apparatus to eliminate misalignment. Experience has shown the later option is difficult to accomplish in practice, so emphasis is placed on the use of compliant linkages to eliminate unwanted load coupling. Therefore:

Test Setup Recommendation 1: Whenever possible, incorporate compliant linkages between the test specimen and the test apparatus to insure clean application of the desired load (i.e., no secondary load coupling).

Depending on the load condition being applied (e.g., pure tension/compression versus bending), it is possible that the displacement response of interest cannot be measured directly and must be inferred by comparing the measurements made from multiple sensors arrayed around the specimen. For example, centerline displacement cannot typically be measured directly; it can only be inferred from displacement measurements made adjacent to the centerline of the specimen. In setting up such an array of displacement sensors, allowances must be made for the fact that small misalignments or asymmetries in the specimen can give rise to elastic coupling (e.g., bending-extension coupling) that can easily confound the results taken from any single displacement sensor

(e.g., ref. 36). Hence, although a single displacement measurement of the specimen is typically all that is desired:

Test Setup Recommendation 2: Multiple displacement sensors must be arrayed appropriately around the specimen, and measurements from these sensors must be average or compare in order to fully account for unavoidable elastic coupling effects.

Finally, experience has shown that the use of non-contacting displacement sensors (e.g., capacitive or fiber-optic gap sensors) instead of more traditional contacting sensors (e.g., electro-mechanical displacement transducers) can eliminate unwanted sources of hysteresis in the test setup. Also, load cells should be calibrated prior to testing to ensure that their hysteretic response is insignificant relative to that of the specimen. In practice, it has been found that, in addition to calibrating all instrumentation individually, a “calibration” specimen can be very useful in qualifying the entire test setup by quantifying any hysteretic or other nonlinear response effects inherent in the setup. Ideally, such a calibration specimen should be identical to the deployment mechanism in size and stiffness, and have the same mechanical features on it for interfacing to the test apparatus. However, a calibration specimen should be fabricated from a single piece of material such that its load-displacement response is perfectly linear. In other words:

Test Setup Recommendation 3: Great care must be taken to ensure that all instrumentation exhibits adequately low inherent hysteresis, and the use of a calibration specimen as a means of final validation of the test setup is highly encouraged.

A.3 Data Reduction and Hysteresis Calculation

A critical issue in the gathering and reduction of data from hysteresis-response testing is that of time synchronization of the data channels. Quite simply, any discrepancy in synchronization between the load and displacement data channels will incorrectly be interpreted as hysteresis in the response of the specimen. For example, a slight lag of the load data relative to the displacement data will be interpreted as *negative* hysteresis! Therefore:

Data Reduction Recommendation 1: Great care must be taken to ensure that all data channels are time synchronized (i.e., multiplexing and analog-to-digital conversion hardware and routines must be checked to ensure they introduce no significant time lag between data channels).

Typical data from a load-cycle test (conducted in accordance with the criteria of sections A.1 and A.2 and Data Reduction Recommendation 1) of a high-precision deployment mechanism are presented in Fig. A-1 (ref. 36). The raw (i.e., unfiltered) response is presented in Fig. A-1(a), and the corresponding raw hysteretic response (derived by subtracting the best-fit straight line from the total response) is presented in Fig. A-1(b). Note that the displacement data presented in this figure were derived from averaging the measurements from two displacement sensors as suggested by Test Setup Recommendation 2 (section A.2).

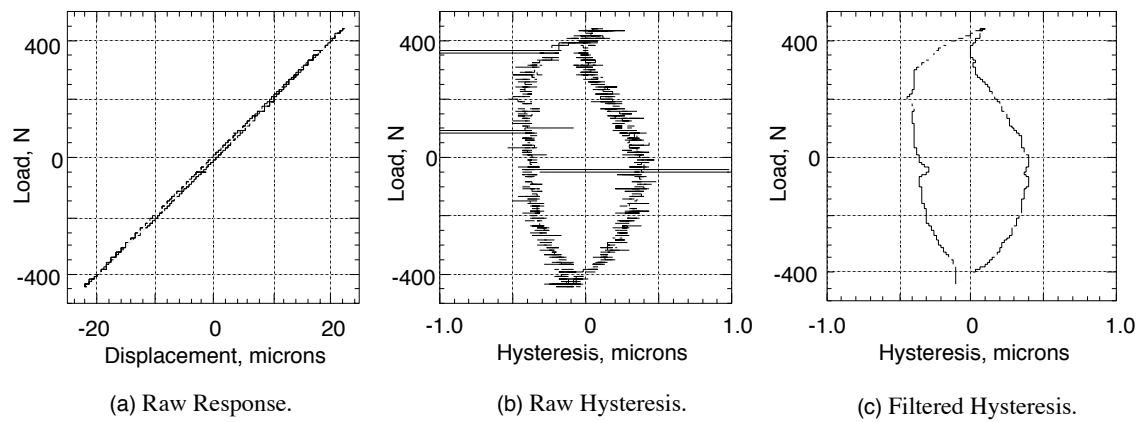


Figure A-1. Typical load-displacement response of a high-precision mechanism.

The data in Fig. A-1 are considered to be typical in the sense that high-precision deployment mechanisms should exhibit nearly linear load-displacement response with very small hysteretic response (i.e., typically less than a micron in absolute magnitude). Furthermore, one should expect instrumentation noise in such data to be significant and potentially to obscure the hysteretic response. Consequently, efforts should be made to filter the data in order to improve measurement resolution. For example, Fig. A-1(c) presents the data from Fig. A-1(b) after numerical filtering has been applied. In this case, it can be seen that filtering effectively reduced instrumentation noise by more than an order of magnitude. However:

Data Reduction Criterion 2: Great care must be taken to ensure that filtering algorithms do not induce biases that might affect time synchronization of the data.

For example, acceptable results have been achieved by using so-called “forward-backward” filtering algorithms that are specifically designed to induce no bias by operating twice on the data streams – once in the forward (temporal) direction and once in the backward direction. However, most tradition one-direction filters should probably be avoided.

A data-reduction strategy that might be considered as an alternative to traditional filtering schemes is to pass the unfiltered load-displacement data through a numerical integrator that explicitly calculates accumulated strain energy. Applying such an algorithm to a set of data from an entire load cycle will result in computation of energy loss within the hysteresis loop. Then, the normalized or percent hysteresis can be computed by dividing the total energy loss by the maximum elastic strain energy at the given load-cycle magnitude. One advantage of this data-reduction approach is that data biasing is not typically an issue with numerical integration routines, and all forms of instrumentation noise can be simultaneously “filtered” out.

Another advantage of using computed energy loss as the basis for calculating hysteresis is that this method leads to a very rational definition for normalized or percent hysteresis. Figure A-2 presents a simplified representation of a hysteresis loop that can be used to define normalized or percent hysteresis. As mentioned previously, the area

within the hysteresis loop is, by definition, the energy loss during a single load cycle. For the hysteresis loop shown in Fig. A-2, this energy loss is calculated by:

$$U_{\text{hys}} = \frac{1}{2} (P_{\text{ten}} + P_{\text{com}}) \delta_{\text{hys}} \quad (\text{A-1})$$

The peak strain energy in tension (denoted U_{ten}) and the peak strain energy in compression (denoted U_{com}) are calculated by:

$$U_{\text{ten}} = \frac{1}{2} P_{\text{ten}} \delta_{\text{ten}} \quad , \quad U_{\text{com}} = \frac{1}{2} P_{\text{com}} \delta_{\text{com}} \quad (\text{A-2})$$

The normalized hysteresis, η_{hys} , can be now defined as:

$$\eta_{\text{hys}} = \frac{U_{\text{hys}}}{U_{\text{ten}} + U_{\text{com}}} = \frac{(P_{\text{ten}} + P_{\text{com}}) \delta_{\text{hys}}}{P_{\text{ten}} \delta_{\text{ten}} + P_{\text{com}} \delta_{\text{com}}} \quad (\text{A-3})$$

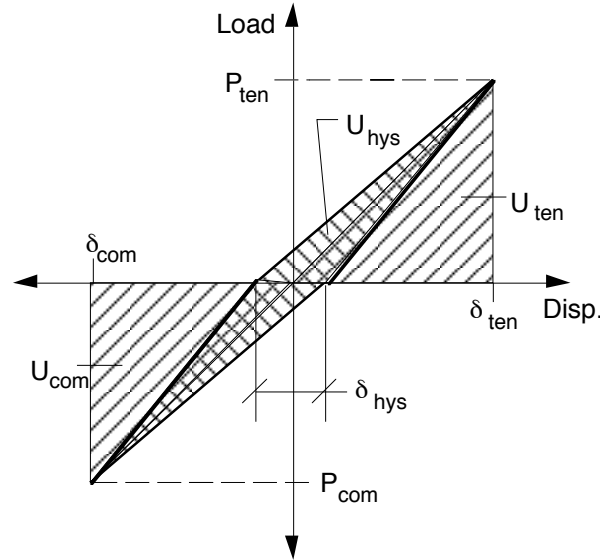


Figure A-1. Simplified response plot for defining hysteretic energy-loss calculation.

The reason that Eq. (A-3) represents a rational definition for normalized hysteresis can be better understood by assuming that the peak tension and compression loads (P_{ten} and P_{com} , respectively) have the same value (as is usually the case during load-cycle testing). In this case, Eq. (A-3) simplifies to:

$$\eta_{\text{hys}} = \frac{U_{\text{hys}}}{U_{\text{ten}} + U_{\text{com}}} = \frac{\delta_{\text{hys}}}{(\delta_{\text{ten}} + \delta_{\text{com}})/2} \quad (\text{A-4})$$

Eq. (A-4) demonstrates that, for the simplified hysteresis case depicted in Fig. A-2, normalizing the hysteresis using the strain-energy calculations gives identically the same result as simply dividing the maximum width of the hysteresis loop by the average of the peak displacements in tension and compression. This later method of computing normalized hysteresis using (i.e., using only the measured displacements) is a simple

approach that is often used in practice. As mentioned previously, the method of using strain-energy calculations from the data is a more involved calculation, but one that can also be implemented in a numerical integration routine that automatically compensates for noise in the load-displacement data. Therefore:

Data Reduction Recommendation 3: The standard definition for percent hysteresis is hereby established to be the total energy loss per load cycle divided by the sum of the peak elastic strain energies in tension and compression. For consistency, all hysteretic-loss should be computed using this standard definition.

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13. ABSTRACT (Maximum 200 words) This paper is intended to facilitate the development of deployable, optical instruments by providing a rational approach for the design, testing, and qualification of high-precision (i.e., low-hysteresis) deployment mechanisms for these instruments. Many of the guidelines included herein come directly from the field of optomechanical engineering, and are, therefore, neither newly developed guidelines, nor are they uniquely applicable to the design of high-precision deployment mechanisms. 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. However, it is expected that this paper, with revisions as experience may indicate to be desirable, might eventually form the basis for a set of uniform design requirements for high-precision deployment mechanisms on future NASA space-based science instruments.			
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