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Limitations of Flexures

By Kevin McCarthy, Chief Technology Officer

When one surveys the field of precision positioners intended for use in photonic automation, flexural guideways show up in a substantial number of product offerings. At first glance, they seem well suited to the task; they are simple to build, low in cost, and their limited travel seems a good fit to the application. Upon closer examination, however, the weak points of flexures multiply, and their intrinsic limitations stand out. At the outset, we should point out that we have been designing and shipping very high performance flexure based systems for over ten years, and have a substantial depth of experience in flexure design. In the proper application, they can be an effective way technology, but we see many cases in which either the wrong flexure design is chosen, or where a good design is used outside its fairly narrow region of good performance.

There are a few good things about flexures, which can be summarized as follows:

1. Their movements are smooth, continuous, and highly repeatable.
2. There is no wear; accordingly, performance is continuous over time.
3. They can be fabricated from a single, monolithic material.

Now for a look at their “dark side”:

1. Flexures are restricted to short travels. Attempts to extend this often backfire.
2. Flexures are frequently made from multiple parts (i.e., sheet metal bands). These designs suffer from low cross-axis and torsional stiffness, limited load capacity, and high sensitivity to dimensional tolerances.
3. Simple flexures suffer from surprising levels of parasitic motion in other axes.
4. Many flexure designs have quite low stiffness for out of plane motions.
5. Flexures, when properly designed, may have relatively high stiffness in the drive direction, which can lead to large motors and unwanted heating.
6. Load capacity can be low, and transient high loads may cause buckling and /or permanent deformation.

To get a better picture of the limitations of flexures, we can begin by recognizing three general classes of flexures: two-band, compound, and four-band. To start, let’s consider the even simpler single band flexure (Fig.1). If we push on the payload at the top of the band, it translates smoothly in the X axis, but suffers from a very large angular tilt (O_Y), as well as a significant movement in the Z axis (along the length of the band). In addition, the single band is very weak in yaw (O_Z , or rotation around the band length). The first solution is to graduate to a double band flexure (Fig.2).

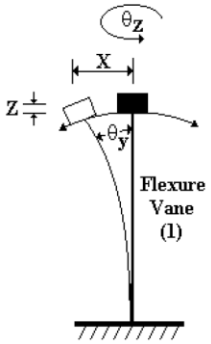

Figure. 1.

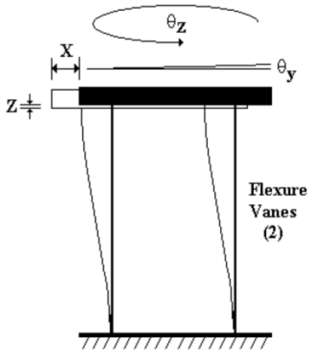

Figure. 2.

This is as far as some flexure stage manufacturers ever get, with two sheets of shim stock forming the flexure. Unfortunately, this design has some serious shortcomings, as we shall see. The rear band does prevent the carriage from tilting in the simple cantilever manner of Fig. 1, and forces the two bands to bend in an “S curve” manner. While the Z axis contraction during X axis motion is considerably reduced over that of the single band, it remains far from negligible, and grows non-linearly with travel. The magnitude of the Z contraction is simply 0.6 * Travel^2/Length. Note that, due to the square term in the Z contraction, a point on the flexure carriage actually traces out a parabolic path. At the extremes of travel of a +/- 2 millimeter flexural guideway of 50 mm. length, the Z axis error is a whopping 48 microns. It is important to note that this error is not a matter of workmanship – it is an intrinsic design feature, reducible only by lengthening the flexure, and/or decreasing its travel. In principal, the length contraction in each flexure band should be identical, and hence O_Y should be eliminated. As it happens, though, the tolerance sensitivities of this design to very minute variations in material thickness and length are surprisingly high, and tilt errors of 30 to 100 arc-seconds are typical. Unit to unit consistency is therefore also an issue. It is a rare optical configuration that would be insensitive to linear and angular errors of this magnitude. Two studies of the errors inherent to two band flexures are presented in detail in reference #1, below. While flexures of this type usually exhibit low stiffness in the drive axis, allowing travels in the multiple millimeter range, any comparable linear bearing would be thrown on the scrap pile if it suffered from linear and angular runouts of this magnitude. An additional weakness is low torsional (O_Z) stiffness and load capacity; when one attempts to build an X-Y-Z axis stack by mounting three of these flexural guideways, one atop the other, the resulting position of the tool tip is, at best, ambiguous.

A clever means of reducing the linear (Z axis) runout described above results if the two-band flexure is modified to produce a compound flexure (Fig. 3). In this case, the intermediate body is allowed to float without hindrance. While this part exhibits the typical (that is, large) Z axisrunouts relative to the stationary base as in the case of the two band flexure, the moving carriage exhibits an equal and opposite Z runout relative to the intermediate body. The result is a near-perfect cancellation of linear (Z axis) error. Despite this improvement, the compound flexure is considerably worse than the two-band with respect to torsional stiffness, has added complexity, and requires more space to implement. The same high sensitivity to component tolerances is present, but with twice the number of flexural bands, operated in series, tilt (O_Y) errors are typically doubled. Like the two-band flexure, it is prone to buckling or deformation under load, and should be implemented, if need be, with a monolithic material, instead of numerous bands of shim stock and bolts. Note that while the compound flexure has four bands, it is better thought of as a serial cascade of two two-band flexures, so as to differentiate it from the true four-band flexure described below.

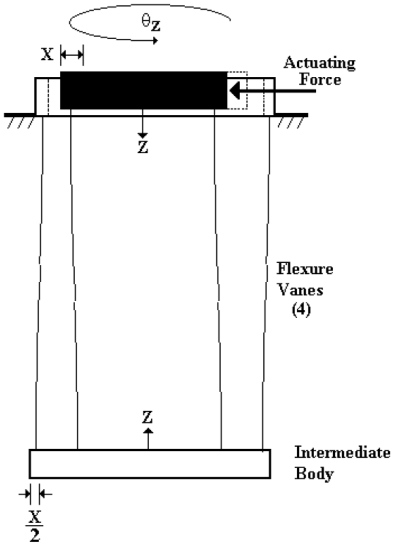

Figure. 3.

The preferred embodiment for successful flexures lies with the last of our three categories: the four-band flexure. This is the only type that we design and build, and we also insist on using EDM methods, so as to allow the entire flexure to be fabricated to high precision from a single, monolithic piece of material. The basic design is shown in Fig. 4, and a photo of a typical single axis flexure is shown in Fig. 5. The corner of an X-Y (two axis) version of this flexure design is shown in Fig. 6.

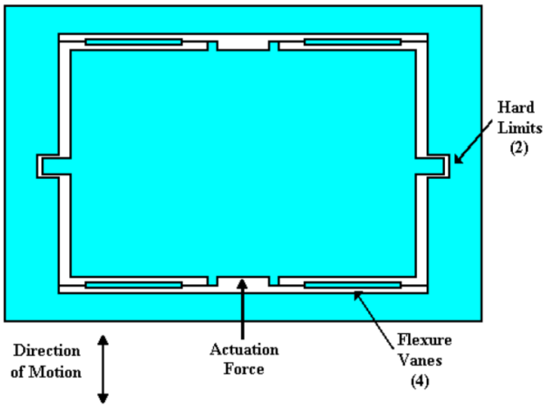

Figure. 4.

Figure. 5.

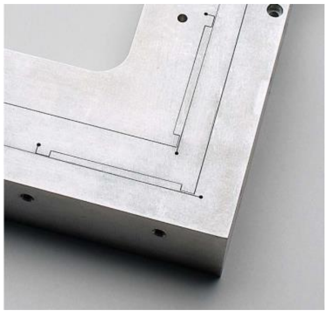

Figure. 6.

In the four-band design, opposing flexural bands constrain motion to a straight line, provide substantially higher torsional stiffness, and essentially eliminate buckling failure. Since the S-curve bending that the flexural vanes experience during motion occurs very near each end, and the vane is essentially straight between the two regions of curvature, our vanes are thickened in the middle. This provides added linear and torsional stiffness, as well as an increase in load capacity, without appreciably increasing the actuation force. Note, as well, that hard stops to prevent deformation due to high loads can be easily added to the EDM routing path. While the performance of this flexure type is nearly ideal, the design operates with each of the four bands in tension, which grows nonlinearly as the travel increases. As a result, conservative designs, even when optimized materials are chosen, will have limited travel capability. One parameter setting the travel limit for a flexure of specified dimensions is the material fatigue strength; another may be the resultant motor heating if a voice coil drive must hold position at the ends of travel. In general, four-band designs can provide exemplary performance, but the proper range of travel for the best designs is typically < 0.1% of the overall dimension – i.e., for a 150 mm square part, 50 to 100 microns of total travel is about it. Needless to say, this limitation, as well as the those of designs other than the four-band flexure, may not satisfy marketing or customer demands. Nevertheless, the physics is real, and since larger travels are legitimately required in photonic applications, the proper course in many cases is to employ air bearing guideways, which are frictionless, and can support single nanometer resolution and extreme straightness over travels of hundreds of millimeters.

[1] Jones R.V., 1951, "Parallel and rectilinear spring movements", J. Sci. Instrum., 28, 38-41. Also: Jones R.V., 1956, "Some parasitic deflections in parallel spring movements", J. Sci. Instrum., 33, 11-15.

Further reading: Smith S.T., "Flexures – Elements of elastic mechanisms", also Smith S.T., and Chetwynd D.G., "Foundations of ultraprecision mechanism design", Gordon and Breach, Amsterdam.

ABOUT THE AUTHOR...
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