Proc. of SPIE Vol. 2542, Optomechanical and Precision Instrument Design, ed. A E Hatheway (Sep 1995) Copyright SPIE

Alignment of Flexure Stages for Best Rectilinear Performance

Alson E. Hatheway

Alson E. Hatheway Inc.
595 East Colorado Boulevard, Suite 400
Pasadena, California 91101

1.0 ABSTRACT

Flat blade flexure stages have attracted attention for many years because of their potential for producing smooth, large-stroke and hysteresis-free motions, especially close approximations of rectilinear motion. Although smooth operation has been demonstrated many times over, close approximations of rectilinear motion have been rare and achieved at relatively great cost in time and facilities. This has been true because most investigators have relied on "composite flexures" in which each flexure element is assembled from as many as five separate pieces, all of which need to be aligned to each other as well as the rest of the instrument in which each flexure is only one element. "Monolithic flexures" have just recently become commercially available and make possible for the first time the precise assembly and alignment of blade flexure systems with a modest investment in time and resources. This research investigates the precision behavior of a simple flexure stage constructed with monolithic blade flexures and compares it with the behavior attainable from a similar stage using composite flexures. The natural trajectory of such a flexure stage is described and equations for the deviations from the ideal trajectory, parasitic motions, are developed. The results of the analysis are compared to the actual results from testing a commercially available simple monolithic flexure stage.

Key Words: Flexure, rectilinear, linear,

2.0 INTRODUCTION

A simple flexure stage is composed of two flat blade flexures, fore and aft, a table that moves under the influence of the flexures

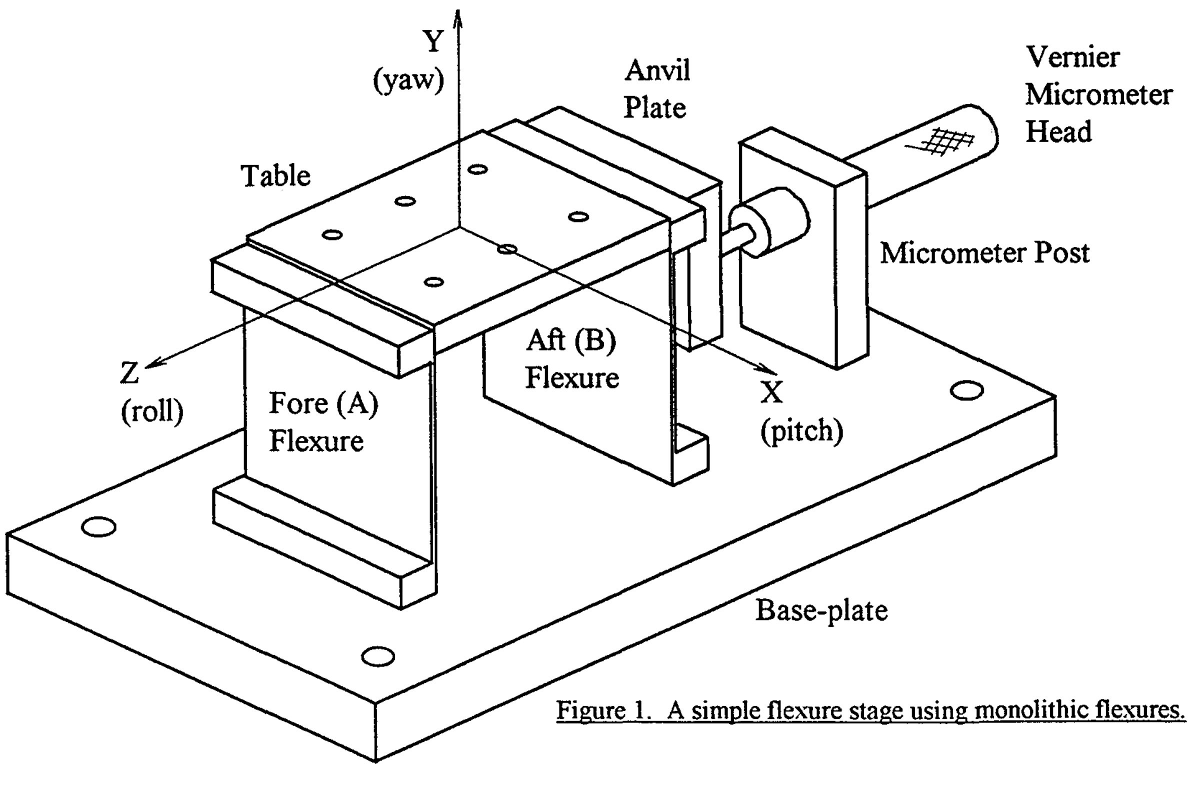


Figure 1. A simple flexure stage using monolithic flexures.

and a base-plate to which the fixed ends of the flexures are attached, Figure 1. The stage may be driven by any number of devices but probably the simplest and most common is a vernier micrometer head. The micrometer will need a post to mount it to the base-plate and some kind of anvil plate to receive the micrometer tip loads. Such simple flexure stages are highly repeatable, hysteresis free and generate very smooth motions. They are often used to simulate rectilinear motion over distances that are small with respect to the length of the flexure blades.

The relationships among the motions produced by simple flexure stages are not well understood by most users and potential users. This is partly due to the widely understood fact that a simple flexure stage cannot produce pure rectilinear (straight-line) motion, a fact that deters many from further investigations into the mechanics of the mechanism. Nonetheless, the simple flexure stage is a fundamental building block in more complicated systems that may produce straight-line motion, or any other motion the designer may desire. This fact makes an understanding of the behavior of the simple flexure stage a convenient start for understanding the behavior of elastic mechanisms in general.

Designers are further deterred from using flexures because of the large number of individual parts required to make a composite flexure assembly from available stock materials; a flat blade (shim stock), a base flange (bar stock), a base clamp (another stock bar), and a flange and clamp for the table end, five separate metal pieces for each flexure, Figure 2.a). Precision assembly of two such flexures into a simple flexure stage is reasonably difficult and repeatability of the assembly is problematic. Further, individual adjustments after assembly are nearly impossible without loosing the original precision.

Commercial availability of monolithic blade-type flexures, Figure 2.b), have greatly simplified the assembly process and made adjustment and alignment repeatability truly possible in flexure mechanisms. This research work analyzes the source of motion errors (parasitics) and compares the anticipated parasitic motions with an actual simple flexure stage assembled from commercially available parts.

3.0 THE KINETICS OF A SIMPLE FLEXURE STAGE.

Study of the kinetics of a well aligned and properly driven simple flexure stage indicate that over small ranges of fore-and-aft motion the table has a preferred and well defined natural trajectory.

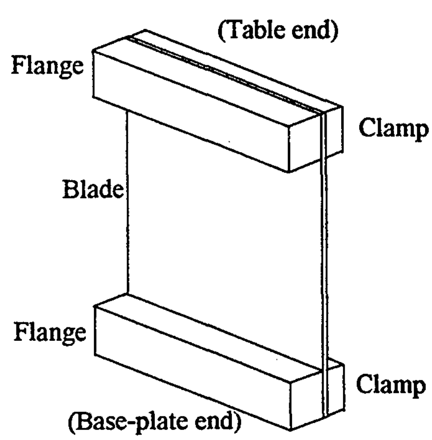
The flat blade flexure in a simple stage behaves structurally like a "double cantilever" beam, that is, a beam that is constrained at both ends but one end moves laterally with respect to the other end causing the beam to bend into an "S" shape while the slopes of the ends remain parallel (but laterally offset). The "double cantilever" beam behaves analogously to a simple cantilever beam of half the length and half the lateral displacement. The kinetics of the "double cantilever" may therefore be understood from the kinetics of the simple cantilever.

The displacement, y_{max}, and slope, θ_{max} of the end of a simple cantilever beam, assuming small deflections, are available in any fundamental textbook on structural mechanics,

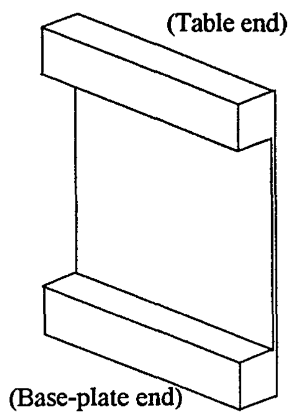
$$y_{max} = PL^3/3EI \text{ and} \qquad (1)$$
$$\theta_{max} = PL^2/2EI.$$

If we ratio deflection, y_{max}, by the angle, θ_{max}, we will have the radius of curvature, ρ, of the locus of points described by the free end of the beam,

$\rho = 2L/3,$



a) Composite flexure.



b) Monolithic flexure.

<u>Figure 2. Flat blade flexure types.</u>

i.e., over small displacements the free end will behave as if it is rotating about an instantaneous center that is located 2/3 of the way from the free end to the fixed end; the beam may be kinetically replaced, for small motions, by a rigid link of length 2L/3 and a torsion spring to simulate the stiffness.

Since a "double cantilever" beam behaves analogously to the simple cantilever we may assume that it also behaves, over small displacements, as if it were a rigid link of 2L/3 (L now being the full length of the double cantilever beam) and two torsion springs. This small displacement assumption is rigorously valid only at the position of zero deflection. It can be shown that for finite displacements the equations (1) need to be expanded by Mclaurin's method into infinite power series. Such series accurately describe the behavior of beams with large deflections. For our purposes it will not be necessary to calculate these series expressions nor evaluate their coefficients but rather to know that they exist and that the deflection of a double cantilever beam is determinate over its whole range of potential values.

Assume that we have two identical "double cantilever" beams to be used as flexures. Assume also that we can mount these flexures into a simple stage with exactly the same span between them at both the base-plate and the table. Further assume that when the flexures are installed their neutral axes are parallel to each other and the principal axes of their moments of inertia are also parallel. These assumptions describe the goals of a *"well aligned" simple flexure stage*. They also define the variables that need to be controlled to achieve the kinetics of the "well aligned" flexure stage, i. e.,

L = flexure length,
S = span between flexures,
α = angular misalignment in the vertical, XY, plane of the fore and aft neutral axes and
β = angular mis-orientation in the horizontal, ZX, plane of the principal axes of inertia.

The criterion of "properly driven" concerns how the drive mechanism interfaces with the flexure stage. First we need to define a point in space which is called the "kinetic center" of the flexure stage. Ideally it will be at the geometric center of the flexures; half span, half length and half breadth. In most real stages the kinetic center will be near the geometric center but may vary slightly from that location due to differences in the flexures and misalignments between them. In some stages the flexures are deliberately different and the location of the kinetic center may need to be determined by elastic structural analysis or experiment. *A "properly driven" stage is driven by a force resultant that acts on a line through the kinetic center, wherever it may be.*

A "well aligned" and "properly driven" simple flexure stage will move through a trajectory defined by the kinetics of the two flexures. For small displacements the trajectory will appear to be a circular arc of radius 2L/3. For larger displacements the trajectory will be seen to be a higher order polynomial associated with the elastic deflection of the flexures. In either case the trajectory will be a curve in a vertical plane; out-of-plane motion will not be observed. Furthermore, the loading on both flexures will be nearly identical and the motions of both ends of the table, fore and aft, will be the same; the table will not rotate with respect to the base, but remain parallel to it. This last property of a "well aligned" and "properly driven" flexure stage is quite important because angular motions of the table, representing parasitic motions, may be observed over large displacements using an autocollimator and the angular deviations may be used to assess related parasitic displacements that may be otherwise difficult to observe and measure.

4.0 MISALIGNMENTS IN A SIMPLE FLEXURE STAGE.

The influences of misalignments in flexure stages are dimensional so we will need to select proportions for a real stage in order to evaluate the influences of variations in the individual dimensions. We will study the Series 5001 simple flexure stage (Figure 1). This stage is constructed from two Series 6000, 2.00 inch size, C flange, monolithic instrument flexures, a table and a base-plate. The drive mechanism is a Model 2001-1 vernier micrometer head which drives a stainless steel anvil plate. The micrometer is mounted to the base-plate with a simple post. A Model 7007 micrometer tip cushion is between the micrometer tip and the anvil plate. The use of the Series 6000 monolithic flexures facilitates the assembly and alignment of the simple flexure stage (the assembly time is typically five minutes).

The characteristic dimensions of the Series 5001 flexure stage are

L = flexure length = 2.00 inches
S = span between flexures = 3.00 inches.

The micrometer driver nominally drives through the kinetic center of the flexures but may be adjusted in a horizontal plane through small angles and displacements. We shall now determine the influence of four types of misalignments on the trajectory of the table. We shall take the misalignments one at a time assuming that in all other respects the flexure stage is "well aligned" and "properly driven." The misalignment variables are,

1) unequal flexure lengths,
2) unequal span lengths,
3) non-parallel neutral axes and
4) non-parallel principal axes of inertia.

We shall then determine the influences of the micrometer's position and orientation on the table's trajectory.

4.1 The influence of unequal flexure lengths

If the flexure lengths are unequal the forward stroke of the table will be accompanied by a slight rotation about the X (pitch) axis, Rx. If flexure A is shorter than flexure B (Figure 1) and the difference is defined as ΔL we may calculate the pitch motion accompanying the stroke.

Assuming small motions so the kinetics of the flexures may be represented by circular arcs of radius 2L/3 and further assuming that the stroke is given by Tz we may say that the instantaneous center of flexure A will move through an angle, θ_A, equal to,

$\theta_A = 3Tz/2(L_A - \Delta L)$

and this will result in the end of the table near A moving vertically, y_A, equal to,

$y_A = -3Tz^2/4(L_A - \Delta L)$.

Similarly for motion at the end of the table near B,

$\theta_B = 3Tz/2L_B$ and
$y_B = -3Tz^2/4L_B$.

The pitch rotation of a table with a span of S will be,

$$Rx = \Delta y/S = (y_A - y_B)/S = -3\Delta LTz^2/4L^2S \,, \qquad (2)$$

assuming that ΔL is small with respect to L_A and L_B so that

$L \sim (L_A + L_B)/2$.

Inspection of equation (2) discloses that the pitch rotation, Rx, is proportional to the square of the stroke, Tz, of the table.

4.2 The influence of unequal span lengths.

If the span between the flexures at the base-plate is larger, by an amount ΔS, than the span at the table, S, the forward stroke of the table will again be accompanied by a slight rotation of the table about the X (pitch) axis, Rx. Again, assuming small motions so the kinetics of the flexures may be represented by circular arcs of radius 2L/3, the angle between the planes through the instantaneous centers in the two flexures is,

$\gamma = 3\Delta S/2L$

and the associated pitch rotation will be,

$$Rx = -3\Delta STz/2SL \, . \qquad (3)$$

The pitch rotation, Rx, from equation (3) is directly proportional to the stroke of the table, Tz.

4.3 The influence of non-parallel neutral axes.

If the fore and aft flexures are mounted to the base-plate so the neutral axes of their webs are not parallel in a vertical and transverse, XY, plane it will cause the opposite ends of the table to move in slightly different directions in the horizontal, ZX, plane. This motion will manifest itself as a yaw rotation, Ry, when the table is translated. If the angle in the XY plane, between the neutral axes, is α and assuming the angle is equally devided between the fore and aft flexures the forward flexure will translate laterally in the horizontal plane an amount

$$Tx_A = -3Tz^2\alpha/8L$$

and the aft end will translate laterally,

$$Tx_B = 3Tz^2\alpha/8L \, .$$

The net yaw rotation of the table, Ry, is

$$Ry = (Tx_A - Tx_B)/S = -3\alpha Tz^2/4SL \, . \qquad (4)$$

The yaw rotation, Ry, caused by non-parallel neutral axes in the flexures is proportional to the square of the stroke.

4.4 The influence of non-parallel principal axes of inertia.

If the principal axes of inertia of the forward and aft flexures are not parallel they will tend to move the opposite ends of the table laterally in slightly different directions in the horizontal, ZX, plane. This will generate a yaw rotation, Ry, when the table is translated. If the effective angle between the principal axes in the two flexures is β the table's net yaw rotation will be,

$$Ry = \beta Tz/S \, . \qquad (5)$$

The yaw rotation, Ry, caused by non-parallel principal axes is directly proportional to the stroke of the table.

4.5 The influence of driver misalignments.

The influence of driver misalignments cannot be determined simply from the kinetics of the flexure stage elements. An elastic deflection analysis must be made to determine the equilibrium position of the table for eccentrically applied loads from the drive mechanism. This may be done either by longhand calculations or with a finite element model of the flexure stage. In either case the analysis must accommodate both bending and shear deformations in the flexures, especially in the lateral, X, direction (which is the stiff direction of the flexures).

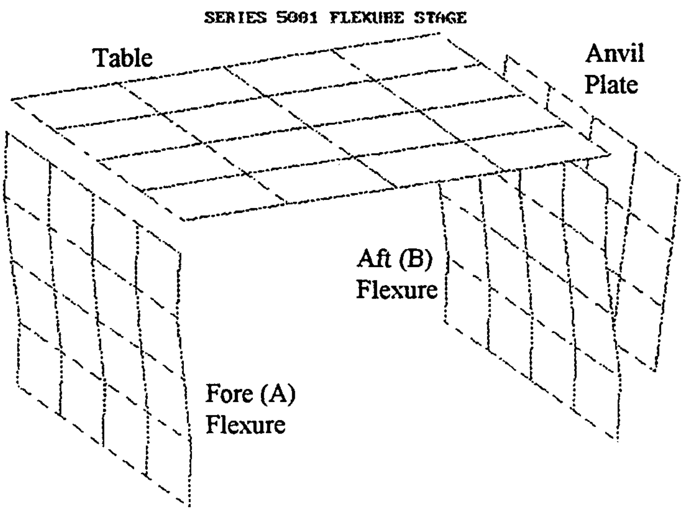


Figure 3. Flexure stage finite element model.

A simple finite element model of the Series 5001 simple flexure stage has been developed to determine the influences of the drive micrometer. A plot of the model is shown in Figure 3 with the flexures deflected. The model simulates the stiffness of the flexure blades, the table and the anvil plate. All other features of the stage are assumed to be rigid and are not shown in the figure.

The micrometer drive in the Series 5001 is designed to impart a force to the anvil plate that is at all times normal to the surface of the plate. At one point on the anvil plate (and only one point) this force will act through the kinetic center of the stage. Under these conditions the force acts as a pure force at the kinetic center. This point is called the anvil's kinetic point. At all other points on the anvil plate the force may be effectively transferred to the kinetic center only if a moment is also applied at the kinetic center; the moment being equal to the product of the force and the offset at the anvil plate from the kinetic point. This transfer is consistent with the principles of static equilibrium.

If the axis of the micrometer is also normal to the anvil plate surface the micrometer will act through a fixed point on the plate. The force will be proportional to the displacement of the flexures by the micrometer and the moment will be proportional to the force and the offset (a constant if the micrometer is normal to the anvil surface). This case *produces a moment that varies linearly with the displacement of the table.*

On the other hand if the micrometer axis is inclined to the surface of the anvil plate the offset from the kinetic center to the point of action of the micrometer will depend upon the amount of deflection of the flexures. Now the offset may vary linearly with the table displacement and the moment varies as the product of the force and the offset *which makes the moment proportional to the square of the table displacement.*

So, the installation geometry of the micrometer offers both a linear term and a second degree term for the moments as functions of table displacement. We may write algebraic expressions to quantitatively describe these effects. In the vertical, YZ, plane,

$$Rx = Tz(\Delta y_o + \psi Tz)(dRx/dTz) \qquad (6)$$

where

Rx is the resulting pitch angle of the table,
Tz is the table stroke in the Z direction,
Δy_o is the vertical offset at zero stroke,
ψ is the vertical component of the micrometers angle of incidence at the anvil plate and
dRx/dTz is the elastic coupling coefficient determined by analysis for a fixed offset.

Similarly in the horizontal, ZX, plane

$$Ry = Tz(\Delta x_o + \phi Tz)(dRy/dTz) \qquad (7)$$

where the new terms are,

Ry is the resulting yaw angle of the table,
Δx_o is the horizontal offset at zero stroke,
ϕ is the horizontal component of the micrometers angle of incidence at the anvil plate and
dRy/dTz is the elastic coupling coefficients determined by analysis for a fixed offset.

The finite element model of the flexure stage was run with offsets Δx_o and Δy_o equal to .010 inches and a stroke, Tz, of .200 inches. From the elastic response of the center of the table the elastic coupling coefficients dRy/dTz and dRx/dTz were determined. The resulting equations defining the micrometer alignment's influence on the pitch and roll of the table become,

$$Rx = Tz(\Delta y_o + \psi Tz)2.57x10^{-6} \qquad (8)$$
$$Ry = Tz(\Delta x_o + \phi Tz)\text{-}3.36x10^{-4}\ . \qquad (9)$$

4.6 A summary of the influences of misalignments on the trajectory.

From the above discussion it may be observed that both the pitch, Rx, and the yaw, Ry, parasitic motions of the table are influenced by both the flexure installation and the driver installation. Furthermore, both the flexure installation and the driver installation contain both linear and second degree terms. We may compare these influences by combining them into a table as shown in Table I:

Table I
General Flexure Stage Parasitic Influences

Parasitic Motion:	Flexure Influences	Driver Influences
Rx (pitch)	$-3\Delta STz/2SL - 3\Delta LTz^2/4L^2S$	$Tz(\Delta y_o + \psi Tz)(dRx/dTz)$
Ry (yaw)	$\beta Tz/S - 3\alpha Tz^2/4SL$	$Tz(\Delta x_o + \phi Tz)(dRy/dTz)$

The above array of influences may be used with a simple flexure stage of any span, S, and with flexures of any length, L.

5.0 ASSEMBLY TOLERANCES

One use of the expressions in Table I is to evaluate the influence of assembly tolerances on the trajectory by estimating the expected parasitic rotations in pitch and yaw. If we substitute into the expressions in Table I the dimensions for the Series 5001 flexure stage we will have expressions governing the parasitic motions of an assembled stage. Table II presents these expressions:

Table II
Series 5001 Flexure Stage Parasitic Influences

Parasitic Motion:	Flexure Influences	Driver Influences
Rx (pitch)	$-.250\Delta STz - .0625\Delta LTz^2$	$2.56\mathrm{x}10^{-6}\Delta y_o Tz + 2.56\mathrm{x}10^{-6}\psi Tz^2$
Ry (yaw)	$.333\beta Tz - .125\alpha Tz^2$	$-3.36\mathrm{x}10^{-4}\Delta x_o Tz - 3.36\mathrm{x}10^{-4}\phi Tz^2$

The Series 5001 simple flexure stage is designed to have a stroke of 5 millimeters, about .200 inches. Now we can compare the magnitudes of the parasitic motions for different assumed assembly tolerances. Let us compare the expected accuracy of a stage made with composite flexures (five separate pieces per flexure) with a stage assembled from monolithic flexures. Furthermore, for the monolithic flexures let is evaluate the effects of worst-case combinations of assembly tolerances as well as controlled assembly trying to minimize the influences of the assembly variables. Table III lists the assumed tolerances and assembly variables for each of these three configurations. At the bottom of Table III are the calculated parasitic motions using the expressions of Table II and assuming worst case combinations of the tolerances in the top of Table III (absolute values of the terms in Table II are summed).

The tolerances in Table III for the composite flexures (column two) assume carefully controlled manufacture of the individual piece-parts and some gaging during the assembly to control the dimensions. The results are parasitic motions of three hundred to four hundred microradians. These values are typical results from mechanisms constructed from composite flexures and a large part of the reason that they are rarely used in precision work. Composite flexures that are not carefully controlled in their manufacture and assembly may perform worse by factors of three or more. It is arguable that composite flexures could approach the tolerances shown in the third column for monolithic flexures (uncontrolled) but at considerable cost in time and tooling; and only reducing the parasitics to the range of one hundred fifty to two hundred microradians.

On the other hand, the tolerances shown in column three of Table III for monolithic flexures represent the worst case limits one may expect from their manufacture and assembly into a stage without special considerations or controls. The resulting parasitic motions are large and should be considered to be an upper limit for untrained or disinterested assembly of a flexure stage. This

Table III
Assembly variables and parasitic motions in flexures

Assembly Variables:	Composite Flexure (controlled)	Monolithic Flexure (uncontrolled)	Monolithic Flexure (controlled)
α (radians)	.010	.002	.0005
β (radians)	.005	.002	.0001
ΔL (inches)	.025	.010	.001
ΔS (inches)	.005	.003	.0001
Δx_o (inches)	.010	.010	.010
Δy_o (inches)	.010	.010	.010
ϕ (radians)	.010	.010	.010
ψ (radians)	.010	.010	.010
Parasitic Motion:			
Rx (μradians)	313.	175.	7.5
Ry (μradians)	383.	144.	10.0

level of performance can rarely be used in precision instruments and hardly recommends the use of flexures at all. These data are given here because they are the results that experimenters occasionally see because *the benefits of greater care in selecting and assembling the parts of a flexure stage are not widely understood.*

The tolerances shown in column four of Table III indicate the redced level fo the alignment variables that is readily attainable in the manufacture and assembly of monolithic flexure stages with some assembly controls. This is a level of precision that is unattainable with composite flexures without extraordinary time and expense. The advantage of monolithic flexures is that these accuracies are readily achievable in practice and the magnitudes of the parasitic motions may be reduced by more than one order of magnitude to seven and one-half to ten microradians. This is a level of precision that is widely usable to experimentalists and instrument designers and is the subject of this discussion.

6.0 MONOLITHIC FLEXURE TESTS

To test these estimates of parasitic motions a Series 5001 simple monolithic flexure stage was assembled from standard parts available from stock using only a simple allen wrench to tighten the screws. This assembly was mounted on an optical bench and a flat mirror mounted on the table. A K&E autocollimator was aligned to the flat mirror and the stage was operated over its full stroke (.200 inches). Pitch and yaw were both measured and the results are shown in Table IV, second column:

Table IV
Parasitic motions in uncontrolled assembly of monolithic flexure stages

Parasitic Motion:	Initial Assembly	Inspection and Re-assembly
Rx (pitch)	1. arcsecond (5. microradians)	18. arcseconds (87. microradians)
Ry (yaw)	8. arcseconds (39. microradians)	4. arcseconds (19. microradians)

The stage was disassembled, inspected, reassembled and tested again. The results are shown in the third column of the table. The purpose of the inspection and re-assembly was to attempt to reduce the magnitude of the yaw motion by dressing the base-plate surfaces and the flexure flanges on a granite surface plate and thereby remove small irregularities in the mating surfaces. Both sets of data were repeatable within the accuracy of the autocollimator, about 1. arcsecond.

As can be observed in the data there was some success in reducing the yaw motion but the assembly process did not reproduce the original small pitch motion. The results of both of these tests are typical of the performance to be expected from "uncontrolled" assemblies (column three in Table III) in which both favorable and unfavorable assembly variables combine to render the results (note that column three of Table III assumes that all the assembly variables have occurred at their worst-case limit).

Achieving the performance associated with "controlled" assembly indicated in column four of Table III requires knowledge and care on the part of the individual assembling the stage. In addition it requires some special controls:

Cleanliness: The mating surfaces must be free of dust, lint and other contaminants. The variation in pitch shown in Table IV requires only one milliradian of change in alignment of the neutral axes, β (see the expressions in Table II). This could easily have been a dust particle or fiber in the joint between the table and the flexures.

Accurate torque measurement on screws: The elastic distortions that occur in structures around screws must be repeatable in order to achieve the highest precision. Controlled assembly often requires disassembly, inspection, modification and re-assembly. The uncertainty in the elastic distortions around the screws must be removed by using accurate torque wrenches.

Accurate angular measurement of pitch and yaw rotations: As we will see in the next section we obtain valuable alignment clues from the measured parasitic motions. With these data and the relationships in Table II we can determine the changes that need to be made to minimize the parasitic motions.

7.0 PRECISION ALIGNMENT

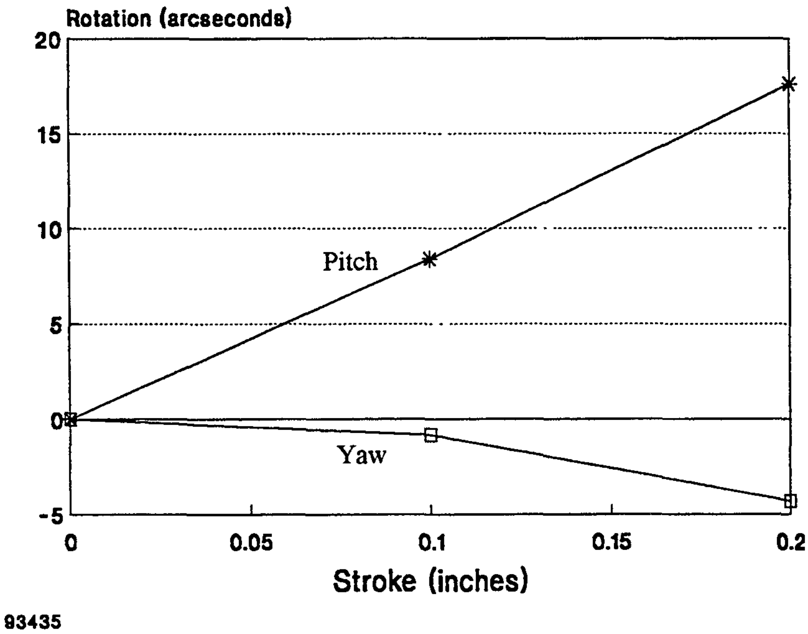
Up to this point we have used the expressions in Table II to predict the influence of the manufacturing and assembly tolerances on the magnitudes of unwanted parasitic rotations, yaw and pitch, in a finished flexure stage assembly. However, the same relationships may be used to determine the desired changes in the assembly variables to minimize these parasitic motions. To illustrate this let us look at the plotted data from the "Inspection and Re-assembly" described in the discussion of Table IV. These data are plotted in Figure 4. The variation of pitch with stroke is very nearly linear. Referring to the flexure influences in Table II the pitch parasitic motion appears to be dominated by ΔS, unequal span length between the table and the base-plate (being a linear expression in Tz). From the flexure influences in Table II it is possible to calculate the magnitude of the inequality, ΔS,

$\Delta S = -Rx/.250Tz$
Rx = 87. microradians (from Table IV)
Tz = .200

so that

$\Delta S = -87.x10^{-6}/.250x.200 = -.00174$ inches.

The span at the table is .00174 inches greater than it should be to eliminate the linear component of pitch. This requires us to either reduce the length of the table (its span) or to increase the span at the base-plate by this amount. Shimming and lapping are both acceptable procedures for making these changes. However, *before modifying any metal parts it would be wise to inspect all the mating surfaces for traces of dust or fibers that may be the source of this misalignment (the original assembly was better by a factor of almost twenty).* Also, check the repeatability of these data after reassembling the



Figuare 4. Observed parasitic motions as functions of stroke

stage with no changes. *If the data are not repeatable you do not have adequate controls on the assembly process.*

Inspection of the yaw data suggests that they are dominated by a second degree function of stroke. From the flexure influences in Table II it may be seen that this must be associated with misalignment of the neutral axes, α. As with the pitch parasitic it is possible to use the influences in Table II to calculate the magnitude of the misalignment.

$\alpha = -Ry/.125Tz^2$
Ry = 19 microradians (from Table IV)
Tz = .200

so that

$\alpha = -19.x10^{-6}/.125x.200^2 = -.0038$ radians.

The screws that mount the flexures to the base-plate are 1.00 inches apart so a .0038 inch shim at one screw location will be sufficient to remove the second degree term observed in Figure III. Removing material by lapping or machining would be a permanent and reliable solution but *before modifying any metal parts check the repeatability of the assembly process.*

Following the above procedure and being careful with cleanliness and accurate with the torque wrench it is possible to reduce the pitch and yaw parasitic motions of the flexure stage to one or two arc seconds, under ten microradians, which is about the limit of resolution for a good autocollimator and we will have realized the performance shown in the fourth column of Table III for monolithic flexures under "controlled" assembly. But we are not yet at the limit for reducing the parasitic motions, we have only outrun our ability to measure the rotations with an autocollimator.

If we have an instrument (perhaps a Fizeau interferometer) for measuring the pitch and yaw angles with greater precision we can further reduce them using the influences of the driver installation. The influences are shown in the third column of Table II and contain both linear and second degree terms for correcting the linear and second degree components of the parasitic motions. From the magnitudes of the coefficients in these expressions we can see that they are weak compared to the flexure influences. We can estimate the driver offset necessary to correct a two arcsecond yaw motion. If the motion is purely a linear function,

$\Delta x_o = -10x10^{-6}/3.36x10^{-4}x.200 = -.149$ inches.

If the motion is purely a second degree function,

$\phi = -10x10^{-6}/3.36x10^{-4}x.200^2 = -.744$ radians (42.6 degrees).

By adjusting the offset and the angle of incidence together it is possible to correct any combination of linear and second degree functions observed in the observed parasitic motion.

From Table II it can be seen that the driver influences on pitch motions are about two orders of magnitude smaller than the comparable terms for yaw motions. These expressions are dominated by the stiffness of the flexures in the vertical direction (compression and tension in the blades) although they are influenced somewhat by the span. For comparable driver alignment parameters, Δy_o equal to .149 inches and $\psi = 42.6$ degrees the driver could only correct for pitch motion amplitudes of .0076 microradians (for a linear function) and .0076 microradians (for a second degree function). It is unlikely that the span inequalities and length inequalities of this flexure stage could be adjusted finely enough to get some overlap of the driver influences and the flexure influences on the pitch motion. If pitch parasitic motion must be reduced below about one arcsecond it may be necessary to re-proportion the flexure stage to achieve overlap between the flexure influences and the drive influences.

8.0 CONCLUSION

Monolithic flat blade flexures are distinctively more accurate and predictable than composite flexures. Monolithic flat blade flexures are capable of significantly reducing (by factors between three and thirty) the parasitic motions in simple flexure stage. This is partly due to the five-fold reduction in the number of separate piece-parts that need to be controlled in both dimension and alignment in order to control their kinetics (compared to traditional composite flexures).

The kinetic behavior of monolithic flexures may be predicted by conventional simple beam theory. In simple flexure stages the theory predicts both linear and second order effects in the deviations, parasitics, from the stage's natural trajectory. The linear and second order effects are due to both the alignment of the flexure blades themselves and the alignment of the driver mechanism with respect to the kinetic center of the blades.

The equations for the magnitudes of the parasitic motions have been developed for a simple flexure stage. These equations allow the prediction of the parasitic rotations from assumed misalignments. Estimates of the parasitic rotations for both composite and monolithic flexure stages have been made based upon assumed manufacture and assembly tolerances. A commercially available simple monolithic flexure stage has been assembled and tested and found to agree reasonably well with the estimated values.

Guidance for further reducing the parasitic motions has been given. The pitch and yaw motions may be reduced to about two arcseconds by adjusting the alignment of the fore and aft flexures. Reducing the parasitics beyond this point may require adjustments of the position and orientation of the drive mechanism (a vernier micrometer in the design evaluated here). In this later case it may be desirable to design the flexure stage so the flexure influences and the driver influences have regions of overlap in order to provide a continuous range of alignment possibilities.