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ABSTRACT

A Fourier Transform X-ray Spectrometer has been designed and built for use at the Advanced light source at Lawrence Berkeley National Laboratory. The design requires a total rectilinear motion of 15 mm with a maximum pitch error of the stage below ±0.4 μradians, to achieve this we chose to build the entire machine as a single monolithic flexure. A hydraulic driver with sliding O-ring seals was developed with the intention to provide motion with a stick-slip position error of less than 0.8 nm at a uniform velocity of 20 μm/sec. The machine is comprised of two pairs of nested linear motion flexures, all of which are articulated by specially designed flexural hinges. The behavior of the machine has been measured and is well explained by means of a theory published earlier by Hathaway. Certain manufacturing errors were successfully corrected by an extra weak-link feature in the monolith frame. The engineering details of all the subsystems of the linear motion machine are described and the measured performance reported.

Keywords: Flexure, Rectilinear, linear, X-ray, Interferometer

1 INTRODUCTION

Fourier-transform spectroscopy has long been practiced in the infrared part of the electromagnetic spectrum and more recently in the visible and near UV[1]. Usually an unequal path interferometer derived from the Michelson geometry is used and the spectral resolution that can be realized is equal to the number of waves of path difference attainable between the two interfering beams. This path difference is determined by the design of the mechanical system that carries the moving mirror producing the path difference.

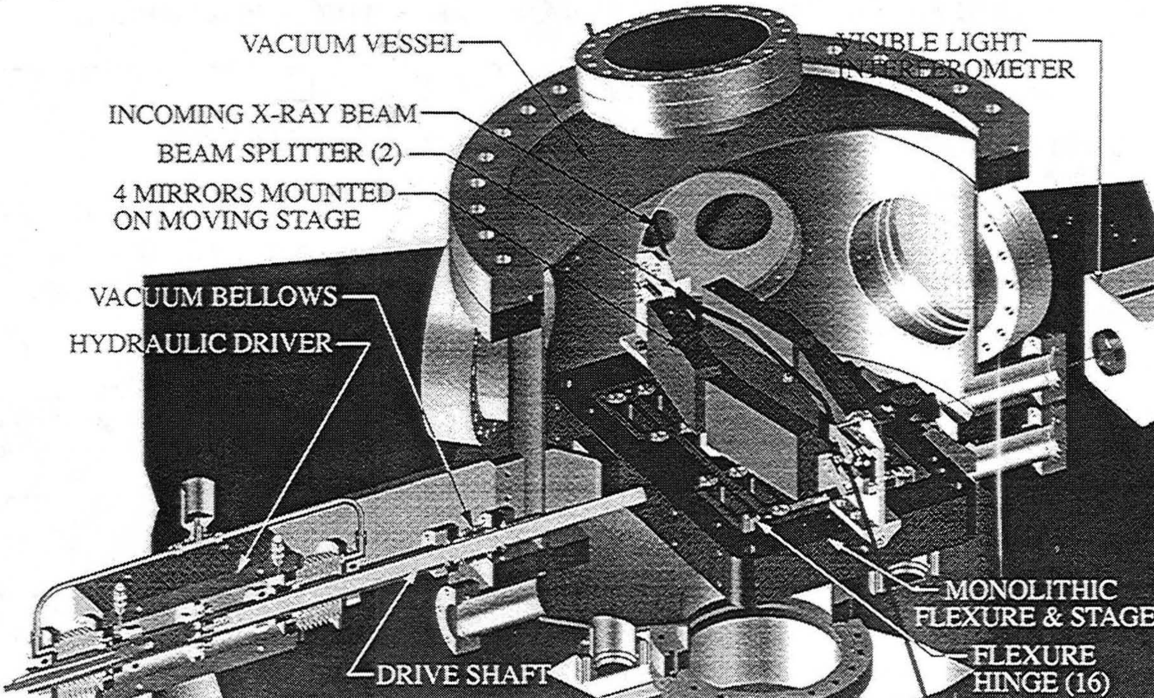

FIGURE 1 INTERFEROMETER ASSEMBLY

The traditional reasons why Fourier-transform spectroscopy is considered advantageous in the infrared (concerning throughput and signal-to-noise ratio) are inapplicable to the experiments that we are interested in, which use soft x-ray beams derived from synchrotrons. However, the diffraction-grating spectrometers which are normally used in soft-x-ray synchrotron-radiation beam lines have a resolving power of about 10^4 (6×10^4 in the best case[2]) which is still insufficient for the experiments that we are interested in. It thus becomes interesting to consider whether one could get better resolution by switching to interference spectroscopy. This possibility appears promising because the path difference needed to get a given resolving power diminishes with wavelength. For example, we are interested in the absorption spectrum of helium in the region of two-electron excitations below the double-ionization threshold which involves measurements near 200 Å. At this wavelength, a 2-cm path difference gives an expected resolving power of one million which would certainly be a worthwhile advance.

Accordingly, the authors have embarked on a program[3] to build a Fourier-transform spectrometer to operate at beam line 9.3.2 at the Advanced Light Source in Berkeley. The interferometer has essentially a Mach-Zehnder geometry (Fig. 1) which has been deformed to a rhomboid shape get the grazing-incidence-reflection angles needed for our experiment. The four mirrors in Fig. 2 are mounted on a stage and an overall path difference of 2 cm between the beams is achieved by translating the stage by ±7.5 mm. The experiment consists of measuring the intensity of the combined beams exiting the instrument as a function of the path difference. The linear-motion machine that accomplishes the translation is somewhat unusual and is subject to some quite challenging mechanical tolerances. The design considerations and measured performance of this machine are described in the sections that follow. The starting point for the design is the set of mechanical tolerances derived from the optical requirements which we describe in the next section.

2 MECHANICAL REQUIREMENTS AND TOLERANCES

The requirements on the motion of the stage that carries the mirrors are listed below (Table 1). with some indication of the logic involved. Further details of the tolerance calculations are available in an Advanced Light Source report[4]. The tolerance for unintended rotations of the stage can be assessed in relation to other machines used in short-wavelength interferometry where rotations need to be minimized: 0.001 μradian over 0.1 mm for hard x-ray interferometry[5], 0.3 μradian over 15 mm (our requirement) and 3 μradian over 400 mm for UV interferometry[1]. One can see that the tolerable rotation is roughly proportional to the amount of travel so, although our requirement is qualitatively new, it is not out of line with what has been achieved in the past.

TABLE 1: MECHANICAL TOLERANCES FOR THE TABLE MOTION

Quantity	Tolerance	Achieved	Explanation
Stage tilts to maintain wavefront tilts below ±1.5 μradian: - pitch	 0.5	 0.38	The other tilts are not critical
Tilt of one interfering wavefront with respect to the other (μradian): - due to mirror slopes - due to beam-splitter slopes - due to stage tilt - quadratic sum of all four tilts	 ±0.8 ±1.0 ±0.8 ±1.5	 1.0 0.6 0.57 1.3	The total tilt budget of 1.5 μradian gives 10% loss of fringe contrast.
Stage-motion indexing error (nm)	0.8	4 Å (noise limit)	To have no larger effect on fringe contrast than the shot noise
Driver stick-slip: position error over 1 sec relative to that expected from a uniform velocity of 20 μm/sec (nm)	0.8	4 Å (over 1 sec.)	

3 GENERAL DESIGN STRATEGY

We have chosen to build a classical rectilinear-motion machine based on flexural principles (Figs. 2 and 3). Machines of this general type have often been used before for scientific measurements[6] and interferometry[7] and have been analyzed by Smith and Chetwynd[8], Hoffrogge[9] and Hatheway[10]. Following Hoffrogge we have elected to build the entire machine as a single monolithic flexure. The machine will operate in vacuum so we assume that a high degree of thermal isolation will be easily achievable and consequently we have not limited our materials choices to low-expansion materials (see section 4).

We have chosen to use a hydraulic driver and the design of such a unit requires compromises between good stick-slip properties and good guidance. We report below our successful research-and-development efforts in this area (see section 6). The coupling of the driver to the moving stage is accomplished by a shaft linking the center point of the hydraulic piston to the center point of the stage. Due to the high symmetry of both the piston and the stage, these two points are at or close to the center of force (or resistance in the case of the stage), the center of friction and the center of inertia (gravity) of the two structures. Such a driving scheme allows one to respect the principle that the line of the drive should pass through all three of these centers of the driven object. One purpose of the principle is to ensure that errors in the position or angle of the line of drive do not translate into tilt errors that increase with drive distance. Similarly random departures from constant velocity (stick-slip) are prevented from producing random time-dependent tilts. Since the symmetries are not exact, the three centers do not exactly coincide and we made the position and angle of the drive adjustable to allow us to search for the optimum driving geometry.

Although we make efforts to drive the stage with an exactly constant velocity, we do not rely on this in measuring the path difference. We do that by means of a commercial laser interferometer that measures the motion of the moving stage with respect to the outer frame of the flexure (Fig. 4). To avoid Abbé errors, the line of the laser measurement beam is in the symmetry plane of the flexure and in fact coincides with the nominal line of drive.

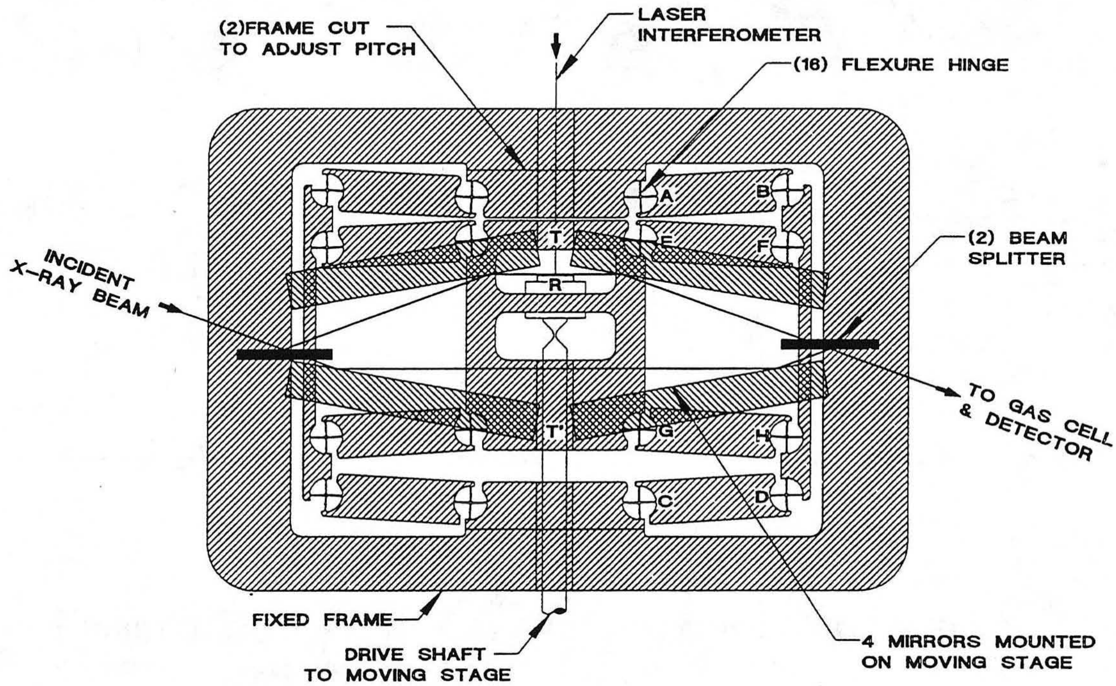

FIGURE 2 FLEXURE PLAN VIEW (SHOWN DEFORMED)

4 FLEXURE DESIGN

A travel of ±7.5 mm is rather large to be achieved with a flexural system and this has a profound effect on the design. At issue is the angle that will be turned by the sixteen flexural hinges that are used at each end of the eight moving levers of the flexure (Fig. 2). An obvious possibility would be to build these hinges as "notch" hinges (Fig. 3) as was done by Hoffrogge et al. However, for levers of reasonable length (ours are 75 mm long which leads to a flexure about the size of a notebook computer) the angle turned in moving ±7.5 mm is about three degrees. Now to design a notch hinge for this angle, without exceeding the stress limits of realistic materials and without making it too thin for the loading, we end up with a very large radius so that the flexing part of the hinge is almost a flat strip. However, although a strip hinge has sufficient range and strength, it has a bad center shift and unpredictable non-circular bending may occur depending on the loading. The traditional response to this is to employ overlapping pairs of strip hinges as used in the commercial flexural hinges produced by Lucas (Fig. 3). This approach provides an improved resistance to non-circular bending compared to a single strip although at some cost in compactness. A further disadvantage of overlapping-pair strip hinges is that they cannot be incorporated into a monolithic design implemented by electric-discharge-machining (EDM) techniques. Such a design is desirable because it allows us to machine the entire flexure in one setup, hopefully eliminating the need to individually align the sixteen hinges involved in the machine. All these considerations have lead us to the "monolithic crossed-strip hinge" shown in Fig 3. This geometry provides improved center shift and compactness compared to overlapping-pair strip hinges It also has the advantage of being manufacturable by EDM techniques and has greatly increased resistance to buckling of the flexing members.

The properties of the monolithic crossed-strip hinge are thus suitable for use in a long-travel mechanism such as the one shown in Fig. 2. Their principal disadvantage, according to conventional reasoning, is their higher stiffness compared to single or overlapping-pair strip hinges of the same dimensions. However in producing an ultraprecise mechanism, stiffness is generally an advantage since it helps to raise the resonant frequencies above the range where environmental disturbances have their highest amplitudes. An extensive analysis of the monolithic-crossed-strip geometry was done by the LBNL group[11] using beam theory, energy methods and finite-element analysis (FEA). Some of the results are shown in Table 2 which compares the three different flexure types of interest to us. The results of the FEA calculation of the von Mises stress[11] were consistently within 3% of the beam theory values. FEA optimization also showed that in our case the minimum stress was achieved when the radii at the hub were approximately equal to the thickness of the flexing strips.

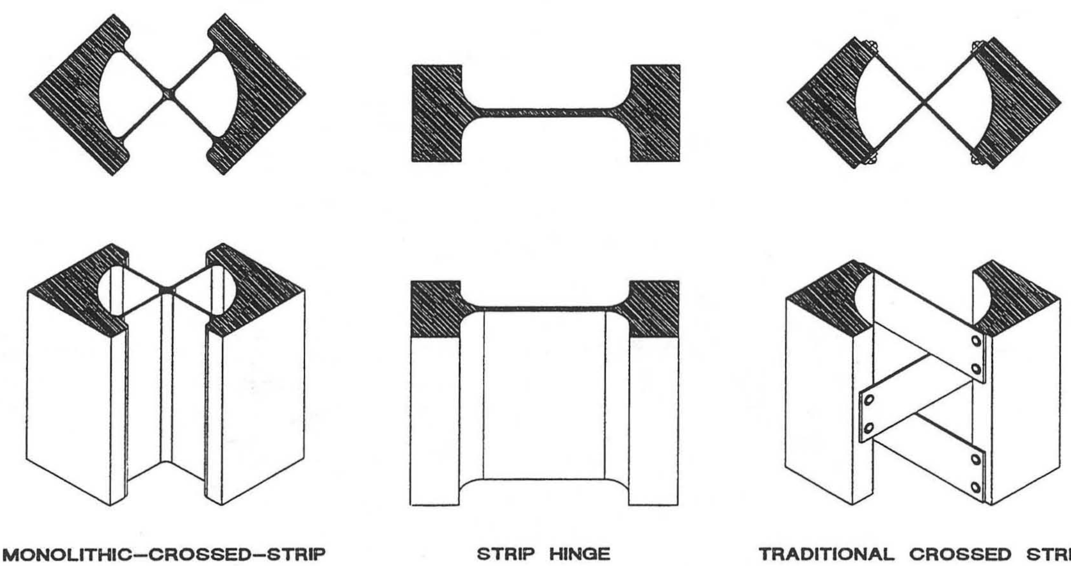

FIGURE 3 FLEXURE TYPES

TABLE 2
SUMMARY OF PROPERTIES OF STRIP-TYPE FLEXURAL HINGES[11]

	Cartwheel flexure of diameter 2ρ	Single strip hinge of length 2ρ**	Crossed-strip hinge of diameter 2ρ**
Torque per unit angle (in.lbs/rad)	$\frac{4EI}{\rho}$	$\frac{EI}{2\rho}$	$\frac{EI}{\rho}$
Center shift (in) at angle θ	$-\frac{\sqrt{2}\rho\theta^2}{30}$	$\frac{\rho\theta^2}{6}$	$\frac{\sqrt{2}\rho\theta^2}{6}$
Maximum stress at angle θ radian (lb/in^2)*	$\frac{E\theta t}{\rho}$	$\frac{E\theta t}{4\rho}$	$\frac{E\theta t}{4\rho}$
Maximum curvature (in^{-1}) at angle θ	$\frac{2\theta}{\rho}$	$\frac{\theta}{2\rho}$	$\frac{\theta}{2\rho}$

* t represents the strip thickness

** loaded by a pure couple

5 FATIGUE AND MATERIALS SELECTION

The material chosen for our flexure was Maraging steel C-300 because it possesses high toughness and ductility at yield strengths of over 300,000 lb/in^2. Since Maraging steel only requires a simple low-temperature age-hardening (900 F for 3-6 hrs. then air cool) this results in extremely low dimensional changes (≈0.08%) making it insensitive to large crossectional changes or special shapes. Since this material is readily manufacturable by conventional methods in the annealed state our manufacturing strategy was to machine the part to shape externally and drill and tap holes for attachments and for wire threading for the EDM cuts and then age harden. The very small EDM cuts needed to define the walls of all of the hinges were then made first, the intent being to complete all the hinge walls (which are the items with the tightest tolerances) before any large cuts carrying a risk of dimensional change were started.

To design our flexure for essentially infinite life ($> 10^6$)[11] we use the theoretically predicted fatigue curve of the material based on the "method of universal slopes"[12]. For application to flexures, it is the *strain* which is prescribed so we choose the form

$$\Delta\varepsilon = 3.5\frac{\sigma_u}{E}\left(N_f\right)^{-0.12} + \left[\ln\left(\frac{1}{1-RA}\right)\right]^{0.6}\left(N_f\right)^{-0.6} \tag{1}$$

where

$\Delta\varepsilon$ = the strain range (the strain *change* from maximum compression to maximum extension)

σ_u = the ultimate tensile strength (lb/in^2)

RA = reduction in area at failure

N_f = number of cycles to failure

The first term represents elastic strain and the second one plastic strain. Setting σ_u=350000 lb/in^2, RA=0.33, we get the calculated values of the strain range that would lead to the required lifetime of the flexure $> 10^6$ cycles. Two of our concerns that would affect the fatigue lifetime of our flexure were hydrogen embrittlement of the material during the EDM process which can be relieved by baking immediately after machining[13], or the possibility that a brittle fracture could begin in the "recast layer" of material that is created on the machined surface due to the localized melting and then quenching of the surface by the EDM process (which is also the location of the maximum stress of our flexure).

After manufacturing several test flexures and cycling them in a test apparatus that rotated them by our required ± 3° we found that the early samples failed long before predicted by theory. Performing the baking processes to alleviate hydrogen embrittlement yielded only small gains in the fatigue life of our test samples. We then created a fixture for lapping all of the flexing surfaces in our mechanism to remove approximately .0015 in. per surface, this along with the baking to remove hydrogen resulted in test specimens that lasted greater than the required 10^6 cycles at ± 3°.

6 DRIVER

The design constraints that drove the design of our driver was that it must have no periodic errors (this has repercussions in the data collection and analysis) such as those possessed by a lead screw or a gear driven mechanism, It must have a stick-slip of no greater than 0.03-0.05λ (~8 Å) and a Stage-motion indexing error of the same (Table 1). Our testing showed that the main source of stick-slip in the driver was due to the o-ring not being fully bathed in oil. We were able to meet our stick-slip requirements by maintaining a reservoir of oil on the outside of the sealing o-rings on the cylinder and on both sides of the piston. This creates a wedge of oil under both sides of the O-ring at all times and eliminates the static friction that develops between the O-ring and a dry cylinder wall or drive shaft.

The point of application of the force to the stage should be the center of resistance and the direction of the force should be the direction of motion (Fig. 4). The force is resisted by the spring and by friction we would like this center of resistance to be the same as the center of mass. Since the resistance and force result from the same thing, namely the spring itself they will be coincident, but because the stage carries a roughly 8-lb load of optics on top, it follows that a counterweight of similar mass and position will be needed underneath. We believed that errors in the point of application of the force would be the leading cause of the most important parasitic motion of the stage which is a variation of pitch angle with position (i.e. with force). Therefore we made the point of application of the force adjustable. However, in practice, we found that the error in the pitch angle was very insensitive to the point of application of the force and appeared to depend more critically on departures from the ideal shape of the flexure.

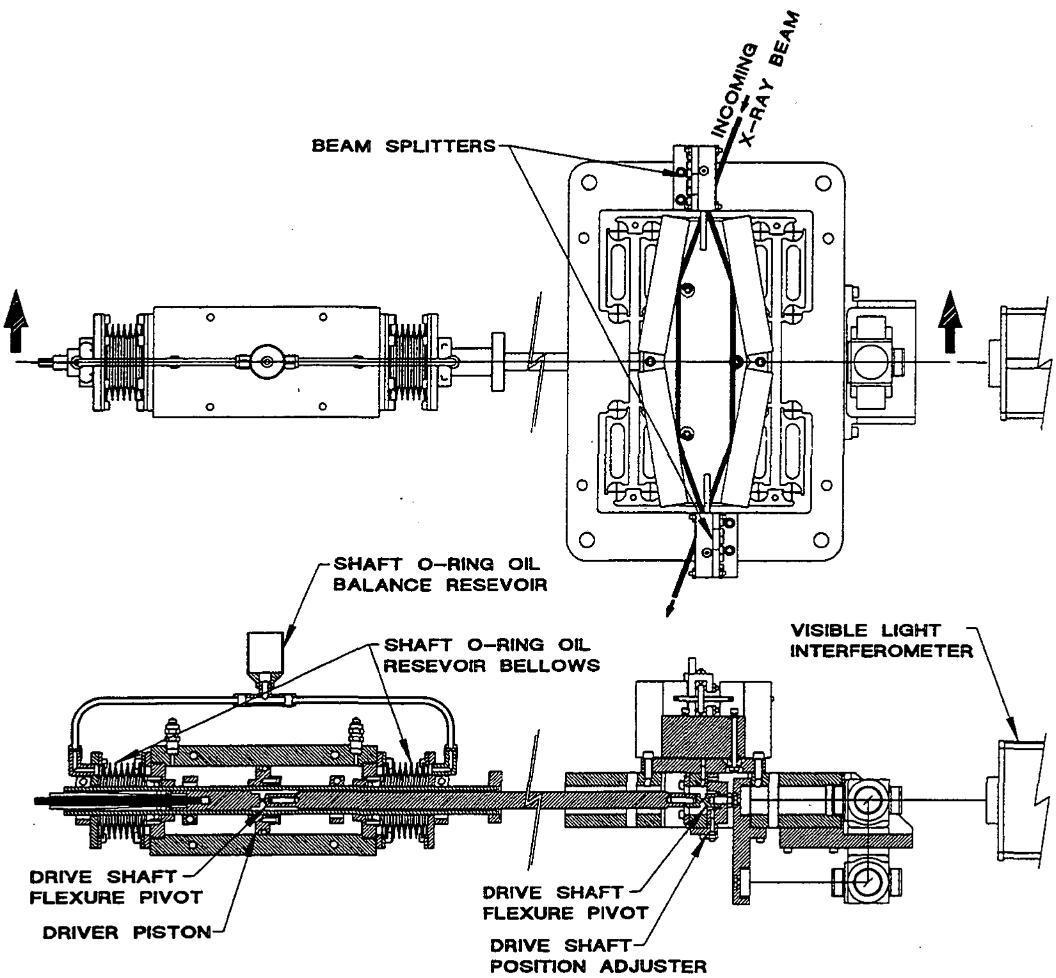

FIGURE 4 SECTION VIEW (NO VESSEL)

7 BEAM SPLITTERS & MIRRORS

The beam splitters which are essentially a an x-ray quality mirror with slots[14] are mounted at a 20° grazing angle with respect to the incoming and outgoing beam and can be adjusted in vacuum by means of a piezoelectrically-driven micrometer. The splitters have been fabricated from silicon by photolithography and directional etching. The slots produced are 50 μm wide by 15 mm long with a period of 100 μm. The reflecting surfaces obtained were flat within 0.6 μradian rms. and smooth within 2 Å rms. The two beam splitters were produced by Boeing North American [15].

The mirror assembly (Fig. 1) which is carried on the moving stage is constructed of ultra low expansion glass and consists of a special prism on which all four mirrors are optically contacted, thus eliminating the need for subsequent alignment of the mirrors. By means of this approach, the required mirror alignment accuracy of 2.5 μradians was achieved once and for all. Before assembly the reflecting surfaces of the mirrors were coated with molybdenum.

8 FINAL ALIGNMENT & OPERATION

After manufacture of the monolithic flexure by an outside vendor we inspected it using the coordinate measuring machine (CMM) at LBNL. We were able to accurately locate the points of intersection of the center lines of the hinges with the upper and lower surfaces of the flexure plate. In a perfectly-made flexure the points on the upper surface would project exactly on to the ones on the lower surface. In fact the projected points had displacement errors up to 0.0035 inch, far greater than our fabrication tolerance which was 0.0003 inch. Upon measuring the pitch angle of the moving table we found an error of about 25 μradian at maximum displacement compared to a goal of less than 0.5 μradian. In order to evaluate our options we first needed to verify that the observed pitch errors were indeed caused by the manufacturing errors.

The key to analyzing the behavior of the mechanism is to regard it as made up of four simple rectangle mechanisms each capable of being deformed into a parallelogram. For example in Fig. 2, the rectangle ABDC forms one such mechanism of which the corners A and C are fixed to the frame. The rectangle EFHG forms a second similar mechanism which is "riding" on the moving member BD of the first. By using techniques based on the paper by Hathaway[10] which gives an analysis of a single rectangle mechanism, we were able to model each half of our flexure as a pair of nested rectangular mechanisms. By using the data from the coordinate measuring machine as input to the Hatheway theory, we were able to classify the errors in the geometry of our rectangle flexures and to calculate their contribution to the overall pitch error of the mechanism. It became apparent that our largest errors were due to tilts of the center lines of the hinges and not to hinge location errors that would result in unequal span lengths. There are two types of tilt errors considered by Hatheway: non-parallel neutral axis (non-zero net value of α in Fig. 5), and non-parallel principal axes of inertia (non-zero value of β in Fig. 5). It was determined that the effect of non-zero β was by far the larger of the two in our case. This causes opposite ends of each nested rectangle to move out of the plane of the system by different amounts.

Influence of non-parallel neutral axis (1)[10].

$$R_y = -3\alpha Tz^2/5SL \tag{2}$$

Influence of non-parallel principal axis (2)[10].

$$R_y = \beta Tz/S \tag{3}$$

where

Ry= resultant pitch rotation of table.
L= Length of the flexure arm.
S = Span between flexures.
α= angular misalignment of the flexures measured perpendicular to the translation axis.
β= angular misalignment of the flexures measured parallel to the translation axis.
Tz= travel of table in required direction.

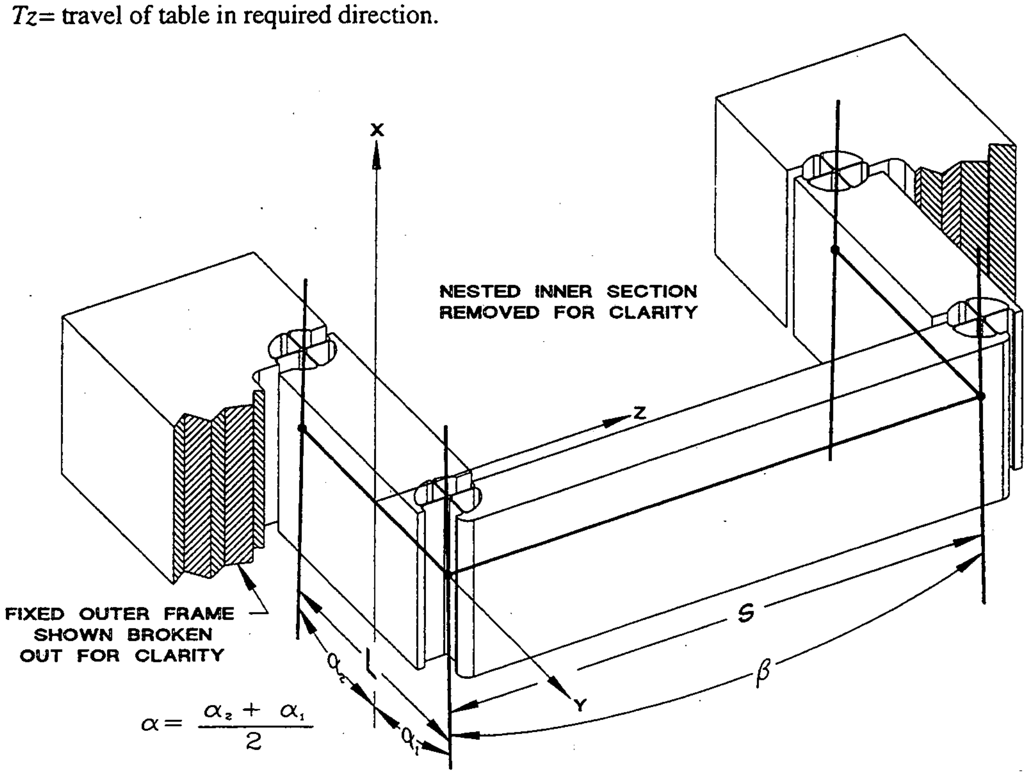

FIGURE 5 FLEXURE MISALIGNMENTS

where R_y is the out-of-plane rotation which causes the observed pitch error and the other quantities are defined in Fig. 5. It is thus expected that the β error is dominant because it is lower order in the displacement z. The result of the calculations was that the table pitch error calculated from the CMM-measured hinge-centerline tilts agreed with the measured one to better than 10%.

Since the undesirable pitch error was an accurate linear function of position, it appeared that a correction should be possible. We therefore added a weak link between the outer frame and each of the two pairs of hinges mounted on the frame. The extra link was adjustable and, once adjusted, fixable by wedges (Fig 3). This idea was very successful and after adjustment and wedging we were able to reduce the pitch errors to 0.38μradian rms (Fig. 6). The out of plane stresses that result from this correction of the manufacturing errors may result in a somewhat reduced lifetime for the flexure but our rectilinear motion requirements have been met.

9 CONCLUSIONS

The monolithic approach to constructing a precise linear motion machine for 15 mm total motion has been shown to work successfully. Installation of an additional adjustment was required to cancel the principal error; a linear variation of pitch angle with displacement which was due principally to a failure to meet manufacturing tolerances. The driver smoothness and indexing accuracy specified in Table 1 were achieved using an inexpensive hydraulic driver with "O"-ring seals bathed in oil on all sides and a commercial laser interferometer system.

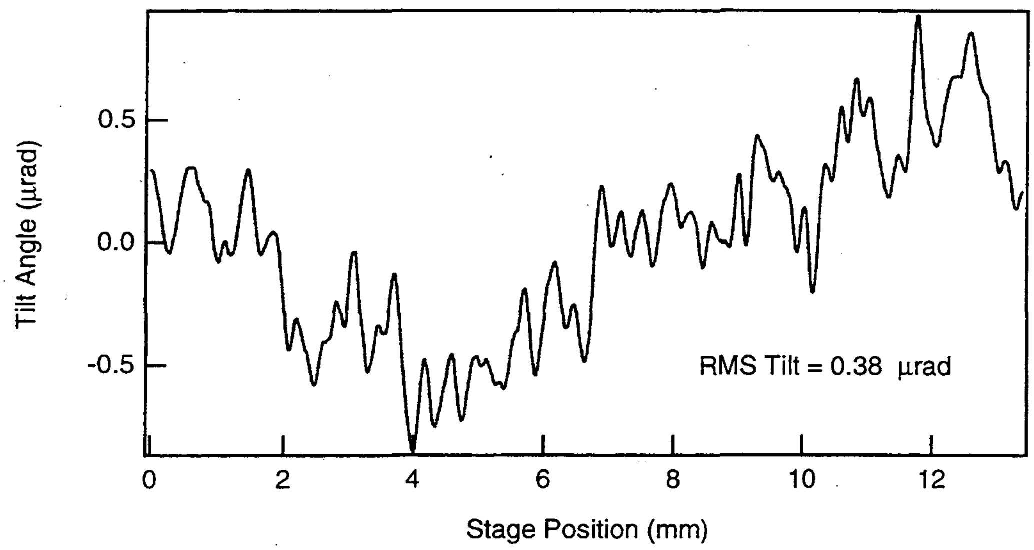

FIGURE 6 (RMS. TILT ERRORS)

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