A Reactionless, Bearingless Linear Shutter Mechanism
for the Multispectral Pushbroom Imaging Radiometer

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Abstract

A shutter mechanism has been developed and flown as part of an airborne imaging radiometer having application to spacecraft or other applications requiring low vibration, high reliability, and long life. The device could be employed in other cases where a reciprocating platform is needed. Typical shutters and choppers utilize a spinning disc, or in very small instruments, a vibrating vane to continually interrupt incident light or radiation that enters the system. A spinning disk requires some sort of bearings that usually have limited life, and at a minimum introduce issues of reliability. Friction, lubrication and contamination always remain critical areas of concern, as well as the need for power to operate. Dual vibrating vanes may be dynamically well balanced as a set and are frictionless. However, these are limited by size in a practical sense. In addition, multiples of these devices are difficult to synchronize.

Introduction

MASTER

SERDP / ARM - UAV

The Atmospheric Radiation Measurement Program is a multi-laboratory, interagency program as part of DOE's principal entry into the U.S. Global Change Research Program. Two issues addressed are the radiation budget and its spectral dependence, and radiative and other properties of clouds. Measures of 'solar flux divergence' and energy exchanges between clouds, the earth, its oceans, and the atmosphere through various altitudes are sought. Additionally, the program seeks to provide measurements to calibrate satellite radiance products and validate their associated flux retrieval algorithms. Unmanned Aerospace Vehicles fly long, extended missions. MPIR is one of the primary instruments on the ARM-UAV campaigns.

System Description

MPIR is a Multispectral Pushbroom Imaging Radiometer, designed to provide multiple images of a scene as viewed through various wavelengths simultaneously. Each detector array scans in crosstrack to record a line image. Utilizing the forward flight path of the host platform, this line is 'swept' over an area; the pushbroom effect allows for construction of a 2D image. As a multispectral system, information is obtained through multiple, parallel channels. The radiation spectrum is sectioned into various bands of interest, and independent modules can be assembled for specific wavelengths. Each channel therefore, is a dedicated module made up of an optics assembly, a sensor assembly, and associated electronics. The modular concept provides for

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reconfiguration and interchangeability. As a whole, the system includes up to 11 microprocessors, optics, and sensors with cryogenic cooling, and a chopper mechanism used for continuous, on-board calibration. It was intended that the chopper design would be transferable to space based applications, such that it would possess the desirable features of vibration isolation, reliability and long life.
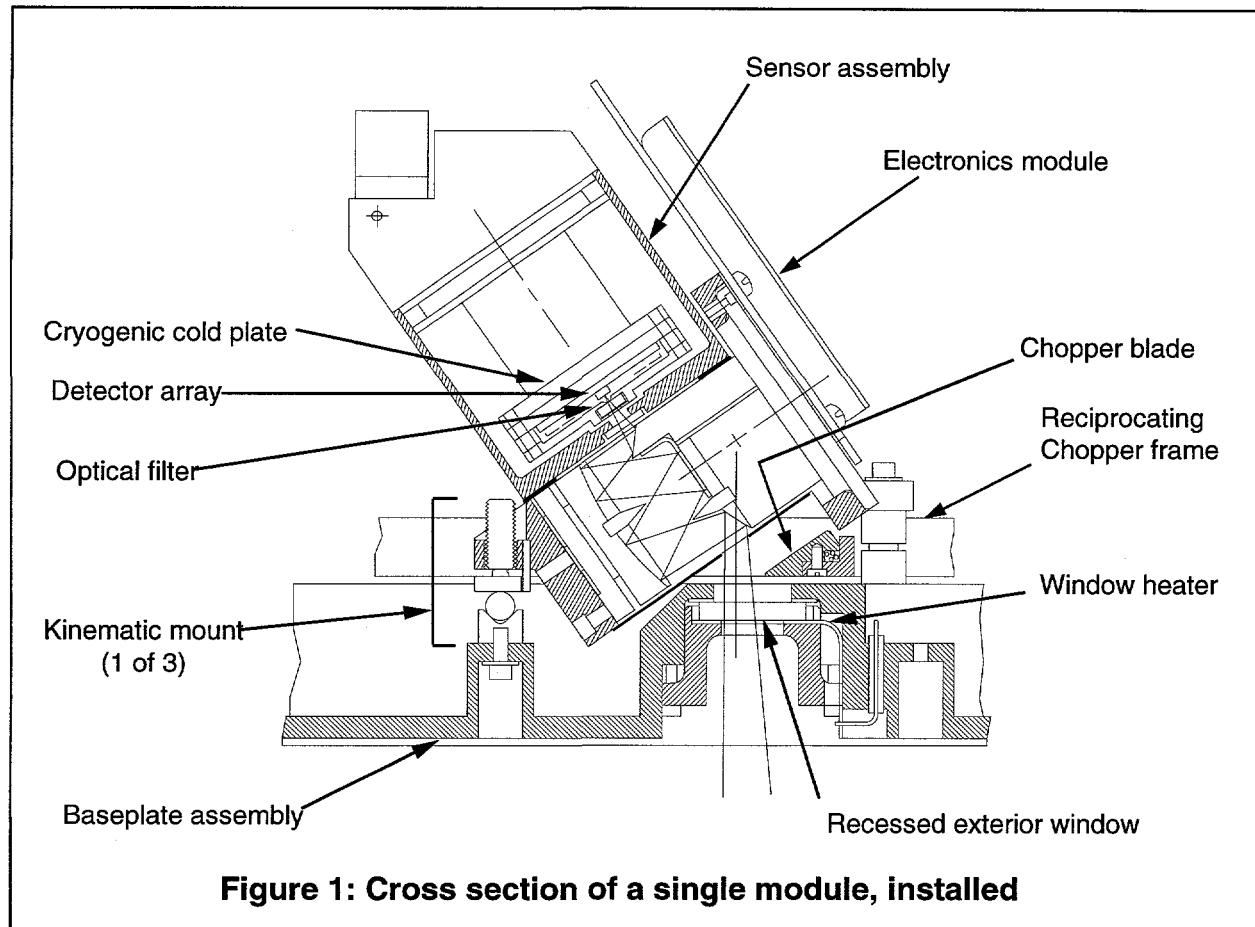
Figure 1: Cross section of a single module, installed

Chopper

The chopper is used for continuous on-board calibration. Incident radiation is scanned by the detector of each module as the chopper blades are retracted (open). At closure, the beam is interrupted, and additional recordings indicate 'dark current' in the visible detectors. To the IR detectors, chopper blades serve as blackbody references. Upon interrogation by the main board processor, each module delivers its data sequentially for formatting with other statistics for transmission via telemetry link.

Typically, a spinning wheel with cutouts is employed as a chopper. For MPIR each module had to be 'chopped' identically and the size of a sufficiently large wheel was prohibitive. In addition, each module must be oriented in parallel for imaging identical scenes; chopping with radial blades with a fixed center would not create similar effects

on each module. Blades could be articulated for parallel motion only with increased complexity. The only solution would be to use a wheel of large radius and to group all the modules into a small general area, making the blade motion seemingly equal and parallel to each module. Actually, only a few blades are necessary, without the need to navigate a full circle. With some logic, this leads to a reciprocating mechanism having only a virtual center that is located at infinity. The concept simply becomes a swinging frame that only oscillates back and forth. Individual chopper blades are attached to a moving grid placed before the 3 x 3 array of modules.
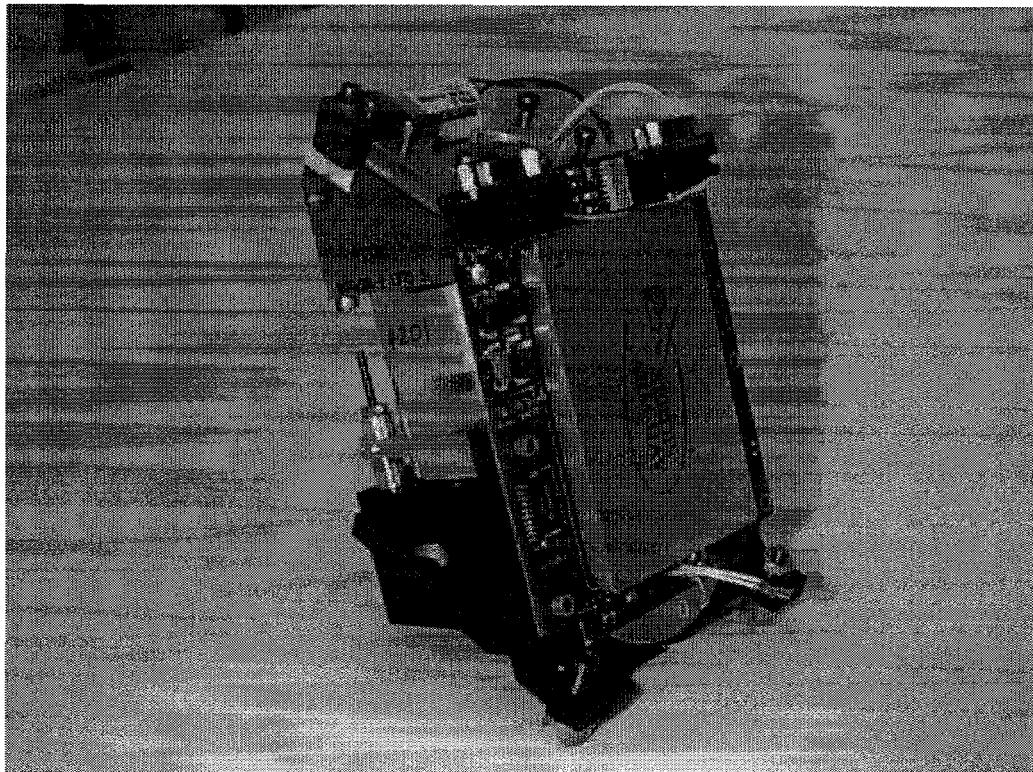
Figure 2: A single integrated sensor module.

Chopper Mechanism
The mechanism involves an oscillating frame, or grid, to which individual chopper blades are attached, a countermass assembly, and a linear actuator. The uniqueness the mechanism is the employment of a reactionless, fully flexural, translating shutter without any rubbing or rolling surface contacts. It operates in resonant mode and requires very little power. Being virtually frictionless, it seems to run on its own once set into motion. Dual, reversing countermasses eliminate inertial forces to make MPIR a relatively quiescent payload.

Support flexures (2)

Countermass assemblies

Crossbeam
(from chassis)

Flexural strap

Chopper frame

Oscillation

Chopper blades (9)

Electro-optic
Position detectors

Drive actuator

Chassis attachment

Support flexures (2)

Electrical flex circuits

Baseplate assembly
(chassis)

**Figure 3: Chopper mechanism**
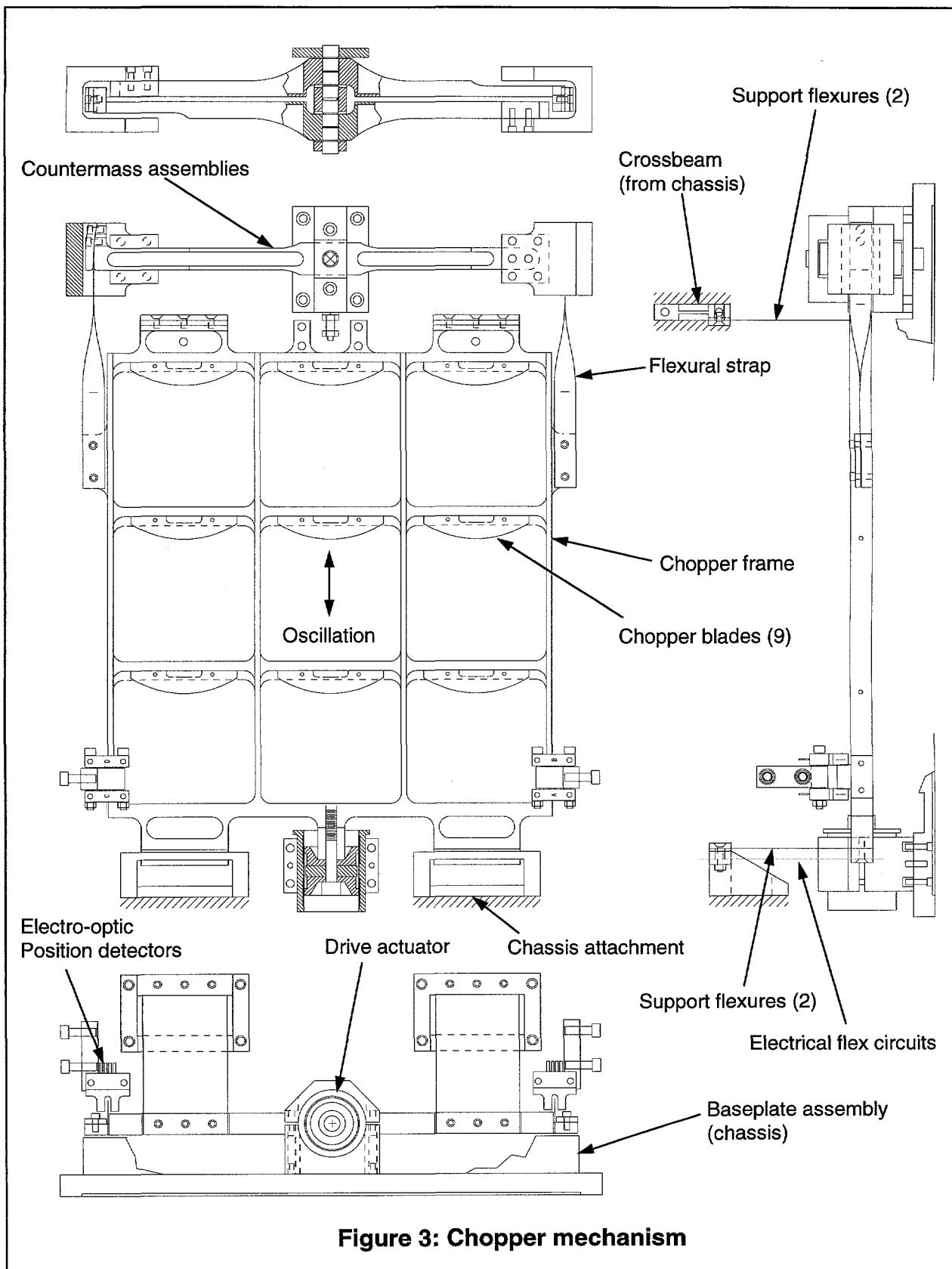
Chopper Frame and Support Flexures

The chopper frame is suspended by four flat flexures, one mounted at each corner. The chopper acts as the coupler of a four-bar mechanism; flexures are of equal length to maintain parallel motion of the chopper. Deflection of cantilevered springs is almost parabolic, so the dimension between opposite ends of a particular flexure is not constant as it swings. Integrating along the parabolic curves, the height change for the chopper is calculated to be .5 mm (.020 in). Considerable width is required for stiffness against side loads, and inverted flight. One other degree of freedom to be constrained was that of rotation within the plane of the chopper around the yaw axis.

For engineering purposes, these support flexures are viewed as two half-length, flat cantilevered springs *(a-b)* & *(b-c)*, butted together at the center. The inflection point *(b)* is assumed to have zero stress, and no bending moment. Figure 4 shows two parallel flexures *(a-c)* & *(d-e)*. The chopper mechanism actually uses four vertical, parallel elements (one on each corner). The flexures not only support the chopper, but also perform as springs.
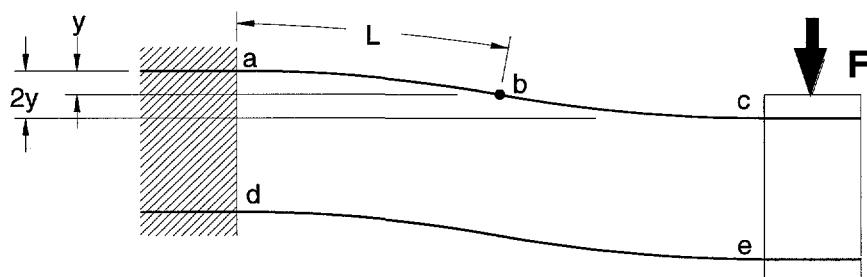

Figure 4: Model for flexural supports

A cantilevered spring (a-b), with force F at point *b* would cause deflection y. On a spring *(b-c)*, by itself, the same deflection would be caused by an equivalent force. Consequently, the force F would cause deflection on both parts *(a-c)* an amount 2y. Stiffness is then calculated starting with the well-known equation for deflection of a single cantilevered spring. The 'spring' has rectangular cross-section so that for *a-b:*

$$I = \frac{bt^3}{12} \tag{1}$$

$$y = \frac{FL^3}{3EI} \tag{2}$$

$$= \frac{4F}{Eb}\left(\frac{L}{t}\right)^3 \tag{3}$$

E is the modulus of elasticity, t the thickness of the spring, b the width, and L the length. Substituting deflection y into the formula for stiffness, k = F / y:

$$k_{ab} = \frac{Eb}{4}\left(\frac{t}{L}\right)^3 \qquad k_{ac} = \frac{Eb}{8}\left(\frac{t}{L}\right)^3 \tag{4a,b}$$

Because the flexure *(a-c)* has twice the deflection (forces being equal), stiffness is only half that amount.

Note the ratio of thickness to length as a sensitive aspect, being a cubed entity. Although the length, 2L, remains constant, separation between ends along the x-axis will decrease with more curvature. Each half of the spline is parabolic and separation between ends can be calculated by integration. The difference from that value and one of a flat condition gives height change of the chopper as it swings through the arc.

Failure Analysis of Flexures
To assure long life and reliability fatigue must be considered. Maximum bending stress at the root of each flexure is simply determined by applying the maximum force of full deflection to the common formula s = M·c/ I. Values used on MPIR show a maximum stress of about 13.8 Mpa (2 ksi) which is far below any endurance limit. In any case, edges of the flexures were polished and inspected to eliminate stress risers and cracks before acceptance and installation.

Chopper Actuator and Drive Mechanism
With the effort to develop a fully flexural design, the selection of a prime mover soon fell to that of a voice coil actuator. Conventional motors would require some sort of bearings, gears, etc., which involve inherent sources of friction. Designing without vibration specifications, use of mechanisms with direct physical linkages were considered as a contingency plan in the untested state of MPIR. Although not frictionless, they provide direct physical control of position and displacement. Two precursor concepts are described. The integrity of an imaging system is obviously compromised by vibration. Likewise, any application to spaceborn platforms would prohibit MPIR from inducing vibration to other instruments or systems. The MPIR chopper therefore needed to be designed as a reactionless system. First to be speculated was the obvious slider-crank mechanism. Considering a chopper (slider) of mass m, a crank of length r, angular velocity ω, and connecting rod of length L, inertial forcing imparted to the frame is:

$$F_i = -m\left[r\omega^2 \cos\theta - r\omega^2\left(\frac{r}{L}\right)\cos 2\theta\right] \tag{5}$$

Dynamic balance of this mechanism is typically accomplished by using a countermass on the crank. The method is only approximate because of the uncompensated second term (r/L)cos2θ. This can be mitigated by using a long connecting rod, minimizing the factor, r/L. Another would be to include an additional rotating countermass with a gear of half the diametral pitch, running off of the crank at the first harmonic frequency. To offset the lateral force of these rotating masses, the rotating elements (gears) may be installed in complimentary pairs.

A scotch yoke mechanism (figure 5) has also been considered where the second term and harmonic effects of the connecting rod is simply eliminated. In equation (5), length

is infinite. Once again, complimentary pairs are employed for lateral balance; two scotch yoke mechanisms with a common yoke are joined together. The pivot at center reciprocates with the load. In this way, crank pin forces are equalized on the yoke for load sharing and smooth operation.

However, the double scotch yoke was abandoned. The drive of choice was the linear actuator. Without gears or cranks, etc., it basically consists of a voice coil and a magnet similar to devices found in loudspeakers. Although less efficient than moving coil devices, these actuators use a moving magnet with a fixed coil, which allows for rigidized wiring. The magnet, cantilevered at one end of the chopper passes through the coil, where it is acted upon only by magnetic forces. This was the compromise to be made over other mechanisms, in that relatively weak physical restraint exists to control displacements of the moving parts. Various masses are free to move as far as the flexible elements will allow. Previously described mechanisms better dictate absolute displacement and amplitude of the chopper. Magnetic coupling is weak in comparison to rigid links, but the linear actuator is avails itself to a system without parts in contact. Full PID closed loop control systems and larger, stronger magnetic circuits are an option to enhance control of this motion. However, the real driver was philosophical, that is, to choose simplicity. A design decision was made, recognizing that external disturbances were acceptable on occasion in that the system would tend to restabilize itself.
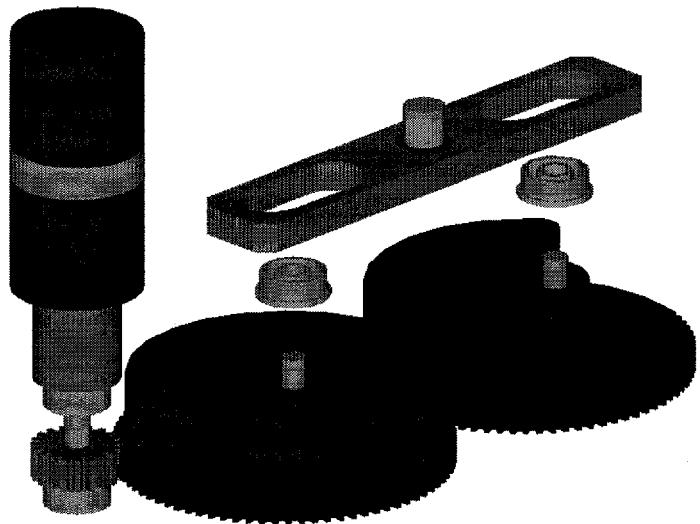
Figure 5: Balanced double scotch yoke

Model No. LA16-19-001A was selected from a line of linear actuators manufactured by BEI Motion Systems Co., having a samarium-cobalt rare earth magnet. The magnetic circuit is comprised of an axial magnet, free to slide within a cylindrical, steel housing (the return path between poles), having sufficient annular gap. The coil is mounted to the inside surface of the housing, situated over the magnet. A cross-section of the

device also reveals steel ends integral with the moving magnet acting as flux directors on the pole faces. Close examination shows that two coils are actually employed in series, each centered over the localized air gap of the flux directors. This configuration provides induced magnetic fields where most influential and flux is more densely located. As the chopper swings in its short arc, allowance for some vertical motion is necessary, the diameter of the flux-director ends was reduced to 31 mm (1.219 in) to allow more radial clearance. Increasing the annular air gap was made at the expense of some efficiency, and yet this loss was unnoticeable.

Countermass Assembly
With the linear actuator a separate means was developed to balance the chopper for lack of an actual drive 'mechanism.' It simply involved adding mass of opposite motion to counteract shaking forces imparted to the instrument. The motion is achieved by a reversing lever, or beam. The 'countermass' is attached to one end; its other end is attached to the chopper. A pivot near the beam center acts as a fulcrum. The pivot is a key component, which is also a flexural member, having no rolling or wearing parts or associated friction. Generally, they can be described as criss-crossed flexures packaged as cylindrical components. For the most part, depending on loading and cycling, angles of rotation are limited to less than 10°. For small displacements, the angle of rotation for the beam is small enough (about +/- 2.7°) that the motion of the countermass can be approximated as being linear.

To provide better centering of the c.g., and to balance any lateral motion, the countermass apparatus is actually divided into two symmetrical devices. They are nested into each other for a compact design, where the pivots share the same axis. Laterally, any errors developed are mirrored and eliminated by cancellation. The key aspect in developing a suitable reactionless system is to cancel out induced vibrations in all degrees of freedom. For the chopper system described, this requires opposing forces to be maintained on the same plane. Hence, the countermasses are nested vertically and lie co-planar with the chopper. Admittedly, the chopper has a small .69 mm (.027 in) vertical displacement as it swings through its arc. Its motion is not purely planar, yet this is negligible for the MPIR application.

No flexural pivot is without spring rate. The countermass assembly is therefore not purely an inertial device, and will also produce a torque opposing its displacement. The additional spring rate is simply added to that of the others in the system. A stiff, flexural pivot is desirable in at least one respect in that it must support the countermass. Considerable torque is presented to the pivots along a transverse axis. Upright, as installed, the pivots are better suited for shear in the horizontal plane. To counteract the bending moment due to vertical accelerations, two pivots are used in support of each beam. Tension at the top of the beam is reacted by horizontal shear of the upper pivot. Compression at the bottom of the beam is reacted by a similar horizontal shear in the lower pivot.

'Free-Flex' pivots by Lucas Aerospace were selected. With proper orientation, they posses a maximum shear capacity of 51 kg (113 lb) perpendicular to their axis, and a spring rate of 5.4 mN·m (.048 in·lb) per degree. Installed at the root of each beam, two of these, with centers essentially separated 15.2 mm (.600 in) can support a 53.4 N (12 lb) force at the beam's end. This value is then reduced with a safety factor and 3 g-loadings taken into account. Flexural pivots are rated with respect to number of cycles, loading and deflection.
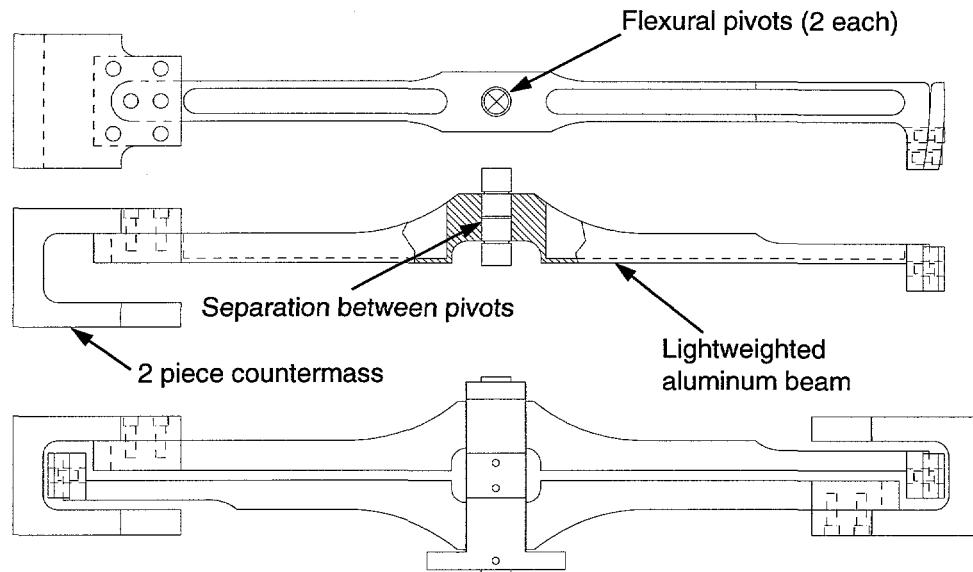

Figure 6: Nested Countermass Assemblies

Deflections are typically less than 10°. In the plane of rotation, the countermass beam for MPIR is required to turn only 2.64° each way. Lifecycle charts indicate that this will bear an infinite lifetime. For the stated springrate and deflection angle, a peak torque of 14.2 mN·m (0.126 in·lb) is calculated for each. Using the approximation of linear motion for these small angles, 1.86 N/mm (10.62 lb/in) is the additional spring rate applied by the countermass assembly. As a protective measure against out of plane bending of the flexural pivots, adjustable nylon-tipped hardstops and .5 mm (.020 in) clearance limit any would-be vertical motion on the countermasses.

Flexural Straps
Connections between the chopper and the countermass assembly are also made with flexural members. Any link between the two assemblies must be compliant in both the vertical and horizontal planes. As described previously, a small vertical displacement occurs with the pendulous motion of the chopper causes a shifting of its plane. Meanwhile, the countermass beams swing a short horizontal arc. While providing congruence between the chopper and countermass motions, the link must allow for conversion between the planar and rotary motions as well. Many ideas were contrived which led to a unique, yet simple, twisted strap. The 'flexural strap' is intended to be

compliant, yet also serve as a compression member as the chopper is accelerated in two instances throughout its cycle. Buckling was not a great concern since the system operates at resonance and parts move in unison. Energy is stored/released by spring elements in both the chopper and the countermass assembly. Hardstops were located so that only tension would occur in the straps in event that action got out of hand. The chopper frame limits motion in one direction while the countermass assemblies limit motion in the other.
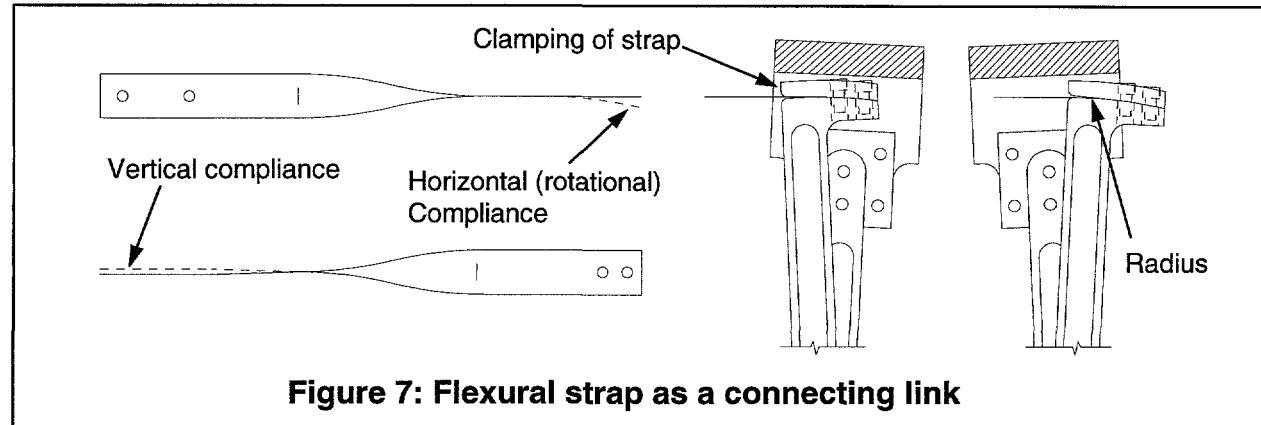

Figure 7: Flexural strap as a connecting link

There was no plan to optimize the connecting link. A narrow thickness was sought through several iterations simply to find one of low bending stress where the material would be utilized within its elastic limit. Links were made of .25 mm (.010 in) 304L stainless steel. An assumption was made that any stored energy in the flexural strap would be released back into the system and in this way compliance would be tolerated. The clamping surfaces at the end of each countermass beam were specifically shaped to accommodate the strap. The inner surface is radiused so that action of the strap would be that of a metal band over a pulley. The outer surface is simply a flat relief allowing the strap to remain straight in the 'chopper closed' position. The strap thus rolls over a constant radius with smooth motion.

Dynamic analysis and design
An interesting discovery came about during the analysis, in that the countermass assemblies can be applied in two different ways. Foremost is need to react against generated inertial forces of the chopper. This effectively cancels vibration that otherwise would be induced to the host platform. On space-borne systems this would be of paramount importance. Such jitter could quite effectively impair imagers such as MPIR and other instruments and attitude controls systems. The other application utilizes the inverse philosophy that rather conserves motion of the chopper, as it is susceptible to accelerations and impulses imparted by the host. In other terms, it's possible to optimize the dynamics, with preference given to either the instrument or the host. The difference lies in the residual inertia of the countermass beams. It's very subtle, probably having little consequence; yet the distinction exists.

Figure 10 depicts the free-body diagrams. This shows a springless system with the chopper free to oscillate horizontally in the plane of the page. It is suspended by long, swinging links conceivably mounted above the plane so that for short arcs the motion is practically along a straight line. Countermass assemblies also are free to rotate within the plane as shown.
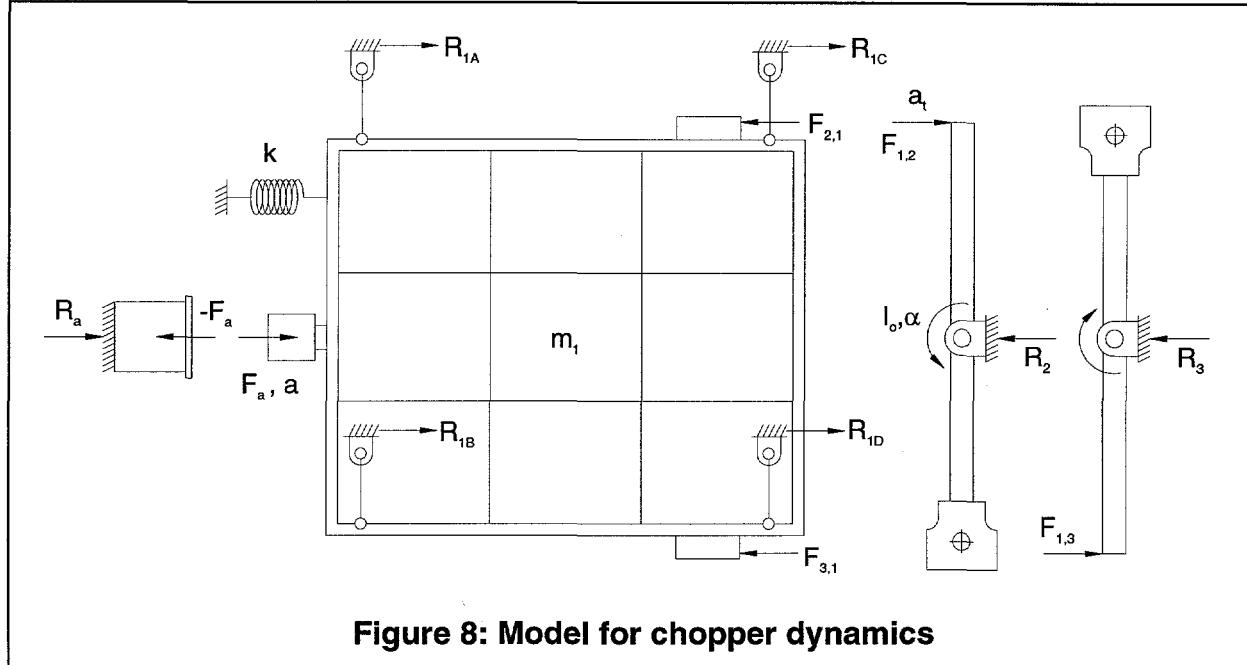
Figure 8: Model for chopper dynamics

Beside the actuator and the frame of the system, three links are indicated, being the chopper, and two countermass assemblies. These carry the subscripts 1, 2, 3, respectively. Reaction forces due to acceleration of the chopper are created in the frame that suspends it. These forces, R_{1A}, R_{1B}, R_{1c}, and R_{1D} may be summed as a whole, designated simply as R_1. Mass m, includes that of the chopper frame, chopper blades, the actuator magnet, etc. Once operational, the forcing function, $F_a = A \sin(\omega t)$ is substantially reduced to provide only a restoring force. Also indicated are the spring forces (F_{flex}) of the support flexures and torques (T_{pivot}) of the flexural pivots. For the current discussion these are not applied.

To size components for the countermass assemblies an analysis is made from a springless system that employs only the inertial forces. With the assemblies installed, opposing forces F_2, and F_3, are applied to the chopper, with consequent reflections imparting R_2 and R_3 to the frame by purely inertial effects. As with any flight system, its optimization would require the minimization of mass. In this case, m, is controlled as an independent variable. Afterward, with a given set of dimensions the problem becomes one in finding the correct countermasses, m_2 and m_3 for reactionless operation. Finally, by including 'springs' in the model, the frequency is determined with the stiffness parameter, k, of the flexures. Because of the necessary tolerances allowed for in manufacturing, small correction masses are attached to the chopper during final

assembly, testing and alignment. This may require subsequent corrections in the countermass assemblies. The process may be listed as follows:

1. Optimize the chopper to determine minimum mass, m_1.
2. Determine values for countermasses, m_2 and m_3, and size for manufacture.
3. Calculate necessary overall stiffness, k, and design flexures for proper frequency of operation.
4. Assemble, align, and test, making corrections in mass for chopper and countermasses.

To continue, the intent is to provide an equal but opposite reactionary shaking force (the sum of R_2 and R_3) to cancel the whole of R_1. This application optimizes the dynamics to eliminate shaking forces imparted to the host. As shown in the previous diagram, both countermass assemblies are made symmetrical and dynamically equivalent. Furthermore, with small angles of rotation, it can be assumed that tangential acceleration of the countermass beam is comparable to the acceleration of the chopper imparted by the actuator: $\alpha_t = -a/r_1$. Any motion of the chopper is mirrored by the countermass assemblies so that the problem is simplified as a quasi-static analysis. The reactionary force R_2 results from the inertial effect of the countermass m_2 being levered on the end of the countermass beam as $F_{1,2}$ is applied. The mass moment of inertia of the beam itself, I_b, is distinguished from that of the countermass, mr^2. Terms r, and r_2 denote radial distances to the point of contact on the beam and the c.g. of the countermass from the pivot point. By design, the countermass assemblies are dynamically equivalent where $m_3 = m_2$ and I_b is the same for each. In selecting a value for the countermasses, inertial effects of the countermass beams always help to counter that of m_1 so that:

$\sum F_{chassis}:$ $\qquad\qquad R_a + R_1 - R_2 - R_3 = 0$ $\qquad\qquad$ (6)

$\sum F_{chopper}:$ $\qquad\qquad F_a + R_1 + m_1 a - F_{2,1} - F_{3,1} = 0$ $\qquad\qquad$ (7)

$\sum F_{C'mass\ Ass'y}:$ $\qquad\qquad F_{1,2} - \dfrac{I_o \alpha}{r_1} - R_2 = 0$ $\qquad\qquad$ (8)

Use (3) to subst. R_2 in (1):

$$R_a + R_1 - 2\left(F_{1,2} - \frac{I_o \alpha}{r_1}\right) = 0 \qquad\qquad (9)$$

Use (4) to subst. R_1 in (2):

$$2\left(F_{1,2} - \frac{I_o \alpha}{r_1}\right) + m_1 a - F_{2,1} - F_{3,1} = 0 \qquad\qquad (10)$$

$$m_1 a = 2\left(I_b + m_2 r_2^2\right)\frac{a}{r_1^2} \qquad\qquad (11)$$

$$m_2 = \left(\frac{m_1}{2}\right)\left(\frac{r_1}{r_2}\right)^2 - \frac{I_b}{r_2^2} \qquad\qquad (12)$$

The effect of inertia in the beam I_b helps reduces the requirements for the countermass in a short term dynamics sense. However, in a constant acceleration or gravity field other than normal to the mechanism (this is a 'planar' mechanism), the countermasses alone must balance the chopper in a statics sense. Inertia of the beam ultimately represents a weight penalty so it should be minimized.

As previously stated, this mechanism may be applied to mitigate external disturbance and vibrations. This is accomplished by the static balance just described, where the countermasses alone contradict the mass of the chopper. To use the equation, the difference in application is whether or not to subtract the term with I_b. Now it is evident that with minimized inertia of the beam the system may work in both applications.

In order to utilize a small angle of rotation and maintain the assumption of 'pure' translation for small angles, values for r_1 should be large. Yet to minimum total mass, r_2 should be maximized. The decision was made to make r, and r_2 relatively equivalent. For a compact design their sum was also limited to being no larger than the width of the chopper, while utilizing a common axis for the flex pivots. In cases where these lengths are equal and the beam has little inertia, the value for m2 and m3 becomes half of m1, as would be expected. For efficiency, the countermass assemblies were optimized for minimum overall weight. The beams themselves are therefore lightweighted and the savings in mass is applied to the countermass on the end.

Tuning
To use minimum power, the spring-mass system is tuned to operate at resonance, the natural frequency being equivalent to the desired operating frequency (4 Hz). A simplified relationship may be utilized where *effective* values for stiffness and mass are employed.

$$f_n = \frac{1}{2\pi} \sqrt{\frac{k_e}{m_e}} \tag{13}$$

Consequently, the model is further simplified to resemble a single spring-mass system consisting of elements m_e and k_e. The two countermasses can be included with the mass of the chopper to describe the *effective* mass of the system:

$$m_e = m_1 + 2\left[m_2\left(\frac{r_2}{r_1}\right)^2 + \frac{I_b}{r_1^2} \right] \tag{14}$$

All flexures working in parallel, the effective combined stiffness is simply a summation of their spring constants. At the end of the beam, rotational stiffness of the flex pivots can be approximated to be linear over small angles. Units of k_{pivot} are N·m / degree. Including the support flexures, the flexural straps, and the flex-pivots in parallel, the overall stiffness is:

$$k_e = 4\left[\frac{Eb}{8}\left(\frac{t}{L}\right)^3 \right] + 2k_{strap} + 4\left[\frac{k_{pivot}}{r_1^2}\left(\frac{180}{\pi}\right) \right] \tag{15}$$

Since the operating frequency is usually first to be specified and the various masses are minimized for weight, only the parameters for spring rate remain to tune the system. The pivots were chosen for bearing capacity. The remaining options lie in specifying the support flexures. These are the elastic modulus of the material 'E', and the dimensions for width 'b', half-length 'L', and thickness 't'. Note that the effects of thickness and length are most significant. Substitutions of mg and k_e into the equation for natural frequency are easily made.

In fabricating the flexural supports, an overall length of 2L is used. By definition, L is the length to midspan. As mentioned previously, long length flexures are desirable to minimize variations in height throughout the arc of motion. A suitable compromise between width and length was found. For thickness, size was selected by commercially available shim stock, available in discrete values.

A particular lesson to learn was finding a reasonable dimension for the width 'b.' Kinematics would disallow the use of four flexures to prevent over-constrained conditions in support of the shutter. The result would be a 'tin canning,' or buckling effect where a flexure can snap between two possible positions due to misalignment of the installation. The problem is alleviated by loosening the clamps and re-aligning the flexures so all are perfectly flat and straight in their 'at rest' position. If possible, the next increment in thickness and narrower widths would mitigate the difficulty. Later versions of the flexures incorporated extra width and a central cutout to provide relief. The effective number of flexures increases yet the proclivity toward bi-stable condition is reduced. At minimum, an aspect ratio of length to width seems to be roughly 3:1, simply by experience. Designing with good lateral separation between flexures also makes an improvement; and of course, a three part suspension would be even better.

In parallel with the flexural supports are two electrical flex-circuits. They posses practically no stiffness, but could be heard ever slightly on occasion against the air. Better designs would use rolling motion.

Fine tuning was accomplished with the least sensitive parameter, the width b. 'On the bench' corrections are also possible with modification to masses m_1, m_2, & m_3, so the system was designed for slightly higher frequency that could then be reduced. Corresponding piece-parts were intentionally made with features to install these corrective masses. The most difficulty lies in minimizing overall mass while using mass for ultimate fine tuning. Seemingly small tolerances on the flexures have pronounced effect. The preventative is good analysis beforehand.

Chopper Motion Control
Except for some amplitude control the drive system is open loop and the chopper mechanism is expected to respond accordingly. As a system of spring-masses, the mechanism is tuned to resonate at the same frequency with sharp cutoffs. Other excitations have little effect, and at resonance very little power is required. Albeit not

directly controlled, the mechanism is loosely synchronous. As a tunable system, it has frequency stability as it attempts to follow the drive excitation. Drive electronics include an 8 bit, 256 level A/D converter operating from the main microprocessor, and a lookup table for the sinusoidal waveshape. This is coupled to filtering and a Class-C push pull amplifier utilizing darlington-pair transistor stages. A control loop in both the A/D converter and the amplifier circuits ensure a very stable signal to the actuator.

Amplitude Control

A set of four electro-optical detectors is employed for position/timing information. Two are positioned to detect an upper and lower limit (UL, LL) of the positive half-cycle that give indication to amplitude. The tolerance band is +/-.18 mm (+/-.007 in), centered at the peak displacement of +6.35 mm (.250 in). Beyond these intended limits, set screws with nylon inserts are used as hardstops.

It is assumed that the frequency is generally well regulated, so the only remaining uncertainty is one of bias. Some difficulty lies in the fact that the spring/mass system has no detent at center. Centering is determined by the force balance of the spring/mass system superimposed by the coil and magnet. This 'zero crossing' can be adjusted by moving any one of the components. Any bias error is seen by unequal duty cycles given by the F.O. and F.C. interrupts, measured accurately by a digital oscilloscope.


Figure 9: Displacement and Timing scheme for chopper
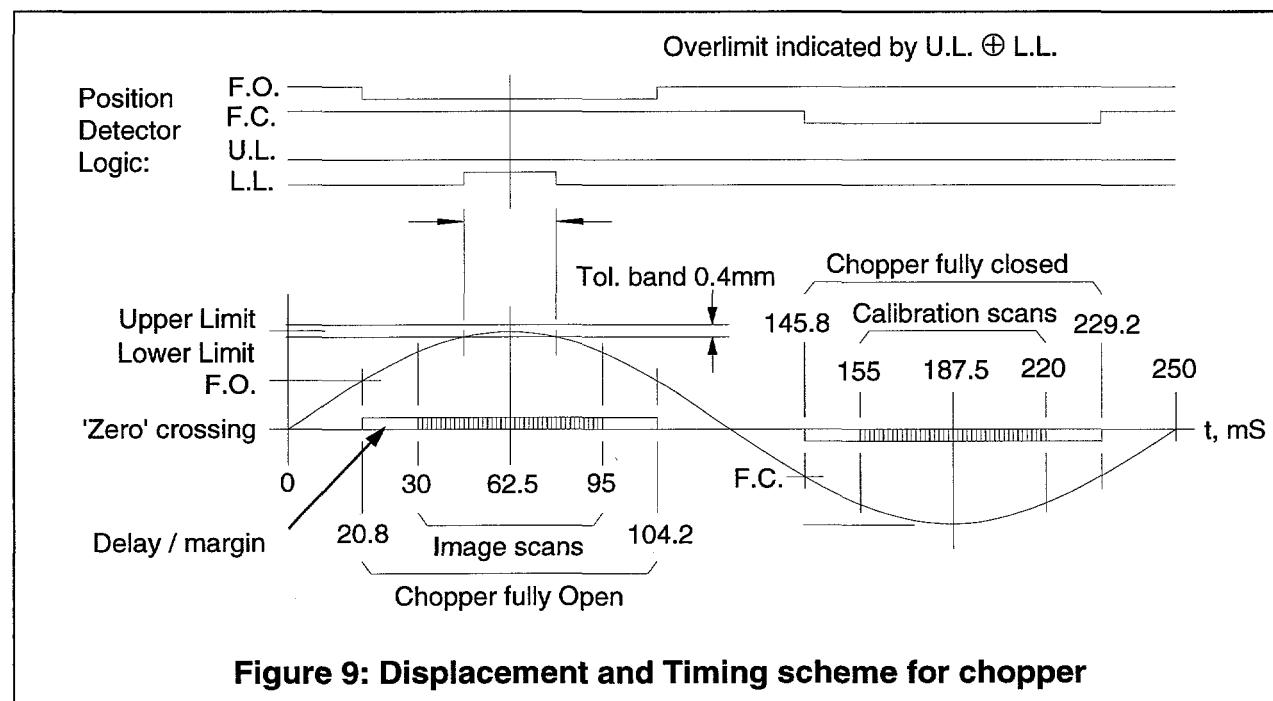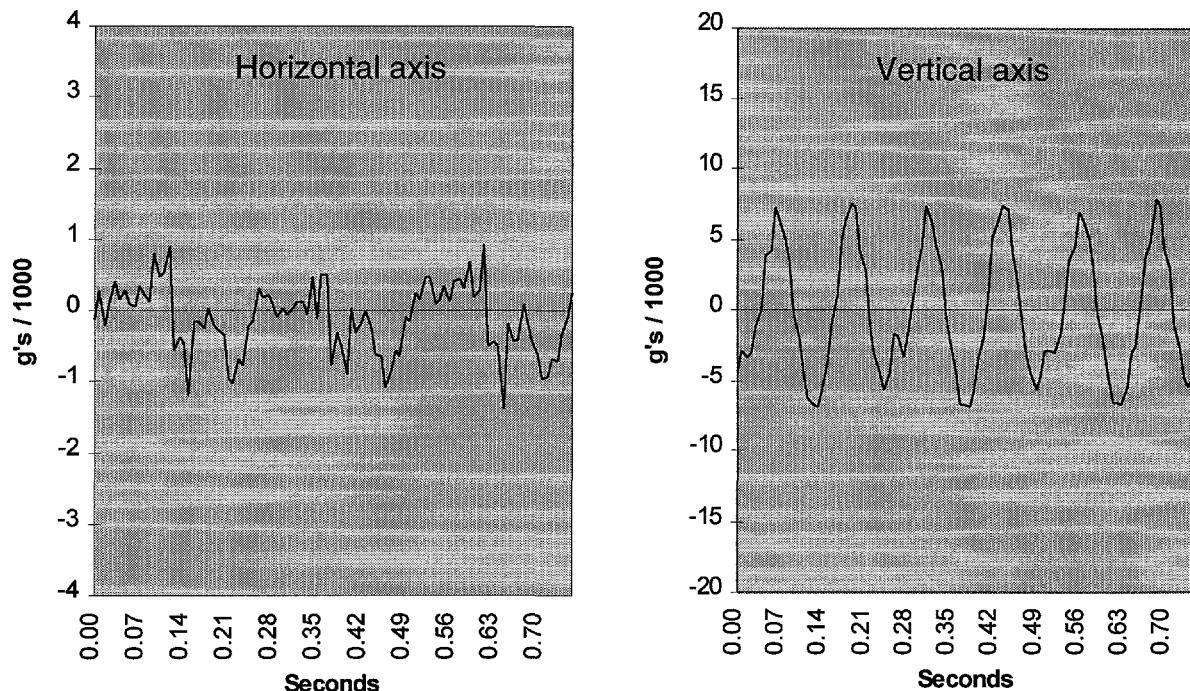
Figure 10: Accelerometer test on chassis

Conclusion

A chopper without countermass assembly would impose a horizontal shaking force upon the chassis an amount $F = -mr\omega^2\sin(\omega t)$. Accelerations would reach 0.4 g's. Compare this to the reactionless system having peaks of about .001 g, as indicated in figure 10. The difference is a reduction of 400:1, or -26 dB. Additionally, a vertical component of about .006 g's exists from the pendulous suspension. Its frequency is double that of the drive signal as a peak occurs at each end of the swing. No compensation was made for this axis; efforts were concentrated upon the horizontal, which was critical to eliminate blur upon imaging. Obviously, long flexures for suspension would reduce the height of swing and vertical accelerations. A set of flexures made as reversed pairs is proposed that could more truly prescribe linear motion for light loads over a small range.

To date, the MPIR system has successfully flown on several campaigns. The chopper continues to operate smoothly without problems. Images are now being reconstructed and geo-rectified from flight data and look very clear.