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SCANNING MIRROR FOR INFRARED SENSORS

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ABSTRACT

A high-resolution, long-life, angle-encoded scanning mirror, built for application in an infrared attitude sensor, is described in this report. The mirror uses a Moire' fringe type optical encoder and a unique torsion-bar suspension together with a magnetic drive to meet stringent operational and environmental requirements at a minimum weight and with minimum power consumption. Details of the specifications, design, and construction are presented with an analysis of the mirror suspension that allows accurate prediction of performance. The emphasis is on mechanical design considerations, and brief discussions are included on the encoder and magnetic drive to provide a complete view of the mirror system and its capabilities.

INTRODUCTION

The encoded scanning mirror is a fully integrated assembly that incorporates a controlled torsional-pendulum scanner that is coupled optically to an angle encoder that has an 87-microradian resolution over a scan angle of 0.35 radian. The mirror suspension uses a one-piece torsion bar to scan at a rate of 3.5 hertz in the resonant torsional-vibration mode. Also, the design allows the mirror to survive a severe initial shock and vibration environment, in which a 60-g level in shock and a 40-g level in vibration can be reached, without caging.

The sensor requirements specified that operation would be in deep space in a zero-g environment, with the enclosure temperature varying from 233° to 338° K, for a minimum operational life of 20 000 hours without the need for maintenance. Furthermore, the entire mirror assembly was limited in weight to 0.680 kilogram and in power consumption to 0.5 watt. Thus, to satisfy all of the requirements, an inherently simple mirror was needed.

Standard flex pivots, coupled with a magnetic drive, were investigated and rejected because their stiffness and radial runout over the ±0.22 radian scan angle would not satisfy the system requirements. In general, a servo-driven pivoted mirror would be heavier and would consume more power than was allowed by specification. In addition, the pivot design would be complex in order to limit the radial runout over the life

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of the mirror to 3.05 micrometers in 0.44 radian of mirror rotation (the tolerance for reliable encoder operation). Therefore, it was decided that the torsional-pendulum concept, which was mechanically simple and required little drive power, could be developed into a viable scanning mirror ideally suited for this application.

In the torsional-pendulum design, the basic parameters are the scan frequency, the scan amplitude, and the moment of inertia of the suspended mirror. The moment of inertia is a function of the geometry, whereas the frequency and amplitude are determined by the sensor-system requirements. Given constraints on the configuration of the torsion bar and its material or fabrication, the higher the frequency (for a given moment of inertia) the smaller the scan amplitude must be to keep the torsional shear stress developed in the bar below the endurance limit. However, if the moment of inertia can be altered, large scan amplitudes (with increasing frequency) still can be obtained without sacrificing reliability or life. This freedom in design through the manipulation of the suspension parameters allows this type of mirror to be applied in a whole spectrum of sensors that have different system requirements. Interplay between these parameters will be made more evident in the section on the suspension analysis.

Nine encoded scanning mirrors have been built and are flight qualified within project specifications. All of these mirrors have passed the environmental-testing program that is to be described in this report.

SYMBOLS

A	cross-sectional area of active section of torsion bar
d	diameter of active section of torsion bar
E	modulus of elasticity
f_n	natural frequency of suspended mirror
g	gravitational acceleration
G	shear modulus
I	area moment of inertia of active section of torsion bar
J	mass moment of inertia of suspended mirror without weights
J_s	mass moment of inertia of complete suspended mirror
k	torsional spring constant of active section of torsion bar
L	length of active section of torsion bar
l_w	length of trim weights (4.128 centimeters)

P tensile preload force

Q $\frac{1}{2 \times \text{damping factor}} = \frac{\pi}{\text{logarithmic decrement}}$

R_o half width of mirror (3.969 centimeters)

t_w thickness of trim weights (0.478 centimeter)

W weight of complete suspended mirror

w_W width of trim weights

δ_m maximum transverse deflection of bar

δ_o maximum deflection of bar without preload ($WL^3/24EI$)

$\bar{\rho}$ mass density of Mallory 1000 (17.28×10^{-3} g-sec^2/cm^4)

σ_m maximum normal stress in active section of torsion bar

τ_m maximum torsional shear stress in active section of torsion bar

φ angle of twist of the torsion bar

GENERAL DESCRIPTION OF ENCODED SCANNING MIRROR

The encoded scanning mirror assembly is shown in figures 1 and 2. The mirror (8.89 by 5.72 centimeters) is gold plated on one side and is suspended within the frame by a one-piece torsion bar, the ends of which are captured by the tension nuts and clamps. The glass encoder reticle plate is attached directly to the mirror by means of the encoder hub. Below the reticle plate and attached to one side of the frame is the encoder illumination source. The remainder of the encoder optics are contained within the encoder housing that forms the base leg of the frame. Behind the mirror (supported by the housing) are the magnet coils that drive the mirror. The frame is closed on the side opposite the encoder housing by an infrared telescope support plate.

The mirror is driven by the magnetic interaction between the short permanent magnets that are mounted on the back of the mirror and the coils. Only one of the coils is used to start and drive the mirror. The other coil senses the mirror motion and generates a synchronous feedback signal to the input of the electronic control circuit.

The mirror and its suspension form a torsional pendulum that has a mechanical Q of approximately 2500 in vacuum. This pendulum is tuned by using trim weights to a frequency of 3.5 hertz. The very high Q ensures a sharp resonance that is free of any perturbing influences from nonlinearities in the drive system and reduces the required drive power to only 0.5 milliwatt.

The sag of the torsion bar, caused by the mirror load, is limited by applying a tensile preload to the bar with the tension nuts. Excessive transverse deflection under high inertial loads is prevented by mechanical stops located at the bumper and the end of the encoder hub. These stops are Teflon bushings that contact the rigid end hubs of the torsion bar when the mirror deflects beyond 0.254 millimeter. Because the mirror "floats" during normal operation, there are no frictional surfaces in the system. Therefore, mechanical wear is eliminated.

The mirror mates with the torsion bar at its center hub. Mating is achieved primarily by chemical bonding with Loctite. This method of attachment is desirable here, because the thinness of the mirror wall in this area precludes the use of mechanical fasteners that would distort the mirror face.

The angular swing amplitude is limited by the mirror stops located at each end of the coil mounting plate. Nominally, contact with the stops will occur at 0.235 radian of swing. The face of the stop is made of sponge rubber to cushion the impacts resulting from vibration-induced torsional oscillation. During normal operation, there is no contact between the mirror and the stop because of the amplitude control exercised by the drive electronics.

Initially, the mirror is excited by the passage of the drive-amplifier noise current through the drive coil. The magnetic interaction causes the mirror to move slightly. This motion is sensed by the pickup coil that generates a synchronous signal to the drive-amplifier input. In turn, this increases the current to the drive coil, thereby increasing the mirror motion. Because the suspended mirror is a torsional pendulum, the amplitude of swing builds up rapidly. When the pickup coil signal reaches a preset value, a control circuit in the drive electronics is activated, stopping the drive current in order to maintain the mirror amplitude at 0.218 radian. Any change in this amplitude is compensated for automatically by the circuit.

ANALYSIS OF THE MIRROR SUSPENSION

The addition of trim weights in the mirror-suspension design was necessary to increase the mass moment of inertia so that the 3.5-hertz scan frequency can be obtained with a torsion bar of reasonable diameter. The following calculations show the iterations required for design. In actual production, all width dimensions for the weights were calculated from actual measurements of the torsion-bar diameter. Then, a matched pair was machined and used with the specific bar. Of the 11 mirrors assembled, all had a natural frequency within 0.1 hertz of the nominal value, demonstrating the validity of the design procedure.

The natural frequency of a torsional pendulum of the type being considered (fig. 3(a)) is given by

$$f_n = \frac{1}{2\pi}\sqrt{\frac{2k}{J_s}} \tag{1}$$

where $k = \pi G d^4/32L$. If f_n, J_s, G, and L are known, the required diameter of the active section of the torsion bar can be calculated from the formula

$$d = \left(\frac{64\pi f_n^{2} L J_s}{G}\right)^{1/4} \tag{2}$$

It is assumed here that the torsion bar is symmetrical in configuration (that is, the active sections are the same).

The value of J_s computed for the suspended mirror minus the trim weights is $J_s \equiv J = 0.397$ g-cm-sec^2. Then, substituting this value along with $L = 3.016$ centimeters, $G = 8.86 \times 10^5$ kg/cm^2 (shear modulus for Elgiloy, the torsion bar material), and $f_n = 3.5$ hertz into equation (2), the value $d = 0.432$ millimeter. This value is too small for practical fabrication of the bar as a one-piece unit. However, it is possible to fabricate (on a production basis) sections that have a diameter of 0.533 millimeter. Then, by using the trim weights to increase J_s, the frequency can be maintained.

The trim weights are located at the outer edges of the mirror to minimize their weight and to significantly increase J_s. Further gains are obtained by making the weights from a heavy material like Mallory 1000 (which has a density of 17 g/cm^3).

Selection of $d = 0.533$ millimeter raises the value of k to 0.234 kg-cm. Therefore, for $f_n = 3.5$ hertz, J_s must now be equal to 0.968 g-cm-sec^2. The required J_s is 2.44 times greater than J. If a rectangular shape of length l_w and thickness t_w is selected for the weights, the required weight width (for example, the dimension in the mirror radial direction) can be calculated by using the formula

$$w_W = R_o \left\{ \left[1 + \frac{3\left(J_s - J\right)}{2\bar{\rho} t_w l_w R_o^3} \right]^{1/3} - 1 \right\} = 0.470 \text{ cm} \tag{3}$$

This width is certainly acceptable within the design envelope, and the suspended mirror weight is increased by only 35 percent.

The final configuration of the torsion bar is shown in figure 3(b). The overall length was determined by the allowable size of the mirror assembly. The hub lengths were dictated by the mirror-mating and deflection-limiting requirements, and the large fillets at each end of the active sections reduce the stress concentrations at the changes in section to a minimum.

The torsion bar is preloaded to control the transverse sag under the weight of the mirror. A suitable analytic model for calculating the stresses and deflection is shown in figure 4. It is reasonable to assume that the hubs are perfectly rigid and the mirror weight acts as a concentrated load at the center of a bar of length 2L.

The maximum transverse deflection will occur at the center and represents the actual mirror deflection. Its value can be calculated from the approximate formula

$$\delta_m = \frac{\delta_o}{1 + \left(PL^2/\pi^2 EI\right)} \tag{4}$$

Solving equation (4) for P, with δ_m limited to 0.102 millimeter and W = 0.121 kilogram, gives the required preload P = 13.99 kilograms.

The normal stress in the torsion bar is the algebraic sum of the axial and bending stresses induced by the P and W loads. The maximum normal stress is given by

$$\sigma_m = \frac{P}{A} + \frac{Wd}{4I}\sqrt{\frac{EI}{P}} \tag{5}$$

At room temperature, $\sigma_m = 7.252 \times 10^3$ kg/cm^2. Over the operating temperature range, caused by changes in the mirror-frame dimensions, this stress can increase by as much as 1.645×10^3 kg/cm^2. The proportional limit for Elgiloy (based on tests) is 5.455×10^3 kg/cm^2 in the non-heat-treated case and 11.998×10^3 kg/cm^2 when heat-treated. Clearly, to preserve the elastic behavior of the torsion bar under load, it is essential that only heat-treated bars be used.

The stresses that developed in the torsion bar when the mirror was subjected to a 60-g level of both transverse and axial inertial loading were analyzed. It was found that if, in the transverse case, the deflection was not limited by the mechanical stops, failure would occur because the ultimate strength of Elgiloy would be exceeded. However, in the purely axial case, the stress would not exceed the proportional limit. The maximum allowable transverse deflection before the proportional limit is exceeded in a worst case situation has been questioned. Knowing this, the maximum clearance between the Teflon bushings and the torsion-bar hubs could be established. By means of analysis, it was shown that, without an axial load component, the maximum deflection would be 0.711 millimeter. However, when both a 60-g axial component and a transverse component were acting simultaneously to stress the bar to its proportional limit, only a 0.127-millimeter deflection would be tolerable. To establish this amount of clearance in the mechanical stops is impractical. Therefore, it was decided that, if the clearance was 0.254 millimeter, the likelihood of failure would be small in any actual situation that was within the performance specifications. This judgment was upheld throughout the environmental testing of the complete mirror assembly.

The maximum torsional shear stress was investigated to ascertain whether this stress would exceed the estimated torsional endurance limit of Elgiloy. Using the formula

$$\tau_m = \frac{Gd}{2L}\varphi \tag{6}$$

it was found that $\tau_m = 1.709 \times 10^3\ \mathrm{kg/cm^2}$ for $\varphi = 0.218$ radian. A conservative estimate of the endurance limit indicated a value of $2.285 \times 10^3\ \mathrm{kg/cm^2}$ when the bar was preloaded at the level $P = 13.99$ kilograms. Without preload, tests have indicated that the endurance limit is raised to $4.218 \times 10^3\ \mathrm{kg/cm^2}$. Therefore, it is seen that torsional fatigue failure, caused by overstressing, does not appear probable under normal operating conditions. The life tests, discussed in the next section, supports this conclusion.

TORSION-BAR CONSIDERATIONS

Torsion-bar-grade Elgiloy was selected for the bar material because it has a very high proportional limit, shear strength, and torsional endurance. Furthermore, it is nonmagnetic and highly corrosion resistant; it also has low thermal conductivity, low torsional hysteresis loss, a modulus of elasticity equivalent to steel, and a notch sensitivity of approximately zero. The grade chosen is select in that it is free of macroscopic inhomogeneities and core pipe throughout its volume.

When the mirror development began, the physical properties for Elgiloy wire were not available. Therefore, it was desirable to perform a series of tests on specimens conforming to the torsion-bar design in support of the theoretical calculations and strength estimates. Tests were conducted on both heat-treated and "as received" specimens for comparison purposes and to establish the true advantage of heat-treating the bars. The results of the project are summarized in table I.

In general, machining Elgiloy is not difficult if carbide-tipped tools are used. However, to machine a rod to the torsion-bar configuration, a special lathe setup is required. The tool pressure on the workpiece is high; therefore, a backup rest that travels with the cutter is necessary when machining the thin active sections. A technique was developed that facilitates the fabrication of as many as four torsion bars in 1 man-day.

The bars are machined to within 51 micrometers of the final diameter. Finishing to specifications is achieved by electropolishing. This process not only allows precise control of the final diameter but facilitates removal of all surface marks, minimizing the chance of failure for this reason.

The reliability of the bar has been proven in both environmental tests and in accelerated life tests. The environmental tests were conducted on complete mirror assemblies and consisted of subjecting them, along three mutually perpendicular axes, to the following environments: a 30-g sustained acceleration for 300 seconds, a 60-g half-sine-wave shock for 8 milliseconds (three impacts), 25-g sinusoidal vibrations to 400 hertz followed by 40 g to 2000 hertz, random vibration for 180 seconds, and temperature cycling while operating in the range 233° to 338° K. In no case did the suspension fail with these flight-quality mirrors.

The accelerated life test involved torquing 12 preloaded bars at a frequency of 60 hertz, for a minimum of 10^9 cycles of scan or to failure, whichever occurred first. Three of the bars were torqued through 0.524 radian and the remainder through 0.436 radian, simulating the actual mirror operation. Ten of the bars were operated over 2.5×10^9 cycles and the other two were operated over 1.9×10^9 cycles before the test was voluntarily terminated with no failures. The discrepancy in cycles is caused by a test-apparatus problem that resulted in the failure of two bars. The results of this test are indicative that, if the design torsional stress is kept below 2.11×10^3 kg/cm^2, there will be no fatigue failure caused by overstressing.

ENCODER-SYSTEM DESIGN AND OPERATION

The optical angle encoder system was unique in that the nature of the application and the limitations imposed by the space and weight requirements called for complete integration of the encoder with the basic mirror assembly design. A schematic of the encoder system is shown in figure 5(a). The light from a miniature lamp is divided into two beams by the optics contained in the illuminator assembly. One beam illuminates the encoder reticle with collimated light and the other beam is focused into a bright line at the center reference reticle. These reticles are printed photographically on the reticle plate which is fastened rigidly to the mirror.

The encoder reticle is made up of a series of evenly spaced lines that radiate from the mirror axis. This pattern is imaged onto a similar pattern at the fixed reticle by the image-transfer optics. Behind the fixed reticle, four silicon detectors form a quadrature array that has two output channels. If there are n lines in 2π radians on the encoder reticle, the fixed reticle contains $n + 1$ lines. The fixed reticle pattern is skewed slightly with respect to the encoder pattern in order to generate Moire' fringes (at the detectors) as the mirror moves. The fringe width is adjusted to the width of one detector and the detectors are wired back-to-back in alternate pairs. The result of this arrangement is the same as having an encoder with full disks of n and $n + 1$ lines and detectors at $\frac{\pi}{2}$, π, $\frac{3\pi}{2}$, and 2π radians.

The A channel is connected to the detectors corresponding to the π and 2π points, and the B channel is connected to the other two detectors. Each pair of detectors produces a sinusoidal signal, the period of which corresponds to 0.698 milliradian of scan angle, with a $\frac{\pi}{2}$ radian phase difference between them. The direct-current component of the signals is eliminated by the back-to-back wiring arrangement. These signals enter the encoder logic circuit where the B signal is added to and subtracted from the A signal to produce sum and difference signals that are $\frac{\pi}{4}$ and $\frac{3\pi}{4}$ radians out of phase with A (fig. 4(b)). The four signals A, B, A + B, and A - B are combined logically to produce data pulses every $\frac{\pi}{4}$ radian within each period. A factor of 8 subdivision is achieved, thereby increasing the encoder resolution to 87 microradians.

The center reference reticle appears as two rows of arc segments, alternately clear and opaque about the center lines (fig. 5(a)). The pattern illumination is projected through a narrow slit and falls on two silicon detectors located behind each row. These detectors are wired back-to-back to eliminate the direct-current component. As the mirror moves through its center position the signal from the detectors reverses polarity. Then, this signal is combined logically with one of the encoder pulses to obtain a center reference data pulse that is accurate to within 87 microradians of the center of the mirror scan.

A considerable amount of encoder testing has been performed to demonstrate the adequacy of the design. The glass reticle plate, mounted in the hub, has been shock tested to the 100-g level. The complete working assembly has been temperature cycled without any loss in performance, and the accuracy has been measured over 0.35 radian of scan angle and was indicative that each data pulse was in its true location to within 17.5 microradians.

CONCLUDING REMARKS

The feasibility of constructing a flight-qualified encoded scanning mirror based on the torsional-pendulum concept has been shown. In all respects, the environmental and operational requirements have been met or exceeded by the application of valid engineering analysis to the mirror suspension and encoder system and the development of special fabrication techniques that ensure reliable torsion-bar performance. The latter represents the most important achievement in actually realizing a mirror of this type.

Since the development of the flight-qualified torsion bar and mirror assembly, several new designs that use the same basic concepts have been created to satisfy other configuration requirements that involve differences in frequency, scan angle, and moment of inertia. All of these new designs have been analyzable along the lines outlined here, with the predicted results achieved during manufacture and test.

TABLE I. - MECHANICAL PROPERTIES OF 0.533-MILLIMETER-DIAMETER TORSION-BAR-GRADE ELGILOY WIRE

Property	Specimen[a]	
	As received	Heat treated[b]
Ultimate tensile strength, $kg/cm^2 \times 10^3$	17.08	18.77
Yield strength, 0.2-percent offset, $kg/cm^2 \times 10^3$	13.99	18.56
Elongation, percent	2	1.27
Proportional limit, $kg/cm^2 \times 10^3$	5.455	11.998
Elastic modulus, $kg/cm^2 \times 10^3$	2066.8	2087.9
Hardness, R_c	39	45
Ultimate torsional shear strength, $kg/cm^2 \times 10^3$	14.20	16.24
Torsional shear yield strength, $kg/cm^2 \times 10^3$	8.29	12.44
Torsional shear proportional limit, $kg/cm^2 \times 10^3$	4.22	9.07
Shear modulus, $kg/cm^2 \times 10^3$	773.3	885.8

[a]Test specimens conformed to one active section of the torsion bar; specimens were not electropolished; gage length = 3.05 cm.

[b]Conformed to manufacturing specifications.

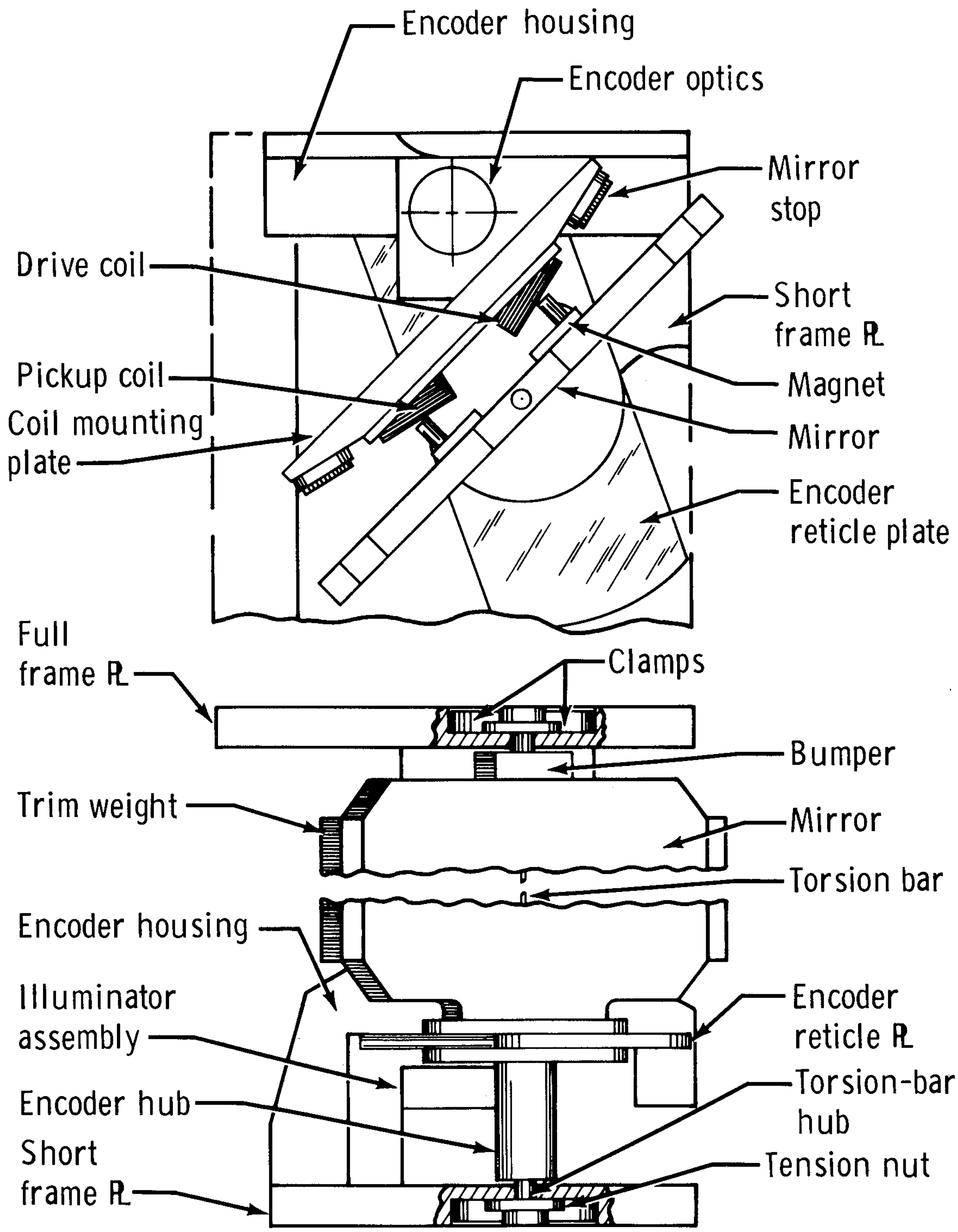


Figure 1. - Encoded scanning mirror configuration.

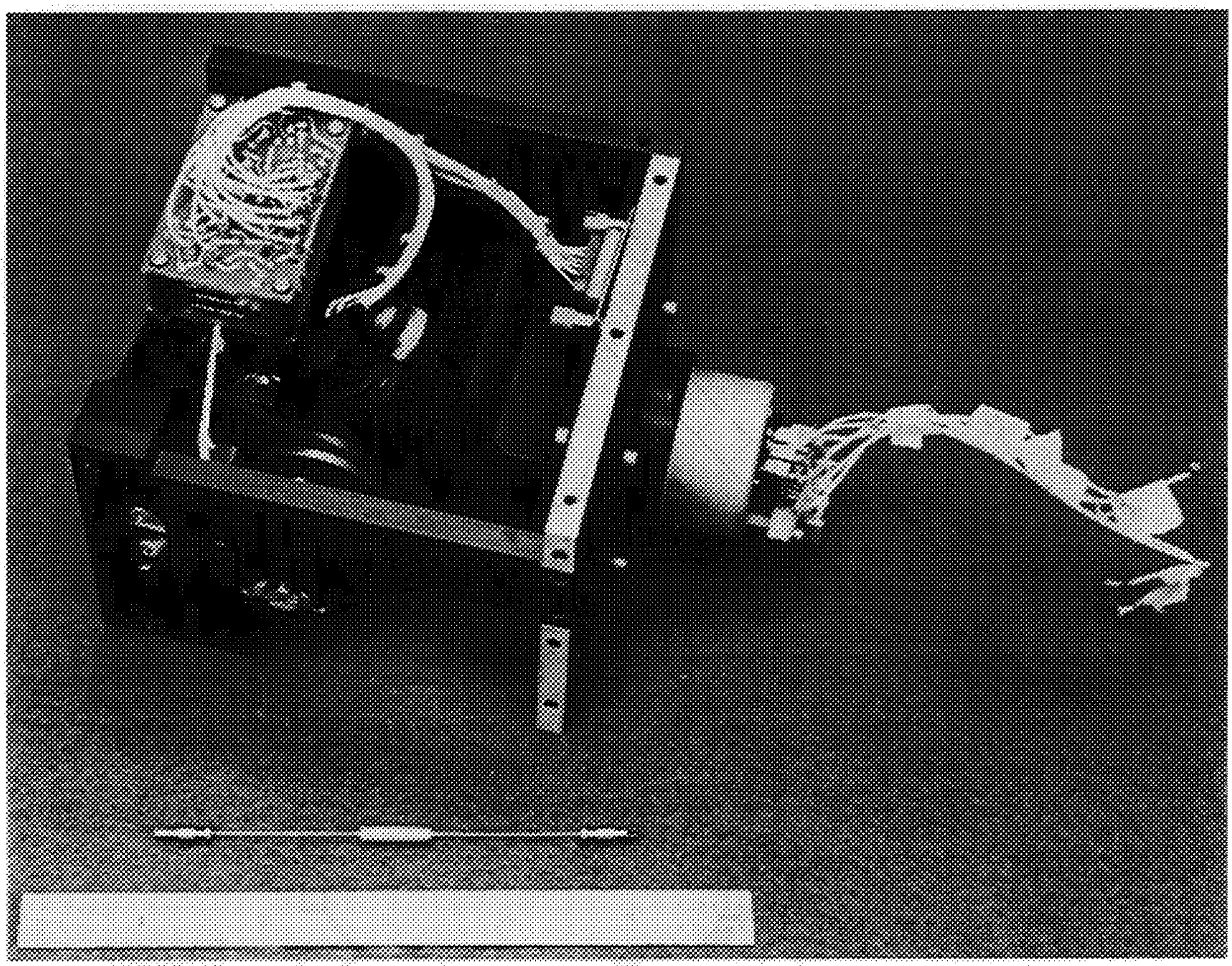
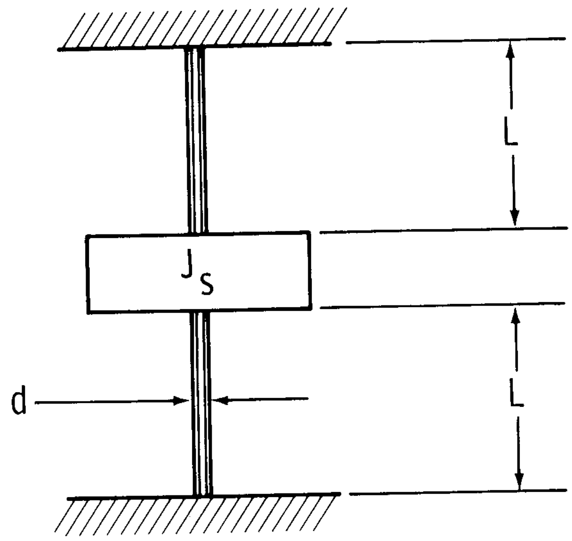
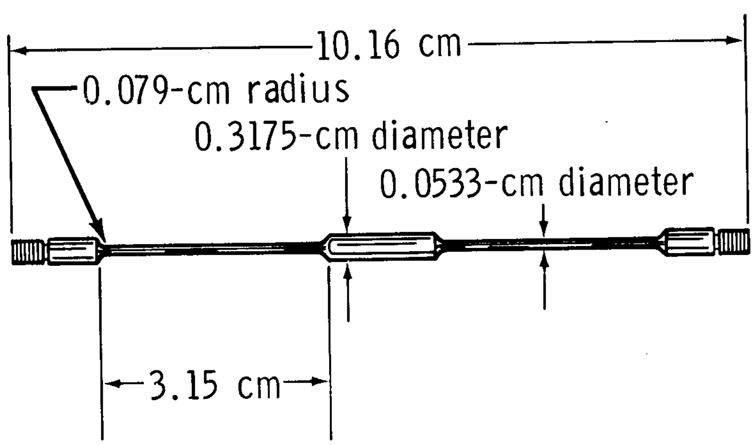


Figure 2. - Encoded scanning mirror assembly and torsion bar.



(a) Torsional-pendulum model for suspended mirror.



(b) Final torsion-bar configuration.

Figure 3. - Torsional-pendulum model and torsion-bar configuration for suspended mirror.

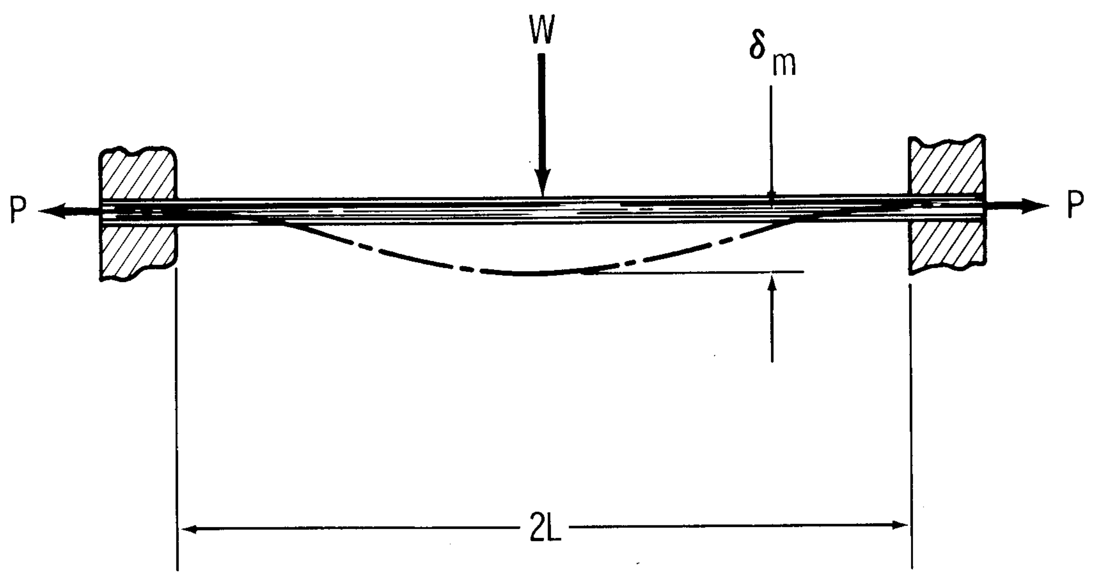
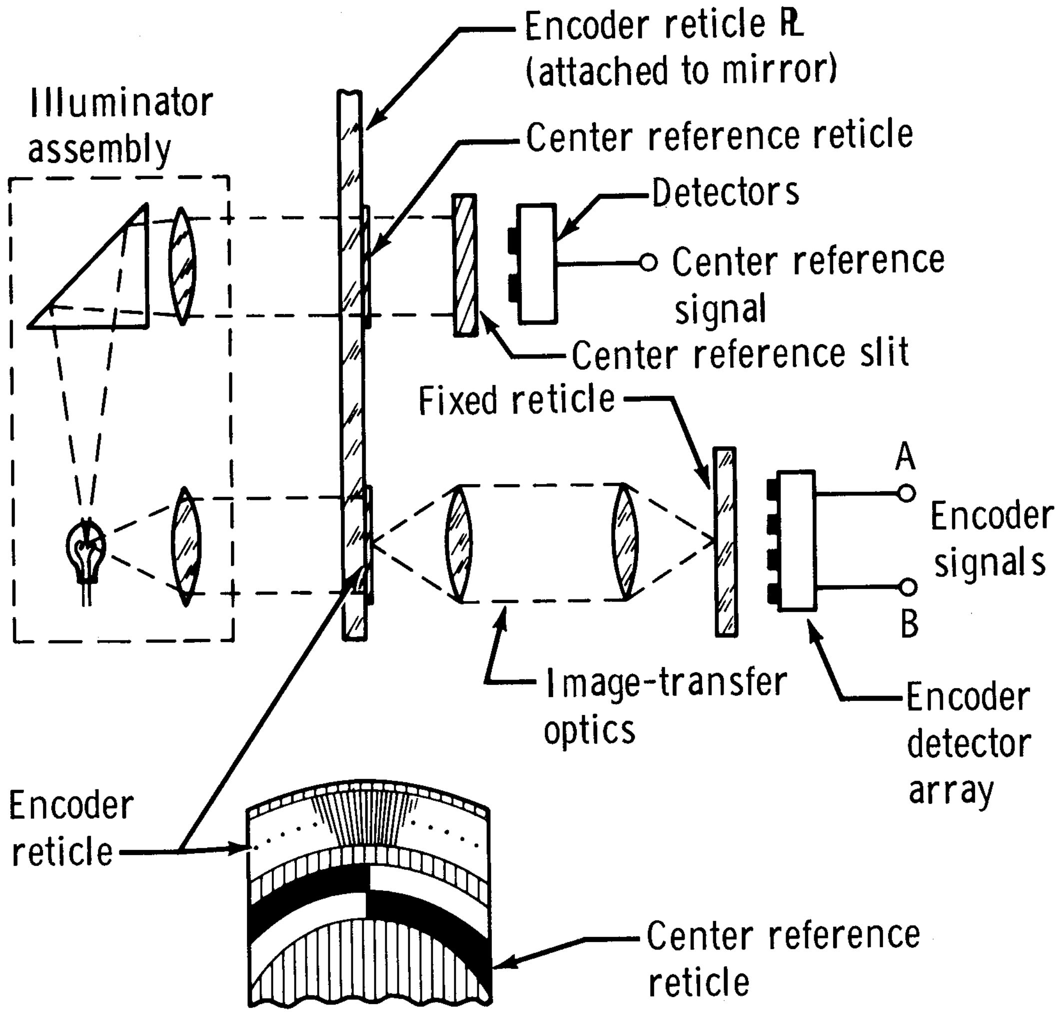
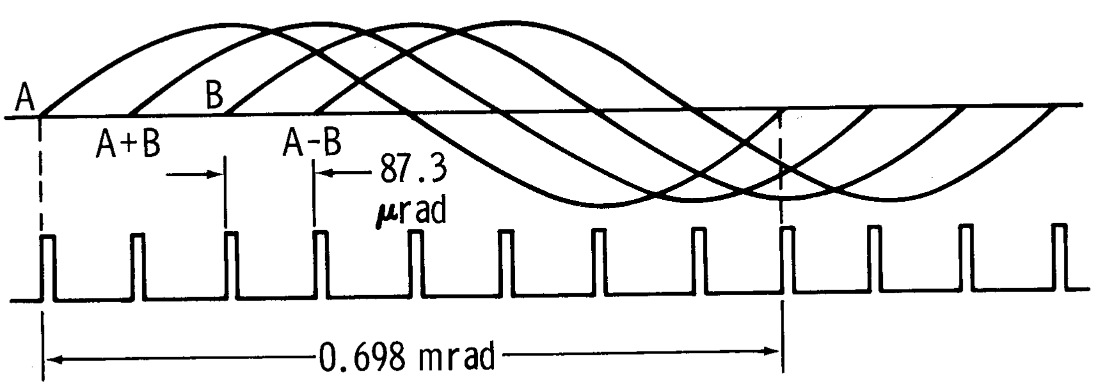


Figure 4. - Analytical model of torsion bar for stress and deflection calculations.



(a) Encoder system.



(b) Encoder-pulse formation.

Figure 5. - Optical-angle encoder system and pulse formation.