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A VIBRATING MIRROR SYSTEM SUITABLE FOR Q-SWITCHING LARGE-APERTU--ETC(U)

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MELBOURNE, VICTORIA

REPORT

MRL-R-700

A VIBRATING MIRROR SYSTEM SUITABLE FOR Q-SWITCHING LARGE-APERTURE LASERS

P.J. Beckwith

Approved for Public Release

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Resonant vibrating mirrors provide a convenient means of Q-switching a laser, but large-aperture versions require careful design if the drive power is not to become excessive. This report outlines the design principles involved in the optimisation of moving-iron galvanometer drivers, and describes a prototype device with an aperture of 40 mm x 80 mm which is capable of beam deflections of ± 40 mrad at 800 Hz. Some suggestions are made concerning more refined designs.

GLOSSARY

Φ	Total magnetic flux (Wb)
B	Magnetic flux density (T)
B_{sat}	Saturation flux density ($\sim$ 0.6 T in iron)
n	Number of turns in magnetic winding
I	Winding current (A)
S	Magnetic reluctance (A Wb^{-1})
μ_o	Permeability of free space (1.26×10^{-6} H m^{-1})
F	Magnetic force on armature poles (N)
U	Energy absorbed over one mechanical cycle (J)
L	Inductance (H)
C_e	Torsional compliance (rad $N^{-1}m^{-1}$)
G	Modulus of rigidity (8×10^{10} N m^{-2} for steel)
J	Polar moment of inertia (kg m^2)

C O N T E N T S

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A VIBRATING MIRROR SYSTEM SUITABLE FOR Q-SWITCHING LARGE-APERTURE LASERS

INTRODUCTION

A series of short, high-power pulses of laser radiation is often found to be more efficacious for the cutting and trimming of materials than the same average power at a constant level (1). Such a pulse train can be derived from a cw laser by sharply modulating the loss factor of the resonator, so that oscillation is possible for only a small proportion of the time. The energy stored in the active medium of the laser between the periods when oscillation is permitted is then quickly released, producing short pulses with high peak power levels. This technique is known as Q-switching.

Q-switching is often achieved by the use of a rotating multifaceted mirror as part of the laser resonator, which is thus periodically aligned for brief intervals. Such switches have a very low insertion loss and high resistance to optically-induced damage. However, they involve the use of precision high-speed bearings and high-quality, multifaceted optical components, both of which are expensive. In addition, bearing wear can be a problem at the high rotational speeds that are often required. These disadvantages are avoided in a system whereby one of the laser mirrors is rotationally vibrated at half the desired repetition rate of the output pulse train (2). Only one figured surface is then necessary, and wear can be eliminated by a torsional suspension, which is normally made resonant so as to increase the amplitude of vibration. However, one obstacle to the use of vibrating mirrors in large-aperture resonators is that their drive power requirement depends on their moment of inertia, and rises very rapidly with increasing size. A mirror of diameter d, thickness t and mass m has a moment of inertia proportional to md^2. Its maximum usable operating frequency depends on the frequency of its fundamental transverse resonance, which is proportional to $t^{\frac{1}{2}} d^{-1}$. If this is held constant, then $t \propto d^2$, $m \propto d^4$ and the moment of inertia of the mirror is proportional to d^6.

This strong dependence means, for example, that a vibrating mirror with a 30-mm aperture will require approximately 1 kW of drive power to equal the performance of a 10-mm mirror driven with 1 W. It follows that for large apertures it becomes increasingly important to maximise the efficiency of the drive unit and to minimise the loss of vibrational energy to the support frame. Furthermore, extreme forces are sustained by both the mirror and the drive unit, and considerable rigidity and mechanical strength are needed by these elements.

MAGNETIC DRIVER

The requirement for mechanical strength precludes the use of moving-coil galvanometer movements, and moving-iron designs are normally chosen for this application. One such drive unit is shown in Fig. 1. This has a simple dipole armature in an unpolarised yoke. The attractive forces across the poles vary as the square of the magnetic flux through the core (3), so that the frequency of the current in the drive coil is one-half of the resonance frequency of the moving system.

Energy is coupled into the moving-iron armature by modulation of the magnetic field and it is desirable that this coupling should be maximised. While it is, in principle, possible to compensate for loose coupling by resonant tuning of the electrical drive circuit, such an approach would require a high electrical Q and large circulating currents. Further, the interaction of two sharply tuned resonant circuits (one electrical and one mechanical) presents tuning problems.

As stated in Appendix 1, good coupling requires that the magnetic reluctance of the air gaps should be greater than that of the rest of the magnetic circuit. The degree of coupling is then shown to be approximately proportional to α, the modulation index of the gap thickness in the useful range of $0 < \alpha < 0.5$. The gap thickness should, therefore, be as small as possible consistent with the requirements (Appendix 1) that $B < B_{sat}$ and that the total reluctance should be principally dependent on the reluctance of the air gaps.

MECHANICAL SYSTEM

The requirements that the loss of energy from the moving elements to the support frame should be minimised has led to the design illustrated in Fig. 2.

The mirror and armature rotate in opposite senses, so that their total angular momentum is nearly zero at all times and thus the reaction forces transmitted to the frame are small. They are coupled together by a steel rod which locates and supports the mirror and which serves as the compliance element in the resonant system. Vertical support is provided by a thinner and more compliant steel rod mounted coaxially with the coupling rod as shown in Fig. 2. This rod is also required to stabilise the armature against any lateral perturbations which may arise from asymmetries in the location of the armature in the magnet gap. The principal lateral support is provided by a rubber bush located approximately at the node of the coupling rod. Very little energy is lost through this bush, not only because of the small displacements which occur near it, but also because of the large acoustical impedance mismatch between the steel of the rod and the rubber.

The torsional compliance of the coupling rod is given by (4)

$$C_\theta = \frac{32\ell}{\pi G D^4}$$

where ℓ is the total free length of the rod, D is its diameter, and G is the modulus of rigidity, which may be taken as 80 GN m^{-2} for steel. The resonance frequency ω_o of the moving elements is given by

$$\frac{\omega_o^{\,2} J_A \ell_A}{D^4} = \frac{\omega_o^{\,2} J_m \ell_m}{D^4} = \frac{\pi G}{32}$$

where J_A, J_m are the moments of inertia of the armature and mirror respectively and ℓ_A, ℓ_m are the distances from the armature and mirror to the node.

The length of the rod may be adjusted to allow any convenient choice of D satisfying the relation.

$$\frac{\ell}{D^4} = \frac{\pi G\, C_\theta}{32}$$

It should be noted, however, that as D is decreased the elastic energy per unit volume which must be stored in the metal will increase, and this will eventually lead to loss of Q and failure due to fatigue.

The energy loss per cycle in this system will be minimised if the moment of inertia of the armature is considerably greater than that of the mirror. In this case the armature velocity will be less than the mirror velocity by a factor J_A/J_m, and there will be a correspondingly lower energy loss through the lower support. In addition, since the torsional strain energy per unit length of the coupling rod is constant for a given mirror velocity, the energy lost in internal friction in the rod will be minimised if the total rod length is minimised. Again, this will be achieved by the use of a high-inertia armature.

It will be noted that the requirement of high armature inertia is incompatible with the requirement of high armature deflection derived in Appendix 1. Some compromise has to be reached on this point. The compromise chosen in each individual case should be based on the dimensions of the mirror and its required vibration amplitude. However, in general terms, a system where the armature inertia is between one and two times the inertia of the mirror should perform adequately.

ELECTRICAL SYSTEM

It is shown in Appendix 2 that the degree of magnetic coupling is independent of the inductance of the drive coil, which can therefore be chosen to suit the characteristics of the amplifier to be used. It has been found convenient to incorporate the drive coil into a series L-C resonant circuit which is driven by a square-wave voltage. The alignment problems which may occur with tuned drive systems do not become serious if there is good magnetic coupling to the armature, since maximum efficiency can then be reached with a low electrical Q. Tuning is generally desirable since, firstly, a large quadrature current must otherwise be supplied by the power amplifier, and, secondly, the required sinusoidal field current can be derived from a square-wave voltage drive. This enables a high-efficiency switching power amplifier to be used, and also simplifies the design of a self-excitation circuit.

If required, the system can be self-excited by driving it with a signal dependent on the position or velocity of the mirror or armature. If the phase relationships are correct and there is sufficient loop gain, the system will oscillate with an amplitude dependent on the available drive power. In the present case, the drive frequency is half the feedback frequency, which must therefore be divided by two. This is a straightforward process with a square-wave drive.

DESCRIPTION OF PROTOTYPE

An operating oscillating mirror system has been constructed as in Fig.3. The armature and yoke were formed from c-core laminated stock to minimise eddy current losses. The minimum cross-section of the iron was 480 mm^2 and the air-gaps were each 620 mm^2 in area and 0.30 mm thick, implying a maximum usable drive (given by equation (1-4) Appendix 1) of 220 ampere-turns, at a saturation flux density of 0.6 T. It will be noted that this estimation is only valid for small amplitudes of vibration and should be treated as approximate. The shape of the armature (and stator) poles closely approximates an equiangular spiral such that the angle between the radius through any point on the pole surface and the normal to that point is 0.2 radians. This implies that the armature is allowed a maximum tangential displacement of ± 1.5 mm at its periphery, corresponding to ± 0.1 rad. In the useful range of $0 < \alpha < 0.5$, vibrational amplitudes of up to ± 50 mrad are therefore allowed. The total amount of inertia of the armature and clamping boss is approximately 1.2×10^{-5} kg m^2.

The mirror currently in use was machined from aluminium alloy bar stock, nickel plated and gold-coated on its reflecting surface, and has the overall dimensions 6 mm x 40 mm x 80 mm. It was fixed with cyanoacrylate adhesive to a mounting block which in turn was silver-soldered to the end of the steel rod which couples the mirror to the armature. The support block and mirror together are in dynamic balance around an axis coaxial with the connecting rod, and have a total moment of inertia of approximately 1.6×10^{-5} kg m^2.

The coupling rod was made from ¼ inch (6.3 mm) diameter silver steel. Its free length is 90 mm, so that the predicted resonance frequency of the system is 700 Hz. The maximum torque which the rod must sustain, calculated from its torsional stiffness and peak deflection, is approximately 15 Nm. The support rod is 2.2 mm in diameter and 20 mm long. It is, therefore, more than an order of magnitude more compliant than the connecting rod and only slightly affects the motion of the armature.

The prototype device is sub-optimal in that the mirror inertia is significantly greater than that of the armature, a condition which arose from rigidity problems in the mirror and mounting block. Mirror deflections of 20 mrad (peak) have, however, been achieved with an input power of 100 W rms, the limiting factor being core saturation. The resonance frequency has been measured to be 800 Hz, in satisfactory agreement with the predicted figure.

It will be noted that, since the magnetic forces at the armature poles are at all times attractive, the mirror will experience a static deflection in addition to the vibrational one. The force producing this static deflection is equal in magnitude to the peak oscillatory force, as is shown in equation (1-7), Appendix 1. The average mirror deflection is consequently offset by an amount proportional to the peak deflection, and depends mainly on the compliance of the support rod and the mechanical Q of the resonant system. In Q-switching applications this need not be a problem if the relative offset is small, since it will then only introduce a small modulation of the intervals between pulses, which could (if desired) be compensated for by introducing a contrary offset in the initial orientation. The offset in the prototype is approximately 10% of peak deflection, which would introduce a variation in successive pulse intervals of only ± 6%.

It is possible to eliminate the static deflection effect by the use of more complicated armature designs. Two possible quadrupolar systems are shown in Fig. 4. Both have disadvantages of complexity. The design of Fig. 4(a) requires quadrature drive currents at half the mechanical resonance frequency, while the design of Fig. 4(b) requires permanent magnets sufficiently powerful to bias the iron core to about half its saturation flux density. Either design could, however, be adopted for applications in which the offset inherent in the dipole armature was unacceptable. In this case, the design procedure outlined in this report can be applied to the more complex case with little modification.

CONCLUSION

The design principles involved in the development of a large-aperture vibrating mirror system have been outlined. A description has been given of a prototype device with a reflective surface of 40 mm x 80 mm, which is capable of producing beam deflections of ± 40 mrad at 800 Hz, but which experiences a static deflection of about 10% of peak deflection. Some suggestions have been made for more complex devices which do not exhibit this latter effect.

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APPENDIX 1

MAGNETIC COUPLING TO THE ARMATURE

Good magnetic coupling requires that the core flux should be influenced as much as possible by the position of the armature, and should therefore be principally determined by the reluctance of the air gaps (4), so that

$$\Phi = A_I B_I \simeq \frac{nI}{S_A} \tag{1-1}$$

where A_I is the iron cross-section

B_I is the iron flux density

and the air reluctance is (6)

$$S_A = \frac{2g}{\mu_o A_A} \tag{1-2}$$

g is the air gap thickness

A_A is the air cross-section.

The air gaps are designed to have a uniform thickness, and may be considered to be bounded by magnetic equipotential surfaces (6). The flux density in the gaps will then be uniform, and conditions of continuity require that

$$B_A = B_I \frac{A_I}{A_A} \tag{1-3}$$

and, therefore,

$$B_I = \frac{nI\mu_o}{2g} \frac{A_A}{A_I} < B_{sat} \tag{1-4}$$

The magnetic attractive force on each pole is given by (3)

$$F = \frac{B_A{}^2 A_A}{2\mu_o} \tag{1-5}$$

whence

$$F \simeq \frac{n^2 I^2 A_A \mu_o}{8g^2} \tag{1-6}$$

if $$I = I_o \cos \tfrac{1}{2}\omega_o t$$

$$I^2 = \tfrac{1}{2} I_o^2 (1 + \cos\omega_o t)$$

then $$F \simeq \frac{n^2 I_o^2 A_A \mu_o}{16 g^2} (1 + \cos\omega_o t) \tag{1-7}$$

In a resonant system such as is described in this report, the motion of the rotor may be taken to be simple-harmonic. The velocity of the rotor poles will (at resonance) be in phase with F so that

$$g = g_o (1 - \alpha \sin \omega_o t) \tag{1-8}$$

where $o < \alpha < 1$

$$\dot{g} = -g_o \alpha \omega_o \cos \omega_o t \tag{1-9}$$

Thus the energy absorbed by the rotor over one cycle is

$$U = \int_o^{\frac{2\pi}{\omega_o}} -2 F \dot{g} \, dt$$

$$= \frac{n^2 I_o^2 A_A \mu_o \alpha \omega_o}{8 g_o} \int_o^{\frac{2\pi}{\omega_o}} \frac{\cos^2 \omega_o t}{(1 - \alpha \sin \omega_o t)^2} dt \tag{1-10}$$

which approaches $\dfrac{n^2 I_o^2 A_A \mu_o \alpha \pi}{8 g_o}$ for small α.

The integral term in equation (1-10) has been evaluated numerically, and has been found to be approximately proportional to α below $\alpha = 0.6$, where it has the value $1.4\ \pi/\omega_o$. Above about $\alpha = 0.8$ its value rises rapidly, approaching infinity as $\alpha \to 1$. However in this latter range the rotor is close to collision with the stator poles and in practice its amplitude of vibration usually has to be kept below $g_o/2$. It is, therefore, reasonable to make the approximation

$$U \simeq \frac{n^2 I_o^2 A_A \mu_o}{2 g_o} \alpha \tag{1-11}$$

or, writing (7)

$$L \simeq \frac{n^2}{S_A} \simeq \frac{n^2 \mu_o A_A}{2g_o} ,$$

$$U \simeq I_o{}^2 L\alpha \tag{1-12}$$

This term is a measure of the degree of coupling between the magnetic energy in the stator and the mechanical energy of the armature. It will be maximised when the modulation index α is large.

APPENDIX 2

EFFECT OF COIL INDUCTANCE ON THE MAGNETIC COUPLING

Equation (1-12) Appendix 1 gives the energy coupled into the armature in each cycle as

$$U \simeq I_o^2 L\alpha$$

The condition that the core should be unsaturated was given in equation (1-4) as

$$\frac{nI\mu_o A_A}{2g\ A_I} < B_{sat} \qquad \text{for all I}$$

Since $I \leqslant I_o$, the condition becomes

$$\frac{n^2\ I_o^2}{A_I^2\ S_A^2} < B^2_{sat} \tag{2-1}$$

or, substituting $L \sim \frac{n^2}{S_A}$ (Appendix 1).

$$\frac{I_o^2\ L}{2}\ \frac{\mu_o\ A_A}{g\ A_I^2} < B^2_{sat} \tag{2-2}$$

and hence $$U < \frac{2g\ A_I^2\ \alpha\ B^2_{sat}}{\mu_o\ A_A} \tag{2-3}$$

which is independent of n.

This result indicates that the degree of magnetic coupling of the device does not depend on the inductance of the drive coil.

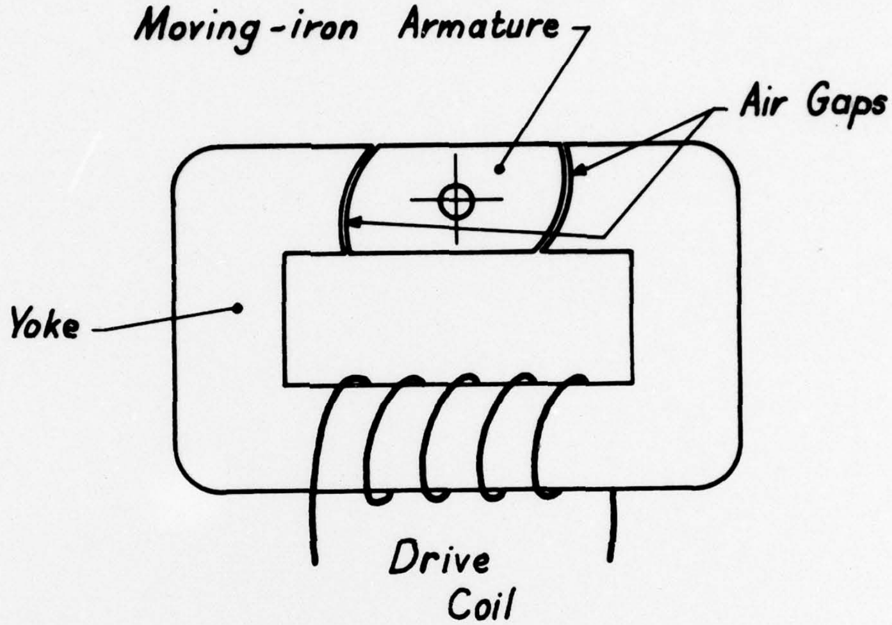


FIG. 1 - Dipolar moving-iron galvanometer movement suitable for use with a vibrating mirror.

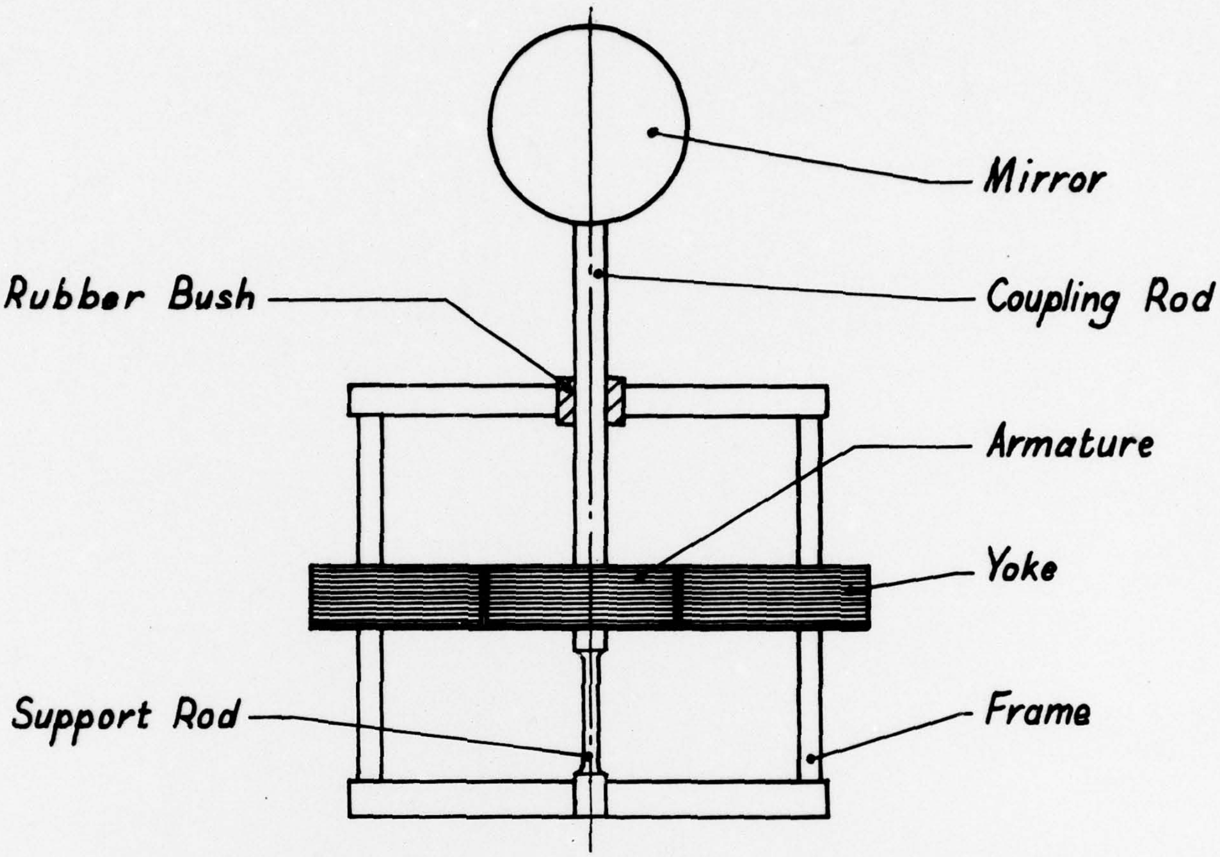


FIG. 2 - Diagram of vibrating mirror system (not to scale).

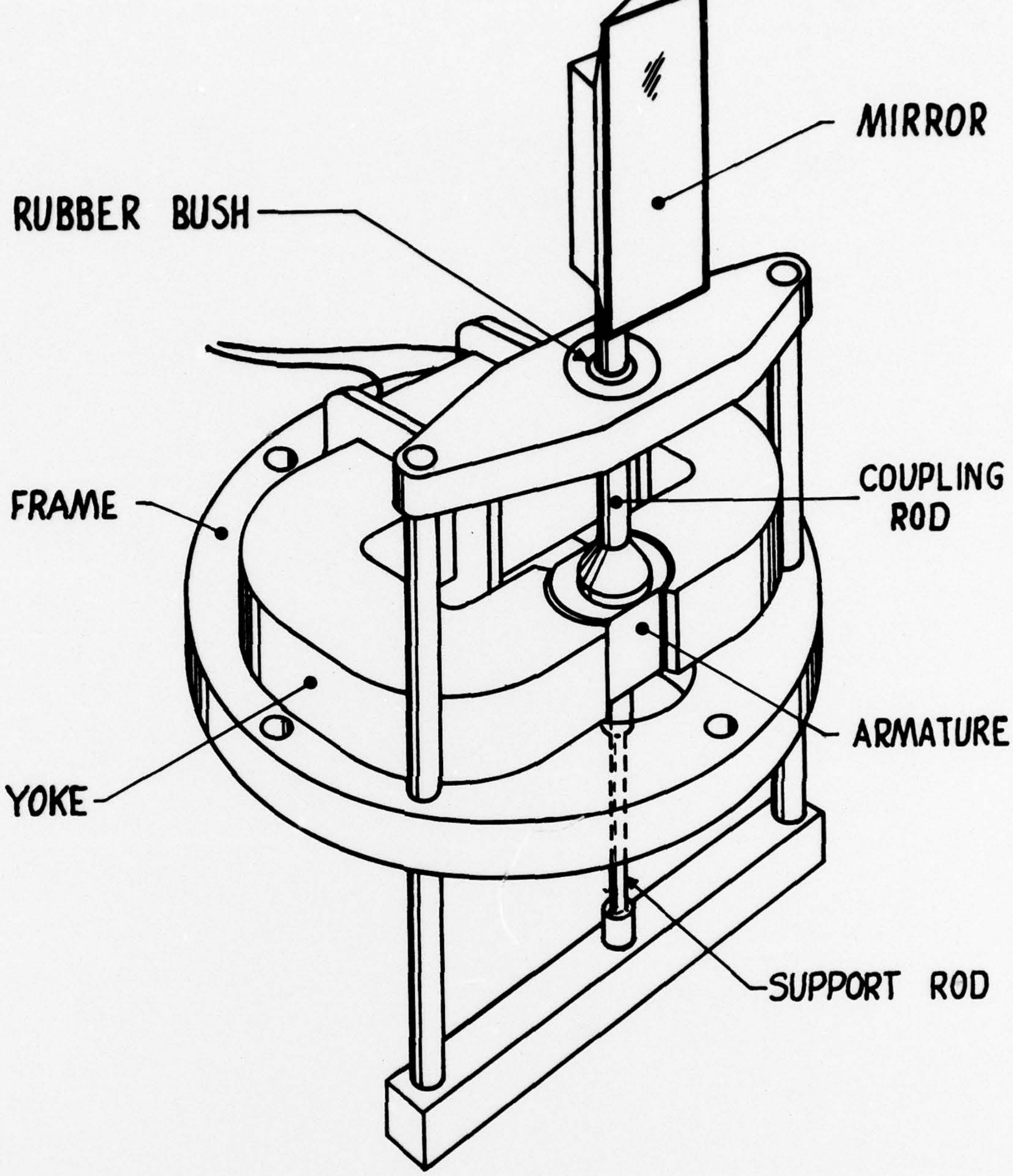
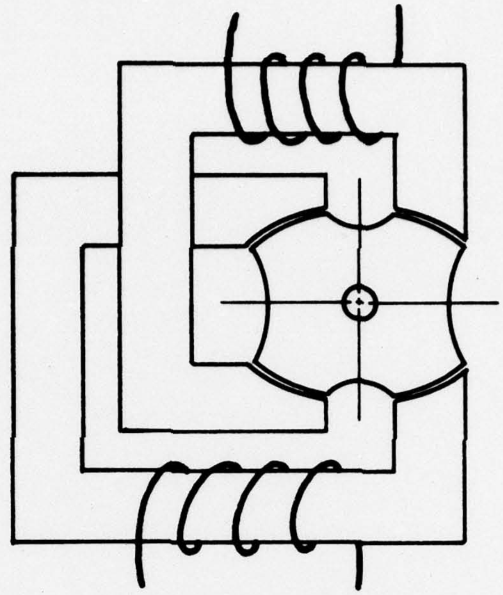


FIG. 3 - Prototype device.

(a)



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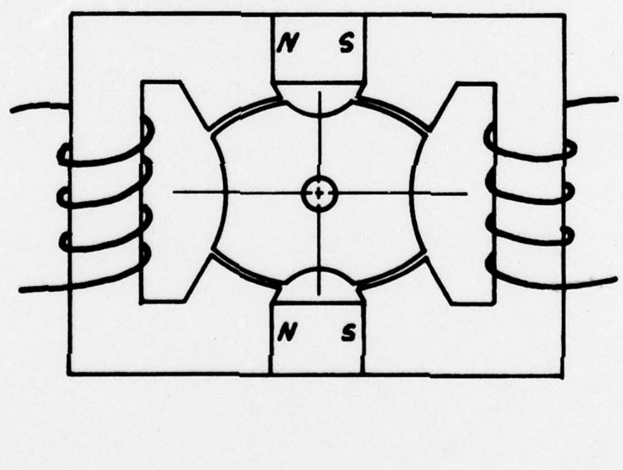

FIG. 4 - Quadrupolar drive systems.

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