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by

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TO MEASURE TRANSIENT FORCES
IN A SMALL ROTATING SHAFT

by

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ABSTRACT

This thesis is concerned with the development of a device which is sensitive enough to measure the transient forces in the rotating propeller shaft of a small (5 or 6 foot) self propelled ship model.

A number of instrumentation possibilities are considered. The more promising schemes are analyzed in some detail.

The project undertaken in the body of the thesis is the development of a thrust pickup using Ferroxcube pot cores as the actuating device.

Two coils are wound inside Ferroxcube pot cores. The pot cores are spring mounted on the rotating shaft in such a manner that thrust will cause an increase in the air gap of one pot core and a decrease in the air gap of the other pot core. The changes in air gaps cause changes in the coil inductances. Leads are taken from the coils via slip rings to two oscillating circuits whose frequencies of oscillation depend on coil inductances. As a thrust is applied, the two frequencies of oscillation change in the opposite direction. The difference in frequency, which is an indication of thrust, is measured.

Test results show that the device will deliver 2 millivolts output per gram of thrust. Physical deflection caused by 100 grams of thrust is too small to be measured by a dial gage. The output is linear over a thrust region from 0 to 300 grams.

This system has very low energy absorbtion, delivers a large output signal, and is linear over the range of interest.

The system, as installed, is not capable of measuring transient forces in a rotating shaft due to the dynamic difficulties apparently arising from the commutator assembly.

Numerous recommendations are given for future research, with special emphasis on a geared torque measuring dynamometer.

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Title: Professor of Naval Architecture

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Mr. Robert Singleton, of David Taylor Model Basin, for his idea and initial construction of a Ferroxcube measuring device.

Numerous classmates, for their help and suggestions in solving analytical problems.

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I. INTRODUCTION

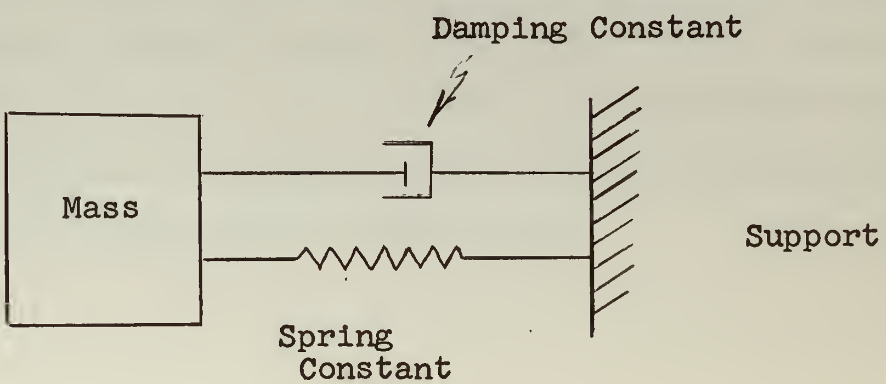
The measurement of force exerted upon a body by some external agent is one of the primary sources of engineering knowledge. Literally hundreds of devices have been designed to do this job on a great variety of structures subjected to an even greater variety of stresses.

The problem is a knotty one because the measuring device must be so designed as to have a negligible effect on the value of the force being measured. In the case of steady state forces this criteria is not difficult to meet; however, under transient conditions this restriction often defeats the best engineering efforts.

In order to decrease the effect of the dynamometer upon the transient force being measured, the energy storing capacity of the measuring device must be small with respect to that of the system. This statement implies a large (stiff) spring constant and a small displacement within the dynamometer. Perhaps the easiest way of visualizing this problem is to note that the majority of physical systems can be represented by a mass attached to a support by means of a spring in parallel with a damping device. (1)

FIGURE I

Single Degree of Freedom, Damped Mass-Spring System



[illegible]

[illegible]

[illegible]

[illegible]

This system has a second order response described by the following equation:

$$\frac{d^2x}{dt^2} + \frac{b}{m} \frac{dx}{dt} + \frac{k}{m} x = 0$$

The Bode plot shows a reasonably flat response out to the resonant frequency, at which point the amount of over or undershoot is determined by the degree of damping. At frequencies above the resonant frequency, the response drops off at 40 decibels per decade (see Appendix A).

Thus in the design of any dynamometer, the highest transient frequency to be measured must be determined and the measuring device constructed so as to make the system response "break" frequency substantially higher than this theoretical maximum. In other words, the complete system must mechanically pass the force frequencies to be measured (see Appendix A).

In the case of a moving or rotating system, an additional problem arises in that the dynamometer must develop signals which are capable of being delivered to a stationary recording instrument. This complication implies the use of slip rings and brushes, or possibly more exotic schemes such as radio linkage or transformer action between a rotating and stationary coil.

This thesis is concerned with the development of a device which is sensitive enough to measure the force in the rotating propeller shaft of a small (5 or 6-foot) self-propelled ship model. Typical values of full load steady state thrust and torque are .25 pounds and 1.0 inch-ounces, respectively.

[illegible]

It was anticipated that an instrument of the following characteristics would be designed, constructed and dynamically tested:

1. High degree of accuracy, linearity and sensitivity.
2. Rugged.
3. Easily installed in a model.
4. Capable of one man operation.

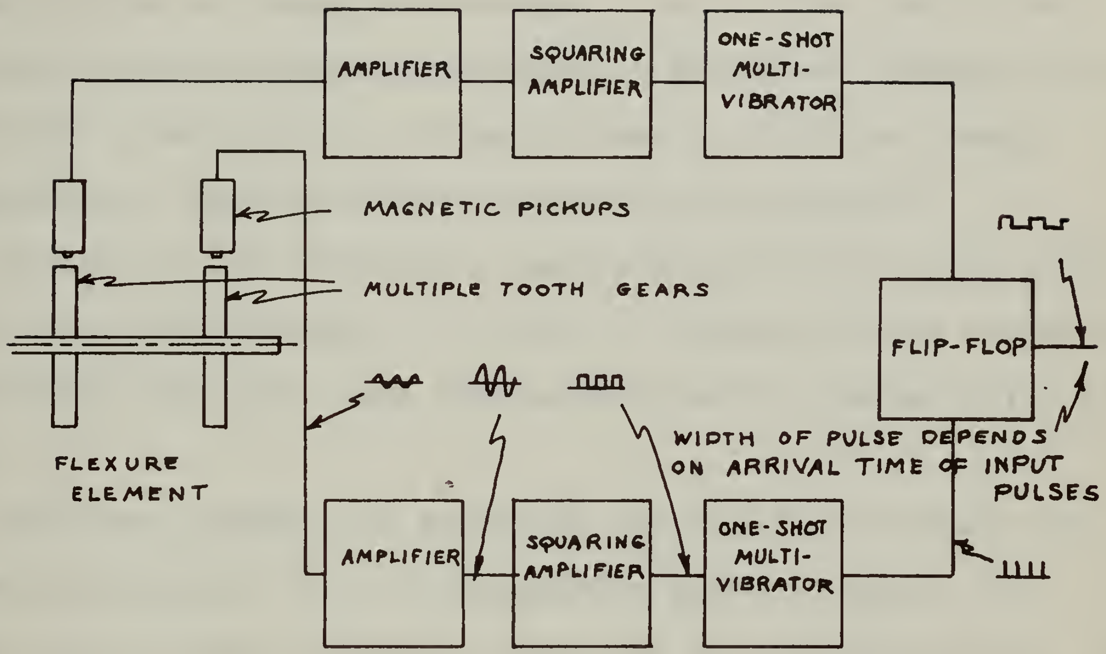
A number of approaches to the problem were considered. Some of the more promising methods will be briefly discussed.

Gears can be used in several configurations to measure thrust and torque. The design in Figure II was proposed by M. Wilson of David Taylor Model Basin. It was "breadboarded" and found to be satisfactory in every respect except sensitivity. Using 100-tooth, 2" diameter gears, a full load deflection of between .01" and .015" was necessary for adequate sensitivity. This amount of deflection led to a spring constant which is much too low for transient force measurements, therefore this idea was dropped. See Appendix A for details.

Reference (2) describes a circuit using magnetic pickups in the configuration shown in Figure III. This idea might work, but time and monetary limitations mitigated against investigations of this method.

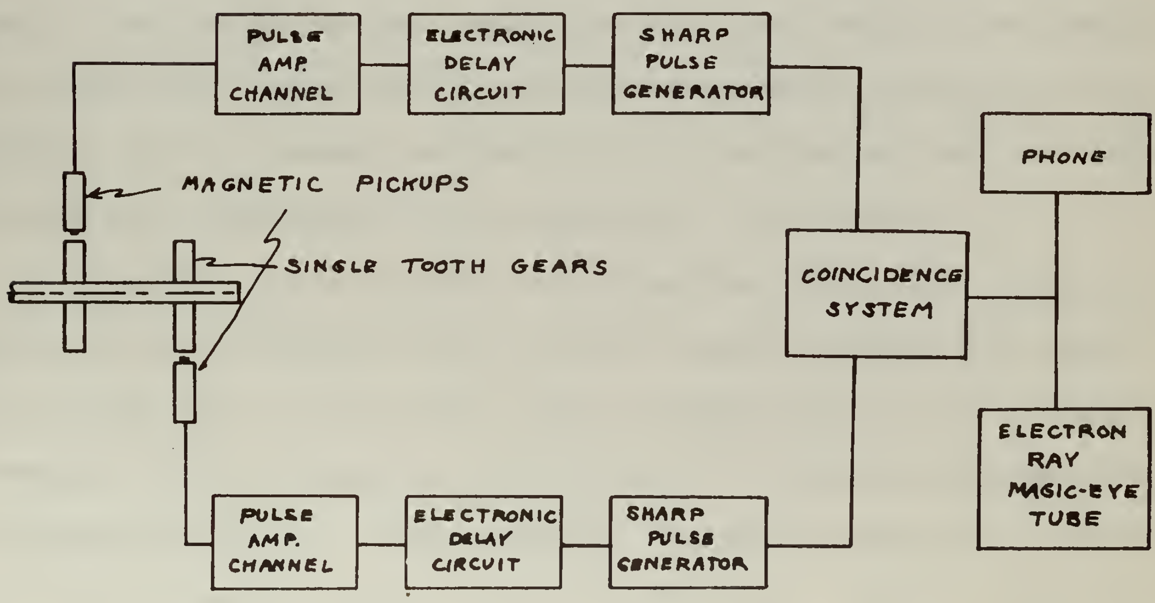
Piezoelectric materials, in which mechanical motion produces a voltage, were considered. These materials were rejected because at low frequencies current masks the response and also because they are adversely affected by temperature and humidity.

FIGURE II



BLOCK DIAGRAM OF GEARED TORQUE DYNAMOMETER

FIGURE III



USE OF SMALL TIME INTERVALS TO MEASURE TORQUE

Reference (3) is an excellent paper relative to the use of magnetostriction in torque measurement. One of the conclusions reached in this discussion is that the Wiedemann Effects have a practical lower limit of 10 inch-ounces in transient torque measurements, therefore this method was not pursued.

Reference (4) describes a semiconductor strain gage with a very large output signal. A letter of inquiry to the author disclosed the fact that these devices are not yet commercially available.

Previous attempts at designing and testing a thrust and torque dynamometer for self-propulsion tests of small ship models (5,6,7) have centered around the use of strain gages, for torque sensing, and a differential transformer, for thrust measurement. These devices convert physical strains and displacements into electrical signals which were removed from the shaft by means of a commutating assembly. No satisfactory results have been obtained. The system spring constants were too small to be useful in an instrument detecting transient frequencies present in the shafting. In all cases, excessive slip ring noise and low signal amplitude have persisted. See Appendix A for details.

In general, these recent investigations have been severely limited by the assumption that torque losses through the stern tube bearing were of the same order of magnitude as the torque to be measured, even though no calculations or data were presented to substantiate this. This required a torque sensing element between the aftermost bearing and the propeller. Such a location

leads to excessive whirling, size limitations, and water leakage problems into the element itself.

A preliminary calculation (see Appendix A) indicated a torque loss of 10^{-5} inch-ounces in a lightly loaded sleeve bearing. It, therefore, appeared reasonable to the present authors that the torque sensing element might be placed inside the model, forward of the aftermost bearing.

Transient fore and aft movement of the shaft was assumed to be so small as not to be restricted by precision ball bearings.

It was concluded that both displacement sensing elements (axial movement for thrust and radial movement for torque) could be placed within the model.

The method finally selected for investigation was originally the idea of Robert Singleton of David Taylor Model Basin. The general idea was to develop a variable inductance device whose value of inductance is determined by an air gap. A small change in the air gap changes the inductance. The coil is in an oscillating circuit whose frequency of oscillation is determined by this inductance. Changes of frequency will then be an indication of displacement.

II. PROCEDURE

The first problem was to find a device whose material and configuration permitted a small change in displacement to produce a large change in inductance. Robert Singleton, of the David Taylor Model Basin, suggested pot cores, produced by the Ferroxcube Corporation of America. He had determined that such a core, when placed in an oscillating circuit (described below), caused a substantial change in the frequency of the circuit for very small changes in air gap.

Accordingly, a variety of pot cores and bobbins were ordered. See reference (9) for details. It was discovered that several of the bobbins would fit entirely inside a half section of one of the pot cores (Figure XVII). With this in mind, a thrust pickup configuration, as shown in Figure IV, was designed and constructed. This pickup design allowed one air gap to increase and the other to decrease, when an axial force was applied. It was hoped that the eventual output signal would be proportional to the difference of the two frequencies in the oscillating circuits.

The electronic instrumentation suffered, in general, from monetary limitations. It was decided that the circuitry should be placed within the model. This meant that transistorized circuits had to be used in lieu of the less expensive tube circuits. Ready built "plug in" units were available, but much too expensive; therefore individual components were purchased, and the circuits constructed therefrom.

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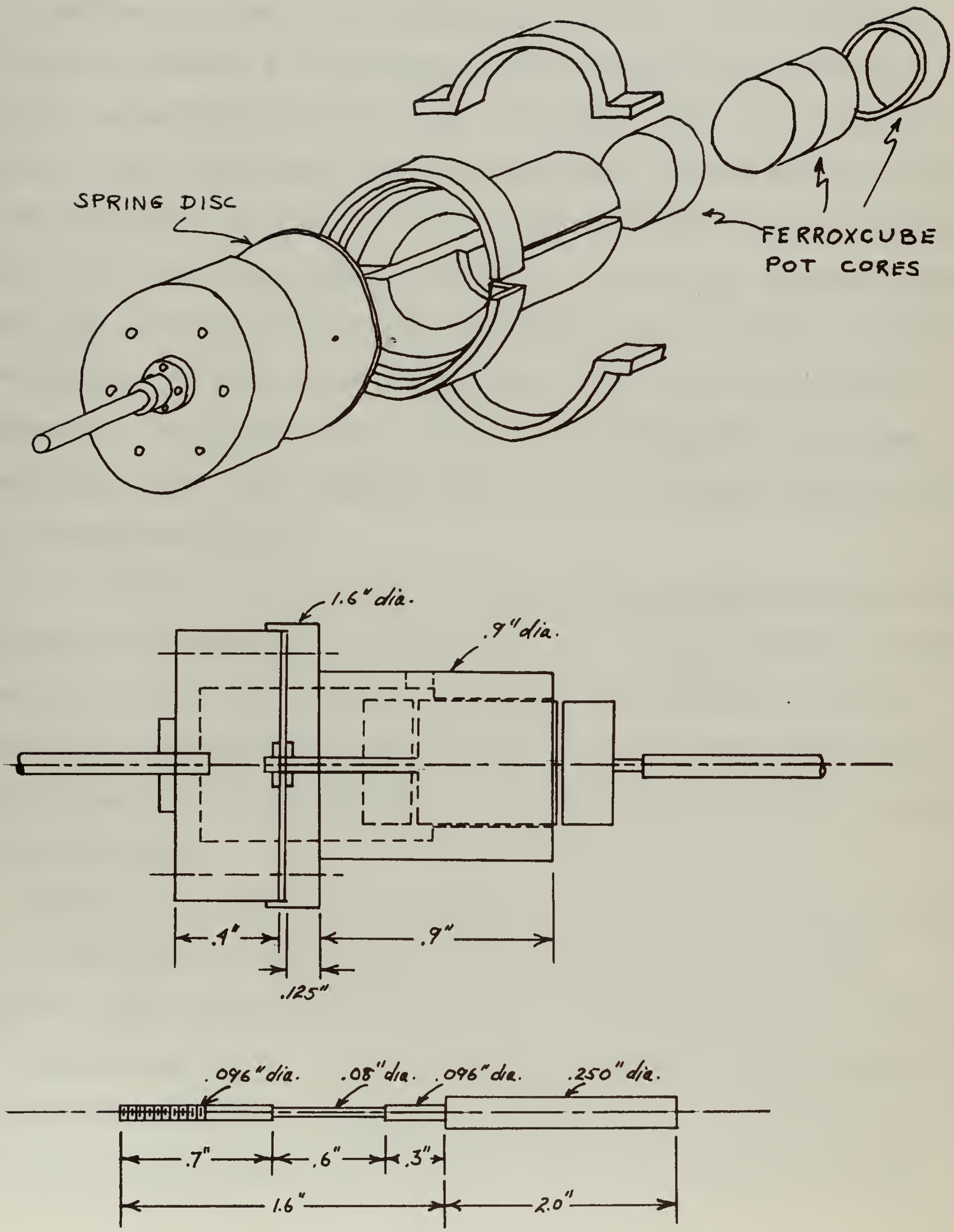
FIGURE IV



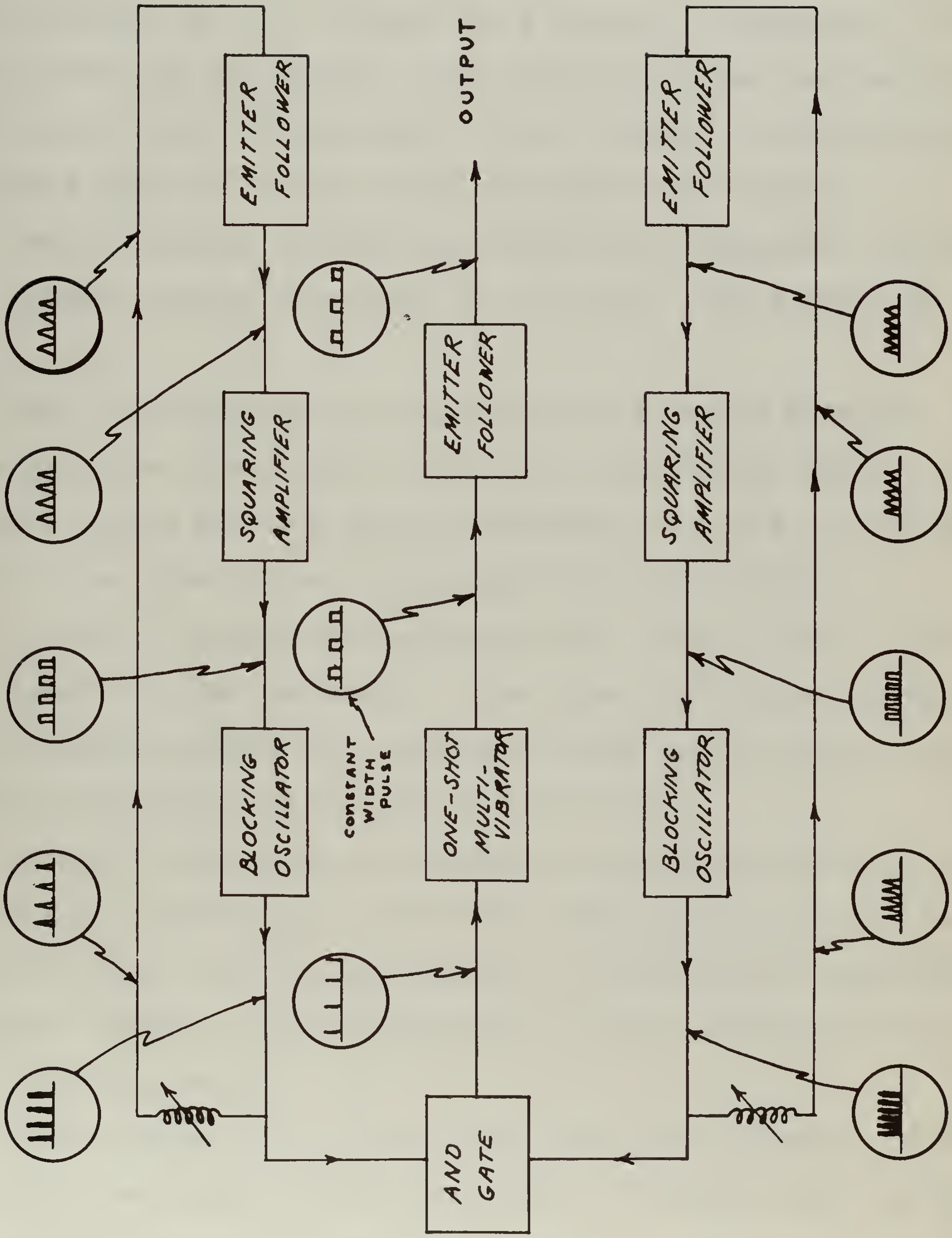
THRUST ELEMENT ENCLOSURE

The oscillating circuit as devised by Mr. Singleton is shown as part of Figure V and is composed of the blocking oscillator, coil, emitter follower, and squaring amplifier. The blocking oscillator produces a high energy pulse every time it receives positive going signal with a fast rise time. The coil acts to delay the pulse, the amount of delay depending on the air gap spacing. The delayed pulse arrives at the emitter follower, which has a gain of one and a very high input impedance. This high impedance should remove the effect of any stray resistance such as might be induced by variations in slip ring resistance. The squaring amplifier "squares up" the pulse form, which has deteriorated in passage through the coil. The squared pulse triggers another pulse from the blocking oscillator.

The problem remained of creating a signal proportional to the difference in frequencies of the two oscillating circuits. After several failures, the central portion of the diagram shown in Figure V was devised and successfully tested. This part of the circuit consists of the AND gate, one-shot multivibrator, and the emitter follower.

The AND gate delivers an output only when the pulses of the two blocking oscillators occur at the same instant in time; therefore, the output of the AND gate is dependent on the relative frequencies of the two oscillating circuits. The one-shot multivibrator delivers a pulse (whose duration is specified by an external capacitor) every time it receives a pulse from the

FIGURE V



BLOCK DIAGRAM (WITH WAVE FORMS)

AND circuit. The frequency of the constant duration pulses leaving the one-shot multivibrator determines the d.c. level of voltage of the output. The pulse duration is so adjusted as to yield a maximum change in d.c. voltage for a change in frequency in the pulses from the AND circuit. The emitter follower was installed to eliminate the loading effect of any external circuits (such as a filter) upon the output of the one-shot multivibrator.

The 9 circuits involved were built from components and wired in a compact housing measuring 3" x 4" x 5". See Figures XX and XXII.

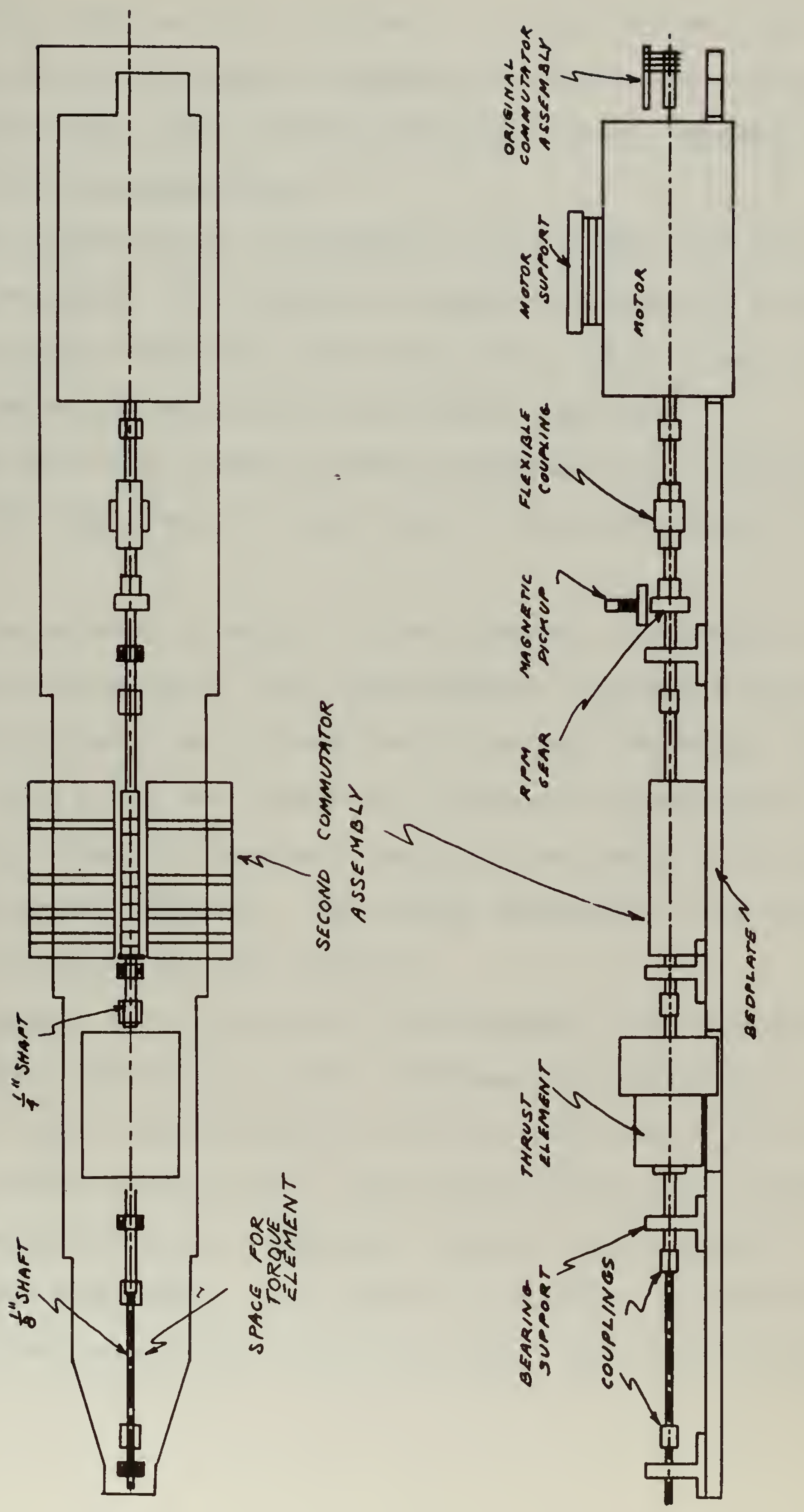
The thrust element was mounted on an aluminum bedplate. The plate was later fitted with a small d.c. propulsion motor. A magnetic pickup and spur gear combination was used to determine shaft r.p.m. (see Figure VI and Appendix B for details).

A pair of bobbins was wound with 250 turns of No. 38 wire and inserted in the pot cores. When these coils were connected to the instrumentation the resulting output varied as much as 6 volts for small hand changes of the air gap.

It was thought that an increase in the number of turns would increase the sensitivity; therefore, 325 turn bobbins were installed in the thrust pickup housing. The miniature commutator (slightly damaged) used in Reference (7) was mounted on the forward end of the motor.

Static tests of the device were quite disappointing in that the output was much less than expected. A dynamic test was attempted, but rotation of the shaft caused the output to vary

FIGURE VI



PHYSICAL LAYOUT

erratically with no force applied. It was concluded that the difficulty lay in the damaged commutator assembly and perhaps in the excessively long leads (2 feet) from the coils, through the shafting, to the instrumentation.

The mechanism was disassembled and the 325 turn coils tested by hand movement. The output was again very small. A third pair of bobbins was wound with 100 turns of No. 34 wire and tested by hand. The output appeared to be quite sensitive.

The slip ring assembly used in Reference (6) was installed, but was discarded when it was found to have more than .070" runout.

After several attempts, a new commutator assembly was constructed and installed (see Figure XXVII and Appendix B for details). The 100 turn coils were installed in conjunction with a .015" spring plate. With a 220 mmf capacitor (externally connected to the one-shot multivibrator) the static sensitivity proved to be excessive for the purpose intended. Any slight movement of the bedplate caused the output to vary greatly.

A dynamic test proved more encouraging. The output oscillated about a mean value with a frequency proportional to r.p.m., but when r.p.m. was increased to such a point where the d.c. meter damped out the oscillations, the output became quite stable.

A calibration was attempted, and the device proved to be very linear (see Figure VII). An effort to determine the deflection under approximately full load (100 grams) conditions was made

using a dial gage. The deflection was so small that no indication could be seen.

It was decided that for the basin run a thinner spring plate would be used. This would allow a larger physical deflection of the pot cores for a full scale meter reading and would degrade the effect of motor vibration and working of the bedplate.

Accordingly, a .005 spring plate was installed. Some static results were obtained, but during the testing the center ferroxcubes were crushed. Time prohibited further physical investigation.

III. RESULTS AND CONCLUSIONS

The oscillating circuits operate in a highly satisfactory manner. A sudden application of the 24 volt supply voltage is all that is required to initiate oscillation. The two circuits are frequency independent of each other. Nearby magnetic materials do not affect the frequency of oscillation. Using the .015" spring diaphragm with a small (.001"-.002") initial air gap, the sensitivity of the system was extremely high in that a 100 gram thrust produced an output of 200 villivolts with no measurable (by dial gage) deflection (see Figure VII). There was no measurable friction loss in the system with the .015" spring diaphragm installed.

Using a softer spring plate, the output became non-linear. It was concluded that the output was linear for only a very small physical displacement.

Fewer turns produced a more sensitive device. Further investigation is necessary in order to determine the optimum number of turns.

The bedplate should be grounded to the power supply.

The output is dependent on variations in the supply voltage.

The device, as constructed, is not suitable for measurement of transient forces in a rotating shaft. This is because of the dependence of the output on the angular position of the shaft. It was concluded that the difficulty must lie in the slip rings, however the phenomena by which the output is caused to vary is not understood.

[illegible]

[illegible]

[illegible]

[illegible]

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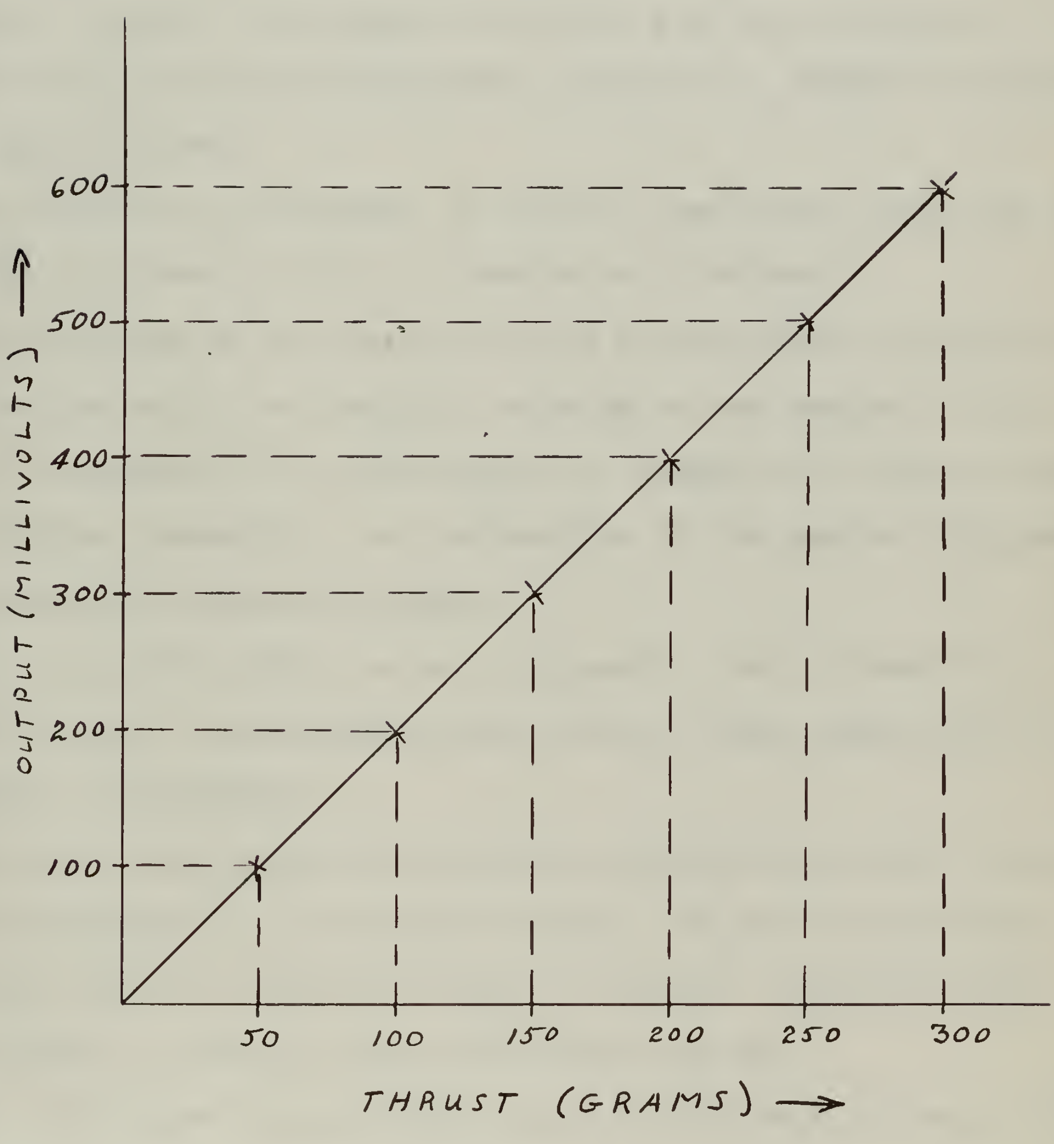
[illegible]

[illegible]

FIGURE VII

OUTPUT VERSUS THRUST



IV. RECOMMENDATIONS FOR FUTURE RESEARCH

It is recommended that several d.c. power supplies be obtained. A good 0-30 volt supply is needed for transistor circuits. A 0-400 volt supply is needed for tube circuits. A 0-30 volt well filtered high ampere output d.c. supply is needed to run small motors.

The effect on the output of varying the initial air gap and number of coil turns should be determined precisely.

The mounting of the pickup should be as stiff as possible to eliminate the effect of bedplate warping on the output. With devices as sensitive to displacement as these pot cores, any warpage, creeping, expansion, or contraction of the system components will produce an erroneous output.

The pot cores should be well aligned. Good alignment was not achieved in this investigation and this may have contributed to the dynamic difficulties.

The pot cores should be mounted so that the air gap can be adjusted by means of a micrometer screw. It was a particular sore point with the authors to have to break a cemented joint and re-cement in order to vary the total air gap.

The center pot cores should be of a different diameter than the outer pot cores. This will prevent any vibration in the shaft from changing the cross sectional area, which, of course, would alter the μ_{eff} and give an erroneous output.

A flywheel should be inserted in the shafting aft of the flexible coupling and forward of the thrust element. This would degrade the effect of the low spring constant present in the flexible coupling.

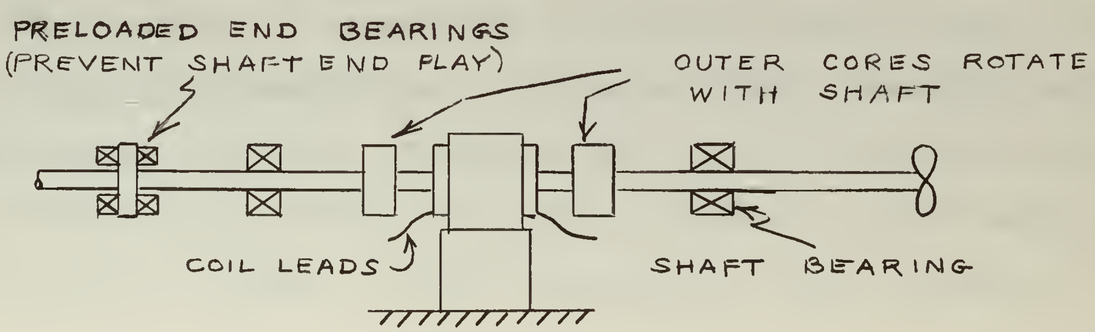
Some more efforts could be made to take the signals from the shafting through the commutator assemblies. As a possible scheme, the commutator rings could rotate in individual mercury baths. This would eliminate brush assemblies and might do away with the noise problem.

The output of the blocking oscillator could be biased to provide zero current through the coils when no pulse is present. The installation, as it now stands, has a small d.c. current passing through the coils. This current might be the source of minute sparking in the slip ring which could possibly initiate unwanted pulses from the blocking oscillator.

A different mechanical mounting of the two center pot cores could be devised which would eliminate the slip ring assembly altogether. The coils would be inserted in the center pot cores which in turn would be placed in a non-rotating housing. (See Figure VIII.)

FIGURE VIII

PROPOSED FERROXCUBE THRUST DYNAMOMETER (NO SLIP RINGS)



The advantages of the proposed installation are obvious. The mass in the rotating portion of the system is very small, and there are no slip rings to contend with. However, some constructional problems may be encountered. The shaft must have absolutely no end play, because this would read as thrust. The opposing faces of the pot cores must be very accurately aligned. Vibration problems could be overcome by making the center pot cores a different size than the outer cores. Indeed, the outer pot cores would not have to be pot cores at all, but merely thin Ferroxcube discs.

The output could be increased by additional amplifiers, or by the use of tube circuits to provide larger voltage swings. Tube circuits would probably be too heavy to be carried in the model, but in the event a carriage is constructed they would become entirely feasible.

The output, as achieved by the authors, is in the form of high frequency pulses. In order to measure transient forces in the shafting, the output will have to be filtered to eliminate the high frequencies. Fortunately, there is a great difference between the highest anticipated transient force frequency (about 900 cycles per second) and the pulse frequency (about 100 kilocycles).

There appear to be several alternative output signals using the present instrumentation. With no air gap, the pulse leaving the blocking oscillator undergoes very little attenuation in passing through coils, however the pulse attenuates quite rapidly as the air gap is opened. After a moderate air gap spacing is

achieved the pulses suffer no further attenuation. An a.c. vacuum tube voltmeter, placed between the emitter follower sides of the two coils, showed a substantial change in voltage for small changes in the air gap. Plans to investigate this phenomena for quantitative results had to be abandoned when the pot cores became damaged.

The output of each of the blocking oscillators could be fed into individual one-shot multivibrators. As the frequency of one circuit increased and the other decreased, the d.c. output of one multivibrator would increase and the d.c. output multivibrator would decrease. The difference in voltage would be a measure of displacement. Space has been allotted within the electronic housing for an additional one-shot multivibrator.

The Ferroxcubes could conceivably be utilized in a torque sensing element by shaping them in the manner shown in the figure below.

FIGURE IX

PROPOSED FERROXCUBE CONFIGURATION FOR TORQUE

The above configuration would have to be obtained by special order.

As mentioned in Chapter I, semi-conductor strain gages were briefly investigated. These devices are not yet commercially available; however, they should be investigated as soon as they appear on the market.

[illegible]

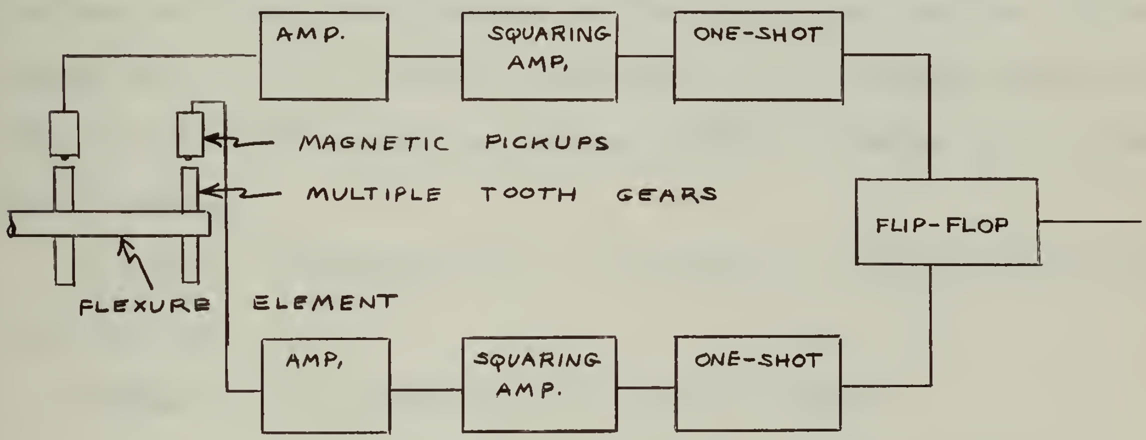
[illegible]

The system of pot cores used in this investigation is readily adapted to thrust measurement; however, in order to measure torque, special configurations of Ferroxcubes must be ordered, and a fairly involved lightweight spring system be constructed.

As the present work drew to a close, new thought was given to the problem of measuring torque. As indicated in the following discussion, it appears that the geared torque measuring configuration (mentioned in Chapter I) proposed by Mr. M. Wilson has real merit. Figure II is repeated here as Figure X.

FIGURE X

BLOCK DIAGRAM OF GEARED TORQUE DYNAMOMETER



The output of the magnetic pickups will approximate a sine wave with a frequency equal to the number of teeth times the shaft r.p.m. The amplifier will increase the amplitude to provide positive triggering of the squaring amplifier. The squaring amplifier produces a constant amplitude square wave of the same frequency as that of the incoming sine wave. The squaring amplifier is triggered when the sine wave crosses its mean value

[illegible]

[illegible]

[illegible]

[illegible]

(i.e., when the sine wave has a maximum slope), therefore there will be little variation in the duration of the pulses leaving the squaring amplifier. The one-shot multivibrator is used to provide a pulse with a very fast rise time. The one-shot may not be necessary if the leading edge of the square wave leaving the squaring amplifier has a sufficiently fast rise time. The flip-flop will turn "on" when it sees a positive going pulse at one input, and will turn "off" when it sees a positive going pulse at the other input.

Thus it can be seen that any twist of one gear with respect to the other will cause a change in the relative arrival time of the two positive going pulses at the flip-flop inputs. This change in arrival time will cause the d.c. voltage level at the output of the flip-flop to change, which, in turn, will be measure of torque.

A sample calculation will indicate the possibilities of the above design.

Assume:
- 2" diameter gear with 200 teeth.
- 800 c.p.s. is the maximum torque frequency encountered.
- T (full load torque) = 1 inch-ounce
- J (mass moment of inertia) = 10^{-2} in.^{2}-lb.
- Tube circuits are used.
- The output of the flip-flop has a 100 volt amplitude.

Using the above assumptions:

- Circumference = πD = 6.28"
- One tooth and space occupy .0314"

Considering the rims of the two gears, a displacement of one gear .0314" with respect to the other will cause a swing of 100 volts in the output of the flip-flop. The following calculations are pertinent:

$$k = \frac{4\pi^2 f^2 J}{g} = \frac{(4\pi^2)(800)^2(10^{-2})}{386} = 652 \ \frac{\text{in-Lb}}{\text{rad.}}$$

$$k = 10{,}200 \ \frac{\text{in-ounce}}{\text{rad}}$$

$$\theta = \frac{T}{k} = \frac{1}{1.02 \times 10^4} \approx 10^{-4} \text{ radians}$$

$$\delta = R\theta = 10^{-4} \text{ inches}$$

where δ = Displacement of rim

R = Radius of gear.

θ = Angular displacement

Finally, if a δ of .0314" produces a 100 volt change in the output:

$$\frac{\delta}{\text{OUTPUT}} = \frac{.0314}{100} = 3.14 \times 10^{-4} \text{ INCHES/VOLT}$$

For $\delta = 10^{-4}$ in.: Output = .318 volts = 318 millivolts

This output could easily be measured by the Sanborn recorder or an oscilloscope.

A great advantage of this system is that no slip rings are required. Also, the output should be perfectly linear.

There are certain design considerations in this proposal which require further comment.

In the sample calculation, the gear teeth are only .015" thick. This means that machining costs will probably be high. It is recommended that the gears be machined as one piece, and then cut in two.

The magnetic pickup will require a very sharp pole piece. Such pickups could be obtained on special order, or pins might be inserted in the pole pieces of more readily available commercial pickups.

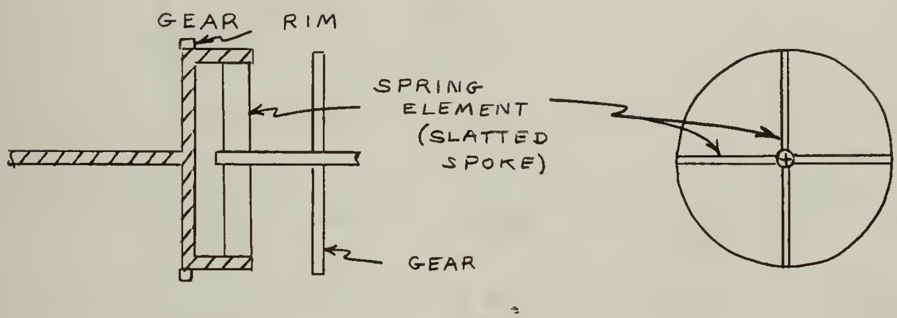
The gears will, of course, have to be machined from a magnetic material. Since the gear weight will have to be kept to a minimum in order to keep the natural frequency high, it is recommended that gear rims be mounted on lightweight discs.

The problem remains of designing a spring element. The spring system must be so constructed as to be sensitive to torque but not to thrust.

A proposed spring system is seen in the figure below:

FIGURE XI

PROPOSED GEAR SPRING SYSTEM

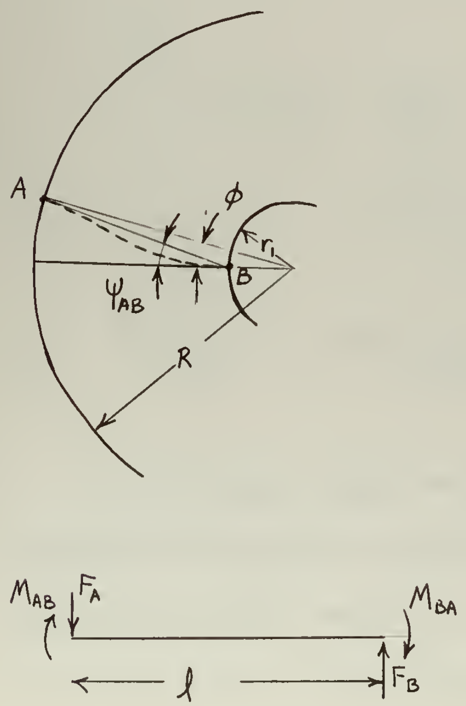


The formulas applying to this configuration are developed below, using the Slope Intercept Method.

The spring elements will deflect as shown in the following figure:

FIGURE XII

PROPOSED SPRING CONFIGURATION



$$l = R - r_1$$

$$M_{AB} = \frac{2EI}{l}\left(2\theta_A + \theta_B - 3\psi_{AB}\right)$$

where: $\theta_A = \theta$

$\theta_B = 0$

$l\,\psi_{AB} = R\phi$

$$M_{AB} = \frac{2EI}{l}\left(2\phi - \frac{3R\phi}{l}\right)$$

$$M_{BA} = \frac{2EI}{l}\left(2\theta_B + \theta_A - 3\psi_{BA}\right) = \frac{2EI}{l}\left(\phi - \frac{3R\phi}{l}\right)$$

$$Fl = M_{AB} + M_{BA}$$

$$F = \frac{2EI}{l^2}\left(3\phi - \frac{6R\phi}{l}\right)$$

$$T - n\,M_{AB} + n F_A R = 0,$$ where n is the number of spring spokes

$$T - n\left[\frac{2EI}{l}\left(2\phi - \frac{3R\phi}{l}\right)\right] + \frac{n\,2EI}{l^2}\left(3\phi - \frac{6R\phi}{l}\right)R = 0$$

$$T = \frac{2EIn}{l}\left[2\phi - \frac{3R\phi}{l} - \frac{3\phi R}{l} + 6\frac{R^2\phi}{l^2}\right]$$

$$T = \frac{2EIn}{l}\left[2\phi - \frac{6R\phi}{l} + \frac{6R^2\phi}{l^2}\right]$$

$$\phi \approx \frac{\delta}{R} : \quad T = \frac{2nEI\delta}{Rl}\left[2 - \frac{6R}{l} + \frac{6R^2}{l^2}\right] \quad \text{in.-lb.}$$

$$T = \frac{32nEI\delta}{Rl}\left[2 - \frac{6R}{l} + \frac{6R^2}{l^2}\right] \quad \text{in-ounces}$$

$$k = \frac{T}{\phi}$$

$$\frac{T}{\phi} = n\left[\frac{2EI}{l}\left(2 - \frac{3R}{l}\right)\right] - \frac{2EIn}{l^2}\left(3 - \frac{6R}{l}\right)R$$

$$\frac{T}{\phi} = \frac{4nEI}{l} - \frac{6REIn}{l^2} - \frac{6nEIR}{l^2} + 12\frac{nEIR^2}{l^3}$$

$$k = \frac{4nEI}{l} - 12\frac{nEIR}{l^2} + \frac{12nEIR^2}{l^3}$$

It is felt that the combination of pot cores to measure thrust and gears to measure torque (mounted in the proposed configurations) would be an extremely satisfactory solution to the problem investigated in this thesis.

APPENDIX

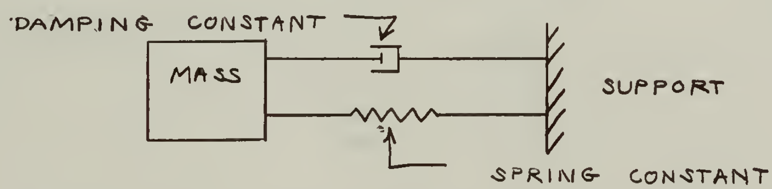
APPENDIX A

DETAILED INTRODUCTION

1. General

The propeller and shafting in the small models under consideration can be represented by the system shown below:

FIGURE XIII



This system has an equation of motion as follows:

$$\frac{d^2x}{dt^2} + \frac{b}{m}\frac{dx}{dt} + \frac{k}{m}x = \frac{F_e(t)}{m}$$

where:

x = Displacement

m = Mass

b = Damping

k = Spring constant

$F_e(t)$ = Excitation force

When placed on a Bode plot, the system response behaves as shown in Figure XIV.

Where:

$x(t)$ = Displacement of system

$F_e(t)$ = Exciting force (sinusoidal)

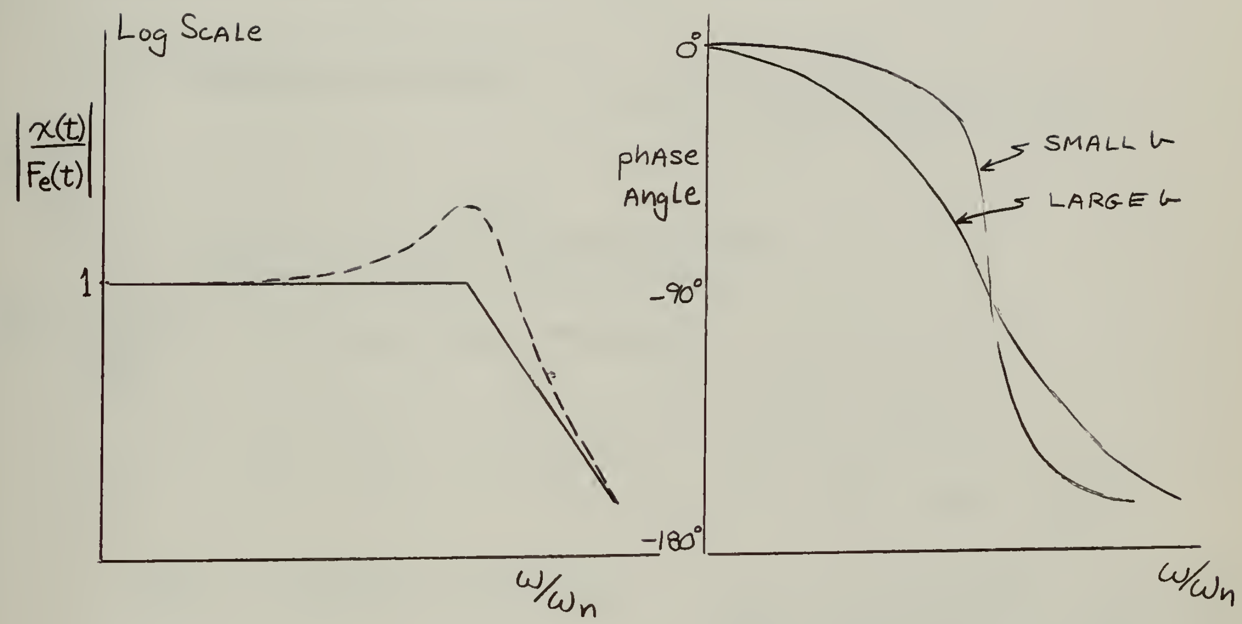
ω = Frequency of system

ω_n = Natural frequency $= \sqrt{k/m}$

Phase Angle = Angular relationship of $x(t)$ to $F_e(t)$

FIGURE XIV

BODE PLOT



NOTE: Dotted line is actual response. Solid line is straight line approximation.

The amount of overshoot (or undershoot) in the magnitude Bode plot is determined by the damping. If the magnitude of b could be adjusted exactly, the actual magnitude response could be made to approach very closely the straight line approximations. The following is an illustration of this procedure:

Define: $b_c = 2\sqrt{mk}$

$$\omega_n^2 = k/m$$

$$\zeta = b/bc$$

With the above definitions, the Laplace Transform of the system

is as follows:

$$\frac{X(s)}{F_e(s)} = \frac{\omega_n^2}{S^2 + 2\omega_n \zeta s + \omega_n^2}$$

For frequency response, let $s = j\omega$ Then:

$$M = \text{Frequency response} = \frac{\omega_n^2}{-\omega^2 + 2\zeta\omega_n(j\omega) + (\omega_n^2)}$$

$$|M| = \frac{\omega_n^2}{\sqrt{(\omega_n^2 - \omega^2)^2 + 4\zeta^2\omega_n^2\omega^2}}$$

Let $\frac{d\,|M|}{dw}$ = 0 for maximum overshoot.

$$= 2(\omega_n^2 - \omega^2)(-2\omega) + 8\zeta^2\omega_n^2\omega$$

$$\omega^2 = \omega_n^2 - 2\zeta^2\omega_n^2 \rightarrow \zeta \leq .707 = \frac{\sqrt{2}}{2}$$

$$M_p = \text{Maximum overshoot} = \frac{\omega^2}{\sqrt{(\omega_n^2 - \omega_n^2 + 2\zeta\omega_n^2)^2 + 4\zeta^2\omega_n^4(1-2\zeta^2)}}$$

$$M_p = \frac{1}{\sqrt{4\zeta^4 + 4\zeta^2 - 8\zeta^4}}$$

$$M_p = \frac{1}{2\zeta\sqrt{(1-\zeta^2)}}, \quad \zeta \leq \frac{\sqrt{2}}{2}$$

Thus if $\zeta = \sqrt{2}/2$, $M_p = 1$ (which means there is no overshoot).

However, there is a substantial phase shift at these values of ω and ζ:

$$\text{For } \zeta = \frac{\sqrt{2}}{2} \quad \omega = \sqrt{\omega_n^2 - .707\ n^2} = \sqrt{.293\,\omega_n^2} = .54\,\omega n$$

$$\text{phase angle} = \measuredangle = \frac{1}{\tan^{-1}\frac{2\zeta\omega_n\omega}{\omega_n^2 - \omega^2}} = \frac{1}{\tan^{-1} 1.08} = 47.1^\circ$$

Thus it can be seen that to specify no overshoot, one is restricted to frequencies below .54 ω_n and also one must accept a relatively large phase shift.

The above equations also hold for torque if moment of inertia is inserted for mass, torsional spring constant for the linear spring constant, and angular displacement for linear displacement.

Since damping in the propeller-shafting system of models is difficult to measure, much less control, the logical procedure is to design a system composed of the propeller, shafting, and measuring pickup with a natural frequency substantially higher than the highest frequency to be measured, and keep the damping low. This will allow a linear response out to the vicinity of the natural frequency, and also keep the phase shift to a minimum.

In the case of 5 foot models, scale r.p.m. may run as high as 3000 r.p.m. With a five bladed propeller, the first harmonic is 250 c.p.s. Reference (1) indicates that for an odd bladed propeller, the second harmonics are important. The second harmonic is 500 c.p.s., and as pointed above, the natural frequency should be higher than this, say 800-1000 c.p.s. Admittedly, these conditions (3000 r.p.m., 5 bladed propeller) are extreme and will seldom be imposed; however, they indicate that in the design of a thrust and torque pickup, the higher the ω_n, the more versatile the instrument will be.

2. Discussion of Previous Work

References (5), (6), and (7) are concerned with previous attempts to measure forces in the shaft of a 5 foot model using

strain gages mounted in Lucite for torque measurement and a Schaevitz linear differential transformer to measure thrust. In all cases difficulty was experienced with slip ring noise which masked the output signal.

The present authors contend that even if the output signal amplitude were large enough to overcome the slip ring noise, these devices would not be acceptable for the measurement of the expected transient forces. The following calculations are presented in support of this contention.

The spring system used in the Schaevitz linear differential transformer was designed for a full scale deflection of .010" for a .25 pound thrust.

$$\text{Thus:} \quad k = \frac{\text{force}}{\text{displacement}} = \frac{.25}{.01} = 25 \ \frac{\text{lb}}{\text{in}} = 300 \ \frac{\text{lb}}{\text{ft}}$$

$$f = \frac{1}{2\pi}\omega, \ \omega_n = \sqrt{k/m} \qquad \therefore \qquad f_n = \frac{1}{2\pi}\sqrt{k/m}$$

Assume an f_n of 500 c.p.s. is specified (this is a low value for 5 foot models).

Substituting in the above formulas: $500 = \frac{1}{2\pi}\sqrt{\frac{300}{m}}$

$$\sqrt{m} = \frac{\sqrt{300}}{1000\pi} = \frac{17.3}{1000\pi} = 5.51 \times 10^{-3}$$

$$m = 30.4 \times 10^{-6} \ \frac{\text{lb}_f - \text{sec}^2}{\text{ft}} = 979 \times 10^{-6} \ \text{lb}_m \approx .001 \ \text{lb}_m$$

Conclusion: For $f_n = 500$ cps, $m \leq .001 \ \text{lb}_m$

The mass m includes, of course, the mass of the entrained water. This extremely small mass could not be realized in the physical system.

The Lucite torque gage had the dimensions shown below:

FIGURE XV
TORSION ELEMENT

The Modulus of Elasticity (E) of Lucite is given as 5.0×10^5 p.s.i. in Reference (5). The relationship between G (Modulus of Rigidity) and E is:

$$G = \frac{E}{2(1+\mu)}$$, where μ is Poisson's Ratio

Neglecting μ, $G = \frac{E}{2} = 2.5 \times 10^5$ p.s.i.; this value of G will be used in the following calculations.

For angular twist in a shaft the following relationship holds:

$\theta = \frac{T\,L}{J\,G}$, where θ = angle of twist (radians)

T = Torque (in-lb)

L = Length of shaft (inches)

J = Polar moment of Inertia (in^4)

G = Modulus of Rigidity (p.s.i.)

Let K_θ = Torsional spring constant

$$K_\theta = \frac{T}{\theta} \quad \frac{lb.-in}{rad.} = \frac{JG}{L}$$

$$J = \frac{\pi}{32}\left(D^4 - d^4\right) = \frac{\pi}{32}\left[\left(\frac{3}{8}\right)^4 - \left(\frac{1}{4}\right)^4\right] = \frac{\pi}{32} \times \frac{1}{256}\left[\frac{81}{16} - 1\right]$$

$$J = \frac{4.06\,\pi}{(32)(256)} = 1.56 \times 10^{-3}\ in^4$$

$$K = \frac{JG}{L} = \frac{(1.56 \times 10^{-3})(2.5 \times 10^5)}{2.5} = 1.56 \times 10^{-2}\ \frac{lb.in.}{rad.}$$

$\omega = \sqrt{\dfrac{K_\theta g}{J_m}}$, where J_m = mass moment of inertia (in^2-lb)

$$f_n = \frac{1}{2\pi}\omega_n = \frac{1}{2\pi}\sqrt{\frac{k_\theta g}{J_m}} \; ; \; g = 386 \frac{\text{in}}{\text{sec}^2}$$

Assume $f_n = 500$ c.p.s.

$$500 = \frac{1}{2\pi}\sqrt{\frac{(156)(386)}{J_m}}$$

Solving:

$$J_m = .0061 \ \text{in}^2\text{-lb}$$

The present authors feel that a J_m of 10^{-2} in^2-lb (including the effects of entrained water) is the lowest feasible mass moment of inertia. This, of course, may be open to debate, as the effect of entrained water can only be grossly estimated.

To summarize; despite the fact that previous work used deflections which were too large (i.e., ω_n was too low) to measure expected frequencies, extreme difficulty was encountered in producing an output signal large enough to overcome slip ring noise.

At this point it became the primary purpose of this thesis to develop a device which was sensitive enough to measure extremely small deflections and yet deliver an output signal of an acceptable magnitude.

Previous workers located the sensitive elements aft of the stern bearing. This placed extreme limitations on the size and wiring of the systems. Accordingly, calculations were made to get an estimation of the losses in the stern tube bearing.

3. Bearing Torque Losses

Using Petroff's Equation, which is valid for a lightly loaded sleeve bearing:

$$T = \frac{4\pi^2 r^3 \mu}{60} \times \frac{LN}{c}$$

where T = Torque loss (inch-ounces)

N = r.p.m.

L = Length of bearing (in)

μ = Viscosity $\frac{\text{(ounce-sec)}}{\text{in}^2}$

r = Radius (in)

c = Clearance (in)

As typical values, assume:

N = 1800 r.p.m.

$$\mu(H_2O @ 70) = 2 \times 10^{-5} \frac{\text{lb-sec}}{\text{ft}^2}$$

$$= 22.2 \times 10^{-7} \frac{\text{ounce-sec}}{\text{in}^2}$$

r = .062"

Then:

$$T = \frac{4\pi^2 (.062)^3 (2.22 \times 10^{-6})\, 1800}{60} \times \frac{L}{c}$$

$$T = 6.42 \times 10^{-7} \times \frac{L}{c} \text{ (inch ounces)}$$

If L = .5" and c = .01"

then $T = 3.21 \times 10^{-5}$ (inch ounces)

This torque loss is negligible compared with the full load torque of 1 inch ounce.

4. <u>Stern Bearing Leakage</u>

The next step was to check the leakage of water past the stern tube bearing. Using the result of problem 5 from Reference (11):

$$-\frac{dp}{dx} = 12\mu\frac{Q}{h^3}$$

Where $\frac{dp}{dx}$ = Pressure gradient along the shaft (lb/in^3)

μ = Viscosity $\left(\frac{\text{lb-sec}}{\text{in}^2}\right)$

h = Clearance (in)

Q = Flow rate per unit of circumference $\left(\frac{\text{in}^2}{\text{sec}}\right)$

As a typical calculation, assume:

Submergence = 2 inches

$h = 10^{-2}$ inches

$\mu = 1.39 \times 10^{-7}\ \frac{\text{lb-sec}}{\text{in}^2}$

Circumference $= 2\pi r = .393$ inches ($r = \frac{1}{16}$ inch)

Density of water $= 3.61 \times 10^{-2}$ lb/in^3

Therefore $\frac{dp}{dx} = \frac{7.2 \times 10^{-2}}{L}\ \frac{\text{lb}}{\text{in}^3}$, where L = Length of shaft

L = .5 inch

$$Q = \frac{1}{12\mu}\ \frac{dp}{dx}\ h^3 = \frac{(7.2 \times 10^{-2})\ (10^{-6})}{(12)(1.39 \times 10^{-7})(.5)} = 8.64 \times 10^{-2}\ \frac{\text{in}^2}{\text{sec}}$$

$$Q \times \text{Circumference} = 8.69 \times 10^{-2} \times .393 = 3.40 \times 10^{-2}\ \frac{\text{in}^3}{\text{sec}}$$

Therefore the total flow $= 3.4 \times 10^{-2}$ in^3/sec, or it will take 29.4 seconds to pass 1 in^3 of water.

This flow was deemed to be too great, even though a much smaller clearance might be more reasonable.

Reference (10) described the successful utilization of a mercury seal to stop the leakage of water around a rotating shaft. Reference (12) listed the viscosity of mercury as very close to that of water; therefore, the losses in the mercury seal could be expected to be of the same magnitude as those in the stern tube bearing. See Detailed Procedure (Appendix B) for the design of this combination seal and bearing.

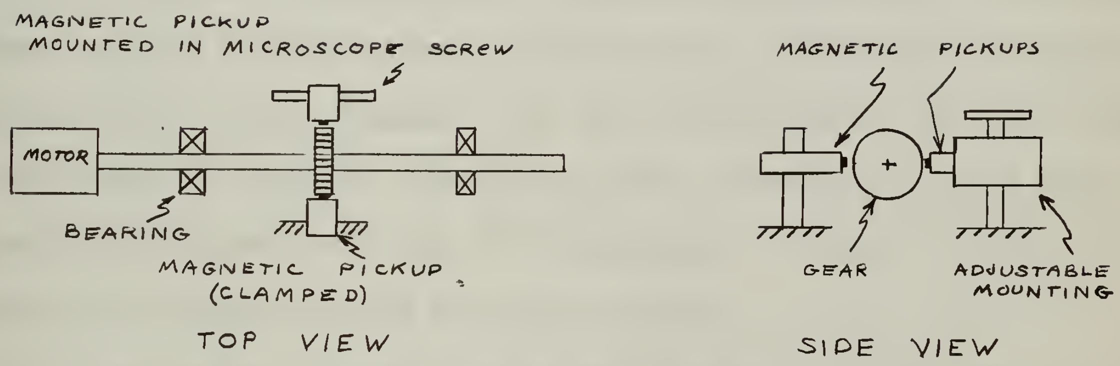
5. Preliminary Investigation

As a result of the above calculations, it was decided that force measuring pickups could be placed forward of the stern bearing. It was recognized that the damping effect of the stern tube bearing and mercury seal is an unknown whose magnitude would have to be determined experimentally if and when a sensitive pickup could be constructed and calibrated. It was hoped that such damping would be small in comparison to the propeller damping.

The actual force sensing scheme had still not been decided upon. A number of possibilities were mentioned in Chapter I. One that actually was "breadboarded" was Mr. M. Wilson's (David Taylor Model Basin) idea, which used gears to measure torque. A single 2 inch diameter (100 tooth) spur gear was utilized in the manner shown in Figure XVI.

FIGURE XVI

BREAD BOARD OF GEARED TORQUE DYNAMOMETER



By turning the microscope screw handle, the clamp holding the magnetic pickup could be moved up and down. Movement of the pick-up relative to the gear simulated twisting of the shaft between two gears (see Figure II).

At the time, it was decided that the machining of the pole piece and gears would be difficult and expensive. It was also concluded that the instrumentation would be costly, the gears heavy (which would increase ω_n), and there was still no assurance that the sensitivity would be adequate. However, upon completion of the present work, Mr. Wilson's idea was again considered and is discussed more thoroughly in Chapter V.

The "breadboarding" of the above circuit was accomplished at David Taylor Model Basin. The electronic circuits consisted of "plug-in" transistorized "cans". These units are a product of

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

Engineered Electronics Company, 506 East First Street, Santa Ana, California. The convenience of these "cans" was unfortunately not available at M.I.T.

The decision to make use of ferrite pot cores (manufactured by the Ferroxcube Corporation of America, Saugerties, New York) stemmed largely from the work of Mr. Robert Singleton, an employee of David Taylor Model Basin. His was the original idea of using the pot cores as part of two oscillating circuits, and he had experimentally determined that the frequency of the circuits was extremely dependent on the air gap spacing.

With the above information as background, the authors launched into the development of the ferrite pot cores as an ultrasensitive force measuring device.

APPENDIX B

DETAILS OF PROCEDURE

The following projects were undertaken:

(a) Size, arrange, and house the pot cores.

(b) Design and build the electronic instrumentation.

(c) Mount the motor, slip rings, shafting, pickup, and instruments in the model.

(d) Evaluate the overall system.

1. Pot Cores

A letter of inquiry was sent to the Ferroxcube Corporation of America, Saugerties, New York, asking about the available sizes of pot cores and bobbins. The company returned a set of Engineering Bulletins describing pot cores in general. A brochure of the available cores was also received.

Engineering Bulletin No. FC-5109 B, dated May 1956, was encouraging in that it contained a graph showing variation of effective permeability (μ_{eff}) with air gap. This plot revealed the fact that μ_{eff} decreased 30-40 per cent for a change in air gap of about .0002 inches. Since L (inductance) of the proposed configuration (see Figure IV) is approximately proportional to μ_{eff} (see the aforementioned bulletin), the prospects looked very encouraging.

Accordingly, a variety of parts was ordered. Upon receipt of the cores and bobbins, it was discovered that the pot core and bobbin shown in Figure XVII were perfect mates in that the bobbin would fit entirely inside one-half of the core. This would

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[illegible]

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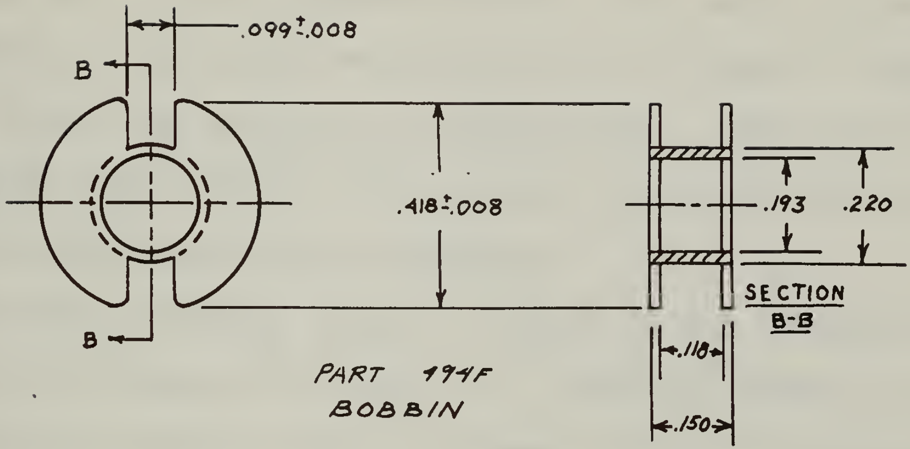
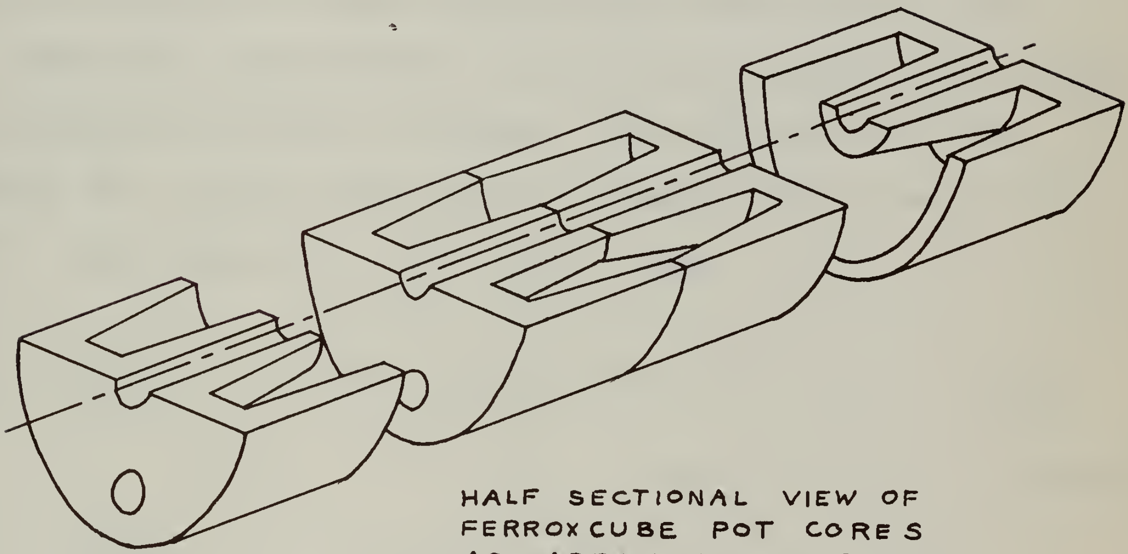
[illegible]

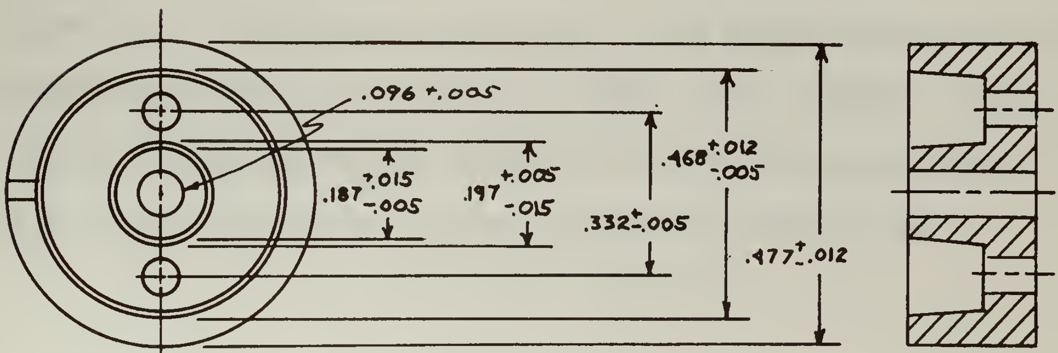
FIGURE XVII



PART 494F
BOBBIN



HALF SECTIONAL VIEW OF FERROXCUBE POT CORES AS ARRANGED IN THRUST HOUSING.



PART 465 P250 B3-3C
POT CORE

DETAILS OF FERROXCUBE POT CORES AND NYLON BOBBINS

eliminate any sliding friction between the center Ferroxcubes and the bobbins. The center Ferroxcubes were turned so that their "backs" faced the bobbins (see Figure XVII). This would prevent the bobbins from working from one pot core to the other while the whole configuration was rotating.

The thrust housing, shown in Figure IV was designed and given to Mr. R. E. Johnson (of the M.I.T. Propeller Tunnel) to construct.

Before the spring plate thickness could be determined, it was necessary to estimate the axial deflections that could be permitted. Assume the smallest mass (including entrained water) that could be attained is about .04 pounds. This allows for future installation of a light torque sensing element aft of the thrust enclosure. Also assume f_n = 500 c.p.s., then:

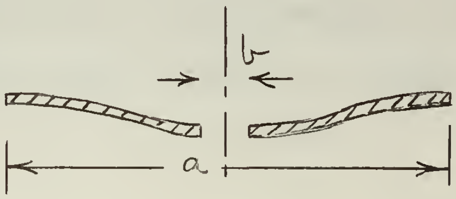
$$k = \frac{4\pi^2 f^2}{g} = \frac{4\pi^2 (500)^2 (.04)}{386} = 1025.0 \ \frac{lb}{in.}$$

$$F_{max} \approx .25 \text{ lb} \ ; \quad \delta = \frac{F}{k}, \text{ where } \delta = \text{displacement}$$

$$\delta = \frac{.25}{1025} = .000245''$$

Page 67 of Reference (13) deals with deflections of circular diaphragms under a variety of loads and boundary conditions. Our spring diaphragm can be assumed to have fixed boundary conditions at the center and outer edges as shown below.

FIGURE XVIII



SPRING DIAPHRAGM

Reference (13) gives the following formula relating the deflection and the applied force:

$$\delta_{max} = k_1 \frac{Pa^2}{Eh^3}$$, where δ = Deflection (in)

P = Force (lb)

a = Center hole diameter (3/8")

E = Modulus of Elasticity $(15 \times 10^6$ p.s.i.)

h = Diaphram thickness (in)

$k_1 = f(a/b)$

$a/b = 3.2$

From (13) an a/b of 3.2 yields a k_1 of .110

Considering $\delta_{max} = .000245"$ at .25 lb.

$$h^3 = \frac{k_1 \; Pa^2}{E \; \delta_{max}} = \frac{(.110)\;(.25)\;(3/8)^2}{(15 \times 10^6)\;(2.45 \times 10^{-4})} = 1.05 \times 10^{-6}$$

$h \simeq 0.01$ inches

As a result of the above calculations, 4 diaphragms were constructed of the following thicknesses: .005", .007", .015", and .017".

2. Electronic Instrumentation

The electronic instrumentation discussed in Chapter II was assembled from components using circuits very similar to those utilized in the aforementioned "cans". This was done so that if the completed pickup element became a production item, the electronic circuits could easily be commercially obtained.

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$$[illegible]$$

[illegible]

$$[illegible]$$

[illegible]

[illegible]

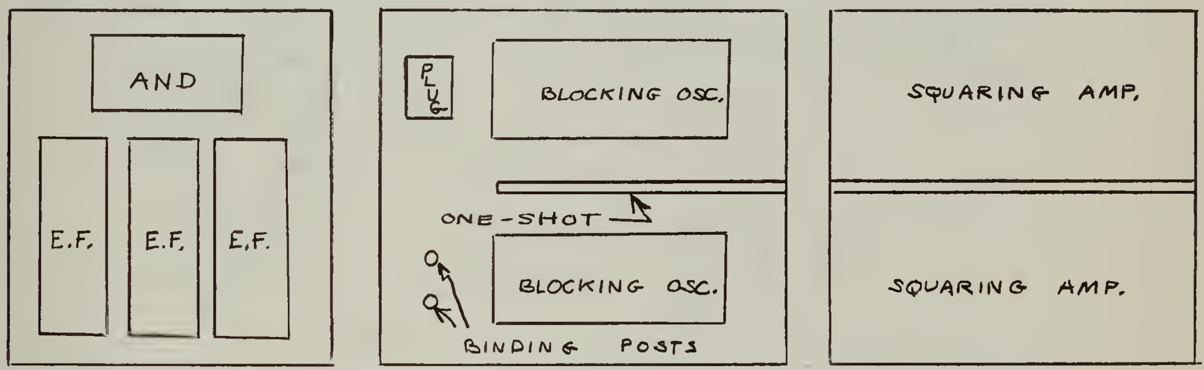
The schematic diagram is shown in Figure XIX. The layout of the inside of the electronic housing is shown in Figures XX and XXII.

There is space allotted in the circuit enclosure for another one-shot multivibrator. The possibility of this additional circuit is discussed in Chapter IV.

The two outside binding posts allow an external capacitor to be connected to the one-shot multivibrator.

FIGURE XXII

LAYOUT OF ELECTRONIC ENCLOSURE



3. <u>Details of Physical Arrangement</u>

A Series 60, Block 60 model was selected for use in this thesis. After concluding that the instrumentation should be placed entirely within the model, the authors were faced with the following problems: the shaft centerline was only $1\frac{1}{4}$" above the baseline and the hole for the stern tube bearing was not parallel with the fore and aft axis of the model.

FIGURE XXIII

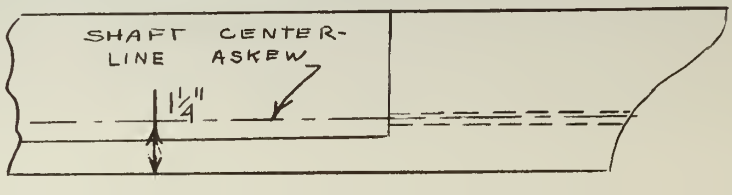
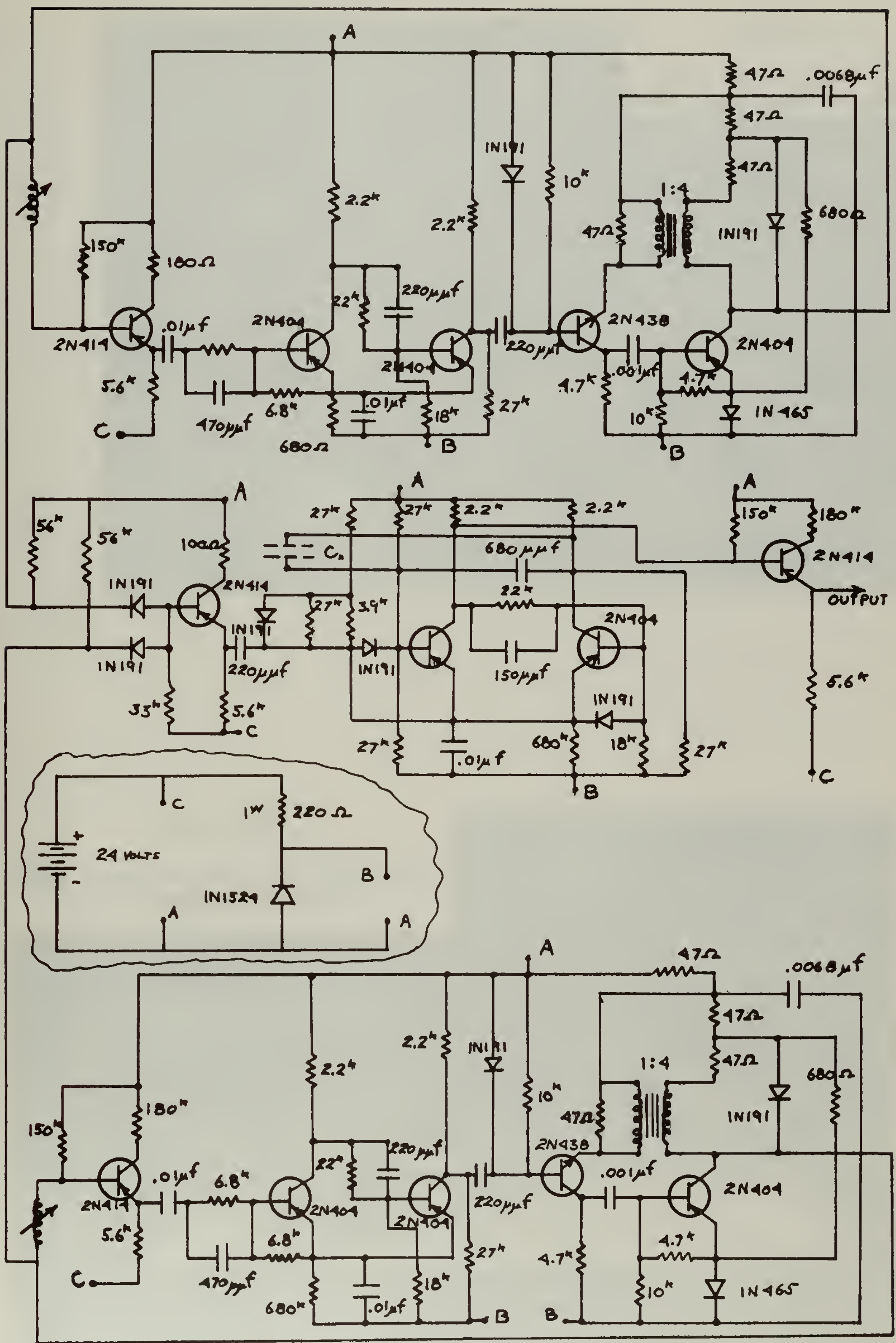


FIGURE XIX



SCHEMATIC

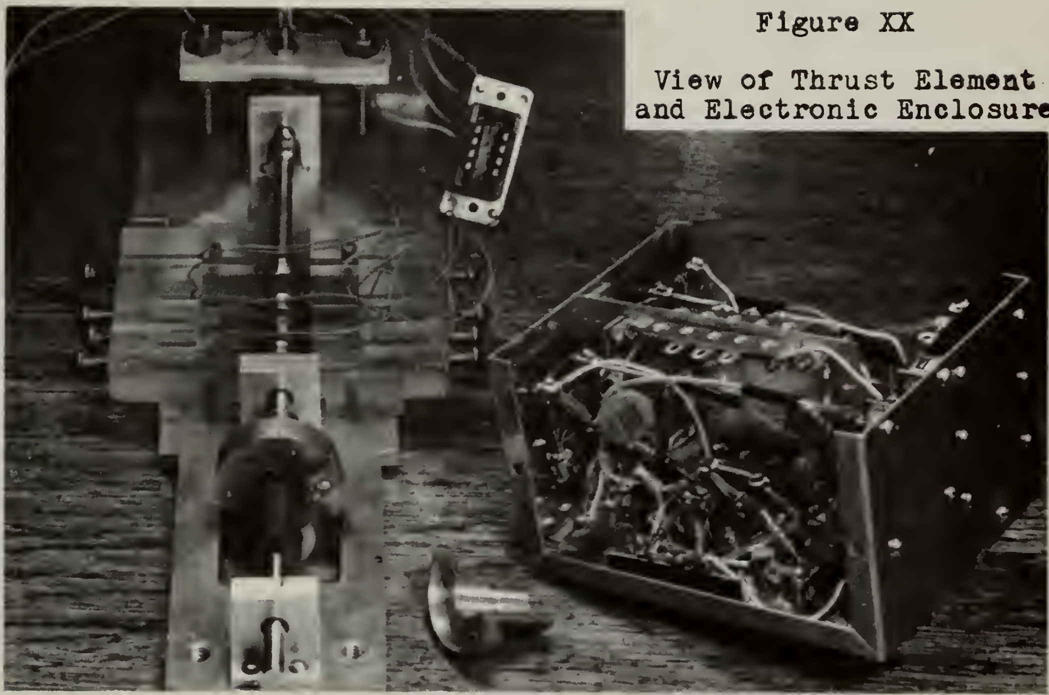
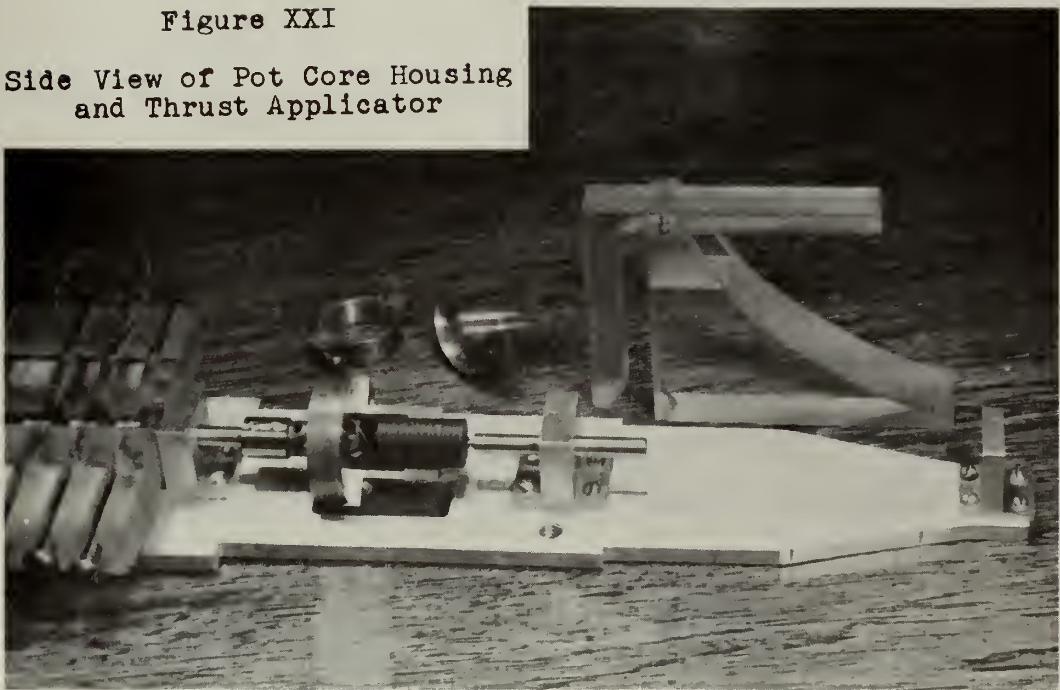


Figure XX

View of Thrust Element
and Electronic Enclosure

Figure XXI

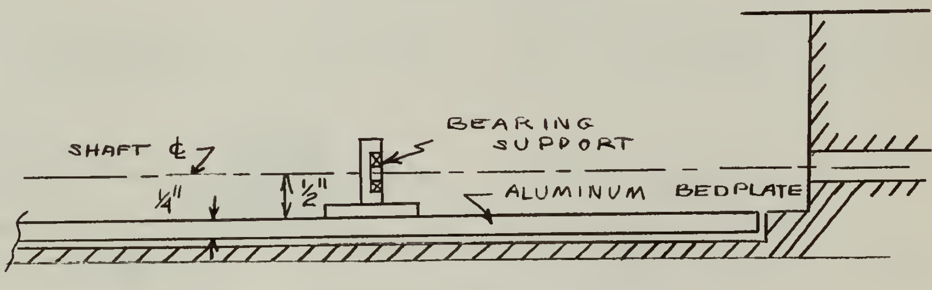
Side View of Pot Core Housing
and Thrust Applicator



The bedplate was designed to accommodate shaft adjustments in both the horizontal and vertical planes (see Figure XXIV).

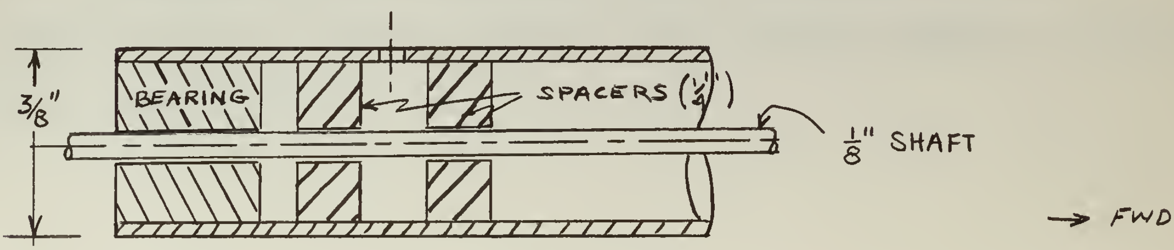
The ball bearings selected were of such a size that ½" clearance between the surface of the bedplate and the shaft centerline was required. The model was modified to allow room for the bedplate. Further alteration in the bottom was required in way of the propulsion motor as the motor casing extends 1-1/8" below the shaft centerline.

FIGURE XXIV
SECTION VIEW OF BEDPLATE



The design in Figure XXV was intended for use as a combination bearing and water seal.

FIGURE XXV
STERN TUBE BEARING AND MERCURY SEAL

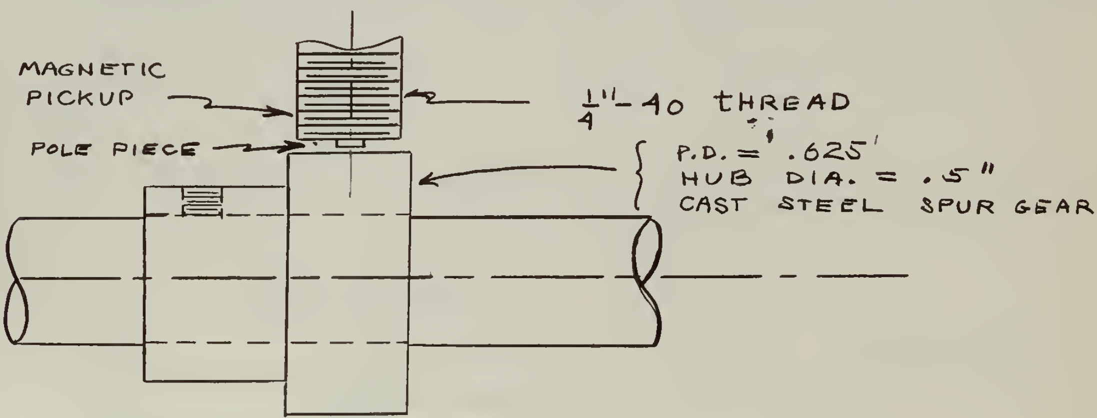


The mercury seal concept (defined in Reference (10)) takes advantage of the high surface tension of mercury to prevent leakage of water around a rotating shaft. A steel shaft must be used in order to preserve this high value of surface tension. A 1/8" hole was bored vertically through the stern section of the model (from the upper deck). This allowed a hollow mercury supply tube to be inserted into the upper part of the mercury chamber. Using Petroff's Equation to determine the magnitude of torque loss in the mercury seal it was seen that torque loss is negligible (12.82×10^{-5} in.-oz.).

The arrangement seen in Figure XXVI was intended for r.p.m. measurement. The output of the magnetic pickup can be fed directly to an electronic counter, or through a frequency to voltage device to the recorder.

FIGURE XXVI

MAGNETIC PICKUP FOR R.P.M. COUNTING



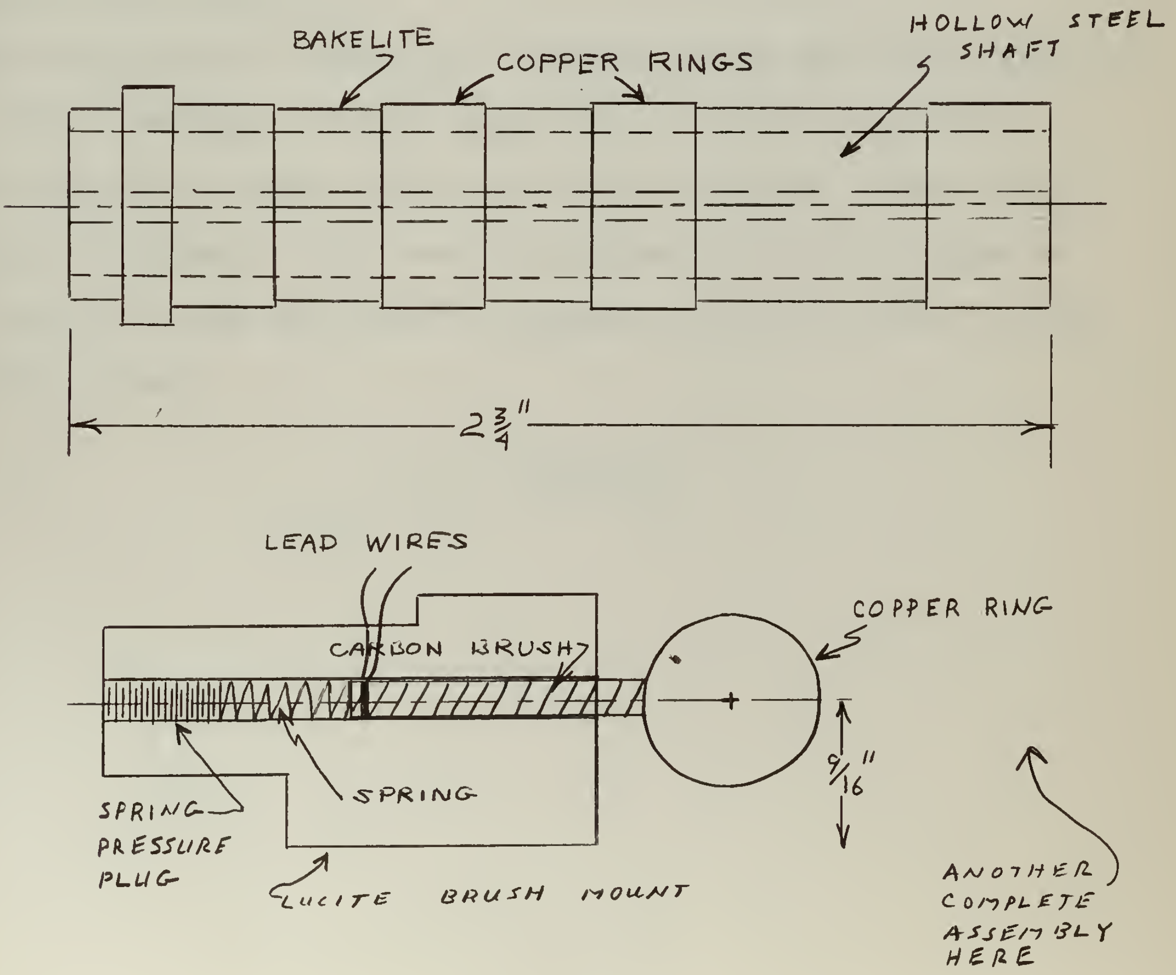
The gear has 15 teeth, therefore the counter reading is multiplied by 4 to get the r.p.m.

[illegible]

The original brush and slip ring assembly used in this investigation was the same as that developed in Reference (7). It was determined in the course of this thesis that this miniature commutator assembly was extremely difficult to mount and adjust. These difficulties led the authors to a revised commutator assembly design (see Figures XXVII and XXI), which utilized the carbon brushes previously investigated in Reference (6). The rings utilized in Reference (6) were unacceptable because of excessive runout.

FIGURE XXVII

REVISED COMMUTATOR ASSEMBLY



4. Evaluation of the Overall System

An EICO model 1020 transistorized power supply was obtained for the electronic supply voltage. This instrument has very low ripple, but lacks good regulation characteristics.

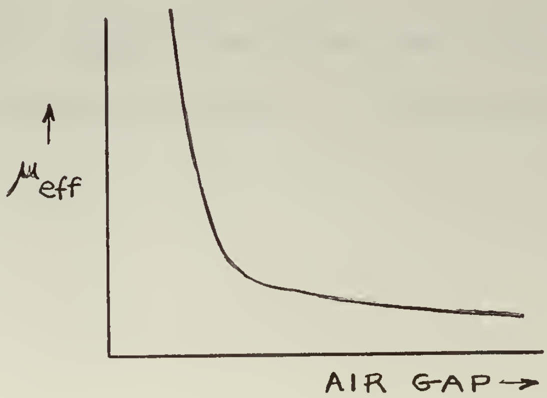
The electronic installation was initially tested by connecting two 250 turn (#38 magnet wire) pot cores to the terminals. The two oscillating circuits were checked out point by point with an oscilloscope. This check was made with no air gap between the pot cores. The output of one of the blocking oscillators appeared to be slightly narrower than the other, but elsewhere the waveforms were virtually identical.

The frequency of each of the oscillating circuits was in the vicinity of 400 kilocycles.

The next step was change the air gap of one pot core only (by hand). The frequency of that oscillating circuit increased rapidly with an increase in air gap up to a point, after which it remained constant. This was in agreement with the graph of μ_{eff} versus air gap mentioned on page 40, which has the following general shape:

FIGURE XXVIII

μ_{eff} vs AIR GAP



[illegible]

[illegible]

[illegible]

[illegible]

During this test, the frequency of the other oscillating circuit did not change, indicating that the two circuits were frequency independent of each other.

A screw driver and some scrap metal were moved around near each of the pot cores with no noticeable effect on the frequency.

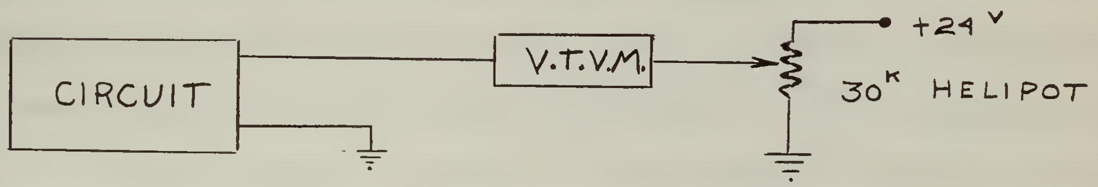
A lead was inserted connecting the inputs of the two emitter followers. This caused the two circuits to be frequency dependent on each other; therefore, this lead was removed.

The AND circuit was then connected to the output of the blocking oscillators. There appeared to be no deterioration of the waveforms in any part of the circuit previously tested.

The output of the complete circuit (which is identical to the output of the one-shot multivibrator) was connected to the oscilloscope, but some difficulty was encountered in synchronizing the signal with the sweep. Accordingly, a vacuum tube volt meter (set on d.c. volts) was connected to the output in the following manner:

FIGURE XXIV

V.T.V.M. CONNECTION FOR OUTPUT MEASUREMENT



The 24 volt supply came from the electronic power supply. The Helipot acted to make the V.T.V.M. a differential d.c. voltmeter.

[illegible]

[illegible]

[illegible]

$$[illegible]$$

[illegible]

A decade capacitor box was connected to the binding posts on the circuit enclosure to determine the duration of the output pulses. The external capacitor set in the vicinity of 100 $\mu\mu f$ to 1000 $\mu\mu f$ produced the most sensitive response.

The air gap of one pot core was varied by hand. The output appeared to be quite sensitive to changes in air gap as long as the air gap spacing was small.

As a final test before installation on the thrust element housing, a 1000 ohm resistor was inserted in the pot core leads. No change in frequency resulted, and the output appeared to be just as sensitive as before. It was therefore concluded that slip ring resistance changes should have no effect on the oscillating frequency.

Some difficulty was encountered in cementing the pot cores to the shafting. An epoxy resin was tried, but it penetrated along the shaft and glued the two center Ferroxcubes to the outer Ferroxcubes. These pot cores had to be destroyed in order that the shaft might be used again. A milder cement (Miracle Black Magic Adhesive) was used with more successful results.

New pot cores (rewound with 325 turns of number 38 magnetic wire) were inserted in the thrust housing. A .007" spring diaphragm was also fastened in place. The housing, shafting, motor, and magnetic pickup were mounted on the bedplate. The miniature commutator assembly developed in Reference (7) was secured to the forward end of the propulsion motor.

The commutator brushes were damaged, and a considerable amount of time was spent attempting to make them operable. When electrical continuity was finally achieved, static and dynamic tests were attempted.

The static tests were disappointing in that the output was less than anticipated when finger pressure was applied to the end of the shaft. The voltmeter changed only 200-300 millivolts for a fairly strong axial force.

A dynamic test was even more discouraging in that the output was strongly dependent on the angular position of the shaft. The variation in output was as much as 500 millivolts. With the motor running, the voltmeter would oscillate about a mean value. With the motor stopped and the shaft rotated slowly by hand, the output would slowly vary throughout the 500 millivolt range.

It was concluded that the damaged commutator brushes were the cause of the difficulty. (Later work indicated that this may not have been the case.)

A new commutator assembly was sought. The assembly used in Reference (6) was located and mounted in the shafting, just forward of the thrust element enclosure. A dial gage indicated that the runout was about .070"; therefore, that commutator assembly was discarded.

After several attempts, the slip ring assembly shown in Figure was constructed. A cylinder of Bakelite was press fitted over a length of hollow 1/4" shaft. Copper rings were heated, placed in position on the Bakelite shaft, and quenched

[illegible]

in water, Holes were drilled in the shafting and leads were inserted. Runout was greatly reduced over the previous assembly.

Meanwhile, hand tests of the 325 turn pot cores indicated that the output was less sensitive than the tests of the 250 turn had been. Therefore, new pot cores wound with 100 turns of #34 magnet wire were prepared. Hand tests produced large voltage swings of the output. These 100 turn pot cores were then mounted in the thrust housing.

Because the deflections of the .007" spring diaphragm appeared comparatively large to the .000245" deflection desired, the housing was reassembled with a .015" spring diaphragm.

The above combination proved to be ultrasensitive. Any working of the bedplate caused a large meter deflection; even the lightest finger pressure on the plate caused a substantial meter deflection.

The new slip ring assembly performed no better than the first. The output proved to be very sensitive to angular position of the shaft.

Dynamic tests were made using a motor r.p.m. high enough to cause the V.T.V.M. to damp out the output oscillations.

Loads varying from 0 to 300 grams were applied to the after end of the rotating shaft in increments of 50 grams. The output proved to be very linear (see Figure VII). When the loads were removed the output returned to the original value, indicating a negligible loss in the bearings. The shaft was lightly pulled (representing a negative thrust) and released. The initial reading

did not change. A dial gage mounted on the thrust enclosure, touching the back of the after pot core, showed no displacement indication for a load of 100 grams. It was concluded that the displacement was well under the desired value of .000245".

The supply voltage was varied plus and minus 3 volts. The output signal varied plus and minus .75 volts. Thus it was seen that the output was quite sensitive to the supply voltage.

For the same reasons indicated in Chapter II the housing was dismantled and reassembled using a .005" spring diaphragm. The deflection of this diaphragm under a 100 gram load appeared to be too great. The tentative calibrations (no numbers were recorded) were non-linear. The output would swing first in one direction and then in the other as the load was increased. Many different settings of the air gap produced the same result.

Unfortunately, while making an air gap adjustment, the clamp holding the center Ferroxcubes was tightened too severely, and the enclosed pot cores were crushed. No additional pot cores were immediately available, therefore, further investigation was discontinued.

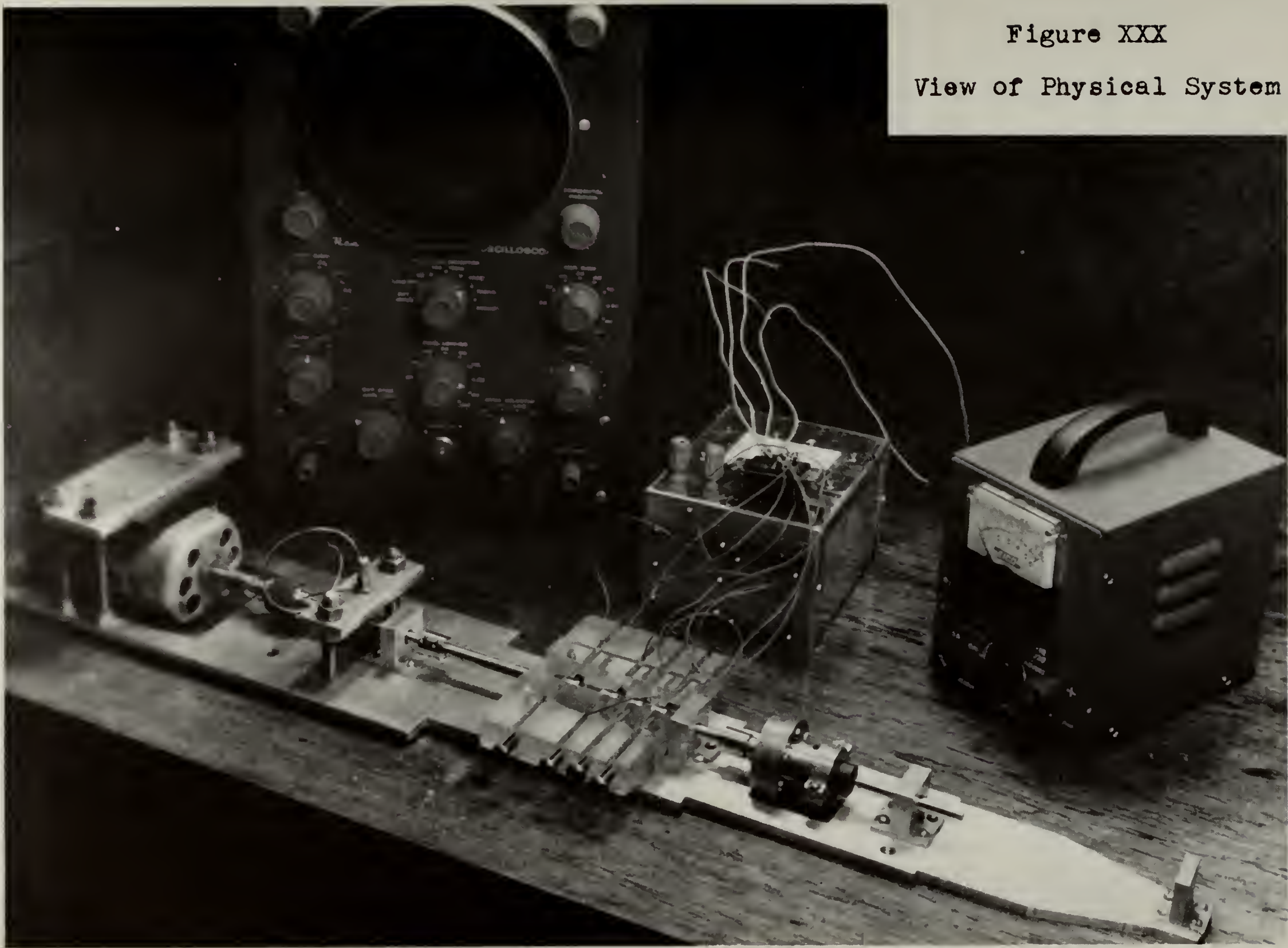


Figure XXX

View of Physical System

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