NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER
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STATIC AND DYNAMIC CALIBRATION OF PROPELLER MODEL FLUCTUATING FORCE BALANCES

by

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The Naval Ship Research and Development Center is a U.S. Navy center for laboratory effort directed at achieving improved sea and air vehicles. It was formed in March 1967 by merging the David Taylor Model Basin at Carderock, Maryland and the Marine Engineering Laboratory at Annapolis, Maryland.

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ABSTRACT

Balance systems and techniques for model-scale measurement of up to six components of the fluctuating propeller forces produced by nonuniform inflow conditions have been developed.

Balances which achieve a broad nonresonant measuring range along with high signal to noise ratio have been developed with two degrees of freedom mechanical designs. Analytic methods employing lumped mass spring and estimated damping representations have aided in balance design.

Devices and methods for statically and dynamically calibrating these balances so that accurate relationships between balance signals and actual propeller forces are described.

A dynamic calibration procedure utilizing small electromechanical force inducers attached in place of the test propeller on the rotating balance shaft is used to determine the frequency response of each of the balances' force transducers. Interaction effects on transducer signals of force components produced in planes other than the principal transducer plane are small on the latest balance designs.

Although relatively satisfactory calibrations have been achieved to date with this scheme, improvements in monitoring amplitude and phase of calibrator input forces should be pursued to assure maximum accuracy in calibration.

ADMINISTRATIVE INFORMATION

This work was performed at the David Taylor Model Basin under Bureau of Ships Project FAN, S-F013-0103, Task 0200, under the direction of Dr. Murray Strasberg.

TABLE OF CONTENTS

NOTATION

C_1	Propeller damping coefficient (viscous)
C_2	Flywheel damping coefficient (viscous)
e_h	Horizontal thrust eccentricity, inch
e_v	Vertical thrust eccentricity, inch
Fh	Horizontal transverse force, lbs
Fv	Vertical transverse force, lbs
F_1	Transverse force sensor No. 1 signal, lbs
F_2	Transverse force sensor No. 2 signal, lbs
f_n	Resonant or natural frequency, cycles per second
J_p	Polar moment of inertia, slug-ft^2
K_1	Transducer stiffness, lbs per inch
K_2	Soft coupling or soft mount stiffness, lbs per inch
Mh	Horizontal bending moment, inch-lbs
Mv	Vertical bending moment, inch-lbs
M_1	Bending moment sensor No. 1 signal, inch-lbs
M_2	Bending moment sensor No. 2 signal, inch-lbs
m_1	Propeller mass, pound (second)^2/inch
m_2	Flywheel mass, pound (second)^2/inch
Q,q	Torque, inch-lbs
T,t	Thrust, lbs
x	Longitudinal distance from a plane passed through the propeller cutting the blade at the 0.7R (normal to the shaft axis) to the centerline of the moment strain gage bridges on the sting balance
x_1,x_2	Longitudinal deflection of propeller and flywheel

ϕ	Angle by which the No. 1 transverse force or bending moment sensor has rotated clockwise from the balance's top dead-center position (vertical centerline)
ω	Circular frequency, radians per second
θ	Phase angle between propeller and flywheel
α	Phase angle between propeller and applied force
DTMB	David Taylor Model Basin
NSRDC	Naval Ship Research and Development Center

INTRODUCTION

An area of much concern today in Naval Architecture is in regard to vibration generated by the hydrodynamic action of propellers. With the desire for increased speed of high-powered vessels, large propeller-induced vibratory forces and moments ensue, causing shipboard vibration which can produce discomfort to passengers, damage to structures, and noise. These vibrations largely result from the fluctuations of the propeller forces generated when a marine propeller operates in a circumferentially nonuniform wake field. A number of papers[1-10] (among others)* have been published over the past several years dealing with attempts to experimentally determine magnitude and character of unsteady propeller forces on model-scale propellers operating either in simulated ship wakes in water tunnels or behind actual ship models in the towing basin. Such forces are transmitted to the ship's hull in several ways.[11]

In conjunction with a sizeable international program involved with the reduction of these forces and alleviation of the resulting detrimental effects, the U.S. Navy has been attempting to perfect theoretical and experimental techniques for predicting and measuring such forces. The forces with which the present discussion is concerned are those transmitted to the hull via the propeller shaft and bearings. In this connection, the Naval Ship Research and Development Center, formerly the David Taylor Model Basin (DTMB) has developed model-scale dynamometers or balances for the measurement of these shaft transmitted fluctuating force components associated with the axial, torsional, and transverse modes of the propeller rigid body vibration. Balance designs which incorporate measurement of dynamic components of torque and transverse force in addition to thrust, generally necessitate some compromise in the usable dynamic range of

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*A bibliography is listed on page 34.

the thrust system. This can occur, for example, in a case where the lateral rigidity of the tail shaft transducer element is reduced to provide adequate (signal to noise ratio) strain in the transverse force and bending moment flexures. These last two components permit the calculation of thrust eccentricity which, together with the transverse forces, produce the bending moments. Knowledge of the magnitude and frequency of these propeller forces can aid both the propeller designer and the ship designer.[2] Early DTMB balances measured only the thrust fluctuation and could be made very rigid in the other vibratory modes. But with the desire to measure more force components, the rigidity problem has persisted although recent advances in strain sensor and multiplicity of bridge development have alleviated it somewhat. Single-screw ship models fitted with such balances have had considerable tank testing at this laboratory.

Various mechanical devices and force transducers have been experimented with during the development. The development of the balances has paralleled similar instrument development being carried out at other research laboratories over the past ten years. Exchanges of technological information have aided in expediting the instrument development.

Often the experimentalist involved with developing such instrumentation finds it an arduous task to ultimately achieve satisfactory test results. In order to accurately relate the signals from each of the balance's component sensors to actual propeller forces while under test, it is necessary to statically and dynamically calibrate the instrument. Static calibration establishes the force-transducer sensitivity factor for each sensor, while the purpose of dynamic calibration is to establish the "flatness" of the response sensitivity over a range of frequency of an applied force.

A brief background of the NSRDC instrumentation development should help to establish the reasons for selection of the devices and techniques in current use.

It is the purpose of this paper to present:

a) A brief discussion of the characteristics required in a balance capable of accurately measuring fluctuating propeller forces.

b) A brief discussion of the dynamic analysis problem.

c) A description of a typical balance currently in use at NSRDC.

d) A description of the techniques and calibration data for the static and dynamic response of the balance in the axial, torsional, and bending modes.

BALANCE DESIGN AND CONSTRUCTION

A basic problem arises in designing successful vibration measuring devices of the type required in this work. Although a certain flexibility of the transducer is required for high signal-to-noise ratios, an opposite requirement - that of having high system rigidity to establish the natural frequency of the balance above the highest desired measurement frequencies - is also of prime importance. The range of force magnitude which a balance will experience depends on the size and speed of model ship hulls tested under conditions of Froude similarity. The frequency range of interest will correspond to the frequency of the number of propeller blades multiplied by the maximum shaft RPM and certain harmonics of that frequency. It is desirable to measure the forces and moments as close to the propeller as possible in order to avoid the necessity of taking the dynamic characteristics of the complete shafting system into account.

BACKGROUND OF DEVELOPMENT

As mentioned in the Introduction, a series of balances has been developed in a continuing program over several years to attempt to perfect the design and meet ever-increasing specialized requirements. Figures 1 and 2 contain photographs of two of these units. The major problem areas were early found to be in excessive shafting and thrust block flexibility which causes mechanical resonances and detracts from the percentage of the dynamic energy absorbed within the transducer section. The unit shown in Figure 1 was designed to alleviate shafting resonance; however, its chief disadvantage was a lack of stiffness in the stationary thrust block. The device shown in Figure 2 was purposely designed for a measurement frequency range from 0 to 60 cps. In the development of the early balances, sources of system noise were found to be the drive motor and shafting, thrust bearing, and the model structure. Methods were found which can isolate the measuring element section from vibrations being transmitted from the hull model or from the drive end of the balance. The latest type, Figure 3, measures six components of the propeller forces including: thrust (T), torque (Q), vertical and horizontal transverse forces (Fv and Fh) (assumed to be acting at 0.7 propeller radius), and vertical and horizontal bending moment in the model propeller shaft (Mv and Mh).

DYNAMIC ANALYSIS

The decision to go to balance designs which incorporate a heavy flywheel in the rotor assembly was based on the results of dynamic analyses carried out during the redesign of some of the early devices. In the United States and in Europe, a balance design which would be simple and rugged but would, in addition, possess a "flat" dynamic response over a broad frequency range and still achieve a satisfactory signal-to-noise ratio was in great demand.

The amplitude of motion of the propeller and transducer element can be calculated by considering a lumped mass-spring system representation as shown from the equations of motion given below.

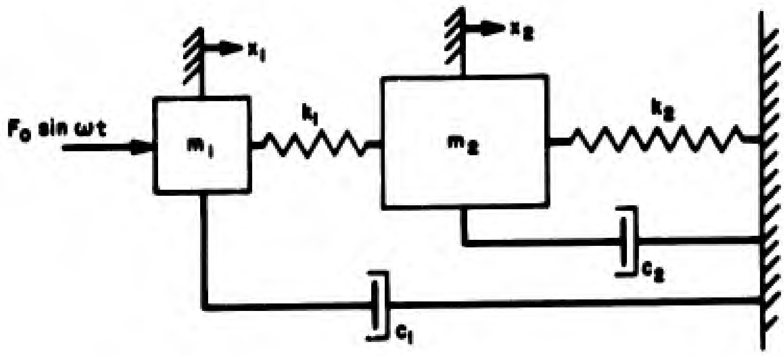


Figure 4. 2-Degree of Freedom Representation

C_1 = propeller damping coefficient

C_2 = flywheel damping coefficient

F_o = amplitude of periodic force

K_1 = spring constant of transducer

K_2 = spring constant of soft coupling

m_1 = propeller mass

m_2 = flywheel mass

t = time in seconds

x_1 = displacement of m_1 from equilibrium

x_2 = displacement of m_2 from equilibrium

ω = circular frequency, radians per second

The effects of changes in mass and stiffness of system components, as well as damping and entrained water, can be studied.

$$m_1\ddot{x}_1 = F_o \sin\omega t - C_1\dot{x}_1 - K_1(x_1 - x_2)$$

$$m_2\ddot{x}_2 = K_1(x_1 - x_2) - C_2\dot{x}_2 - K_2x_2$$

Solution of these equations by impedance methods will yield the amplitudes and phase angles of vibration of m_1 and m_2.

Early balance designs aimed at obtaining high natural frequency had a very stiff (thus low signal-to-noise ratio) transducer between the propeller and the thrust-bearing foundation. This arrangement will produce a single degree-of-freedom system if the foundation stiffness is high. If it is not of comparable stiffness to the transducer, the system natural frequency will be determined essentially by the foundation stiffness and the signal-to-noise ratio will have been sacrificed to no advantage.

To alleviate the need for a very stiff foundation, it was decided to mount a large mass (m_2) on a soft spring (K_2) - a seismic system - ahead of the propeller (m_1) and the transducer element (K_1). The inertia forces of such a mass increase with frequency and the amplitude of motion decreases rapidly as frequency goes up. Thus the "dynamic stiffness" of the foundation is very high at high frequency. The transducer output is proportional to the relative motion ($x_1 - x_2$) between the propeller and the mass or flywheel in this case.

It has been found that representation of this system by only two masses and two massless springs as shown produces calculated response curves very similar to those achieved with many lumped masses and springs. Therefore the more simple representation is utilized.

In connection with design of the latest balances, system dynamic response characteristics were calculated on the NSRDC analog network analyzer, as well as on the NSRDC IBM-7090 computer. In the latter case, the mechanical system was treated as two lumped masses connected together and to ground by springs and dash pots and the response of the transducer sections under the influence of a simple harmonic driving force or moment over the frequency range was determined. The amplitude and phase angle (relative to the input force) of each mass is obtained. A brief description of the computer program can be found in Reference 13.

Values of the system spring constants and masses can be readily calculated or measured; however, accurate estimates of system damping factors and entrained water mass are not easily obtained. Two different damping factors and two different phase angles are considered in the analysis. Damping on the flywheel C_2 is assumed to be viscous in nature but is relatively small compared with propeller damping. Consequently, the phase angle between the propeller and flywheel θ is very small until the flywheel/soft coupling system goes into resonance, and then this phase angle rapidly goes through 90 deg and soon becomes close to 180 deg. The other phase angle, that between the propeller motion and the applied force α is governed by the propeller damping C_1. A good estimate of the propeller damping factor is important to balance response predictions because damping reduces the amplitude of motion over that of an undamped system and, in addition, produces a marked change in the shape of the response curve in the areas adjacent to system resonances. Propeller damping estimates have been proposed by a number of authors, the method outlined in Reference 12 being used in the subject analyses.

Plots of a typical calculated response curve or transfer function for the case of .ero damping and the associated phase angles for a particular set of values of the sys:em damping factors are shown below.

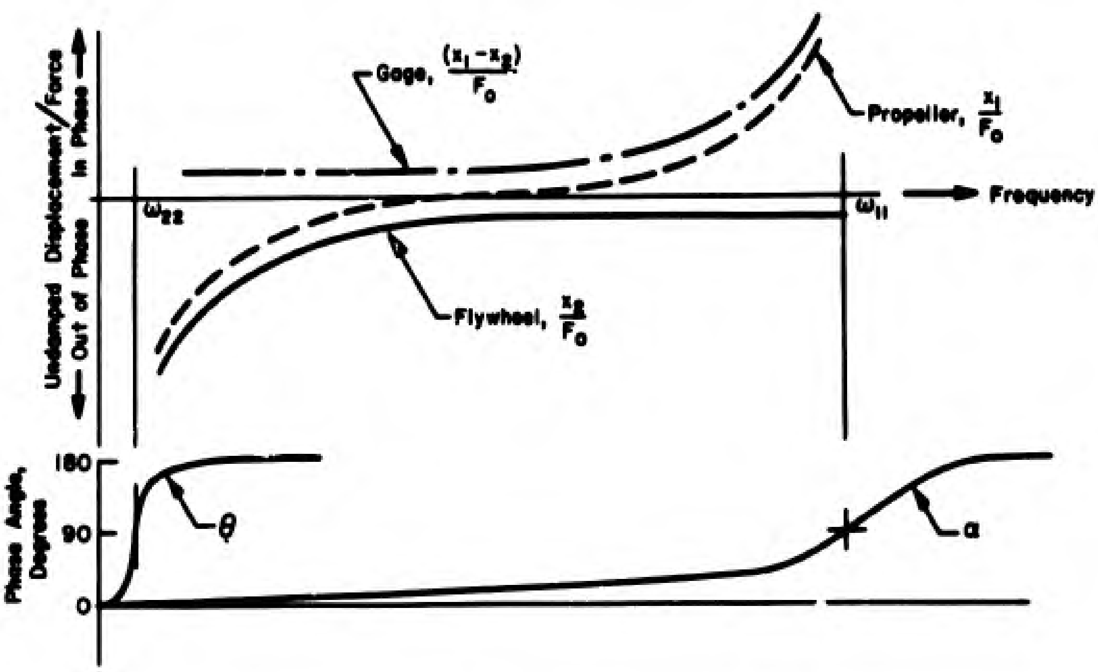


Figure 5. Transfer Function and Phase Angles 2-Degree of Freedom System

The Netherlands Ship Model Basin (NSMB) has developed an experimental device for simulating the propeller damping and entrained water mass on their model-scale fluctuating propeller balances.[16]

Since NSRDC dynamic calibrations are performed with the propeller essentially in air rather than water, some discrepancy can exist between the calibrated response and the system response while under test conditions. However, such discrepancy is only significant in areas of resonance of the system response, since transfer function curves are characteristically flat away from resonances.

BALANCE DESIGNS FOR DESIRED CHARACTERISTICS

In light of the design considerations previously discussed, careful attention should be given to the size, shape, and material for the balance; type of transducer, mounting and drive systems, associated instrumentation (preamplifiers and power supplies), and finally the signal processing and analyzing equipment. (See Appendix.) The most successful NSRDC balances have employed transducers connected to the rear of heavy shaft flywheels mounted on soft bearing supports and driven through soft couplings or rubber timing belts to achieve a two-degree-of-freedom device. Dynamic loading of the tailshaft by fluctuating propeller forces in the axial, torsional, and bending modes will excite the resonant frequency of the primary (flywheel-shaft mass on the soft supports) system which is estimated at 10 to 15 cps in all principal modes. Thus, if the rpm is increased rapidly, the blade-exciting frequency will soon exceed this natural frequency and the 100 + lb flywheel, in conjunction with the "softness" of the supports, will present a high dynamic stiffness such that the forward end of the transducer shaft will be relatively fixed in space. This "soft" supporting of the rotor shaft also tends to reduce the transmission of vibration from the drive motor end into the balance transducers. The secondary mass-spring system which is essentially the propeller-transducer portion is then able to vibrate relative to this relatively fixed flywheel mass.

It is interesting to note that the calculated response characteristics of several balances recently constructed at NSRDC by the method mentioned in the previous section compared favorably with measured response obtained experimentally after construction.

SELECTION OF TRANSDUCERS

Model-scale dynamometers utilizing transducers of the differential transformer, capacitance, piezoelectric and strain gage principles, have been developed at the Model Basin.[7]

In general, piezoelectric gages have a very high sensitivity, are self-generating, but cannot respond to a static load. Capacitance gages can be designed with high sensitivity, are ideal for frequency modulation, and can be used as two-part gages having one part on the rotating shaft and one on the stationary thrust bearing, although such devices require considerable space. Differential transformers and electromagnetic devices also are suitable for two-part transducers. Strain gages of the metal film and semi-conductor type have less sensitivity than the f t two types mentioned but take up very little space, can be installed for multi-direction measurements, and can be used with telemetering or slip-ring signal transmissions.

In view of the above-mentioned characteristics, strain gage sensors were selected for the latest balances.

A number of sting balances, one gaged with metal film and several with semi-conductor gages are now on hand (Figure 6).

BALANCE DESCRIPTION

The Basin Six-Component Submarine and Surface Ship balance (Figure 3), one of several currently in use at NSRDC, will be described here as to design and operation since it is typical of these devices. In addition, it is the balance on which the subject calibrations were performed. (For brief descriptions of the various NSRDC balance systems and data analysis techniques, see the Appendix.)

This instrument is a model scale, softly supported flywheel-type alternating propeller force balance, similar in overall design to the surface ship dynamometer built at the Netherlands Ship Model Basin.[2,3]

Photographs and diagrams of the unit are presented in Figures 3 and 7. It employs strain gages bonded to the propeller tailshaft for force sensing and slipring-brush transmission of signals.

The strain gages are Budd 120 ohm metal film or Microsystems 120 ohm or 350 ohm semi-conductor, the latter having 50 to 60 times the output of metal film gages.

The stainless steel shaft is driven by a shock-mounted direct-current waterproofed electric motor. The drive is through nonintegral speed ratio Morse tooth timing belt sprockets and 100 + lb flywheel, supported on water-lubricated bronze-impregnated teflon bearings. The bearing areas of the shaft are hard-chromed. This rotor assembly is supported in a frame which is suspended by Lord Manufacturing Company shock mounts from the model at the rotor's horizontal and vertical center of gravity. The six-component balance fabricated of 17-4-PH high alloy steel forms a part of the tailshaft and has at its end a tapered section fitted for the propeller hub (Figure 8a). The joint between the transducer and the flywheel tailshaft is also a tapered design, for positive seating at the joint. The requirement that this balance operate in surface ship models necessitates a long tailshaft which results in a compromise of natural frequency in bending. The tailshaft is enclosed by a stiffened tubular housing which is rigidly cantilevered from the flywheel frame. This housing forms the stern tube of the ship model. A "soft" rubber ring is inserted between the after end of the housing and the model hull to retain the soft support of the balance within the hull. In order for the housing to add transverse rigidity to the shaft, the stern tube Babbitt bearing was tightly fitted to the shaft. It was found that this bearing could also be water-lubricated allowing all the sleeve bearings to be pressure-fed from a single submersible-type pump.

The transducer strain-gage leads run through the hollow shaft, the a-c signals being fed to an ACF Electronics Co. transistorized preamplifier package housed inside the flywheel (Figure 8b). This amplifies the six signals by 40 decibels (x 100), producing a minimum signal in the order of several millivolts which is fed to coin-silver slip rings having three silver-graphite brushes per ring. Thirty slip rings are provided in the waterproofed housing. These are utilized for excitation of the preamplifiers, d-c excitation of the strain-gage bridges, the d-c and a-c components of the bridge signals, and grounds. Five spare rings exist (Figure 8c).

The stator leads from the brushes are fed through a waterproof 37 pin connector and thence through a Belden waterproofed 19 shielded pair cable. The cable output leads are connected into a conditioning amplifier which is rack-mounted (Figure 8d) and amplifies the signal to an output level of 2 volts peak to peak for magnetic tape recording. An attenuator-gain switch is provided here. This rack also contains provision for obtaining d-c calibration, a-c calibration, bridge balance, and in addition, has two microvoltmeters (d-c) for monitoring d-c signals and a pulse conditioning circuit for shaping and sizing the tooth-pickup pulses from the probes prior to tape recording.

A single steel gear tooth and a 90-tooth gear are attached to the flywheel shaft to excite two electromagnetic probes as the shaft rotates. The single tooth and one of the 90 teeth are aligned circumferentially with the number one propeller blade for phase retention. The outputs of these pickups are utilized as shaft rotational position reference signals for analysis purposes.[7]

A standard 10-slot disc and galvanometer coil are used to actuate a digital RPM counter.

BALANCE SPECIFICATIONS

Transducers: Thrust, torque, transverse (vertical and horizontal), bending moments (vertical and horizontal).

Transducer pickups: Metal film or semiconductor stain gage.

Measurable dynamic frequency: Thrust = 30 to 1200 cps, torque = 30 to 450 cps, transverse force and bending moment = 30 to 200 cps, also d-c component.

Force range: T = -20 to +100 lb, Q = 0 to 100 in-lb, F_v and F_h = 0 to 10 lb, M_v and M_h = 0 to 100 in-lb, drive d-c motor (0 - 400 volts).

Maximum dimensions: Length = 6 ft 6 in., width = 1 ft 2 in., height = 1 ft 4 in., total weight = 300 lb.

CALIBRATION

CALIBRATION PHILOSOPHY

Determination of the sensitivity of each of a particular balance's transducers is performed both statically and dynamically. Transducer interaction sensitivity between axial, torsional, and transverse forces is obtained as part of this calibration.

Static and dynamic interaction must be determined independently. Static interaction can be minimized during manufacture of a strain-gage transducer by alternate loading, reading, and adjustment of a sensor's output signal. However, after a satisfactory static minimum interaction is obtained, excessive dynamic coupling may still prevail. The flatness of the response curve of, say the thrust mode, may be disturbed in the region of resonance of another mode, say bending. That is, if the device's elastic system is more limber in bending than in thrust, its resonance in bending will be lower than the axial resonant frequency. The interaction of bending on thrust can thus cause a "hump" in the thrust response curve.

Static calibration using calibrated dead-weight forces establishes the force-transducer sensitivity or output of the particular transducer sensor. Dynamic calibration establishes the flatness of the transducer's response over a range of frequencies of the applied force and should follow the static sensitivity and interaction calibration since the static determination is more simple to obtain.

The dynamic calibration should consist of the application of one or more known dynamic forces within the balance's force range over a frequency band which exceeds the resonant frequency of the balance. Any deviation of the sensitivity level from that obtained during static calibration indicates a resonant or antiresonant response and will be an indication of which portion of the frequency range is flat. Thus the user of the balance is made aware of the frequency range in which the static sensitivity calibration is valid, i.e., in the areas far enough removed from resonant amplification that system damping does not appreciably affect the shape of the response curve.

Experience has indicated that reliable signal data can be obtained only when the natural frequency of the measuring balance is above the highest significant propeller excitation frequency. In other words, the dynamic propeller-model measurements should be limited to the frequency range corresponding to the flat portion of the dynamic response curve.

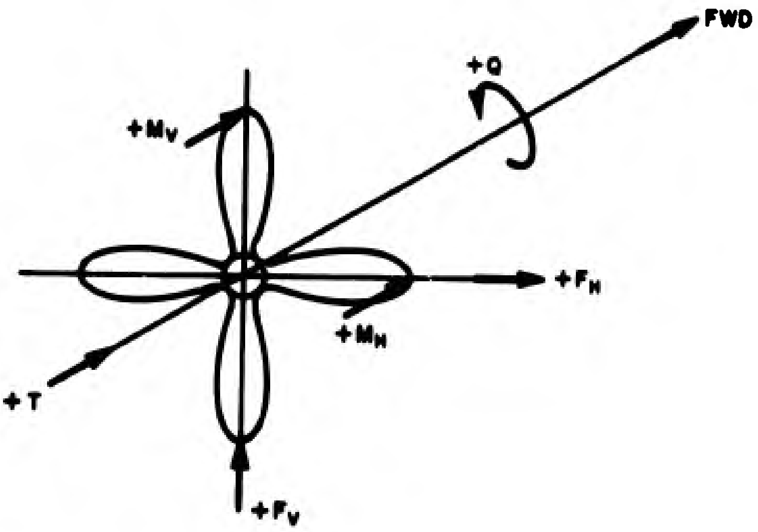
If this dynamic calibration is performed with the balance installed in the test model, the response of the overall elastic system is determined.

That is, in brief, the problem of determining the instrument's response characteristics, information which is essential to the engineer in establishing his measurement accuracy. The procedures and equipment used for the static and dynamic calibrations are described in the following sections.

STATIC CALIBRATION

Static forces are applied to the tailshaft through a loading disc which replaces the test propeller. Calibrated dead weights are used through a system of pulleys and arms to subject the transducer to thrust, torque, transverse force (vertical and horizontal), and bending moment (vertical and horizontal) (Figures 9 and 10).

The polarity of forces and moments measured on NSRDC balances is indicated on the sketch below:



Typical single-screw ship model balance load ranges used currently are given below.

	Estimated to be approximate maximum propeller loadings for a particular test
Thrust	0 - 100 in 10-lb increments
Torque	0 - 100 in-lb in 10-in-lb increments
Transverse Vertical	0 - 10 lb in 1-lb increments
Transverse Horizontal	0 - 10 lb in 1-lb increments
Bending Moment Vertical	0 - 100 in-lb in 10-in-lb increments
Bending Moment Horizontal	0 - 100 in lb in 10-in-lb increments

Increments of 10 percent of the total loading range are used in all calibrations.

Curves of thrust sensitivity to the above loads are shown in Figure 11 which presents at a glance the interaction characteristics for a six-component balance fitted with a semiconductor sting balance. In like manner, sensitivities of the torque, vertical transverse, etc., transducers are shown in Figures 12 through 16 for each loading mode. Since the force sensors are of the strain-gage bridge principle, the sensitivity is directly proportional to the bridge excitation voltage.

The magnitude of the voltage applied to each bridge during calibration depends upon the bridge resistance and is selected such that equal heating effects will be realized for each bridge. The bridge voltages are listed on the calibration figures.

DYNAMIC CALIBRATION

The following techniques, in general, reflect the present dynamic calibration philosophy adopted by NSRDC for this type of work. As stated earlier, it is felt that good calibration practice includes static calibration in increments over the input force range and dynamic calibration with a known input force in increments of driving frequency over a suitable frequency range.

Naturally as the number of force components being measured increases, the problem of interaction between transducers increases. The degree of static and dynamic interaction must be determined so as to enable the experimenter to know just what percentage of the output from any transducer, say thrust, is attributable to the thrust force alone and what percentage to the torque and side forces. Maximum static interactions of about three percent were obtained with the latest strain-gaged transducers.

Some experimenters[2] using this type of balance do not introduce the dynamic calibration force over a range frequency, but rather use a technique of striking the end of the stationary propeller shaft a sharp, light blow. By recording the transient response of the device's transducers, a resonant decay curve of the system's mechanical response can be obtained. The limitation of such a procedure is that only the fundamental or primary resonance magnification factor is obtained. Although the logarithmic decrement of the successive decaying oscillations provides an insight into the mechanical system damping, no indication of the magnification factor of a transducer system throughout the intended useable range of frequency is available. In addition, it is very difficult to determine dynamic interaction effects from one component to another by such a nebulous input force technique.

NSRDC feels a necessity for dynamic calibration over the working frequency range rather than simply the determination of the propeller mass/transducer resonant frequency.[7] Hence, an alternative to this procedure is to introduce, in succession, known input forces at known frequencies and measure the response of each element of the balance to each input force or moment.

This latter procedure consists of connecting one or more small electromagnetic vibration generators to an adapter mounted in place of the test propel und inducing a vibratory force, generally of constant magnitude, er the frequency range (Figure 17). The system response is monitored on an oscilloscope or a-c voltmeter from the output of the balance's transducers and of auxiliary accelerometers and/or force transducers connected to the test balance. The frequency of vibration can be controlled well above 2 KC; however, the absolute value* of the constant force is known only within ± 5 percent. A discussion of the overall accuracy is given in a later section.

Certain practical problems which have faced the dynamic calibration development include the following:

a) Attachment of heavy mechanical or electromechanical shakers to the tailshaft complicating rotation of the shaft and necessitating tailshaft end mass corrections of significant magnitude to the dynamic response curves.

b) Inability to be certain of the exact magnitude of the exciting force of an electromechanical shaker.

c) The introduction of a static preload force required on some balance designs, without introducing mechanical noise from a stationary device which would bear against the tail shaft end.

d) The uncertainty of the validity of using linear superposition of interaction effects from transducer to transducer from single-mode force inputs during calibration.

e) The unknown actual wetted propeller entrained mass and damping coefficient. (Estimates of entrained mass and

* Actually, only magnitude of input force is monitored; phase angle is not.

damping have been calculated by methods suggested in Reference 12.)

A number of devices have been used to introduce the required dynamic forces and perform related functions. These include (Figure 18) compressed air chambers (for static preload), electromagnetic shakers (Figure 19), motor-driven scotch-yoke spring vibrators (Figure 20), force generators with impedance heads (Figure 21), and proposals of using eccentric mass vibration generators or hydraulic piston pulsators have been pursued. Large, heavy electromechanical shakers have been replaced by small light shakers which can be attached to the tailshaft in place of a propeller and rotated during calibration.

FREQUENCY RESPONSE INSTRUMENTATION

Dynamic excitation forces are applied with the small electromagnetic shakers (mini-shakers)* which weigh approximately 1 lb each (Figure 17). The basic force inducers are Mandrel Model EVS-7 shakers with a modification to the moving coil assembly to increase its weight to 78.2 grams, thereby increasing the force output to approximately 2 lbs peak to peak. A schematic diagram of the control circuit is shown in Figure 22. The manufacturer's specifications list the natural frequency of the shaker as 29 cps and with the heavier coil, the natural frequency is about 22 cps.

The mini-shakers are attached to the shaft in place of the propeller on an aluminum fixture which is designed to apply thrust, torque, transverse, and bending forces one at a time, the mode depending on the way the fixture is employed (Figures 23 and 24). The weight and polar moment of inertia of the shaker-fixture package is designed to approximate the weight and inertia characteristics of wet aluminum

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* Mandrel Industries Inc., P.O. Box 36306, Houston 36, Texas.

propellers currently in use at NSRDC. Since the shaft is rotated during calibration, power input to the shakers is by means of a small slipring-brush assembly mounted on the aluminum fixture. This entire assembly weighs approximately 1.75 lb with one shaker and 2.75 lb with 2 shakers attached.

The maximum force output of about 2.0 lb peak-peak is utilized during the calibrations. When moments or torques are being applied, the magnitude of moment or torque is equal to the product of a 2.0-lb force and the applicable moment arm, e.g., in the case of torque, the effective diameter of the circle on which the fixture shakers are mounted is 1.25 inch, thus a torque of 2.50 in-lb peak-peak ensues.

The shakers are driven by an amplified oscillator output. Transducer signals are read on either a calibrated root-mean-square alternating current voltmeter or a calibrated high-frequency galvanometer string oscillograph.

DYNAMIC CALIBRATION PROCEDURE

Final dynamic calibration is accomplished with the balance mounted in a ready-for-use condition in a test model and the rotor shaft rotated at approximately six revolutions per second with normal lubrication of the shaft bearings. A slight static preload of the thrust bearing is obtained during calibration by having the test model tilted with the stern up at about a 10-deg angle.

The frequency range and increment of range is determined by the practical lower driving limit of the mini-shaker (natural frequency of 22 cps) which is about 50 cps and a high limit above the anticipated useful range of a particular sensor of the balance. For example, the thrust transducers and system having a resonant frequency above 1500 cps is excited up to and including 2 KC in order to cover the resonant range. The other channels are treated similarly.

The dynamic calibration is performed in two distinct phases: frequency response and dynamic force determination.

The procedure for determining the balance's frequency response is as follows:*

1) With complete measurement system electronics energized, background noise levels on all channels are read.

2) Shaft is rotated with the mini-shaker and fixture rig in place at a low rate of rotation (approximately 6 revolutions per second) in order to break static friction in bearings, etc. Noise levels on all channels are read.

3) Thrust excitation (approximately 2-lb peak-peak) is applied over a suitable frequency range (from 70 cps to 2 KC in 20-cps increments). Signal levels on all channels are read.

4) Torque excitation (approximately 2 1/2 in-lb peak-peak) is applied over a range of frequencies (from 70 cps to 1 KC in 20-cps increments). Signal levels on all channels are read.

5) Side force excitation (approximately 2-lb peak-peak) is applied over a frequency range (from 70 cps to 1 KC in 20-cps increments). Signal levels on all channels are read for: (a) application on vertical centerline of the propeller plane on the balance, and (b) application on horizontal centerline of the propeller plane on the balance.

6) Bending moment excitation (approximately 2 in-lb peak-peak) is applied over a frequency range (from 70 cps to 1 KC in 20-cps increments). Signal levels are read for all channels for: (a) application on vertical centerline of balance, and (b) application on horizontal centerline of balance.

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* In this calibration procedure, the values of force and frequency magnitudes stated in parentheses apply only to existing instrumentation scheme.

Signal levels for each channel from the principal and interaction excitations are plotted on the ordinate axis and frequency on the abscissa. It is assumed that the various mutual interactions superpose linearly.

DYNAMIC FORCE DETERMINATION

In order to accurately relate the constant magnitude electromechanical dynamic calibration to a known force, an additional step is introduced to the calibration procedure. This consists of attaching a motor-driven mechanical "scotch-yoke spring" device (Figure 20) to the end of the tailshaft to apply axial force (Figure 25). Both the mechanical throw of the yoke (Δ) and the spring rate (k) of the two coil springs are measured as accurately as possible. A discussion of accuracy is presented below. The force ($k\Delta$) is applied to the shaft by rotation of the device. The yoke flywheel is first rotated carefully by hand and then motor-driven at frequencies up to 50 cps, since 60 cps is the design limit for this particular unit. The transducer signals are again read on the oscillograph. Thus all output signals obtained by excitation with the mini-shaker can be referenced to the known force obtained mechanically. An alternate method would consist of using a calibrated piezoelectric force gage to monitor shaker output force during a calibration. Accuracy of force magnitude determination would be approximately the same by both methods. The dynamic calibration curves for the six-component balance are included as Figures 26 through 28.

SHAKER RESONANCE

The procedure outlined above was complicated by a resonance exhibited by the mini-shaker. An investigation of this problem is discussed below.

By monitoring and controlling the driving amperage to the shaker, it should be possible to exert a constant force in the frequency range from 50 cps to above 1 KC. However, the shakers now in use unfortunately have what appears to be a transverse resonance at a frequency between 300 and 400 cps, and the force output in that range rises although satisfactory control can be maintained outside this range. The response curve of the balance thus indicated a "hump" in the range in which the input forces increases. This leave an uncertainty as to the true dynamic response. Consequently, the following procedure was used to investigate this problem.

First, the mini-shakers were checked in the NSRDC Acoustics and Vibrations Laboratory for force constancy versus shaker frequency by driving a soft mounted 100-lb steel plate with each shaker. The mini-shaker was bolted rigidly to the plate on which a NSRDC piezoelectric impedance head (accelerometer-force gage) was mounted.[14] The resonant frequency of the softly supported plate arrangement was 5 cps. Consequently, for frequencies above 20 cps, constant acceleration for a constant force could be expected over the frequency range.

The force amplitude versus frequency curve from this test (Curve A of Figure 29) indicates a large increase in force output between 300 and 400 cps. By varying the alignment of the most sensitive axis of the impedance head, with respect to the longitudinal axis of the mini-shaker, it was determined that the mini-shaker has a transverse resonance in this frequency range. This proves to be detrimental to balance calibration because the transverse resonance produces an increase in the axial force output.

When the balance was excited for frequency response with the mini-shaker, an increase in amplitude was also noted in this same frequency range when the shaker transverse axis was aligned parallel with the sensitive axis of the particular sensor being read (see Curve B of Figure 29).

In order to verify that this effect was inherent in the mini-shaker and not in the balance itself, the balance was excited by means of another electromechanical shaker, Calidyne Model 6.[15] This unit can exert a maximum force of 25 lb, but due to its size and weight (60 lb), it is not readily adaptable for applying all the required forces to the rotating tailshaft. The thrust (t) sensor output curve for this test is the Curve C of Figure 29. It appears as if the "hump" in Curve B is attributable to the mini-shaker. Thus, the calibration curves in Figures 26 through 28 are presented as the response of this balance with the resonant effects of the mini-shakers removed.

Apparently, the thin disc metal diaphragms designed for the coil, axial f_n of 29 cps, have a transverse f_n in this range of 300 to 400 cps. One solution to the problem would thus be to redesign the shaker coil suspension. An attempt is being made to replace the metal disc diaphragms supporting the coil with soft rubber fixtures.

Another solution would be to use a calibrated force gage used in series with the input shaker and transducer to monitor the actual force which the transducer sees (Figure 29, upper curve). This has been done in recent dynamic calibrations.

ACCURACY OF CALIBRATION

The major sources of inaccuracies involved here are connected with the static test stand, the dynamic test fixtures, the balance itself, and the instrumentation.

The following table is intended to indicate the typical inaccuracies of various components of a calibration or a crosscheck on some steps of calibration.

	Probable Deviation from True within
1. True weights of calibrated dead weights	$\pm$ 0.1 Percent
2. Lengths and angles of moment arms and test fixtures	$\pm$ 0.1 Percent
3. Ability of measuring system to repeat static readings	$\pm$ 0.5 Percent of full scale
4. Same D-C voltmeter used in calibration and under test	$\pm$ 0.0 Percent

5. Use of calibrated force gage monitoring shaker input (to alleviate need for scotch yoke if desired) — $\pm$ 5.0 Percent

6. Scotch yoke: Δ = throw of yoke, K_s = spring constancy of spring. Force = $K_s \times \Delta$ — 0.25 in. $\pm$ 3.0 Percent, K_s within $\pm$ 2.0 Percent, $\pm$ 5.0 Percent

7. Frequency accuracy of calibration driving oscillator — $\pm$ 0.2 Percent

8. Accuracy of shaker calibration on 100-lb plate — $\pm$ 5.0 Percent

Thus the overall accuracy of static calibration is within 1.0 percent and of dynamic calibration, is within 7.0 to 8.0 percent.

It should be noted that the calibration readout is obtained at the final dynamometer signal output point and thus the same overall measurement system is involved in both calibration and model testing.

The phase angles of the propeller to the flywheel are assumed to be positive throughout the useable range of balance frequencies; i.e., in the range up to 80 percent of the respective transducer resonances.

SUMMARY AND CONCLUSIONS

Experience in design and use of propeller fluctuating force balances has indicated that such devices should have high system rigidity and as high a transducer sensitivity as possible. The first requirement, based on achieving a mechanical system resonance above the desired measurement frequency range, has been fulfilled by the use of a heavy flywheel mounted between the propeller transducer and the hull model structure. Satisfactory transducer sensitivities have been obtained through the use of either high output per unit load capacitance thrust transducers or semiconductor strain gages. In addition, the use of solid-state preamplifiers located in the balance shafting between transducers and transmission slip rings has greatly helped in attaining high signal to noise ratios.

Design of the balances has been aided by analysis of simplified lumped mass-spring system representations, calculations being carried out on a digital computer. System resonant frequencies by analytical and experimental methods have been in good agreement (within 10 percent).

NSRDC balances are calibrated both statically and dynamically. Static calibration produces the sensitivity of each transducer to forces applied in axial, torsional, and bending modes. Dynamic calibration consists of exciting the measuring system with a known force over a broad range of frequency in order to establish the dynamic response of each transducer's sensitivity over the desired range of measured frequency. This technique is favored over those in which only the primary natural frequency of each transducer is determined. The latter method is not capable of establishing other resonances or anti-resonances which may be produced by the interaction of forces from one mode to another and can significantly affect the response curve of a particular transducer.

Certain limitations exist with the NSRDC calibration technique. Propeller viscous damping coefficients are not accounted for in the experimental dynamic calibration. However, by introducing estimated damping factors into calculations of system response of lumped mass-spring representations of the physical balance, it appears that propeller damping will significantly affect dynamic response only at frequencies corresponding to system resonances or anti-resonances. Experimental determination of propeller damping and entrained mass effects on calibration response could possibly be achieved by utilizing a device of the type recently developed at NSMB.[16] Precise determination of the force output of vibration generators used in dynamic calibration excitation can be accomplished by monitoring the shaker force output with a calibrated force gage of the type utilized in NSRDC Acoustics and Vibrations Laboratory impedance heads. This technique has been successfully used in the calibration of the six-component dynamometer for the 24-inch water tunnel.

In conclusion, it is the author's opinion that the calibration technique described herein is capable of providing satisfactory calibrations of propeller model fluctuating force balances. Adaptation of several modifications to the described method, as discussed above, should be made to improve the accuracy of the calibration.

APPENDIX

DESCRIPTION OF OTHER PROPELLER MODEL FLUCTUATING FORCE BALANCES IN USE IN THE HYDROMECHANICS LABORATORY

DESCRIPTION OF SUBMARINE MODEL FLUCTUATING THRUST BALANCE

This is an inertial mass (flywheel-soft coupling) type balance which measures thrust by means of a capacitance-type transducer. A diagram of the instrument is shown in Figure 30 and photographs are included as Figure 31. One plate of the transducer is effectively the after end of the rotating K-Monel flywheel and the other, the forward end of a short, stiff tailshaft. The cylindrical-shaped capacitor assembly is machined to form a flexure, the axial stiffness of which approaches 1×10^6 pounds per inch. The transducer acts as a frequency-determining element in a 1.5-megacycle gage oscillator circuit. Under the propeller thrust load, the flexure plates move axially varying the plate gap and a frequency shift, proportional to the thrust load occurs in the 1.5-megacycle oscillator. A typical frequency shift approximates 5 KC. The fifth harmonic of the gage oscillator shift is used to detune a superheterodyne frequency-modulated receiver unit (initially tuned to 4.5 KC). The F-M receiver and demodulator yields a voltage output which is a linear function of the carrier frequency deviation which is itself a linear function of the propeller thrust-bearing load.

The flywheel support sleeve bearings are of Kel-F (thermoplastic) material and are water-lubricated. The 6-inch long tailshaft is sufficiently stiff in bending and torsion to preclude any significant signal output of the thrust transducer under lateral or torsional loads. A self-aligning face type Kel-F thrust bearing is included to reduce thrust-bearing noise. The entire rotor assembly is encased in a conical-shaped rigid stainless-steel shell which is designed to conform with the shape of model submarine hulls. Steady thrust up to about 250 lb can be measured and the frequency range of thrust signal measurement is 0 to above 1000 cps. Power and signal transmission to the rotating flywheel is accomplished with precision sliprings. The shell is free-flooding which necessitates complete waterproofing of cables and electrical components contained therein.

24-INCH VARIABLE PRESSURE WATER TUNNEL BALANCE DESCRIPTION

The balance is a six-component design and is basically the same as the Basin six-component unit. It is mounted inside a cylindrical housing on the longitudinal centerline of the NSRDC 24-inch cavitation tunnel (Figure 32). Soft mounting and noise isolation are similar to that employed in the Basin Model system. However, design differences permitted in tailshaft and sting balance (Figures 33a and 33b) dimensions have resulted in an increase in the lateral and torsional high-frequency resonances which extend the dynamic range to about 650 cps in torque and 400 cps in bending.

The totally submerged dynamometer is driven by a d.c. motor through a slipring shaft and amplifier housing, all of which are exterior to the tunnel casing. The connecting shaft (1.75-inch diameter) is hollow to carry the signal cables and passes through a stuffing tube as it enters the tunnel. The transducer section is gaged with semiconductor strain gages, the bridge layouts being shown in Figure 34. A block diagram of this instrumentation circuitry is included as Figure 35. The useable frequency range is:

Thrust	30 to 1300 cps
Torque	30 to 500 cps
Lateral Force Measurements	30 to 400 cps

BASIN MINIATURE TWO-COMPONENT BALANCE DESCRIPTION

This unit is a scaled-down version of the water tunnel balance. However, it is presently limited to measurement of thrust and torque. It was built to allow fluctuating force measurements to be made on twin propeller ship models, especially where stern design entails lengthy outboard tailshaft and strut arrangements.

The balance (Figure 36) is of the heavy (Tungsten carbide) flywheel, soft-mount design, and the housing diameter has been limited to 2 inches. The semiconductor strain-gaged transducer section is adjacent to the propeller hub and all system components such as timing pulse pickups, flywheel, soft mounts, thrust bearings, and soft coupling are contained within the shaft housing. The transistor amplifiers and sliprings are located within the model hull.

The following characteristics of the dynamometer have been obtained experimentally:

	Thrust	Torque
Primary natural frequency (f_n) (15-lb flywheel)*	10 cps	10 cps
Secondary natural frequency (f_n)	1400	850 cps
Lateral f_n	350 cps	
Base sensitivity (without amplification)	0.35 mv/lb	1.05 mv/in-lb
Interaction sensitivity, maximum	2 percent from torque	3 percent from thrust
Fixed amplifier gain	100	100
Bridge excitation voltage	15 volt d-c	15 volt d-c

DATA ANALYSIS

Force transducer and reference signals generated from the subject propeller balances are recorded on magnetic tape.

SPECTRUM ANALYZER

The unfiltered magnetic tape records are processed by two different methods. The first method consists of analyzing the tape "loop" for each rpm run in the NSRDC Seadac facility, an energy spectrum analyzer. This system produces for each test run a plot of amplitude (voltage level)

* Flywheel size limitations necessitate the use of very soft mounts and couplings. Soft, but durable, elastomers used in the manufacture of these components have been difficult to obtain.

versus frequency so that an energy spectrum of the wave form is obtained. Any point plotted represents the data passed through a filter having a bandwidth of 5 cps.

SAMPLING SYSTEM

The second method of analysis consists of digitizing the signal wave form at uniform angular intervals of shaft rotation, say every three degrees. These intervals are marked on adjacent tape channels as the data are being recorded and are generated by two electromagnetic pickups. One pickup generates one pulse per shaft revolution. This pulse is called the start pulse. The second pickup, placed in close proximity to a steel gear, generates one pulse each time a tooth passes under it. Gears of 60, 90, and 120 teeth have been used. By using a 120-tooth gear, a pulse can be generated each three degrees. This pulse is termed the "trigger" pulse.

The operation of this data reduction system is briefly as follows. The tape signal is reproduced into the digitizing electronics. The digitizing equipment remains inoperative until arrival of a start pulse and a coincident trigger pulse. At this time the amplitude of the signal is measured and recorded in digital form. At the next trigger pulse (three degrees later in shaft rotation), another reading of the signal amplitude is recorded until 120 pulses have been counted. The process is then repeated for a specific number of successive shaft cycles (100 to 300) depending on signal stability. The resulting digitized record is then enetered into a digital computer which first computes the average amplitude of the signal wave for each of the 120 selected positions of the shaft. This average wave form which is relatively free from random noise is then analyzed for its Fourier coefficients. The final data normally consist of the alternating torque expressed as a percentage of the mean or steady-state torque and the alternating thrust and transverse forces as a percentage of the mean thrust.

TRANSVERSE FORCE AND MOMENT MEASUREMENT DURING TESTING

Although the primary purpose of this report is to describe calibration, an important operating characteristic of the Basin six-component balance should be noted. The design of the balance is such that the side force and bending moment sensors (bonded strain gages) rotate with the propeller shaft. Thus, their sensitivity to a stationary (in space) transverse force varies sinusoidally. Consequently, the side force sensors which are designated F_1 and F_2 (F_1 being located 90 degrees ahead of F_2 for clockwise rotation) produce two signals which must be combined vectorially to give true vertical and horizontal transverse force. This procedure pertains to the bending moment sensors also. Thus, at any particular angular position of shaft rotation, forces and bending moments are derived by:

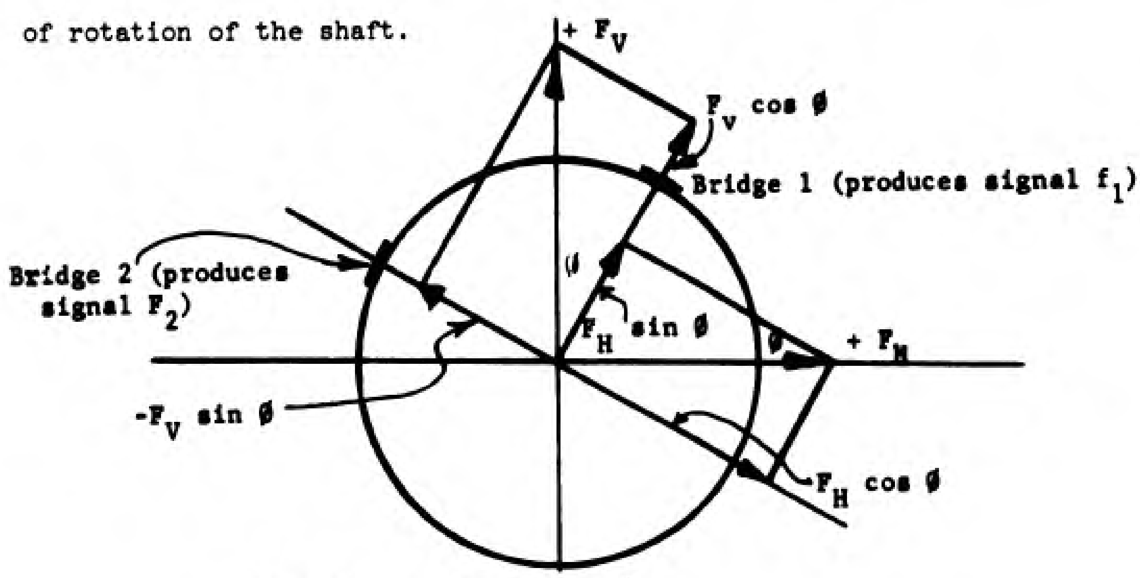
(1) $F_v = F_1 \cos \phi - F_2 \sin \phi$

(2) $F_h = F_1 \sin \phi + F_2 \cos \phi$

(3) $M_v = M_1 \cos \phi - M_2 \sin \phi$

(4) $M_h = M_1 \sin \phi + M_2 \cos \phi$

The diagram below shows graphically the situation at an instant of rotation of the shaft.



$$F_1 = F_v \cos \phi + F_h \sin \phi$$

$$F_2 = -F_v \sin \phi + F_h \cos \phi$$

Multiplying F_1 through by cos $\emptyset$ and F_2 by sin $\emptyset$ and subtracting results in:

$$F_1 \cos \emptyset - F_2 \sin \emptyset = F_v$$

and by multiplying F_1 by sin $\emptyset$ and F_2 by cos $\emptyset$ and adding, the result is:

$$F_1 \sin \emptyset + F_2 \cos \emptyset = F_h$$

THRUST ECCENTRICITY

As mentioned earlier, the Basin six-component balance system allows measurement, in the frequency ranges indicated, of the steady state and unsteady thrust, torque, vertical, and horizontal transverse forces and vertical and horizontal bending moment. In addition, the thrust eccentricity can be calculated by vertical and horizontal components by:

$$e_{v\ (or)\ h} = \frac{M_{v\ (or)\ h} - F_{v\ (or)\ h}\ (x)}{t}$$

for any angular position of shaft rotation where the values of $e_{v \text{ or } h}$, $M_{v \text{ or } h}$, $F_{v \text{ or } h}$, and t are those for the propeller blade frequency harmonic for that particular angular position (i.e., the instantaneous value).

ACKNOWLEDGEMENTS

Many individuals, including those mentioned below, have contributed to this program of instrumentation development. The work was sponsored by the U. S. Navy, Bureau of Ships, under Project FAN, S-F013-0103, Task 0200. Overall direction of the project was provided jointly by the NSRDC Hydromechanics Ship Powering Division and the Project FAN Coordinator. The instrument system developmental efforts of Mr. Warren Kellerberg, Mr. Ronald Kiley, Mr. Florian Frillman, Mr. Fred Schloss, Mr. Marlin Miller, and J. F. Steimer are appreciated. The author is grateful for the contributions of the technical content and editorial format made by Mr. Richard A. Cumming, Naval Architect at NSRDC. Appreciation is expressed to Mrs. Shirley Childers for typing the manuscript.

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Figure 1 — Dynamometer on Test Stand

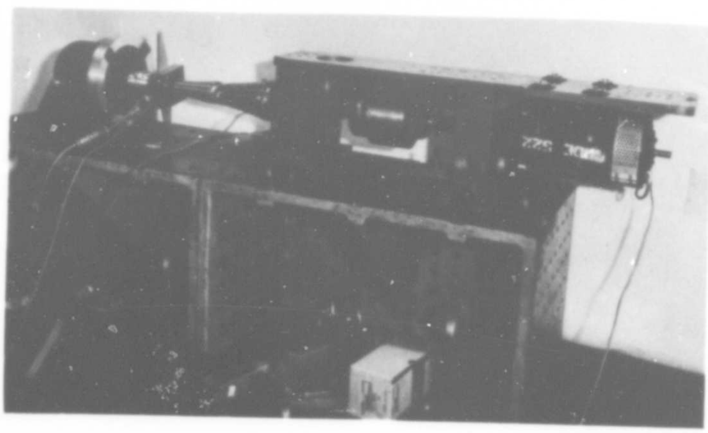
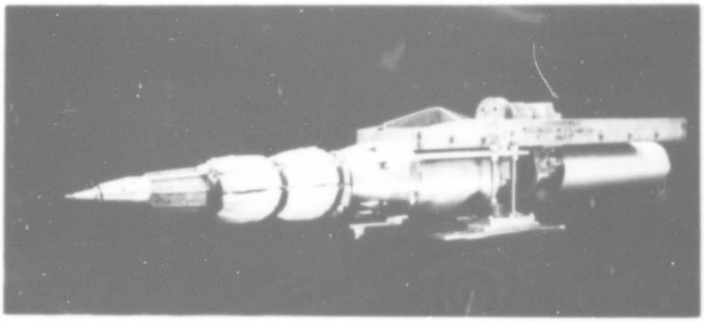
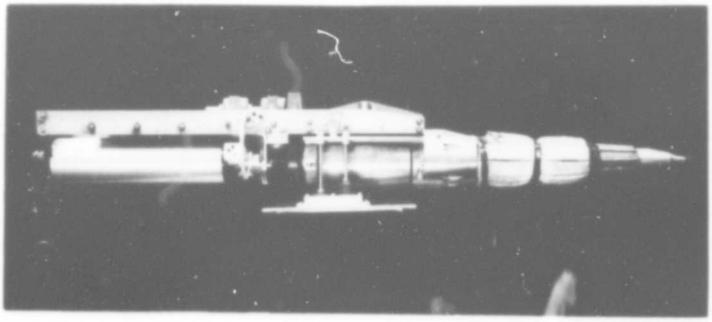


Figure 2 — Dynamometer - 60 CPS Six Component Measurement - Thrust, Torque, Transverse, and Bending Moments



Stern Quarter View



Profile View

Figure 3 – Surface Ship and Submarine Six-Component Dynamometer

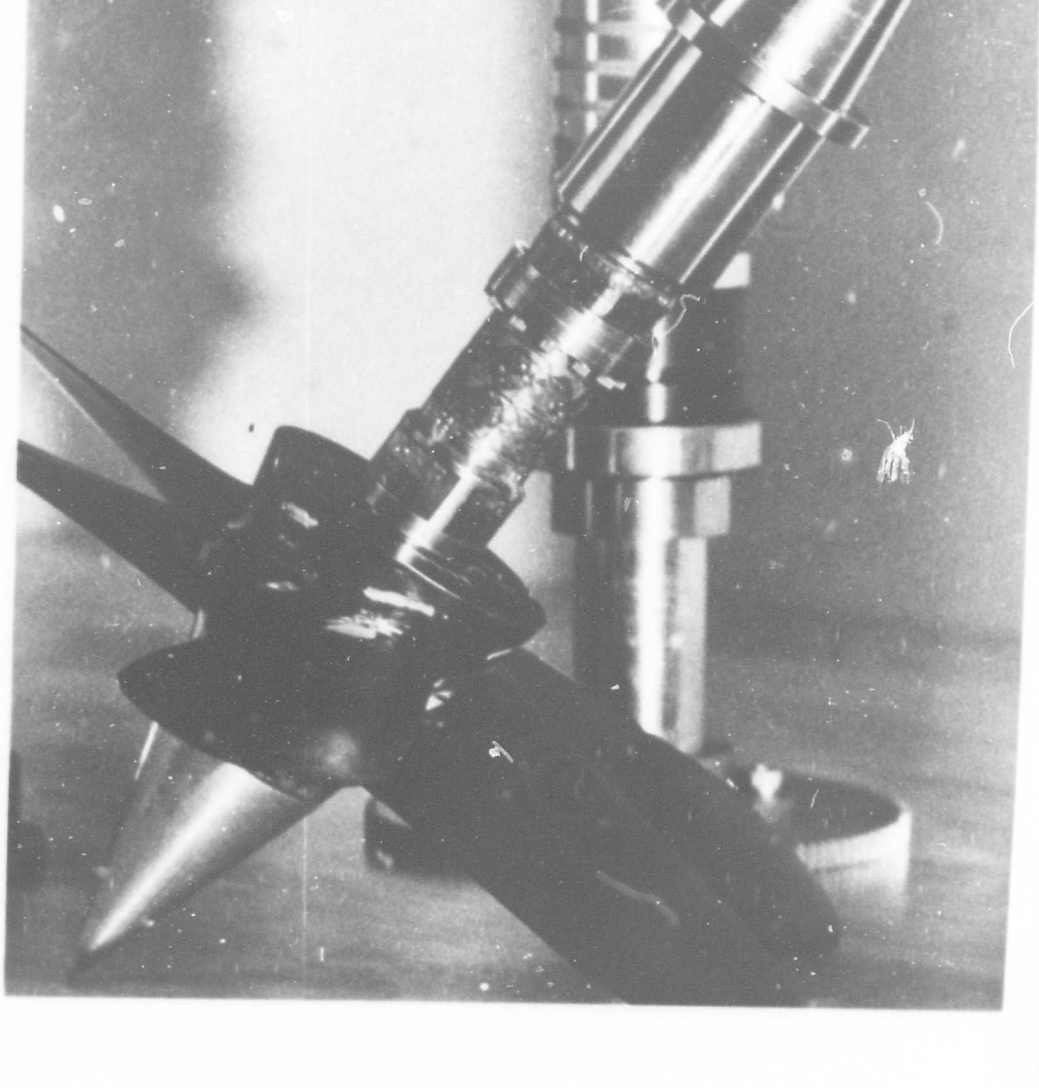


Figure 6 – Typical Six-Component Sting Balance

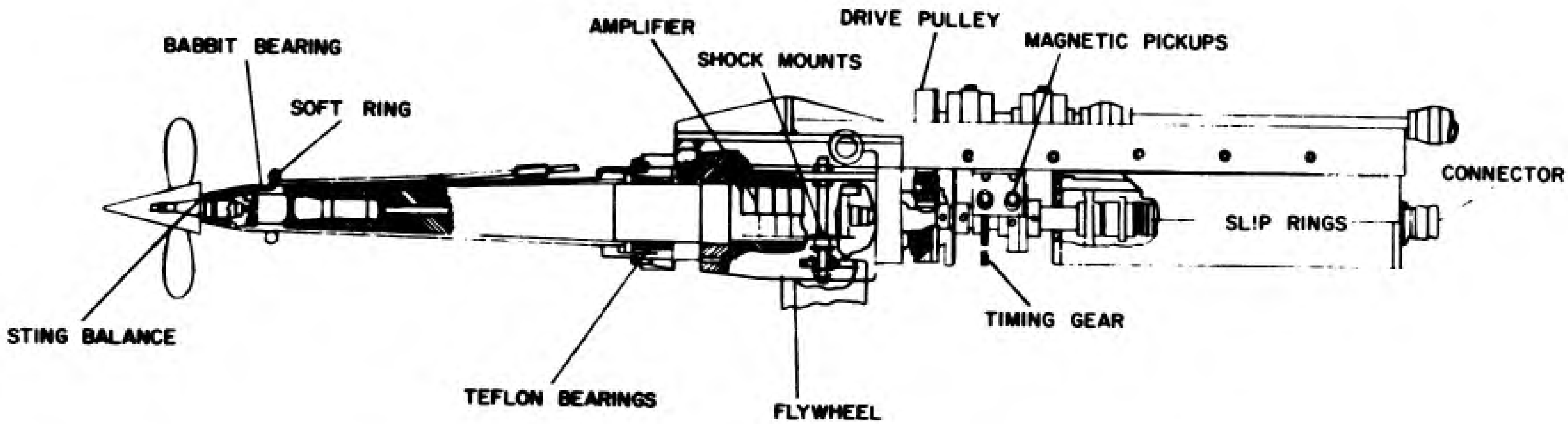
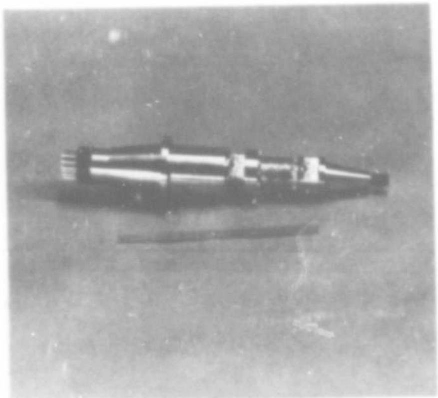
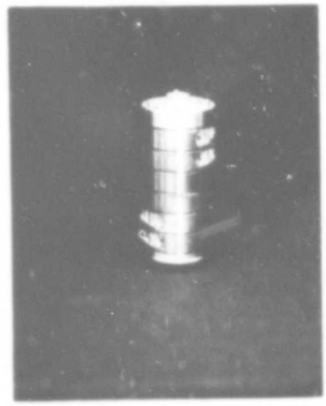


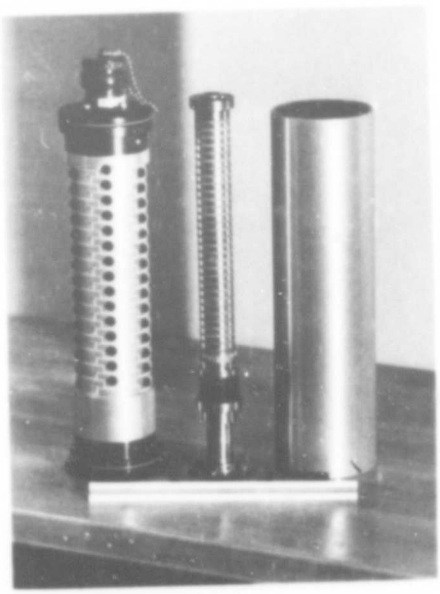
Figure 7 – Diagram of Six-Component (Basin) Dynamometer



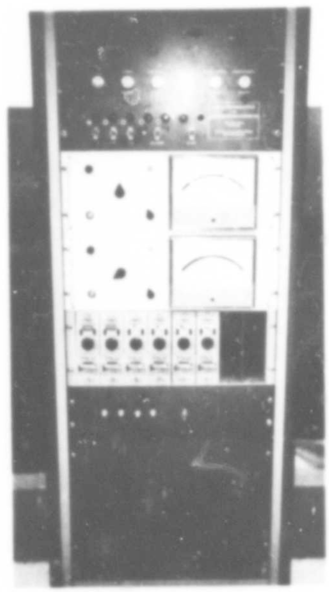
a. Sting Balance



b. Transistor Preamplifier



c. Slip Ring - Brush Unit (disassembled)



d. Control Console

Figure 8 – Six-Component Dynamometer Hardware

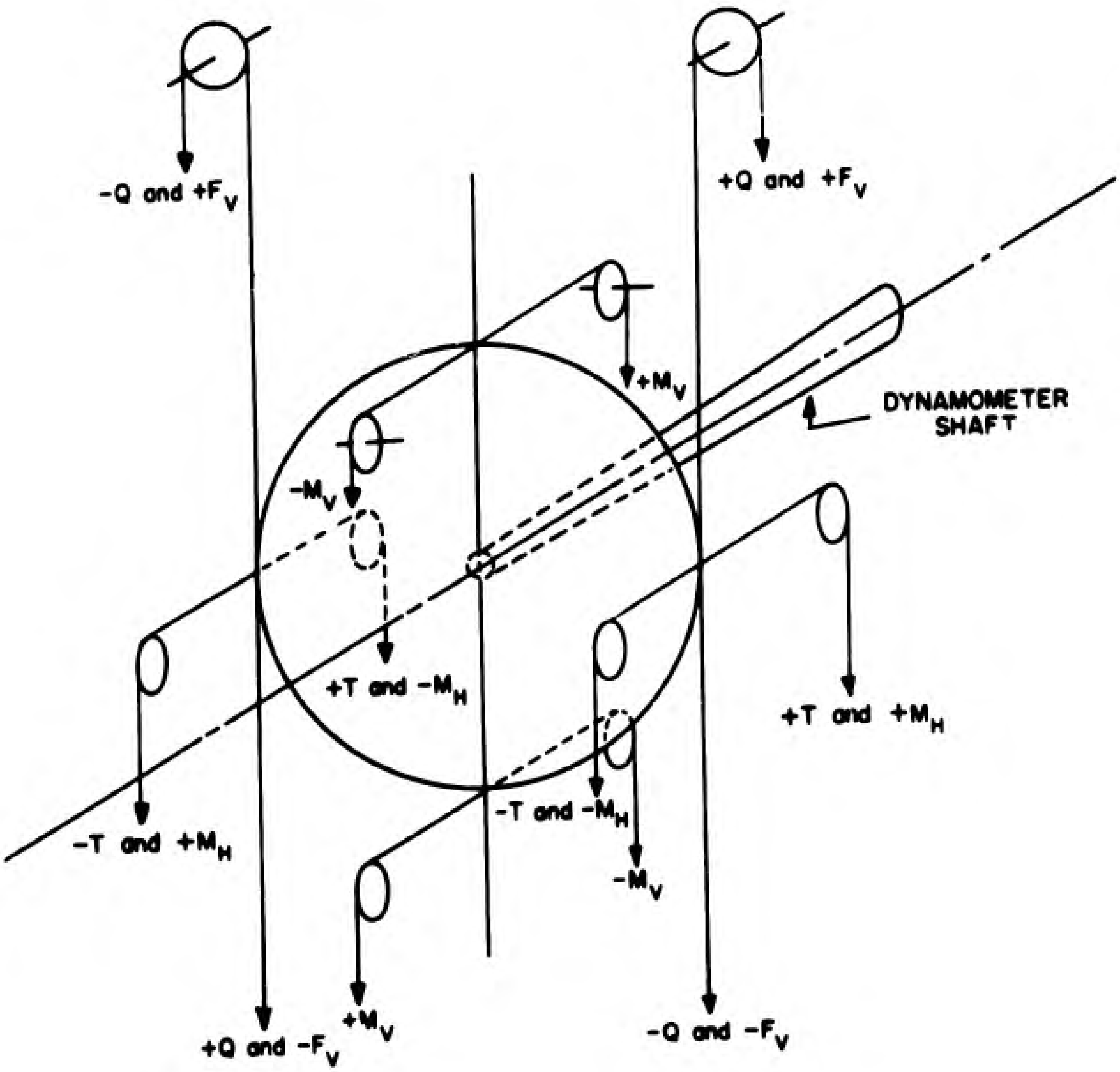
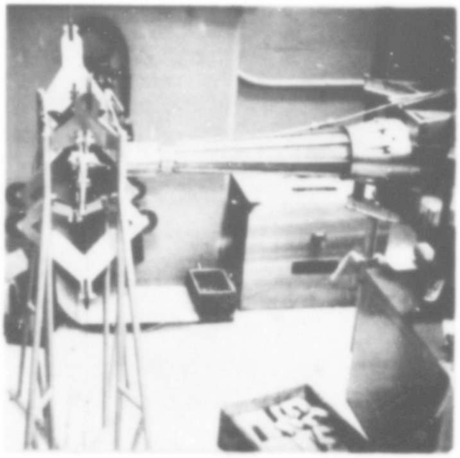
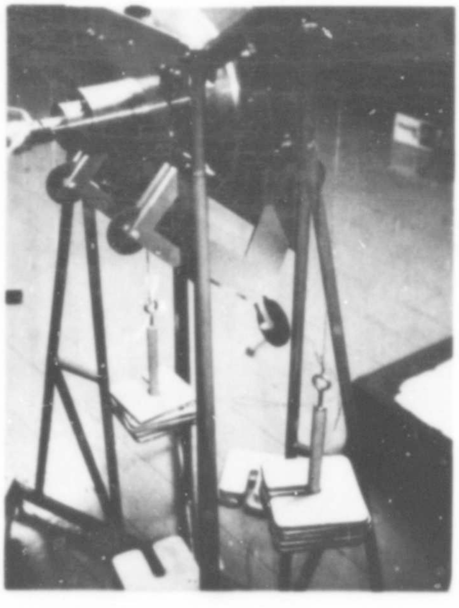


Figure 9 – Static Calibration Loading Diagram



Side Force



Front View

Figure 10 – Six-Component Static Dead-Weight Calibration Stand

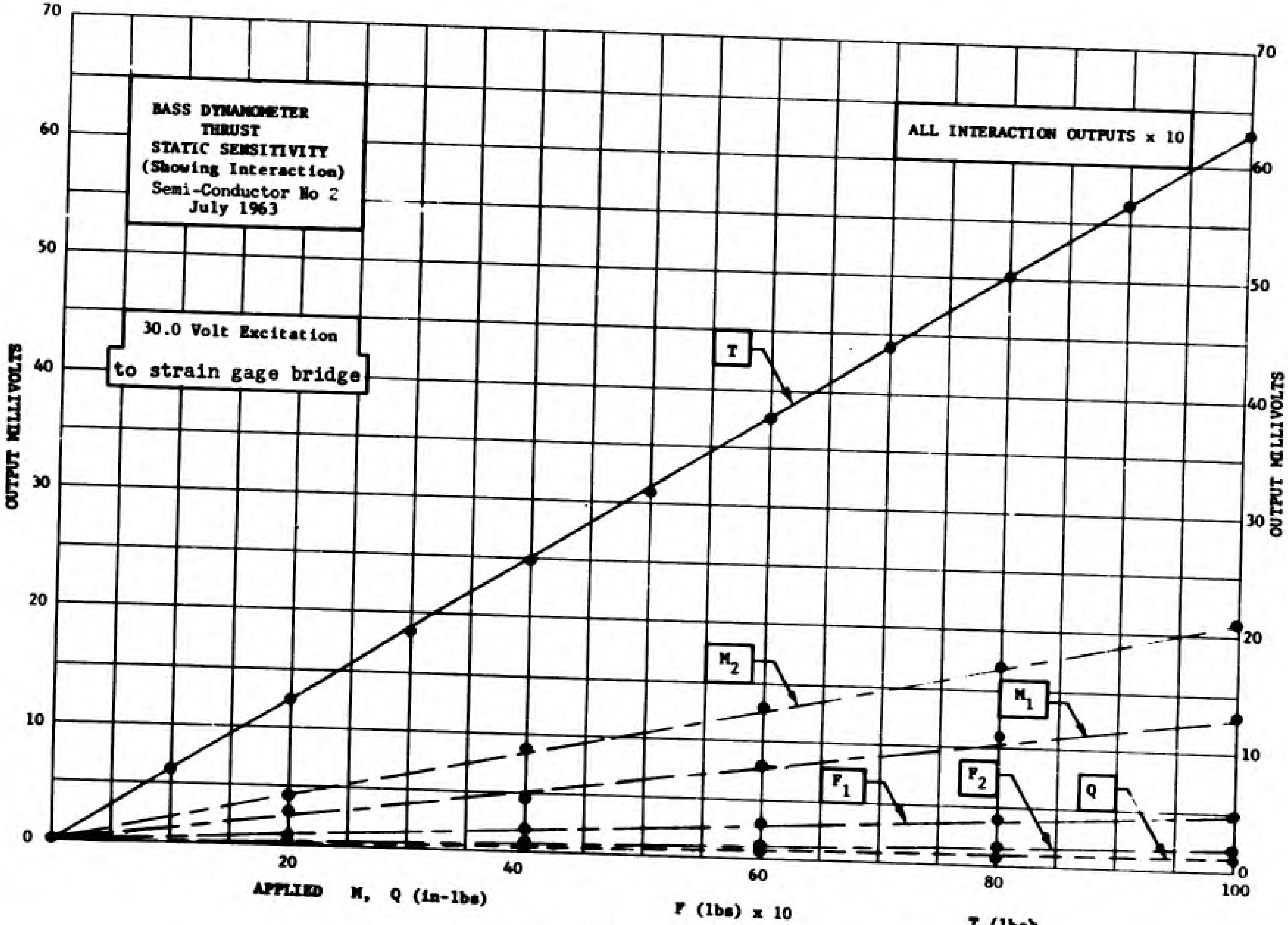


Figure 11 – Typical Static Thrust Sensitivity Plot

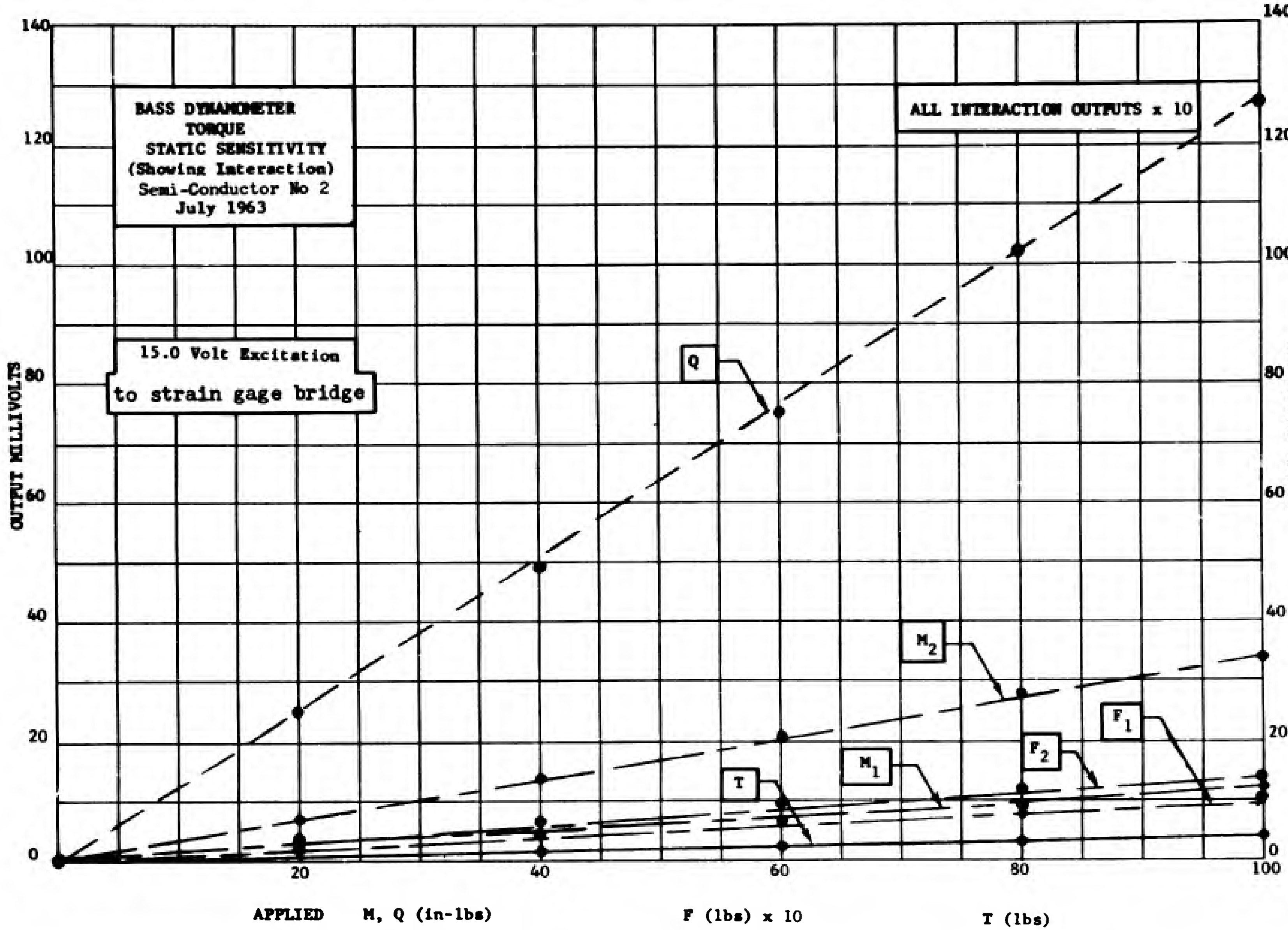


Figure 12 – Typical Static Torque Sensitivity Plot

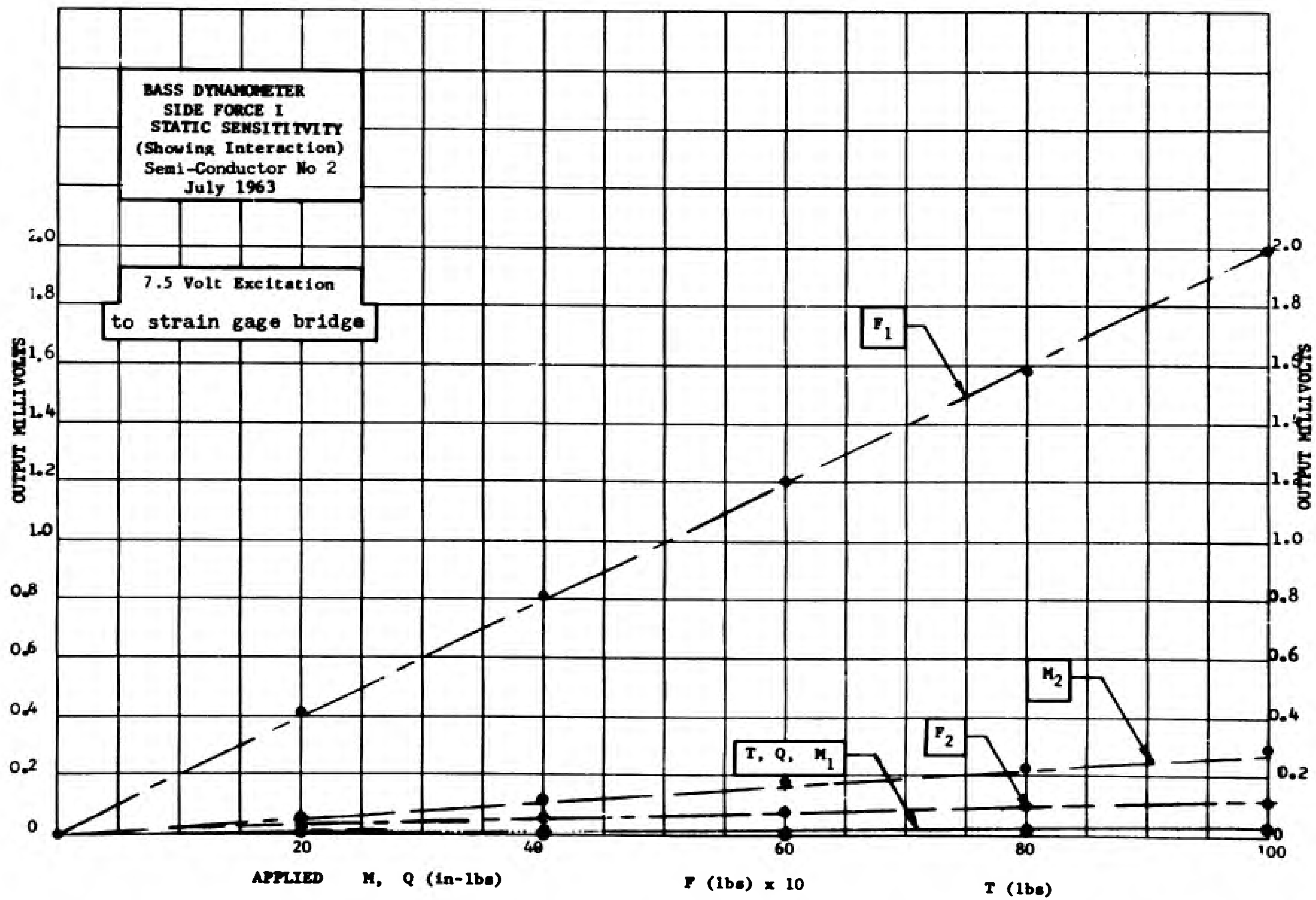


Figure 13 – Typical Static F_1 Sensitivity Plot

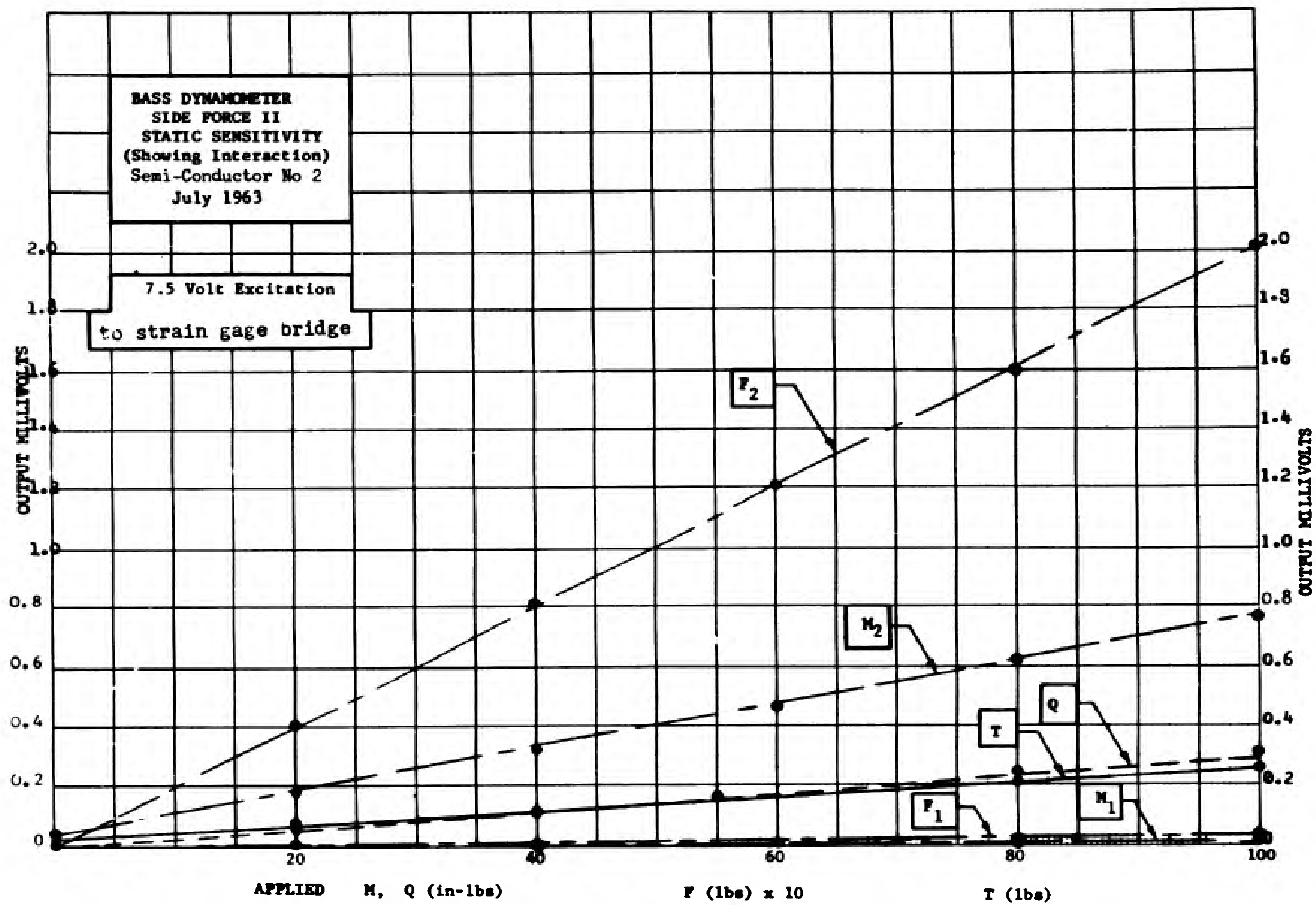


Figure 14 – Typical Static F_2 Sensitivity Plot

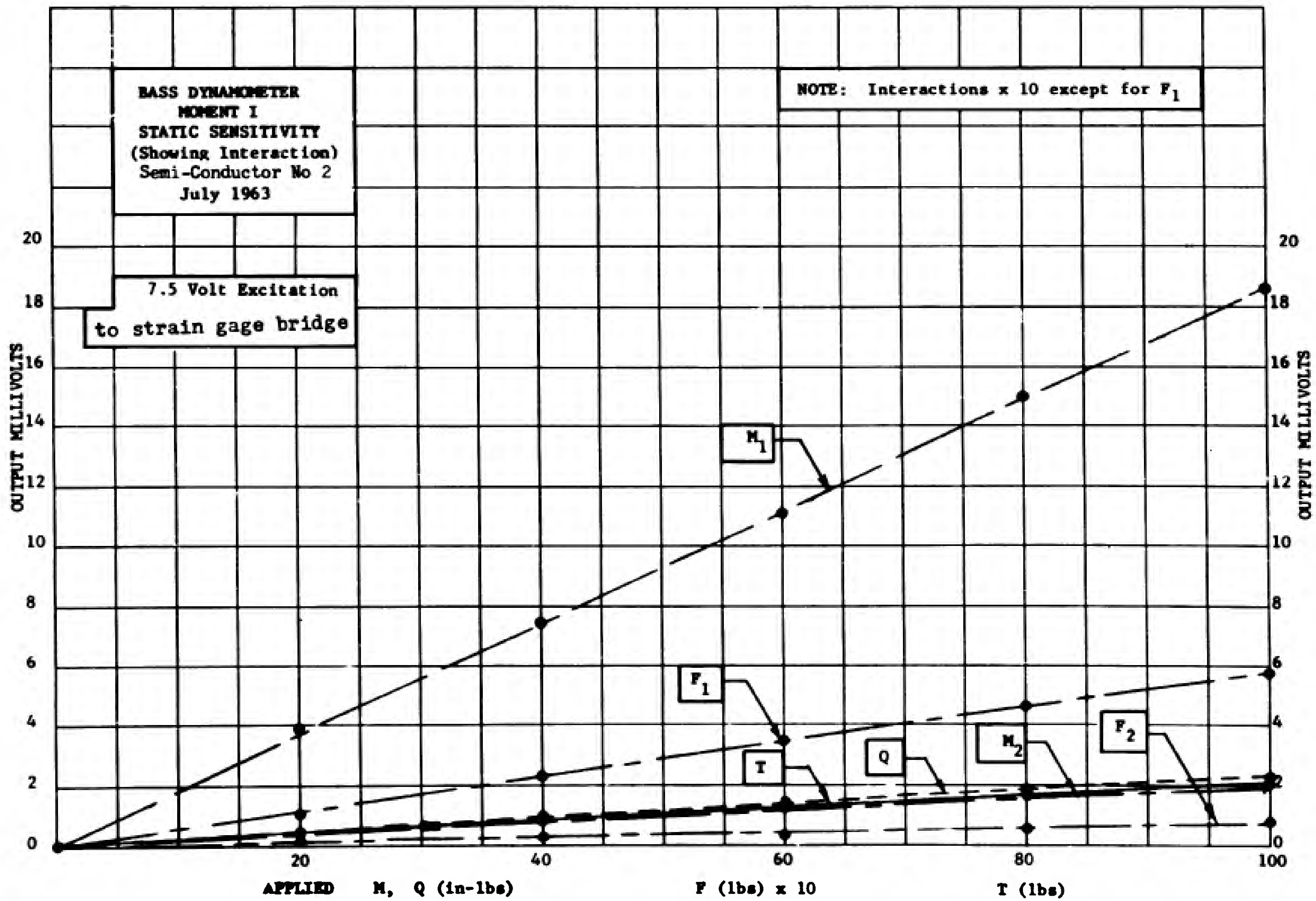


Figure 15 – Typical Static M_1 Sensitivity Plot

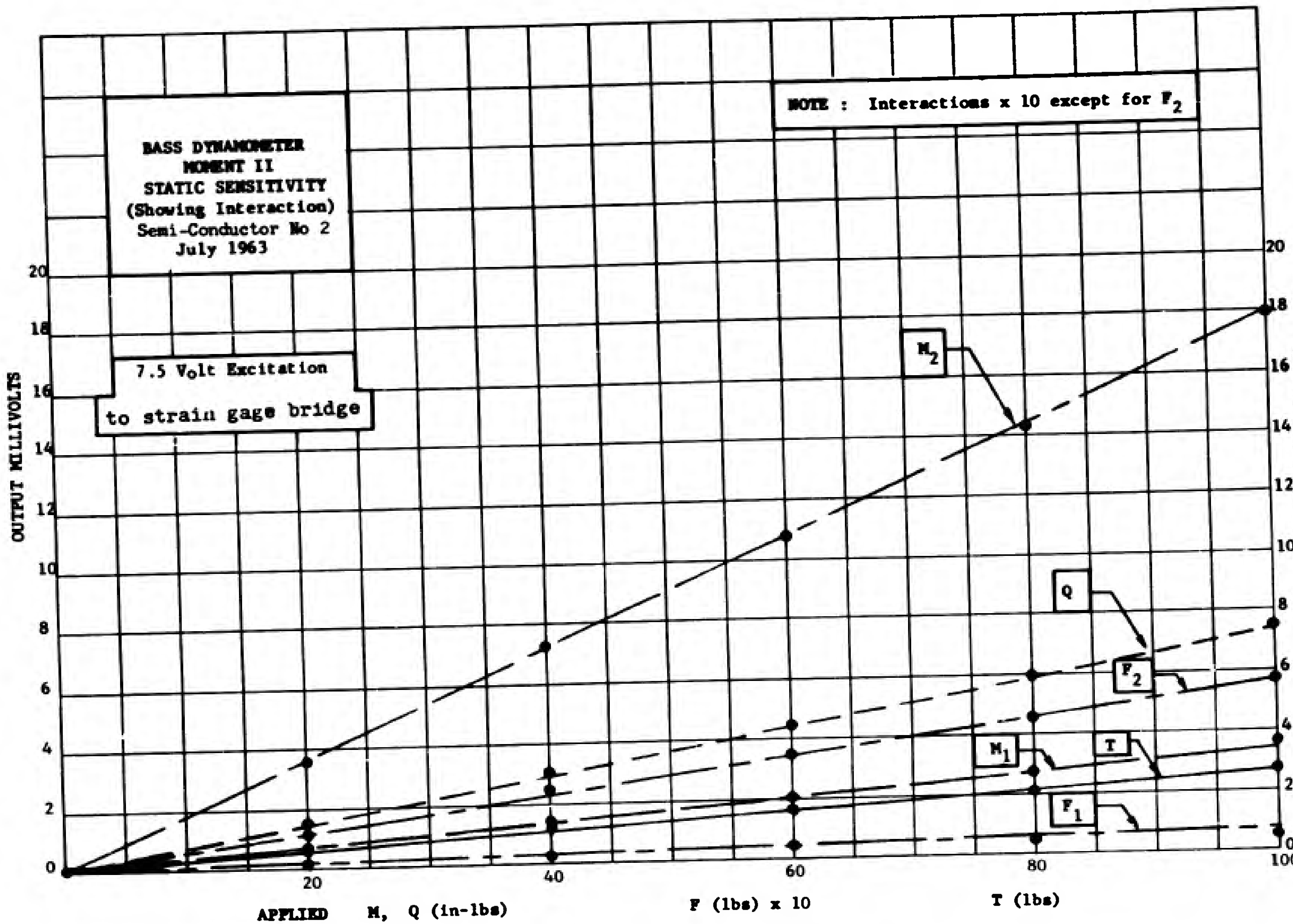
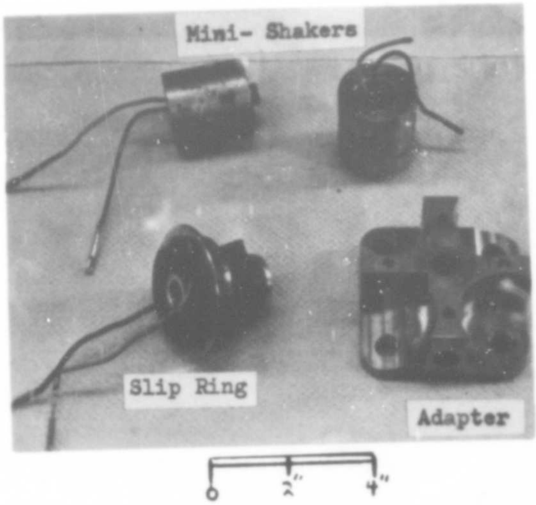


Figure 16 – Typical Static M_2 Sensitivity Plot



Shaker Parts

Figure 17 – Small Electromagnetic Shaker Components

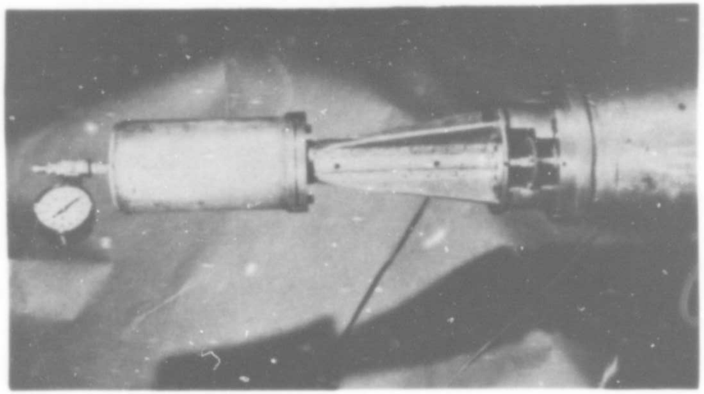


Figure 18 – Air Chamber

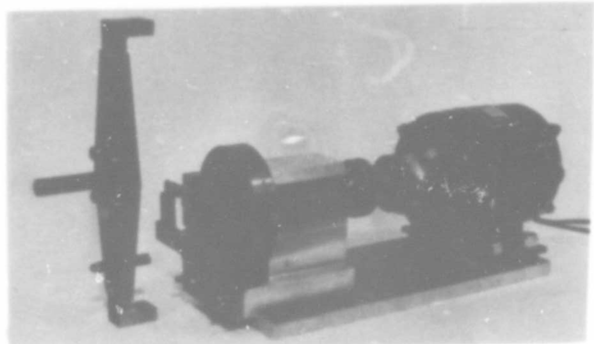


Figure 20 – Scotch Yoke Spring

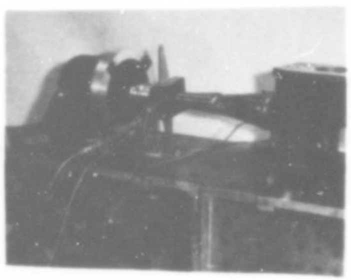


Figure 19 – Electromagnetic Shaker

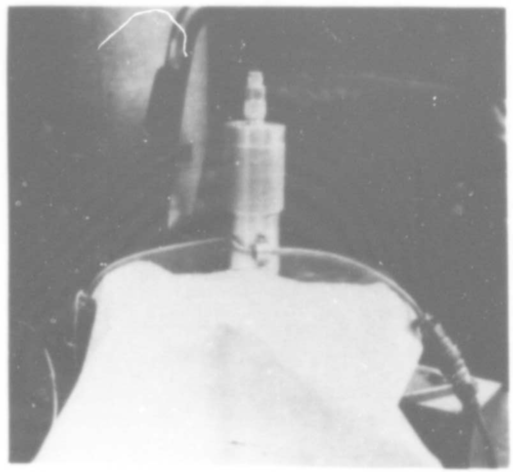


Figure 21 – Force Generator Impedance Head

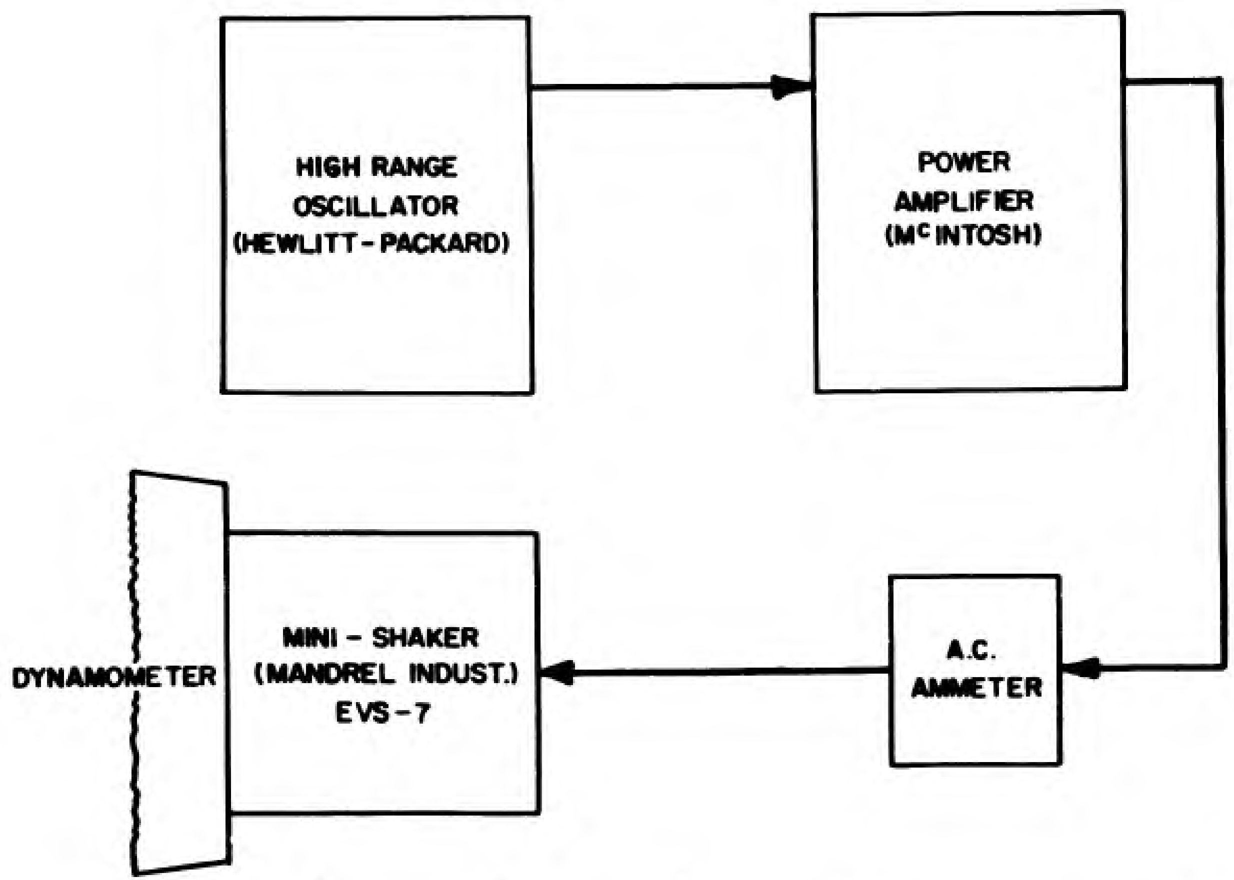
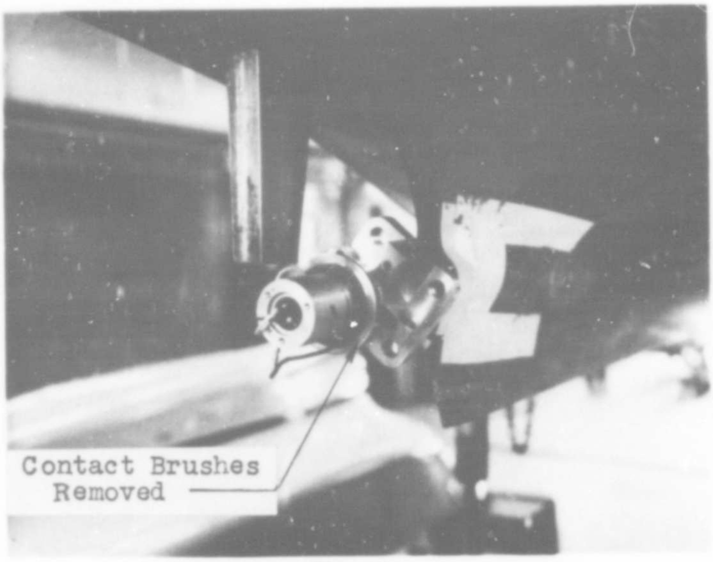
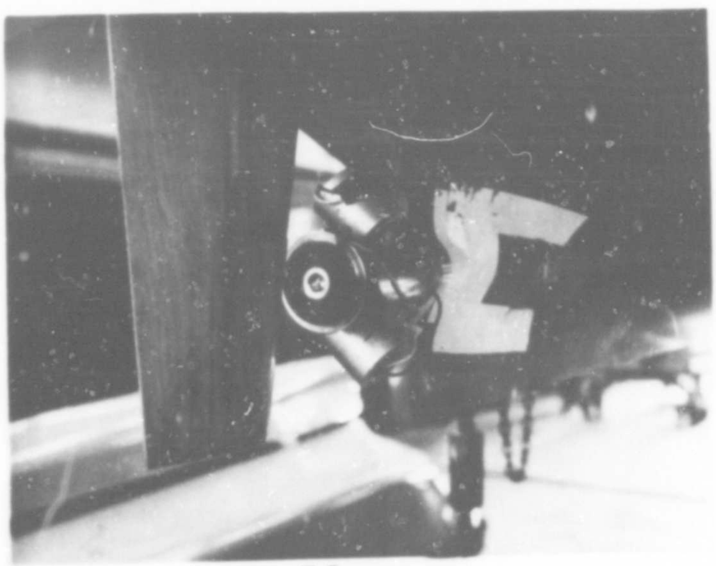


Figure 22 — Schematic Diagram of "Mini-shaker" Control System

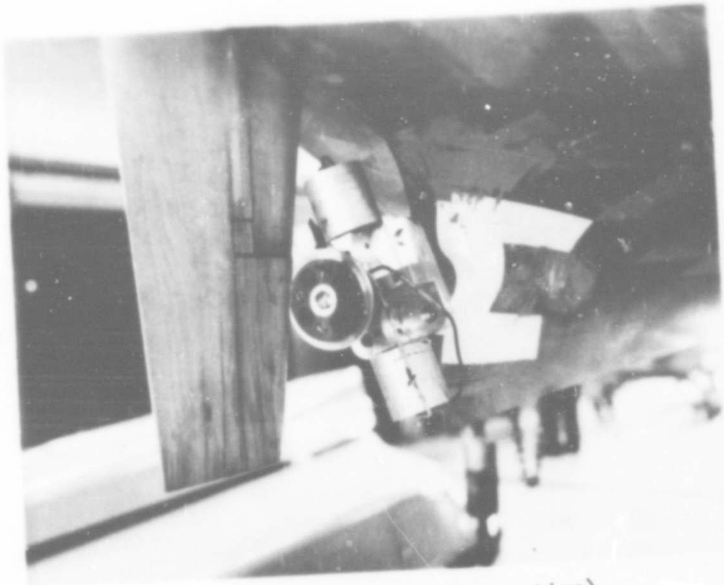


Thrust

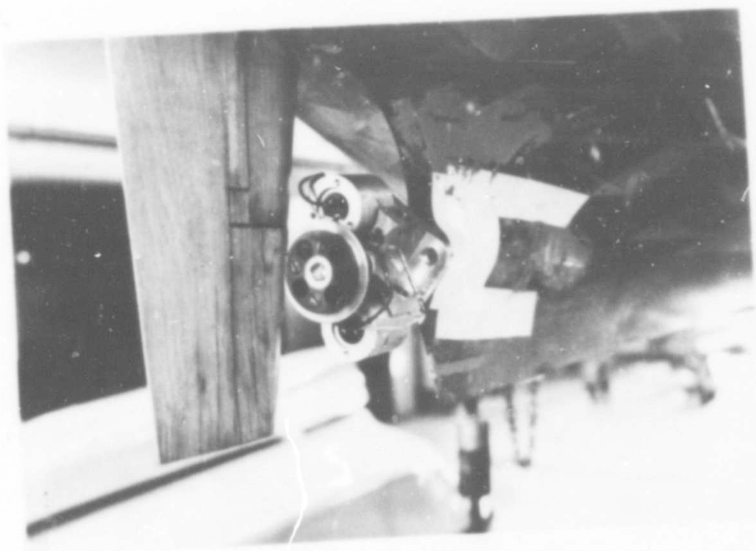


Torque

Figure 23 – Mini-shakers on Fixture for Applying Dynamic T and Q Loads

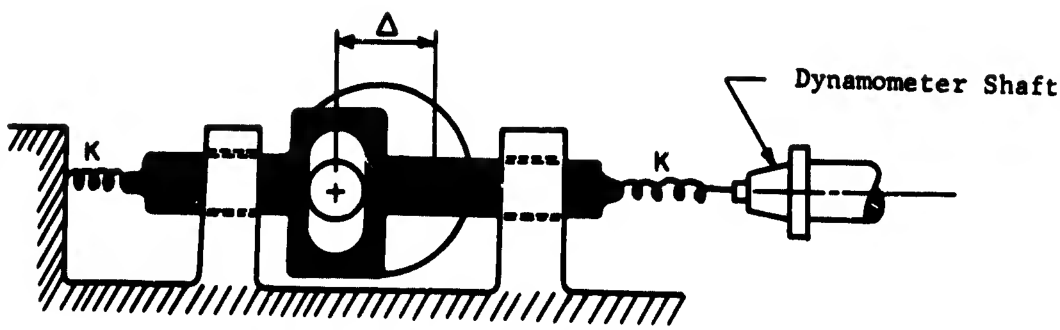


Side Force (one shaker active)



Bending Moment (one shaker active)

Figure 24 – Mini-shakers on Fixture for Applying Dynamic Side Force and Bending Moment



Yoke Mechanism Arrangement

Figure 25 — Scotch Yoke Shaker

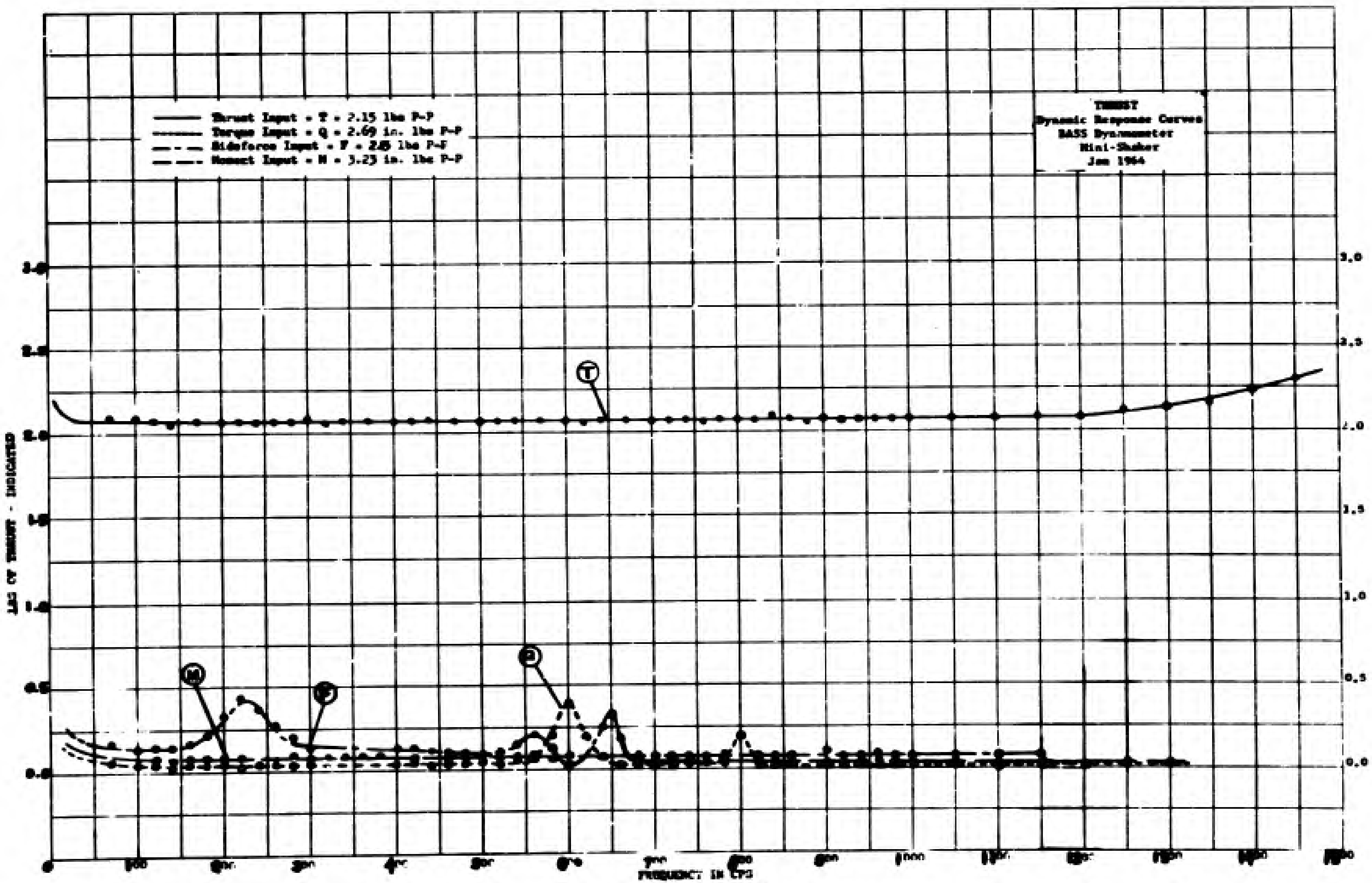


Figure 26 — Typical Dynamic Thrust Response Plot

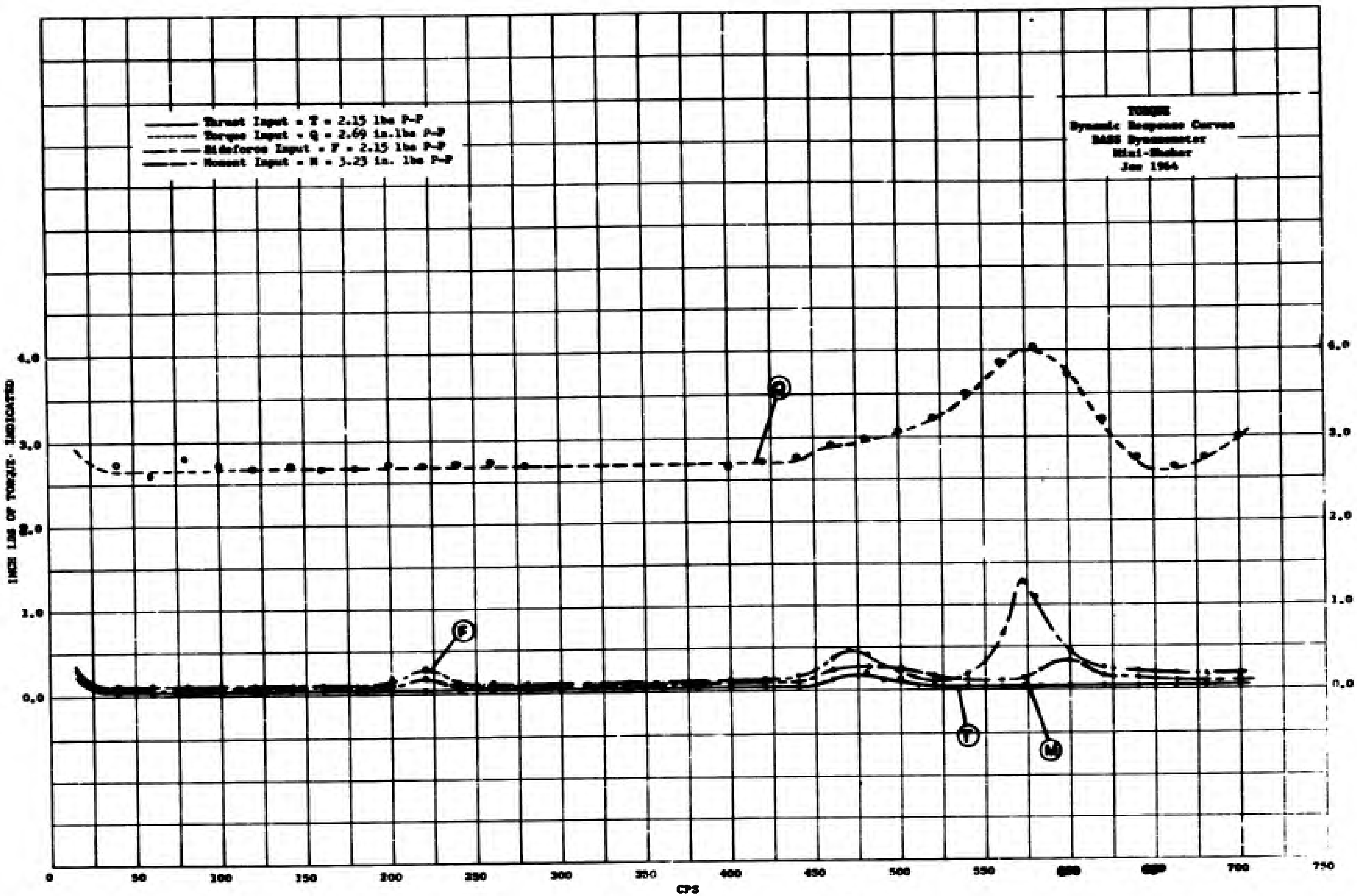


Figure 27 – Typical Dynamic Torque Response Plot

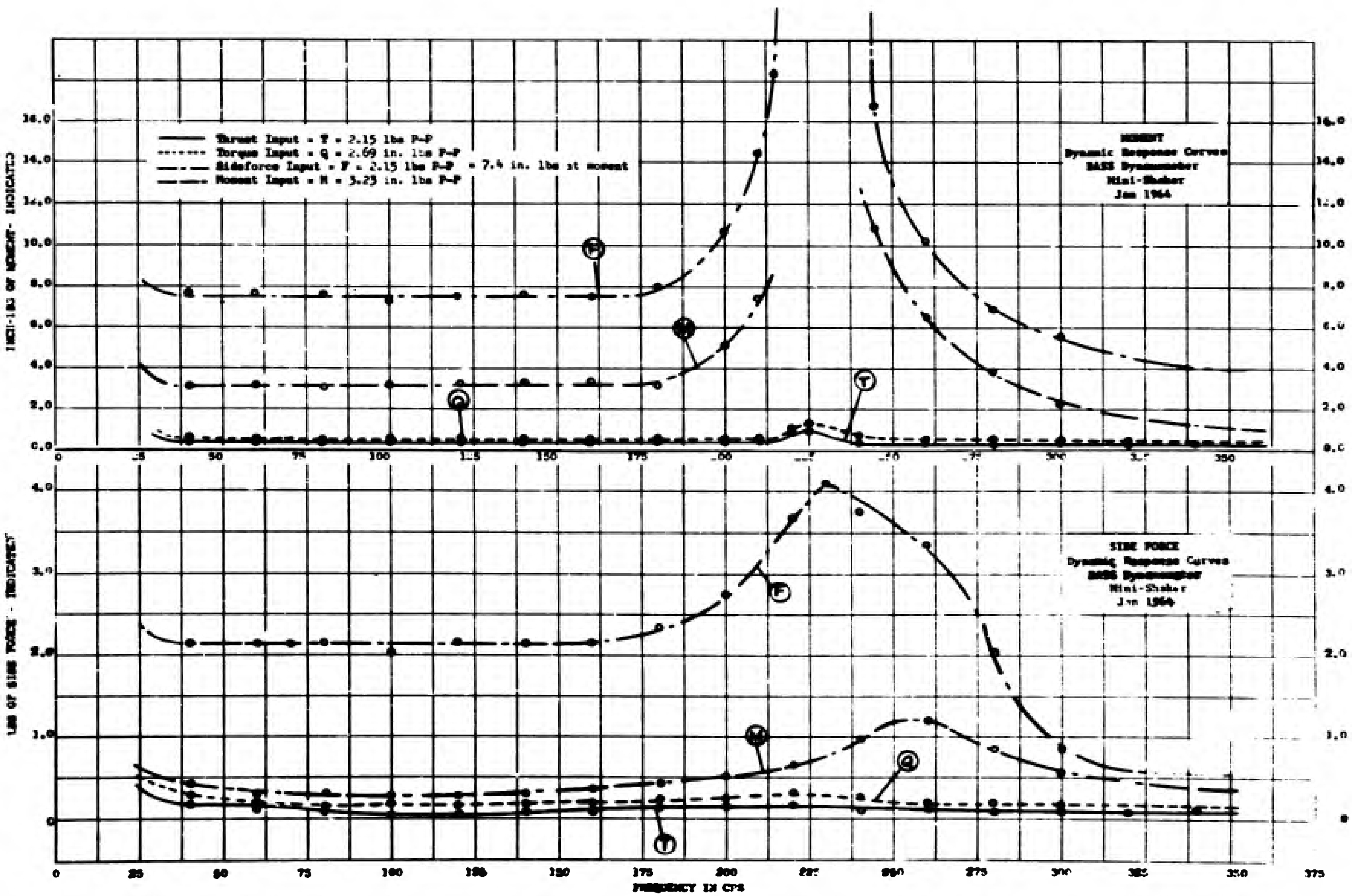


Figure 28 – Typical Dynamic Side Force and Bending Moment Response Plot

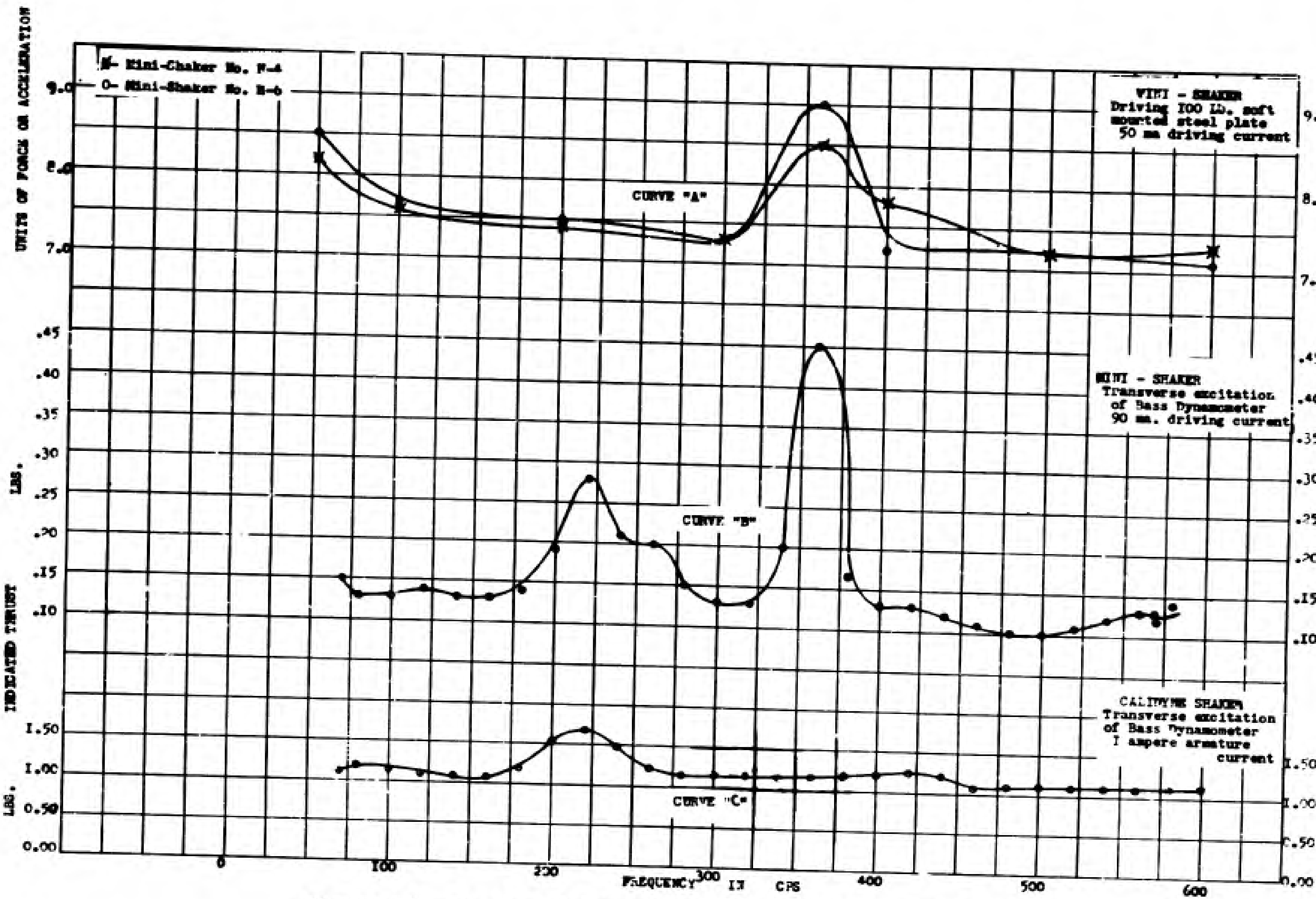


Figure 29 – Curves of Investigation of "Mini-shaker" Response

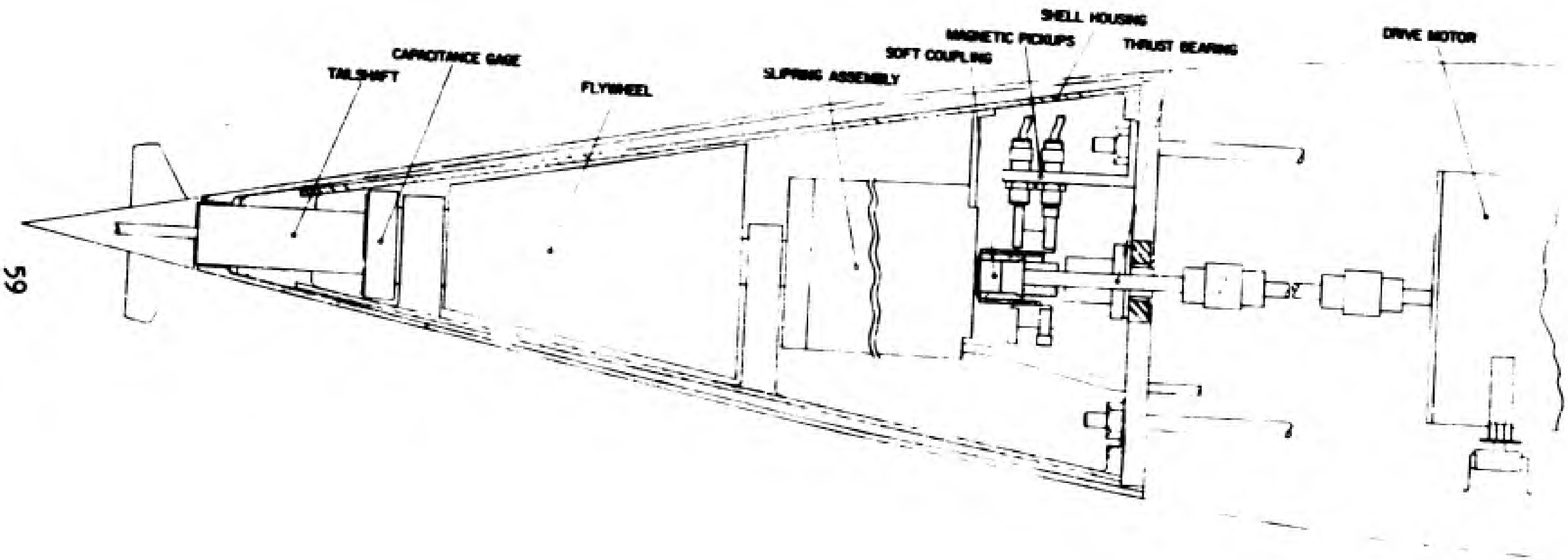
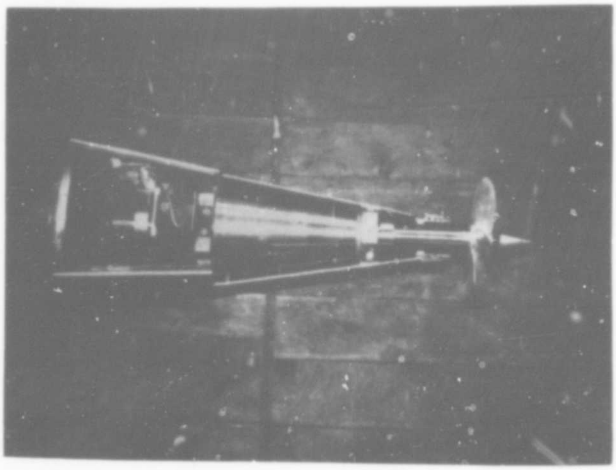


Figure 30 – Submarine Fluctuating Thrust Balance



Housing Cover Plates Removed

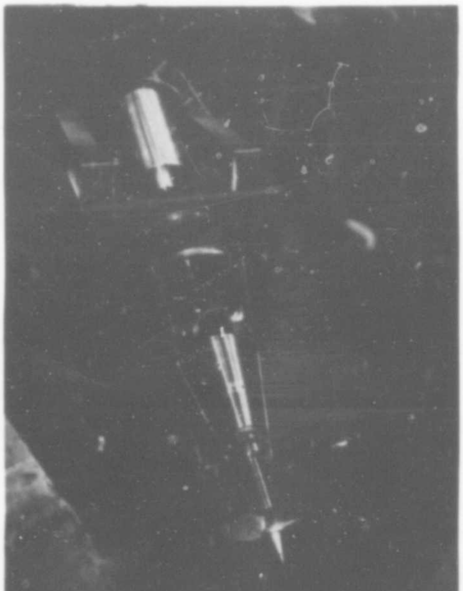


Figure 31 – Photographs of (Basin) Thrust Dynamometer

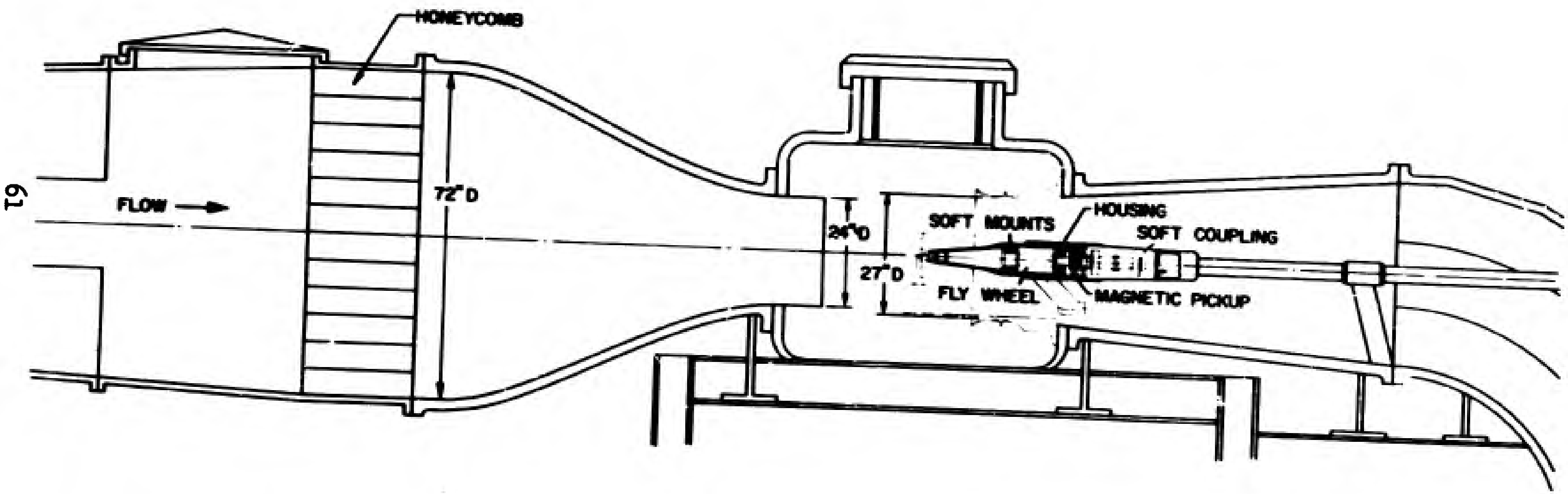
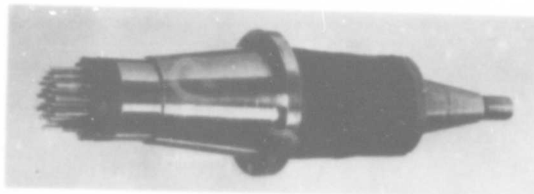
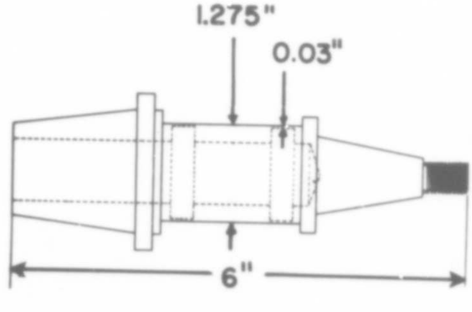


Figure 32 - Six Component Water Tunnel Dynamometer Arrangement



(a)



(b)

Figure 33 – Water Tunnel Six-Component Dynamometer Details

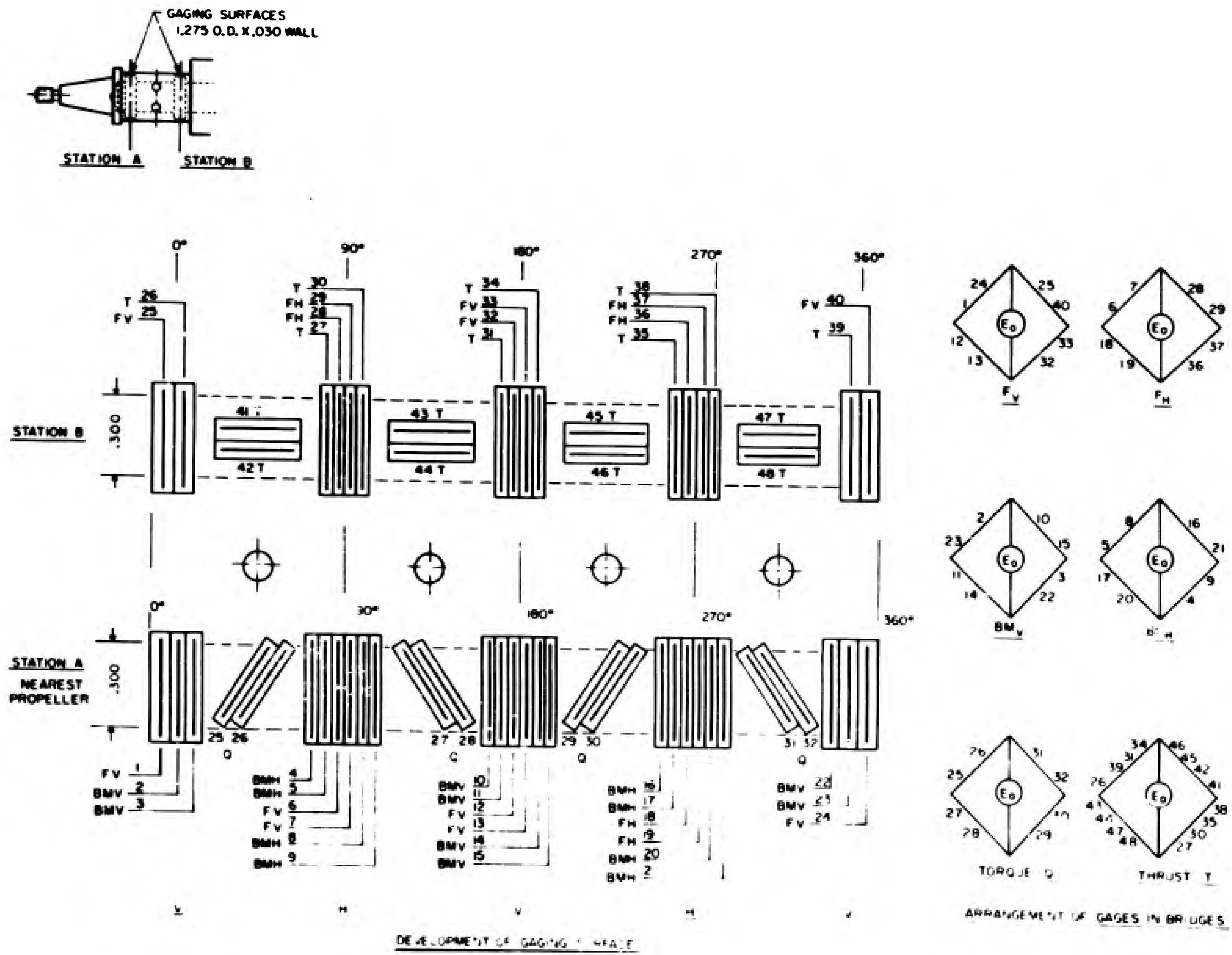
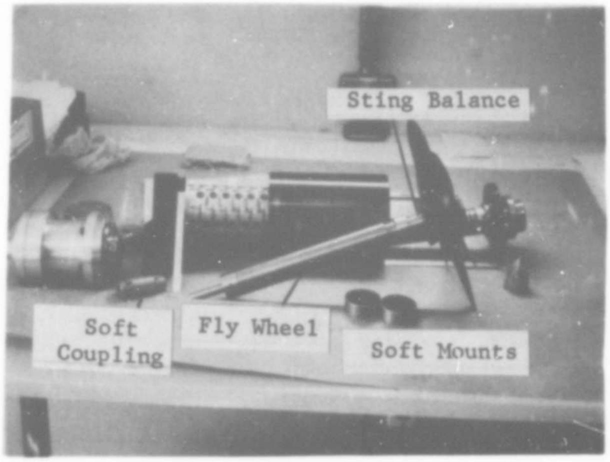
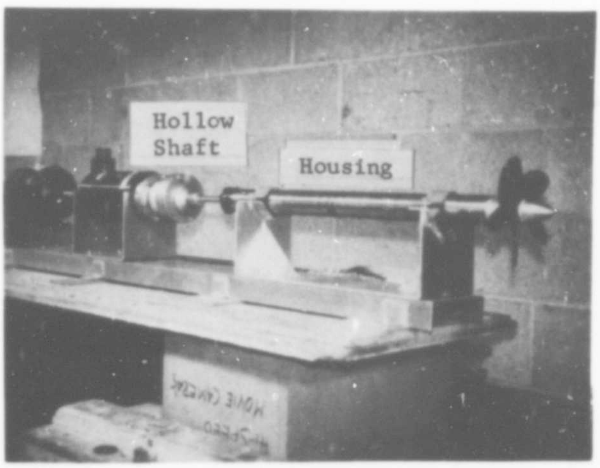


Figure 34 – Proposed Arrangement of Gages with Water Tunnel Six-Component Dynamometer



System Components



Dynamometer Mounted on Test Stand

Figure 36 - Basin Miniature Two-Component Dynamometer on Test Stand