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ACOUSTICS AND VIBRATION

RECENT ADVANCES IN MECHANICAL IMPEDANCE INSTRUMENTATION AND APPLICATIONS

by

Fred Schloss

ACOUSTICS AND VIBRATION LABORATORY
RESEARCH AND DEVELOPMENT REPORT

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RECENT ADVANCES IN MECHANICAL IMPEDANCE INSTRUMENTATION AND APPLICATIONS

by

Fred Schloss

Presented at 34th Symposium on Shock,
Vibration, and Associated Environments,
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ABSTRACT

This report discusses an automatic impedance measurement system of high precision, results of "round robin" evaluation of electronics of various mechanical impedance measurement systems, new developments in mechanical impedance instrumentation such as improved vibration generators and a miniature impedance head, and the application of new impedance measurement techniques. These techniques have been applied to the evaluation of dynamic properties of resilient mountings, couplings, damped structures, and damping and viscoelastic materials and to the determination of the internal damping of materials as a function of vibratory stress, added water mass of propellers, and elastic moduli of heterogeneous materials such as fiberglass.

ADMINISTRATIVE INFORMATION

The work discussed in this report was authorized by Bureau of Ships Code 345, under Project S-F013 11 08, Task 1352.

INTRODUCTION

Advances in instrumentation in the field of mechanical impedance and applications of new impedance techniques are presented. Specifically, this report discusses:

1. An automatic impedance measurement system possessing a high degree of precision.

2. Results of the "round robin" evaluation of the electronics of various mechanical impedance measurement systems.

3. Development of improved vibration generators for use in impedance measurements and a miniature impedance head.

4. Impedance techniques as applied to:

   a. Evaluation of the dynamic properties of resilient mountings, couplings, damped structures, damping and viscoelastic materials.

   b. Determination of the internal damping of materials as a function of vibratory stress, added water mass of propellers, and elastic moduli of heterogeneous materials such as fiberglass.

I. AUTOMATIC IMPEDANCE MEASUREMENT SYSTEM

An automatic mechanical impedance measurement system of high precision was developed under contract to comply with the condensed specifications shown in Table 1. The high phase accuracy, although not necessary for routine impedance measurement, is required to determine accurately the real component of the impedance, as will be discussed later in the applications of this technique. The precision may be obtained by careful design - paying particular attention to details without greatly increasing costs. The large dynamic range is necessary for routine measurements on structures where the ratio of maximum to minimum impedance may be as much as 150 db.

The measurement system which met the above specifications is shown in Figure 1. This system is identical, except for the change in the intermediate frequency from 11 kc to 20.5 kc, to that shown in Figure 23 of Reference 1,* where the principle of operation is discussed. The system was first conceived in 1958 and construction was begun under Navy contract in 1960. After the prototype had been built, considerable time was spent in making approximately 30 modifications to the system to improve its performance. In the interim other commercially available systems were developed and used, but - largely because of a lack of attention to detail or of inherent inaccuracies of the particular system - they do not meet the specifications given in Table 1, or even the specifications given by the manufacturers.

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* References are listed on page 29.

TABLE 1

Condensed Specification for Automatic Impedance Measuring Apparatus

Frequency Range:	10 - 15,000 cps
Filtering:	Filters tracking with audio oscillator, both channels, constant bandwidth of 5 cps or less. All filters to be temperature controlled.
Dynamic Range:	80 db without switching and greater than 20 db signal/noise within the range. 140 db with switching.
Minimum Input:	3 x 10^{-6} volt
Phase Accuracy:	± 0.25 degree 40 - 5,000 cps ± 1 degree 10 - 15,000 cps including high impedance preamplifiers and input attenuators.
Amplitude Accuracy: Noise Rejection:	± 0.5 db Amplitude and phase accuracy maintained for noise outside pass band which is 40 db above signal amplitude.
Noise and Hum:	Less than 10 x 10^{-6} volt at input with capacitive source of 1000 pf and less than 3 x 10^{-7} volt at power frequency and harmonics.

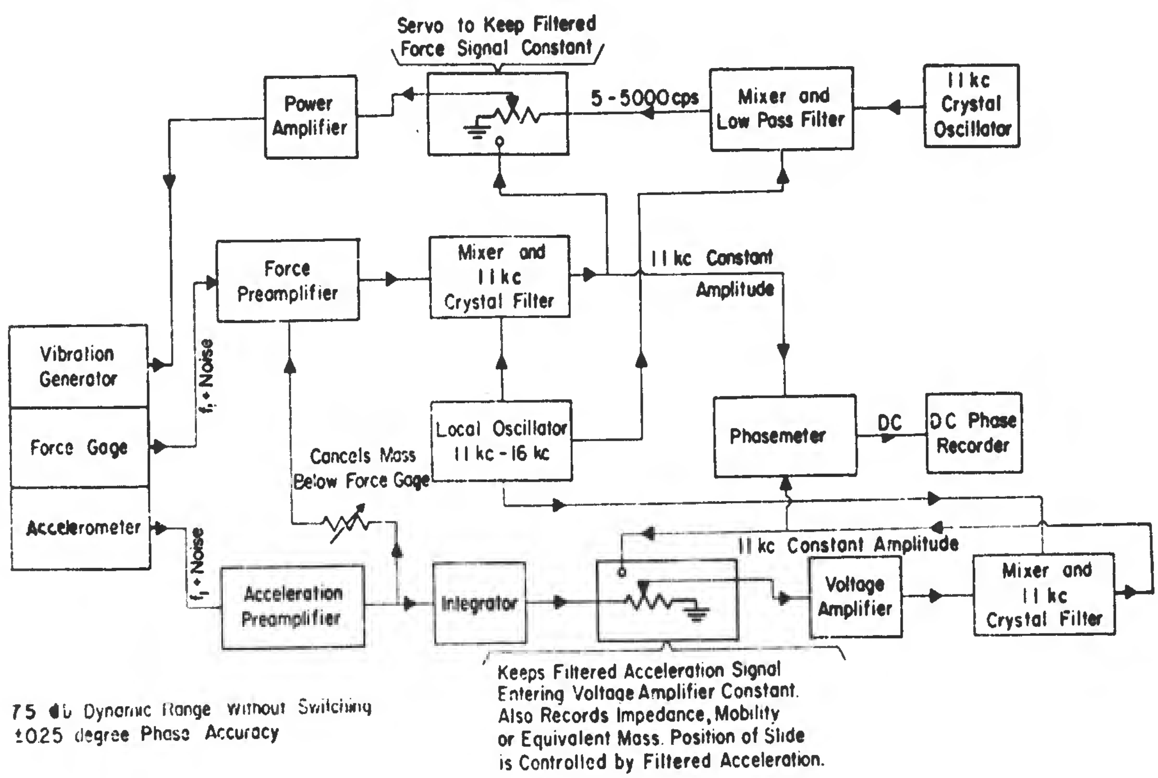


Figure 1 - Automatic Impedance, Mobility, or Equivalent Mass Plotter

This present system has been used for many purposes other than impedance measurements, such as cross-correlation measurements, acoustic intensity measurements, transducer response curves, and determination of structural modal shapes.

II. "ROUND ROBIN" EVALUATION OF MECHANICAL IMPEDANCE MEASUREMENT SYSTEMS

To evaluate the "state of the art" of impedance measurements, the U. S. Naval Research Laboratory in early 1963 enlisted various activities to participate in a "round robin" evaluation of three test structures. Results of these tests will be discussed in a subsequent paper. The NRL round robin is an evaluation of the transducer including the electronics; thereby the transducer errors were not separated from the errors introduced by the electronics. Also, this evaluation test, due to the lack of background noise usually encountered in the field, probably does not strain the system sufficiently. Therefore, a separate electrical evaluation was initiated by the David Taylor Model Basin. Unfortunately, many activities declined to participate, mainly because of their complete trust in the electronics. Others commented, after conducting the three tests, that this electrical evaluation helped them to realize the limitation of their equipment.

A sealed "black box" (as shown in Figure 2) was constructed and evaluated by six activities using eight different systems, including all those presently available commercially. In Test 1 pure sinusoidal signals were used, and in Tests 2 and 3 random and discrete frequency signals were added to one channel. Although Tests 2 and 3 may appear to be quite severe, such conditions are not unusual in practice. The dynamic range of tests on the black box was less than 55 db, which is less than that of "live" structures. Results of the evaluations are presented in Table 2. Data taken by activity F1 were used as reference. The manual system with digital outputs was precisely adjusted for phase and amplitude for each measurement; it has a short time accuracy of better than 0.2 db in amplitude and 0.1 deg in phase. This system is shown in Figure 3.

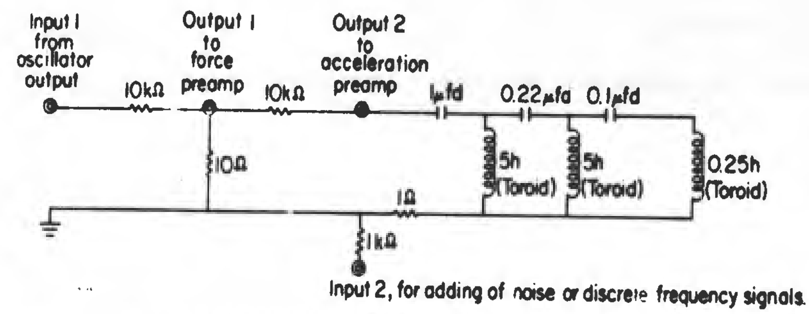
The results of these three tests led to the following conclusions:

1. Four systems, including one automatic system, had a relatively high degree of accuracy and the results had a spread of less than 5 deg and 1 db.

2. Surprisingly, the very expensive systems were inaccurate even in tests using pure signals of as much as 27 deg and 4.4 db.

3. Maximum errors were 30 deg and 23.5 db.

It is felt that the inherent accuracy of Systems A, B, and C is greater than the results show and that, with minor modifications, the precision could be greatly improved. Other sources of phase errors, such as those introduced by changes of attenuation at the preamplifiers, were not evaluated by these tests.



Test 1. 1 volt input 1 from 10—5,000cps plot amplitude ratio and phase between the two outputs

Test 2 Repeat test 1 with 1 volt of random noise into input 2.

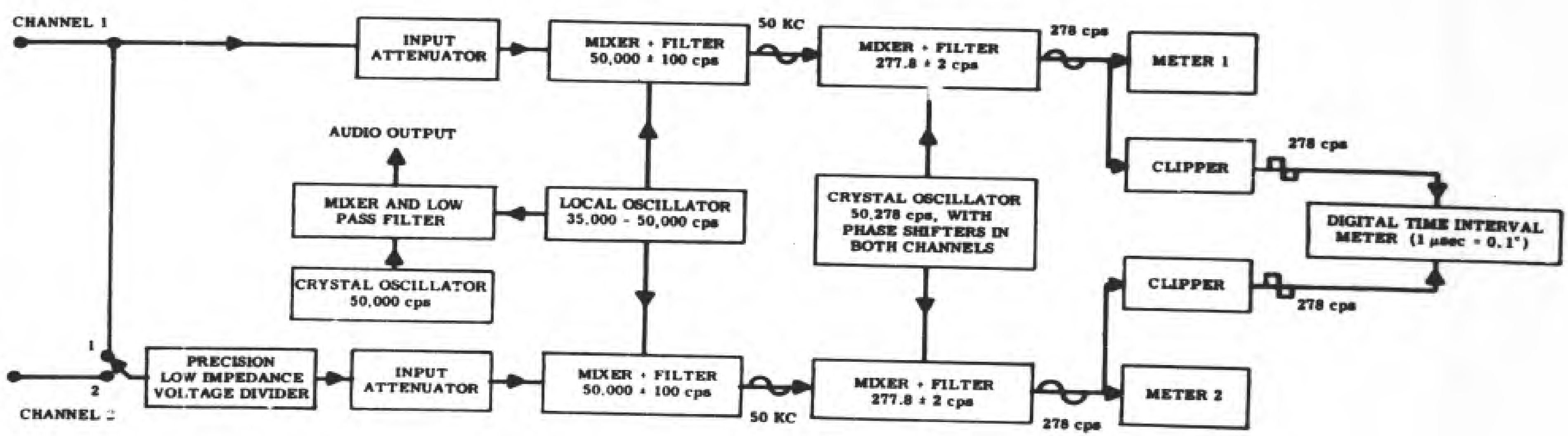
Test 3 Repeat test 1 with 5 volts of a fixed frequency signal of 80cps into input 2.

Figure 2 - "Black Box" for Electrical Evaluation of Mechanical Impedance Instrumentation

TABLE 2

Results of the "Round Robin" Evaluation of the Black Box

Activity	Type of System	Relative Cost	Max. Error, Pure Signal		Max. Error, with Noise	
			Phase	Amplitude	Phase	Amplitude
A	Heterodyne, automatic	Very high	- 25°	- 4.4 db	- 19°	-23.5 db
B	Heterodyne, automatic	Very high	+27°	+3.6 db	+30°	+3.9 db
C	Homodyne, automatic	Very high	- 10°	+ 1.1 db	+20°	+ 1.0 db
D	Audio, automatic	Medium	+ 3.8°	- 0.4 db	+ 6.2°	- 0.6 db
E	Nulling, manual	Low	- 1.9°	< 0.2 db	- 2.0°	< 0.3 db
F1	Heterodyne, automatic	High	+ 0.4°	+0.5 db	+ 0.5°	+0.5 db
F2	Heterodyne, manual	Medium	< 0.1°	< 0.2 db	< 0.1°	< 0.2 db
F3	Nulling, manual	Low	+ 2.8°	< 0.2 db	+ 2.7°	< 0.2 db



STEP 1: Insert larger of the two signals into Channel 1.

STEP 2: With switch in Position 2 and precision attenuator set at 0 attenuation, adjust input attenuators so that both meters read on scale and note reading of Meter 2.

STEP 3: With switch in Position 1 adjust precision attenuator only to obtain same reading as in Step 2 on Meter 2. Attenuator setting is indication of ratio of input signals. Adjust phase shifter for zero digital output.

STEP 4: Use same settings as in Step 2 and read phase from digital output.

Figure 3 - Manual Precision Phase and Ratio Measuring System

III. NEW DEVELOPMENTS IN MECHANICAL IMPEDANCE INSTRUMENTATION

A. VIBRATION GENERATORS

It is known that whenever the rotational impedance is low compared to the lineal impedance, results are somewhat inconsistent when measurements are made with various types of impedance heads and vibration generators. The inconsistencies were wrongly attributed to a high degree of rotational or transverse sensitivity of the transducer; but they are caused solely by changes of the rotational inertia of the transducer and driver system. This phenomenon was first observed when driving point impedance measurements were made on the end of a cantilever, and it was discussed in Reference 1 under subtitle "Change in Boundary Conditions."

It should be pointed out here that it is often a moot question as to which measurements are of value. For example, if a machine is to be mounted solidly to a structure, driving point impedance measurements made on the feet of the unmounted machine without the same rotational restraint as that of the structure are of little value. Similarly, if the machine is to be mounted on vibration isolators, the effects of the mounting plate must be taken into account.

To improve the measuring system, it became necessary to reduce its rotational inertia by decoupling the transducer and driver. One solution under development by the U.S. Navy Marine Engineering Laboratories involves the use of magnetic drivers and very light transducers. This solution, although best for very light structures, has the disadvantage of requiring critical alignment for the relatively low output magnetic drivers. Rather than using this approach, new electromagnetic drivers (needed also for rotational impedance measurements) were developed. Most of the mass of these drivers (shown in Figures 4, 5, and 6) is dynamically decoupled in all modes from the test structure above about 40 cps by having the relatively heavy annular magnet assembly supported by two rubber diaphragms. In addition, the annular arrangement and the rubber diaphragms provide a very low center of gravity of the assembly and also assure freedom from spurious effects such as rocking and high-frequency resonances in the suspension system. The added dynamic weight of the smaller vibration generator with a force output of 0.7 lb is less than 0.1 lb. The force output of the larger of the drivers is 5 lb.

B. NEW MINIATURE IMPEDANCE HEAD

A miniature impedance head and driver was developed for measurements on very light structures, such as a 1/30-scale model of a submarine propeller; see Figure 7. With this head, diameter 1/2 in. and length 1/2 in., an impedance equivalent to a weight of 0.0001 lb may be measured from 20 to 10,000 cps. The stiffness is 500,000 lb/in., which limits the maximum impedance which may be determined with this transducer. The stiffness of light structures is usually

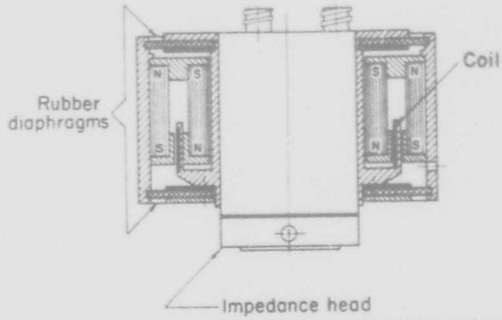


Figure 4 - "Wrap Around" Vibration Generator

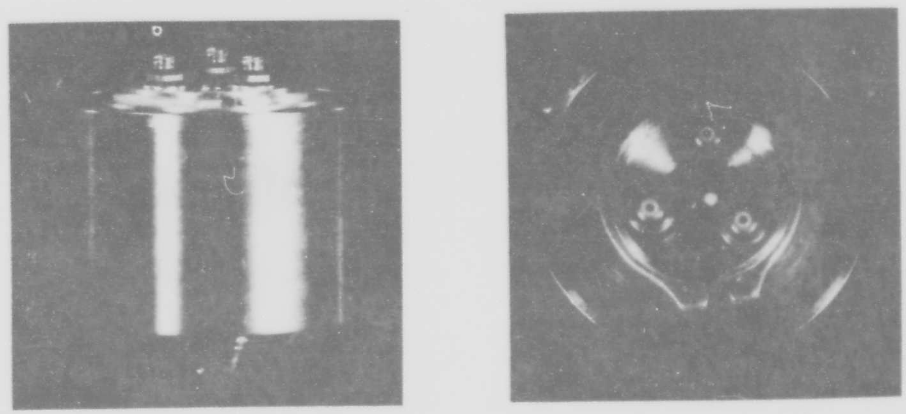


Figure 5 - Small "Wrap Around" Vibration Generator with Small Head

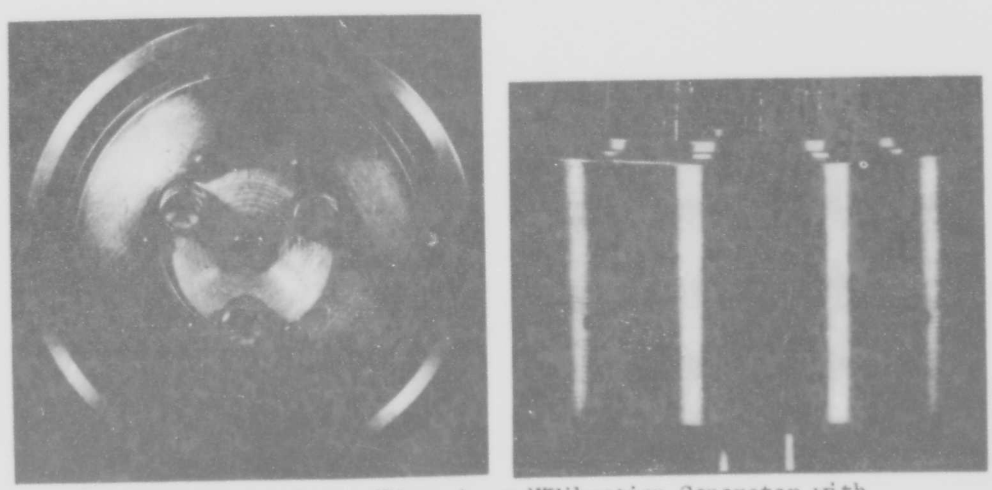


Figure 6 - Large "Wrap Around"Vibration Generator with Large Head

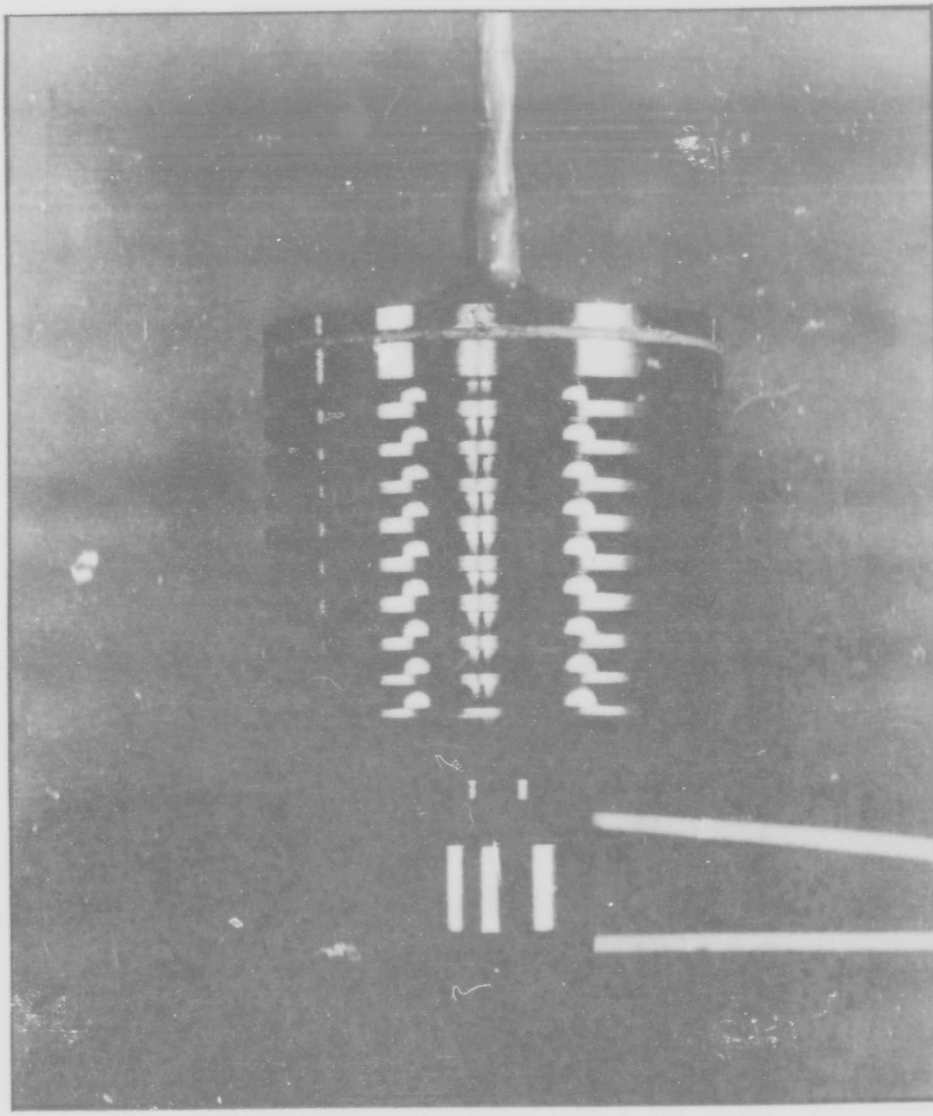


Figure 7 - Miniature Impedance Head and Driver

well below this value, however. The effective stiffness of the head may be increased to over 1,500,000 lb/in. by reversing the specimen and driver attachment points; however the weight below the force gage is increased by a factor of six to 0.01 lb, thus increasing the minimum measurable impedance.

The miniature vibration generator weighing 0.2 lb has a force output of 0.15 lb.

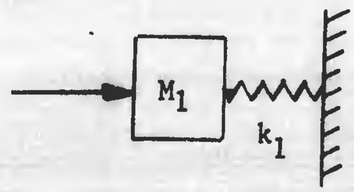
C. COMMENTS REGARDING EFFECTS OF ATTACHMENTS TO THE IMPEDANCE HEAD ON IMPEDANCE MEASUREMENTS

It is hoped that the following discussion will clarify a common misconception in mechanical impedance measurements. The following question is often asked:

> Does the relatively heavy mass of the vibration generators which are directly attached to the impedance heads change the resonances and, therefore, should only relatively light shakers and impedance heads be used in order not to change the characteristics of the structure?

The answer to the first part of this question is yes, the answer to the second part is no. This apparent paradox may be resolved by treating the problem in terms of impedances (the quantity measured) rather than in terms of resonances.

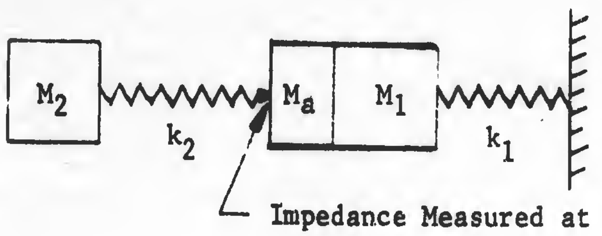
Consider the familiar spring mass system:



SKETCH 1

Resonance occurs at a frequency $\omega_1 = (k_1/M_1)^{\frac{1}{2}}$, the same frequency at which a minimum occurs in the impedance.

Now consider an impedance head attached to the mass M_1:



SKETCH 2

M_a represents the mass below the force gage in the impedance head which includes the accelerometer mass (only frequencies below the resonant frequency of the accelerometer are considered here), k_2 represents the stiffness of the force gage (only frequencies below the standing wave frequencies in the force gage are considered), and M_2 represents the mass of the impedance head housing plus any other mass attached to it such as rods and armatures of shakers or stators of reaction-type shakers. The two natural frequencies of this new system are different from the natural frequency of the system shown in Sketch 1. Yet the minimum in the impedance occurs at $\omega_2 = (k_1/M_1 + M_a)^{\frac{1}{2}}$, the same frequency as the natural frequency of the first system, except for the effect of M_a. Therefore, it is <u>only</u> M_a, the mass below the force gage, which changes the impedance measurements from the true impedance.

The mass below the force gage may easily be subtracted electronically in the actual measurements by adding 180 degrees out of phase (subtracting) a certain portion of the acceleration signal to the force signal, with adjustments being made when driving a zero impedance (no attachments to the specimen end of the impedance head). A 20-db cancellation is easily accomplished over the entire useful frequency range of the impedance head, although much higher cancellations have been obtained by compensating electrically for the increase in response with frequency of the accelerometer.

It should be noted that if the motion transducer is attached to the "wrong side" of the force gage the measured impedance will include the effect of the stiffness of the force gage, thus leading to large errors whenever the impedance of the specimen is of the same order of magnitude or higher than the impedance of the force gage $- jk_2/\omega$.

In general, the masses M_1 and M_2 in Sketch 2 may be replaced by any two impedances for the true impedance to be affected in the measurements only by the mass below the force gage.

To explain this in another way, consider a transmissibility test on a mounting system. In the usual arrangement a motion transducer is placed above and below the mount, and the transmissibility is the ratio of the two with the system placed on a shaker. Now if any other structure is placed between the shaker and the system under test, the transmissibility does not change, assuming linearity in the system, although the magnitudes of the motions and total system resonances will change considerably.

In this discussion, it was assumed that the static load attached to the specimen does not exceed its elastic limit and that the contact area of the impedance head is sufficiently small to prevent stiffening of the specimen. Further, it was assumed that the rotational impedance was high compared to the axial driving point impedance. When the rotational impedance is low, such as at the end of a cantilever or at any unsymmetrical point on a structure, the total inertia of the impedance head and driver about the point of attachment affects the results. For example, the low-frequency stiffness of a cantilever of length b is 3 EI/b^3, where E is the Young's modulus and I the moment of inertia of the cantilever. If the end of the cantilever is restrained from rotating freely, such as by attaching a large inertia to its end, the low-frequency stiffness will approach the stiffness with axial restraint or 12 EI/b^3. In this case the weight and, therefore, the inertia of the impedance head and attachments to shakers do affect results due to changes in boundary conditions. To alleviate this problem, the new drivers discussed in Section III A have been developed.

IV. APPLICATIONS OF MECHANICAL IMPEDANCE TECHNIQUES

Some applications of the mechanical impedance techniques used by the Structural Acoustics Branch at DTMB follow.

A. DYNAMIC PROPERTIES OF RESILIENT MOUNTINGS

In connection with the transmission of vibration through resilient mountings (or, more generally, through any isolator such as flexible shaft couplings), it is necessary to know certain characteristics as a function of frequency. Heretofore, the characteristic commonly named "transmissibility" was usually determined and reported. The transmissibility is the ratio of the blocked force on one end of the mounting to the force acting on the mass load (equal usually to the rated load) on the other end of the mounting; it is also equal to the ratio of the motion of the mass load at one end of the mounting to the applied motion at the other end. In either case, the transmissibility is a function of the mass load and, therefore, characterizes the system and not the mounting. It is difficult to make these transmissibility measurements under the large rated loads - for Navy mountings up to 10,000 lb - since these large loads cannot be regarded as lumped masses over the frequency range of interest, and the design of these loads affects the measurements.

A mounting may be represented as a mechanical black box with two accessible points. If the motion or force of each accessible

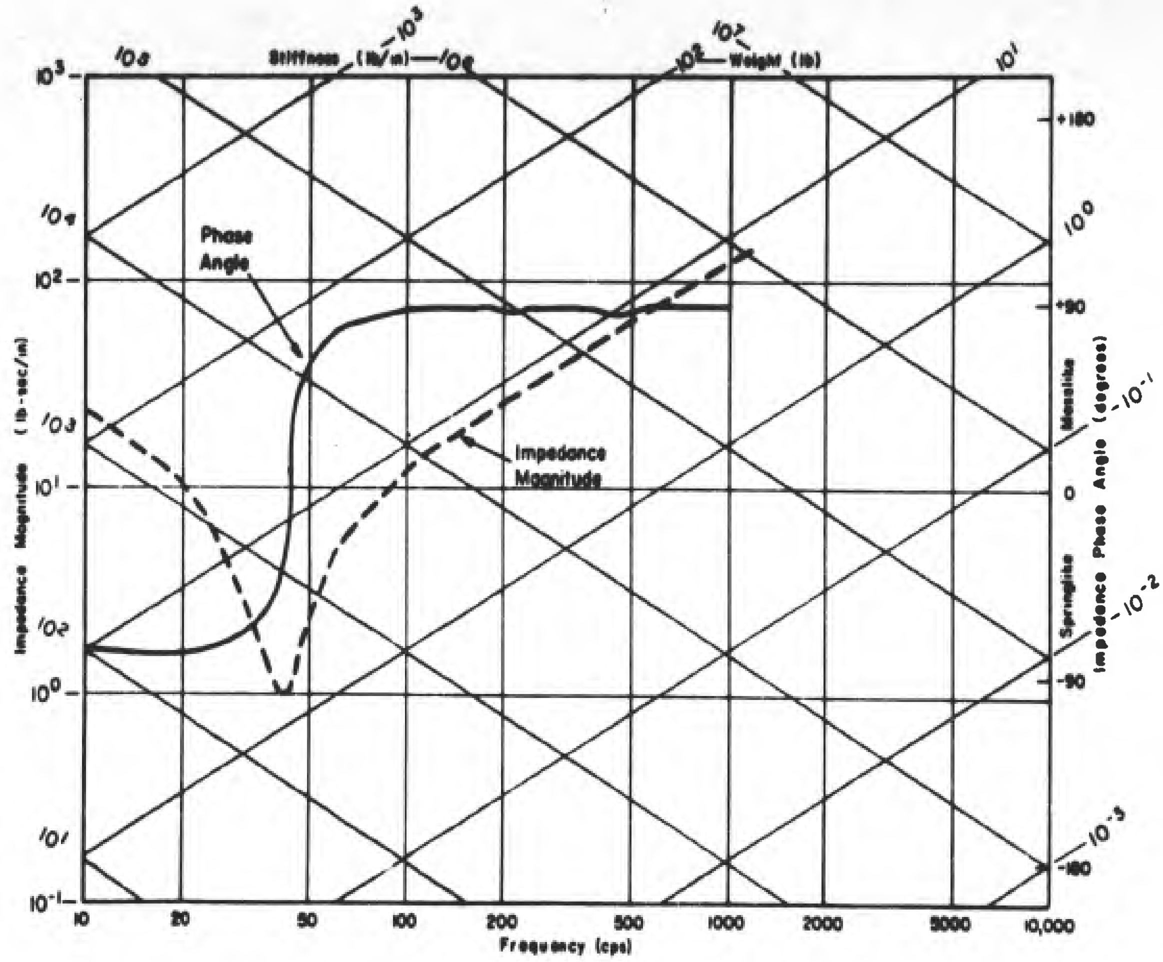


Figure 8 - Driving Point Impedance for a Rubber Mounting

point is restricted so that it can be described by a single space variable, then the mechanical behavior of the black box when connected to any system, including the conventional transmissibility, can be determined completely, provided three impedances measured at the two terminals of the box are known. It seemed most convenient to determine the following impedances:

1. Driving point impedance of end A with end B blocked.

2. Driving point impedance of end B with end A blocked (similar to (1) except for differences in plate weights).

3. Transfer impedance with one end blocked.

Except for wave effects, (1) and (2) are the impedances of a simple damped mass-spring system with the mass of the mounting plate, compliance, and resistance of the resilie element. These driving point impedances are shown for an actual rubber mounting in Figure 8. Wave effects at about 200 and 450 cps are masked in the impedance magnitude curve by the relatively large driving point impedance of the metal plates of the mount and they can only be detected from the more sensitive phase measurements.

Figure 9 shows the transfer impedance measurements under two different static loads. Essentially, this is a measure of the complex dynamic stiffness of the mounting, and with precision phase measurements the loss factor may be determined very easily over a wide range of frequencies as discussed in Reference 2. The measuring apparatus is shown in Figure 10, and the static load is applied by

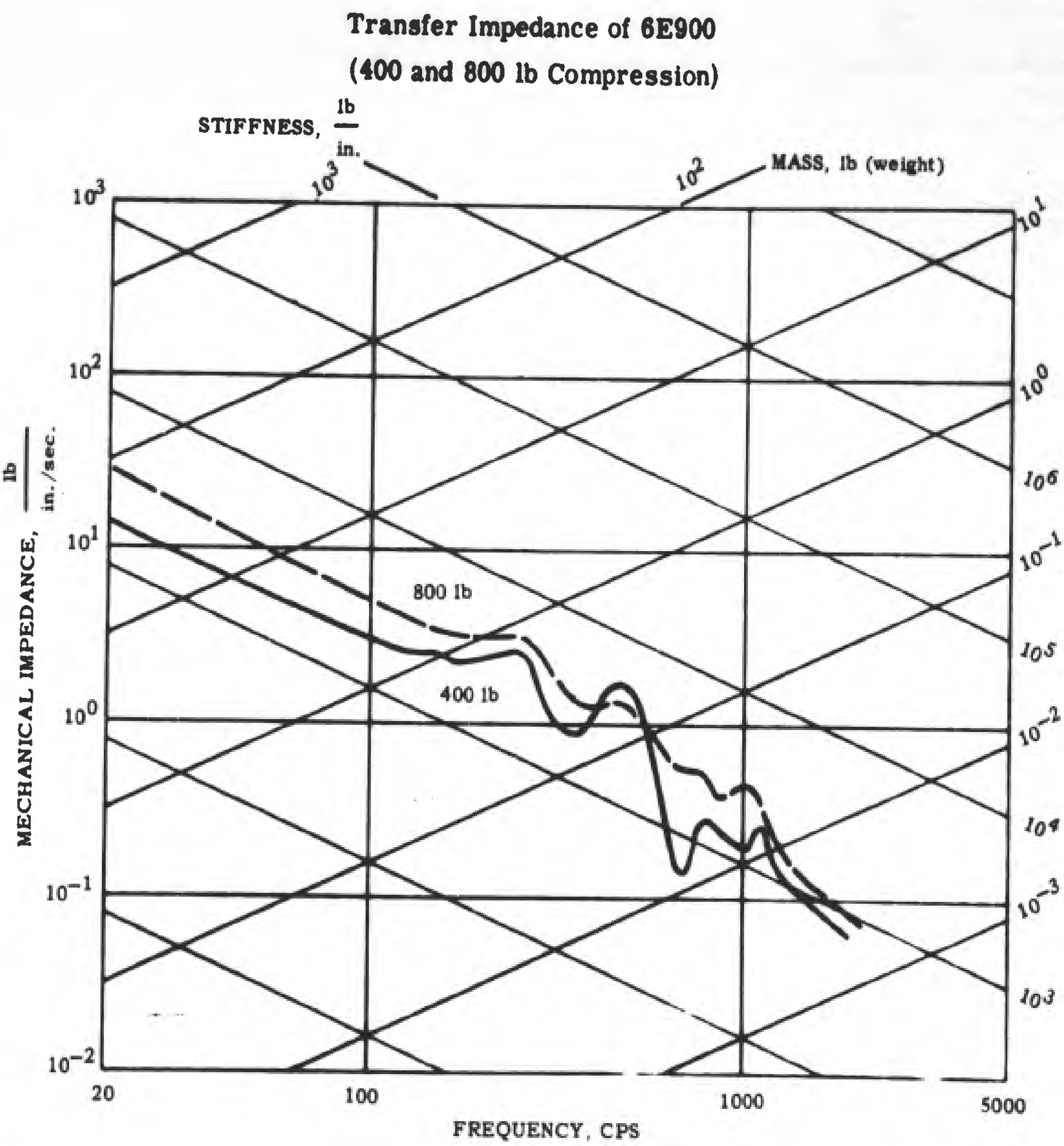


Figure 9 - Transfer Impedance for a Rubber Mounting

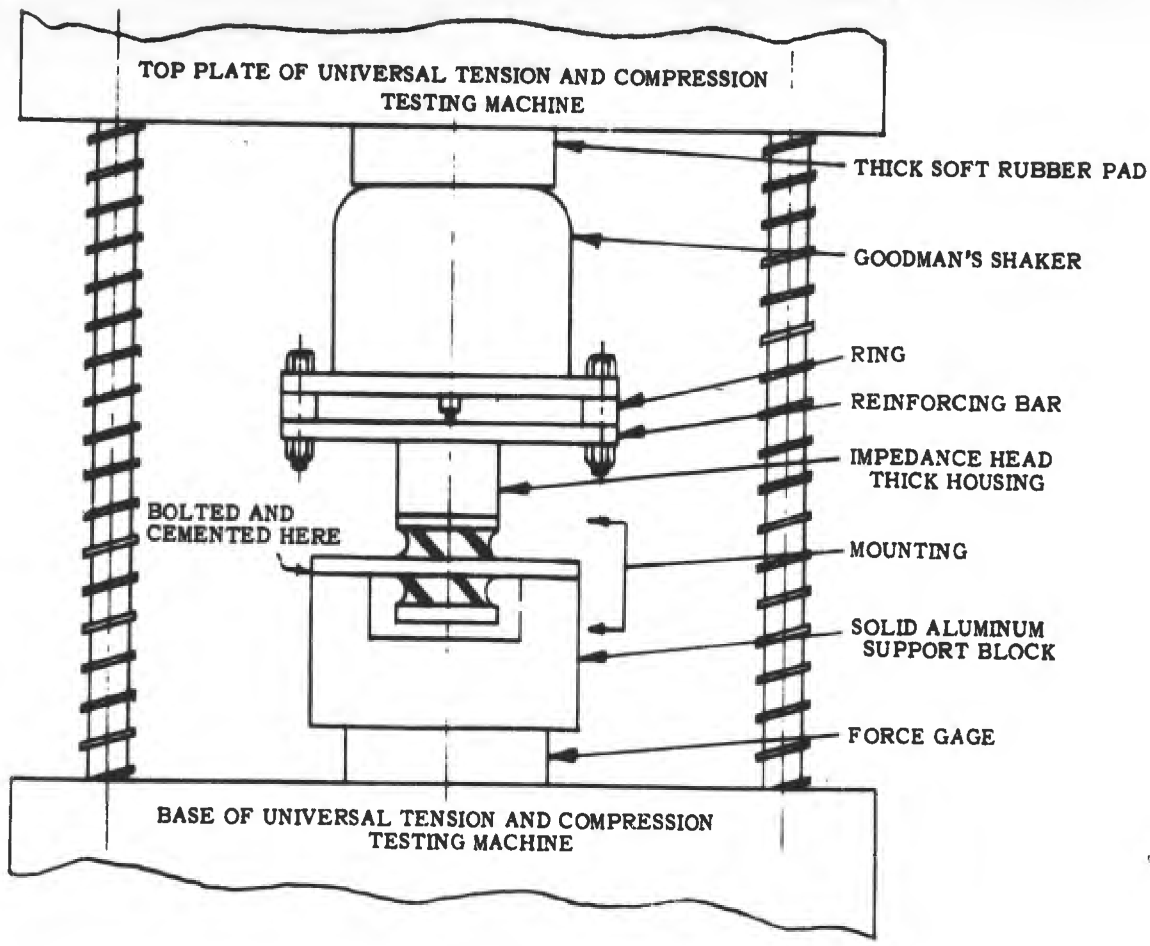


Figure 10 - Mounting Impedance Measuring Apparatus

a Universal Tension and Compression Testing machine. Mountings with rated loads up to 10,000 lb have been evaluated up to 5000 cps using this technique.

The acoustic decoupling properties of special coatings have been evaluated by the use of this method.

B. DYNAMIC PROPERTIES OF DAMPED STRUCTURES AND MATERIALS

A new technique was developed for measuring the loss factor of materials and systems continuously from 10 to 2000 cps or higher. By the use of this method, described in detail in Reference 2, accuracy is increased with increased damping and, therefore, this method complements other methods usually used to determine the damping properties at system resonances, which are more accurate for low values of damping. High damping materials are playing a widening role in the defense effort.

Using essentially the blocked transfer impedance technique given in Section III A, the loss factor is equal in magnitude to the cotangent of the impedance phase angle or the ratio of the real to the imaginary component of the impedance, provided the frequency is low compared to the frequency of the first standing wave in the specimen. The loss factor is defined as $\omega R/k$, where R and k are the resistance and stiffness of the system, respectively, and is equal to the reciprocal of the quality factor Q and 0.02 of the percentage of critical damping.

A high degree of phase precision is required in the instrumentation since, for example, a phase inaccuracy of 1 degree for a material having a loss factor of 0.1 is equivalent to a 17.5 percent error.

The apparatus is shown in Figure 11. A laminated beam blocked at the ends in the vertical direction and hinged to allow rotation is driven in the center. The transfer measurement is made between the force at the end and the motion in the center.

C. DYNAMIC PROPERTIES OF VISCOELASTIC MATERIALS

The blocked transfer technique can also be applied to continuously plot loss factor, and the real and imaginary moduli of viscoelastic materials such as solid fuels for missiles or mounting materials. The apparatus for the determination of the Young's modulus is shown in Figure 12. Sixty columns of viscoelastic material 1/8 in. in height and 1/32 in diameter are bonded at each end to two plates which are 1/2 in. in diameter. Since these materials have a low longitudinal bar velocity of sound, the height of the columns must be relatively short to increase the upper limit of the usable frequency range. The diameter of each column must be small compared to the height to assure that the data are a true measure of the Young's modulus. Many columns are used to provide mechanical stability and to prevent buckling. The apparatus may be inverted so that the columns are statically in tension, and the tension may

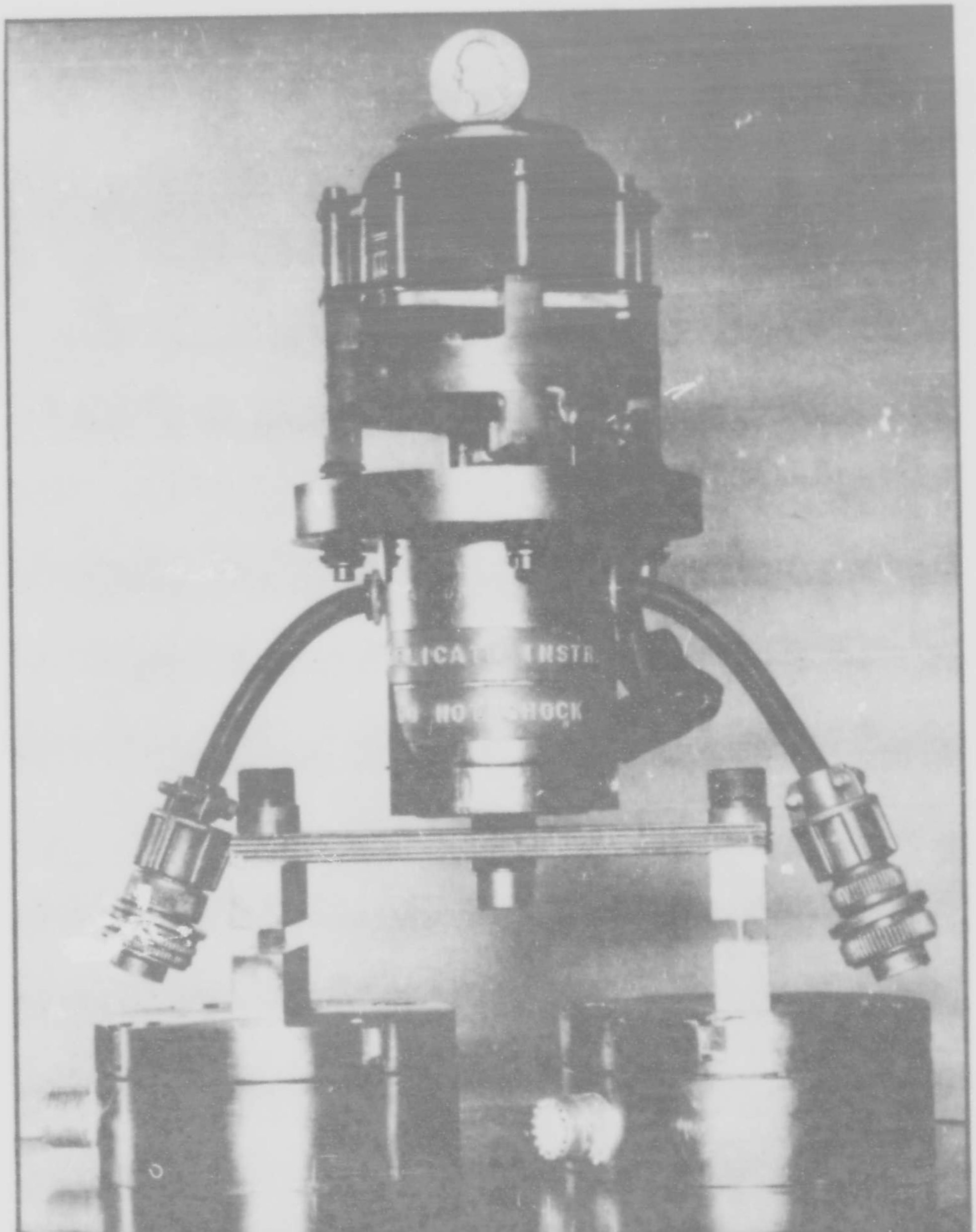


Figure 11 - Apparatus for the Determination of the Loss Factor of Damped Beams

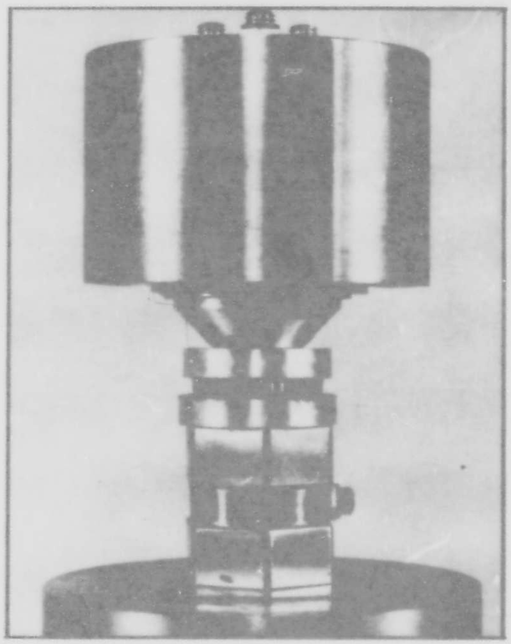


Figure 12 - Apparatus for the Determination of the Dynamic Properties of Viscoelastic Materials

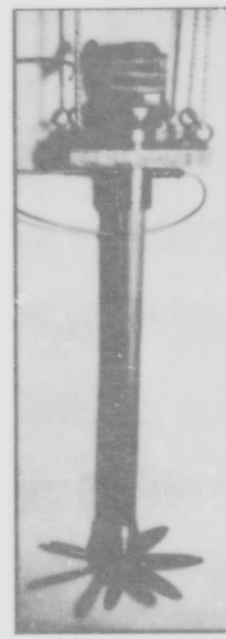


Figure 13 - Apparatus for the Determination of Propeller Resonances and Added Masses

be adjusted.

Similarly, the bulk or shear properties may be evaluated either by placing a thin layer of viscoelastic material between two plates or between two concentric cylinders, respectively.

D. FUNDAMENTAL RESONANT FREQUENCY OF PROPELLERS AND ADDED WATER MASSES

Impedance techniques have been used to determine resonant frequencies of structures without loading the structure or changing the damping of the structure by placing the impedance head at a node; also they have been used as convenient dynamic weighing devices.

Figure 13 shows the apparatus used to determine the fundamental resonant frequency of full-scale submarine propellers using impedance measurements on models. The propeller is driven at its hub through a long rod to eliminate the need for waterproofing the transducer. The first antiresonant frequency (maximum in impedance) is measured in air and water. From these measurements, the added water mass at resonance and the resonance of full-scale propellers may be calculated even though the model propellers are made of a different material.

Viscous effects may also be determined using precision phase measurements. The heavy plate at the top of the impedance head above the force transducer is used to prevent rotation of the propeller in the water since the resonant frequencies of propellers are dependent upon rotational restraint.

This impedance measuring apparatus was also used as a very convenient dynamic weighing device to obtain the added fluid mass of rigid bodies moving in fluids. This work will be reported shortly.

E. MEASUREMENT OF THE INTERNAL DAMPING CHARACTERISTICS OF METALS

In most of the theoretical impedance studies, linear behavior of structures is assumed, which was partially verified by measurements on submarines under relatively small vibratory stress levels. Yet it is known that under high stress levels the internal damping of metals is not independent of the amplitude of vibration. Therefore, experiments were conducted to determine the internal damping characteristics of metals as a function of vibratory stress. Since there is very little internal damping in metals, no additional damping must be introduced by vibrating transducer cables and friction at the points of attachment. A double cantilever beam machined from one piece was driven in the middle through an impedance head at its first antiresonance, as shown in Figure 14. The middle portion, a node at that frequency, was thickened to further reduce frictional effects. The damping properties may be calculated from the impedance magnitude or the shape of the impedance curve at the antiresonance. The results are shown in Figure 15 for an aluminum, steel, and stainless steel bar as a function of the outer fibre stress.

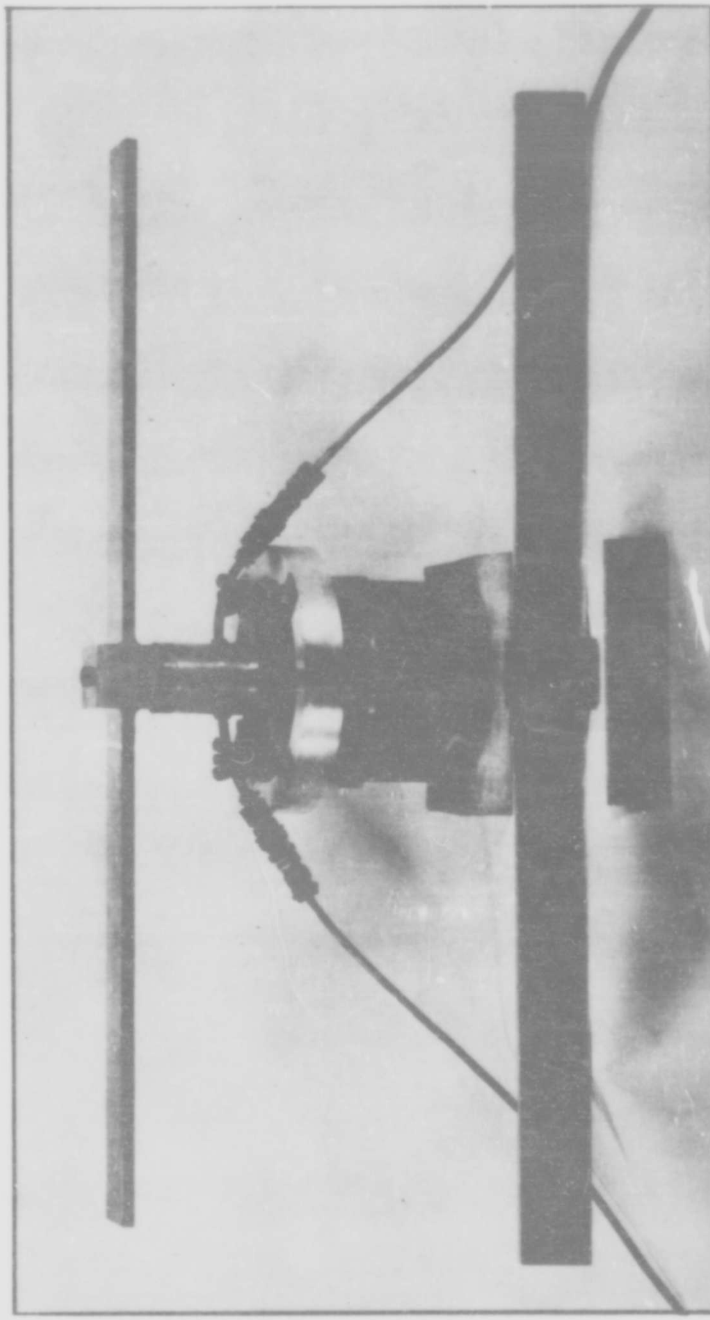


Figure 14 - Apparatus for Determination of Internal Damping of Metals

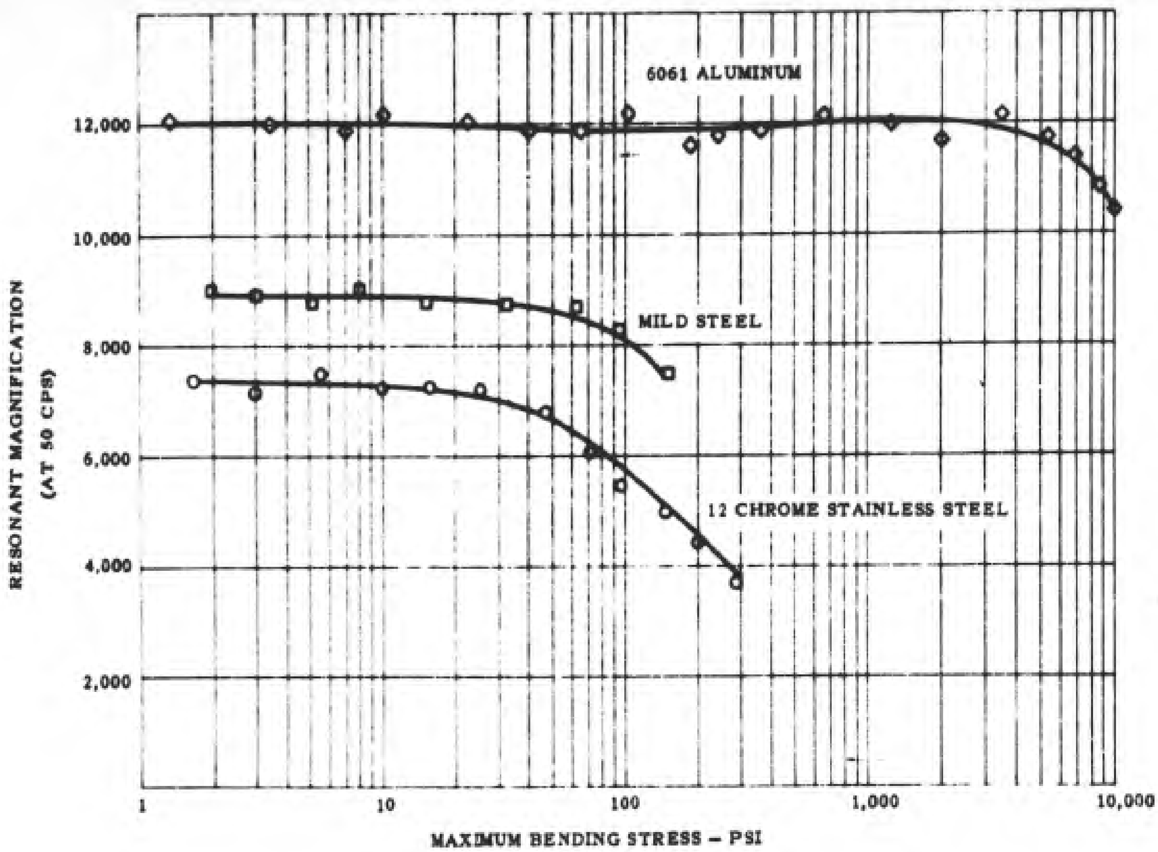


Figure 15 - Resonant Magnification of Various Materials versus Stress Levels

F. OTHER APPLICATIONS

A technique similar to IV E above, namely determining the impedance at a node, was used to obtain the Young's modulus of elasticity of heterogeneous materials such as fibreglass in the various directions. Using this method, time and cost savings of over 80 percent were attained over more conventional static methods using strain gages.

The feasibility of using impedance techniques is presently under study in an attempt to quiet singing propellers.

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