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The Design and Testing of a Low Range Acceleration Transducer with Predictable Response Characteristics

by

I. McLaren

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THE DESIGN AND TESTING OF A LOW RANGE ACCELERATION TRANSDUCER WITH PREDICTABLE RESPONSE CHARACTERISTICS

by

I. McLaren

SUMMARY

A remote indicating accelerometer has been developed for aircraft response measurements in atmospheric turbulence. The instrument is designed to be dynamically linear over a frequency range 0-10 c.p.s. with an acceleration range of 0-2g. Some of the principal factors influencing the design of the instrument are discussed and the unit and the laboratory tests made are described. A theoretical analysis of the system response to sinusoidal inputs of various frequencies is made and verified experimentally.

Tests show that the accelerometer should be satisfactory over the required frequency range.

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1 INTRODUCTION

The increasing emphasis recently placed on aircraft instrumentation for oscillatory techniques in aerodynamic flight testing has led to a demand for an accelerometer with a response which can be described by a linear differential equation with constant coefficients. This is difficult to achieve over a large range of amplitudes and frequencies, and under varying temperatures and pressures.

It is appropriate at this stage to mention that although methods exist to solve some types of differential equations with variable coefficients, the numerical and experimental work involved in determining the complete equation which truly describes the motion of the system may be prohibitive.

Two important factors contributing to dynamic non-linearity are friction and backlash which are generally attributable to the types of pick-off and damping system employed in the design of the instrument. Dry or "Coulomb" friction i.e. friction whose magnitude is independent of velocity, is a drag between two rubbing surfaces and the complete elimination of this type of friction is imperative. Devices such as links, pivots, and bearings can be responsible for poor dynamic performance owing to backlash and friction, and all possible design measures should be taken to eliminate them.

The use of dampers on accelerometers is necessary to provide means of controlling the response of the instrument to dynamic inputs. For high dynamic performance it is important to ensure that the damping is proportional to velocity and does not alter appreciably under environmental conditions experienced in flight. The characteristics of gas and oil damping systems, working in either the viscous-shear or fluid displacement configuration, change appreciably with pressure and temperature respectively. Other disadvantages, more obscure perhaps, but nevertheless important, associated with these systems are secondary mass effects, viscous effects superimposed on shear damping, and compressibility effects.

The instrument described in this note was designed to minimise these sources of errors and to provide an output which could be predicted under dynamic conditions.

2 BASIS OF THE ACCELEROMETER DESIGN

The first stage in the development of the unit was to choose the type of transmission system and the method of damping.

From the considerations of the previous section and from experience with potentiometric types of pick-offs, it was decided to use a variable reactance pick-off and eddy current damping.

The advantages of an a.c. pick-off are that it is frictionless, resolution is infinite, and the phase shift can be made negligible throughout the working range. One of the most important reasons for adopting this type of pick-off is its capability of measuring very small movements. This feature enables the instrument to have a high undamped natural frequency since the latter is inversely proportional to the square root of the displacement of the mass on mass-spring systems. As with most pick-offs, the output is dependent on the input voltage when used with a galvanometer but appreciable changes in supply frequency have negligible effect.

Potentiometers, on the other hand, have inherent disadvantages such as poor resolution, friction, and wiper bounce at high frequencies.

Past experience with some types of silicone oil and air damping techniques suggested that these were not immune from the effects of variation of temperature and pressure. Furthermore, there is a general lack of appreciation of the broad principles governing the design of these systems. It has been known for some time that silicone oil damping, even with careful design, can alter the parameters of a mass and spring system because of changes in the viscosity of the damping fluid. The variation in viscosity of silicone oil with change of temperature is much less than for other oils but is nevertheless large enough to be important in the accurate and reliable prediction of the dynamic performance of flight test instruments. Damping methods employing piston and cylinder arrangements can introduce backlash and friction in the link mechanism and to secure smooth operation of the piston inside the cylinder careful design and construction are essential.

There are two systems of air or gas damping in general use. In one case a piston or vane is attached to the moving system and moves in an air chamber which takes the form of a closed pneumatic circuit or incorporates an adjustable orifice at the closed end. The second method employs a diaphragm and capillary tubing, the former being attached to the moving system and "pumping" air through the capillary. The objective in both cases is to create a frictional drag on the moving system by utilising the resistance to air flow as a damping force. In practice the damping achieved may be associated with laminar and turbulent flow conditions. If the flow pattern remains the same at low frequencies and small amplitudes of the mass-spring system the flow is non-turbulent and the damping force will vary approximately linearly with velocity. However, for increasing speed of the moving system due to increasing amplitude and/or frequency, the flow becomes turbulent if a critical velocity is exceeded. There is thus a change in the damping force and it will now vary approximately as the velocity squared. When these and compressibility effects are taken into consideration it will indeed be fortuitous if the dynamic performance of the accelerometer is linear. The author has not had the opportunity of making investigations to confirm these theories, but it is known that several instruments exist where the air flow for damping purposes is in the turbulent region and for these instruments, peculiar dynamic effects have been apparent.

For high dynamic performance the device that satisfies most requirements for velocity damping under all conditions is the eddy current damper. Difficulty may be experienced in trying to incorporate this type of damping due to the size of magnets required but the rewards of achievement are worthwhile in that the damping force is:-

(a) for all practical purposes, proportional to velocity,

and (b) changes only slightly with temperature and is independent of atmospheric pressure. Furthermore no links or lever arms, which can introduce friction or backlash into the system, are required.

To complete the outline of present treatment of damping problems, it should be noted that there is an arrangement whereby the demand for such precise damping characteristics is not so stringent. An example of this arises if the undamped natural frequency of the instrument is very much greater than the highest impressed frequency to be measured. Changes of the damping factor now become of less importance and by suitable choice of a low damping factor negligible phase shift is achieved over the working range. However great difficulty is encountered in obtaining a high undamped natural frequency for sensitive instruments which are required to operate robust ratiometers without amplifiers. There is also a possibility that the noise level due to unwanted high frequencies may become excessive.

3 CONSTRUCTION OF THE UNIT

The acceleration is measured in the conventional manner by a mass and spring system. The inertia force experienced by the elastically suspended mass housed in the accelerating body causes movement of the mass. This movement is proportional to the acceleration affecting the mass.

Structurally the accelerometer comprises a pair of parallel leaf springs which are rigidly attached at one end to a housing within the case and a mass is attached to the free ends. The springs are designed to have a linear load-deflection characteristic over the working range and to have a high transverse stiffness. For simplicity the mass of the instrument acts as the armature of the pick-off for converting mechanical movement of the mass on its elastic suspension into an electric parameter. The accelerometer is illustrated in Fig.1 and the basic arrangement is depicted in Fig.2.

Referring to Fig.2 it will be seen that the specially designed layout of the instrument makes it exceptionally easy to carry out adjustments and inspection.

The pick-off consists of a pair of E-cores, which are housed and potted with their respective coils in a brass dish. In order to minimise cross effects the springs are biased so that they are straight throughout their entire length when the mass and spring assembly is subjected to an acceleration of 1g in the middle of the range. Damping is effected by means of a copper cup attached to the mass and moving in a radial magnetic field provided by a "Ticonal G" magnet and suitable pole pieces. To preserve symmetry of the moving system a balance weight is attached to the mass.

Basically the pick-off is of the variable area type working in push-pull. Movement of the armature causes a simultaneous change in the two coils on their respective E-cores, increasing the inductance of one and at the same time decreasing that of the other by equal amounts. For small movements the change of inductance of the coils is approximately proportional to the displacement of the armature from the neutral position. It was found by experiment that the maximum movement of the armature to obtain the desired sensitivity within prescribed limits of linearity was of the order of ±0.05 inches. Thus with a given mass $\underline{m}$ and a deflection of 0.05 inches per g the spring stiffness $\underline{k}$ may be determined from the relation

$$kx = ma \tag{1}$$

where $\underline{x}$ = static deflection of mass

and $\underline{a}$ = acceleration.

If $\underline{d}$ is the deflection of the mass for an acceleration of 1g, then

$$k = \frac{mg}{d} . \tag{2}$$

The undamped natural frequency $\underline{f_u}$ associated with this system is given by

$$f_u = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \tag{3}$$

and substituting from (2)

$$f_u = \frac{1}{2\pi} \sqrt{\frac{g}{d}} . \qquad (4)$$

With a deflection of 0.05 inches per g the natural undamped frequency f_u is 14 c.p.s. For the prototype the values of m and k were 0.042 pounds and 231 poundals per foot respectively giving a natural undamped frequency of 11.8 c.p.s. This is considered satisfactory for the particular requirement.

To establish a damping ratio of 0.7 the magnetic system is magnetised up to saturation point, in which condition the damping factor is well above critical. A demagnetising force is then applied in discrete steps by inverting the instrument in the magnetic field of the magnetiser with various air gaps between the instrument and one pole of the machine. Meanwhile step input tests are carried out simultaneously at each stage in the process until the overshoot for 0.7 damping is achieved.

4 ASSOCIATED CIRCUITRY

For use with a galvanometer the two E-core coils are combined in a bridge network which includes a phase discriminating circuit in the form of two silicon junction rectifiers. The output from this bridge contains a large a.c. component at carrier frequency plus a pulsating d.c. signal component and a filter circuit is required to eliminate the former. This low-pass second order filter is also used to extend the usable frequency range of the instrument. The preferable choice of components for the filter is resistance and capacitance elements, but the response curve of this type of network is not suitable because the damping is never less than critical. The circuit is shown in Fig.3.

When the output from the transducer is fed into a ratiometer of the S.F.I.M. type the phase discriminating network and filter are not required. The two voltage coils of the ratiometer are combined with the two pick-off coils to form a bridge circuit, the current coil of the ratiometer forming the "galvo" arm of the bridge.

For the ratiometer configuration the standard 115 volt 400 c.p.s. aircraft supply, suitably transformed down to 30 volts, is adequate. However the supply requirements for the galvanometer case are severe. A highly accurate control of voltage is necessary over a wide range of environmental conditions and the waveform must be clean and free from jitter and noise. For optimum performance it is recommended that a static inverter be used as a source of supply.

5 THEORETICAL ANALYSIS

A procedure that has been found very satisfactory for determining the overall response of the complete system is to perform the analysis on a sinusoidal basis. Use is made of the fact that the magnification factor of the complete system is the product of the magnification factors and that the phase shift is the sum of the phase shifts of each individual section.

In both the ratiometer and galvanometer configurations the accelerometer and output circuitry can be considered as non-interacting adjacent variables, i.e. the displacement of the armature of the pick-off is independent of the output circuitry under any working condition. For the ratiometer case the transfer function of the complete system is composed of two non-interacting second order systems.

However, to analyse the dynamic behaviour of the combined mechanical-electrical system when the output is fed into a galvanometer, it is convenient to use an analogous electrical network for the accelerometer in which all the relevant physical constants are represented. A table of analogous quantities used in this note is included in Fig.4. The network in Fig.4 can be substituted for the mechanical system of the accelerometer.

For various sinusoidal inputs we are concerned only with the steady state response and denoting the differential operator d/dt by Jw_i, we have the impedance,

$$Z = R + Jw_i L - \frac{J}{w_i C} \tag{5}$$

and the current,

$$i = \ell_i \left(R + Jw_i L - \frac{J}{w_i C}\right)^{-1} . \tag{6}$$

Therefore the p.d. across C,

$$\ell_o = - \frac{Ji}{w_i C} \tag{7}$$

and substituting from (6)

$$\ell_o = \ell_i \left(1 - w_i^2 LC + Jw_i RC\right)^{-1} \tag{8}$$

and the ratio of output to input is

$$\frac{\ell_o}{\ell_i} = \left(1 - w_i^2 LC + Jw_i RC\right)^{-1} . \tag{9}$$

Putting
$$LC = \frac{1}{w_N^2} ,$$

$$C = \frac{1}{w_N^2 L} ,$$

and $$\frac{R}{w_N L} = \frac{1}{Q} = 2\zeta$$

where w_N = undamped natural frequency

and ζ = damping factor relative to critical,

we have $$\frac{e_o}{e_i} = \left(1 - \frac{w_i^2}{w_N^2} + J2\zeta\frac{w_i}{w_N}\right)^{-1} . \quad (10)$$

With $$\frac{w_i}{w_N} = N_1 ,$$

$$\frac{e_o}{e_i} = \left(1 - N_1^2 + J2\zeta N_1\right)^{-1} . \quad (11)$$

This is the well known equation for a simple mass, spring, and velocity damped system. Thus the magnification factor and phase shift are given by

$$|A|_1 = \left[\left(1 - N_1^2\right)^2 + 4\zeta^2 N_1^2\right]^{-\frac{1}{2}} \quad (12)$$

and $$\varphi_1 = \tan^{-1} 2\zeta N_1 \left(1 - N_1^2\right)^{-1} \quad (13)$$

where $|A|_1$ is the ratio of the amplitudes of the output to the input.

Before attempting the analysis of the electrical circuit it is important to study the components of the network in detail. Also, unless certain assumptions are formulated and resort is made to graphical representation, it is difficult to calculate the response characteristics of the complete circuit. However, although the representation of the circuit elements in terms of constant parameters for purposes of analysis may appear to lack the elegance of analytical methods, it can be made to yield extremely useful results with little expenditure of effort.

Commencing with the choke the basic requirement of this element is that its inductance should remain reasonably constant over the working ranges of current and frequency. To obtain this feature it is necessary to use mumetal laminations to give high permeability and low hysteresis loss. The laminations acceleration can be considered independent of frequency. With given values of $\underline{L}$ and $\underline{C}$ the problem to be solved is the choice of value of $\underline{R_1}$. Calculation of this factor showed that a value of about 1600Ω would give the optimum in response of the complete system. Accordingly the bridge resistance arm was set at 1200Ω so that the value of $\underline{R_1}$ in the equivalent network shown in Fig.3 was of the order of 1500Ω.

A very convenient and useful means of combining both the phase shift and magnification factor into one composite picture is a polar graph of the amplitude against the phase. A polar frequency plot of the two sections of the system is shown in Fig.5 together with that of the complete system. Experimental tests made on the electrical circuit at various amplitudes of excitation over a wide frequency range showed very good agreement with the calculated results.

7 STATIC CALIBRATION AND TEMPERATURE TESTS

The instrument was statically calibrated by inverting in the earth's gravitational field and on a centrifuge. A curve of acceleration in terms of galvanometer deflection is shown in Fig.6. It is seen that there is little or no hysteresis and the output is practically linear over 70% of full scale, and while this is considered adequate for the present work steps are being taken to improve the linearity.

The complete system was checked in a refrigerator and an oven over the range $-40^{\circ}C$ to $+60^{\circ}C$. In the low temperature region the sensitivity increased by $1\frac{1}{2}$% and there was a zero shift of 2%. At a temperature of $+60^{\circ}C$ the sensitivity increased by 1% with a zero shift of $1\frac{1}{2}$% in the same sense as that encountered at $-40^{\circ}C$. From results of further tests it became apparent that the major part of these discrepancies emanated from the accelerometer itself as the output circuit remained remarkably stable throughout the temperature range.

Flick tests performed at $-20^{\circ}C$ showed that the damping factor had increased from 0.7 critical to approximately 0.9. This variation in damping with change of temperature is mainly due to the large temperature coefficient of resistance of the copper damping ring.

8 METHOD OF DYNAMIC CALIBRATION

It is well known that the evaluation of the dynamic response of an accelerometer consisting of a mass, spring, and velocity damped system can be computed from the differential equation when the two main fundamental parameters, the undamped natural frequency and the damping factor relative to critical are known. However to prove whether the inevitable assumptions used in setting up the equation are justified, experimental dynamic tests made with considerable precision are necessary.

Two well-tried methods suggest themselves for these experiments, first the response to a step-input and second the dynamic calibration of the instrument on a device which is capable of executing sinusoidal oscillations over the desired ranges of amplitude and frequency. Although the transient response yields valuable information, the results obtained from such a test are inconclusive, especially when information is required at low frequencies and it was decided that sinusoidal analysis would provide a better performance criteria.

As the accelerometer to be tested was of the type for measuring normal acceleration, sinusoidal oscillations in the earth's gravitational field were required to study the characteristics in equilibrium about the 1g zero position.

The shaking table normally used for such tests operated in a horizontal plane and was therefore unsuitable for this particular test.

Accordingly a large "shaker" like an accelerometer was built similar in principle to the test accelerometer except for the absence of eddy current damping. To produce a reasonable value of acceleration at low frequencies the springs have to be at least 8 to 10 feet in length. The mass and the length of the springs can be altered to give a wide range of accelerations at various frequencies. The shaker is a free-oscillating mass and spring system executing simple harmonic motion when released from a selected deflected position.

In practice the amplitude of the oscillations does decay very slowly since there is always a certain amount of damping present arising from internal friction in the springs and from the resistance of the air. However in this case the damping is negligibly small and, as will be seen later, very little distortion is introduced as a result of this effect. How lightly damped the vibratory motion is can be clearly seen from the ratio of two successive maxima from a specimen of the recorded traces taken during the tests and shown in Fig. 7.

The test instrument was mounted on the mass of the shaker and great care was taken to ensure that the shaker motion was truly vertical. The amplitude of the shaker motion was measured by a combination of photo-electric cells and a potentiometer, the outputs of which together with the output of the test specimen were fed into 65 c.p.s. galvanometers in a multi-channel optical recorder.

9 THE PERFORMANCE OF THE INSTRUMENT

The derivation of the dynamic calibration curves for phase and amplitude from the recordings is now considered.

For simple harmonic motion the displacement of the shaker mass is given by

$$x = X \cos wt,$$

velocity by $$\dot{x} = -X\,w \sin wt$$

and acceleration by $$\ddot{x} = -X\,w^2 \cos wt.$$

The peak acceleration $\hat{\ddot{x}} = -w^2\,\hat{X}$ ft/sec^2 where $\hat{X} = \frac{1}{2}$ peak-to-peak displacement = amplitude and $w = 2\pi f$ where f = frequency in cycles per second.

The latter expression was the one used during the tests for calibration purposes. It was necessary to consider whether the inherent damping in the shaker would require considerable corrections to be made to this expression and also to the phase shift measurements as interpreted from the records. A careful examination of the test records revealed the maximum damping factor was of the order of 0.006, and as might have been expected, occurred at high frequencies and large amplitudes. Since ζ, the damping factor, is $<<1$, a convenient rule of thumb method to provide correction factors to the amplitude and phase shift characteristics has been devised. It consists of multiplying the calculated acceleration input by the factor $(1-\zeta^2)e^{-2\zeta^2}$ and retarding the phase shift of the output by 2ζ radians. For $\zeta = 0.006$ the error in the acceleration input is less than 0.01% and the phase shift error is 0.7° which shows that the effect of the damping on the shaker is negligible.

In the calibration of the transducer photo-electric cells were placed at equidistant points from the neutral position of the shaker and this peak-to-peak distance was noted. Recording was started when the shaker was released from a deflected position and continued until the photo-cells just ceased to blip due to the decay of amplitude of the shaker. Measurements were taken at the corresponding stage on the recorded traces of amplitude, frequency, and phase-shift. For amplitudes of 8 inches or less a potentiometer was used after it had been calibrated for linear displacement of the shaker with known d.c. potentials. A large range of frequencies was covered at various amplitudes of acceleration.

The interpretation of the records for amplitude measurement is relatively straightforward but the determination of the phase shift is rather more difficult. The method employed in measuring this angle entailed obtaining the time differences between appropriate pairs of adjacent half-cycles and averaging these readings. A standard 50 c.p.s. tuning fork was used to check the timing marks on the recording traces from which time measurements were obtained. To facilitate phase shift measurement the polarity of the accelerometer output was reversed on the galvanometer so that the traces from the potentiometer and accelerometer were in-phase at low frequencies instead of anti-phase.

Tests were made for a number of amplitudes at particular frequencies and amplitude ratios were computed from the records. These have been suitably plotted on a graph in Fig.8. For comparison with the calculated plots the mean values of the amplitude ratios and phase shifts are plotted on the polar graph illustrated in Fig.9. The various amplitude readings are within ±2% and the phase angles within ±1.5° at particular frequencies and these errors are of the same order as those of the test measurements. It will be seen that there is good agreement between the calculated and experimental polar plots of the complete system.

An interesting feature was the surprisingly accurate measurement of phase angle obtained from the records. It was shown in para.9 that the phase relation between the amplitudes of displacement and acceleration of the shaker was of the order of 180.7°. It can be seen from Fig.10 that the mean straight line drawn through the points of the various values of phase angle plotted against frequency goes approximately through the phase angle lead point (0.7°) and not through the origin.

10 CONCLUSIONS

An accelerometer with a 0 to 2g range incorporating the two important features of magnetic damping and no friction has been developed.

Tests on the accelerometer and associated circuitry have shown that

(a) The amplitude ratio and phase lag are independent of the amplitude of input.

(b) The static calibration is free from hysteresis errors and at temperatures between −40°C and +60°C the maximum discrepancy is 3.5%.

(c) The damping factor changes with temperature from 0.7 crit. at 20°C to approximately 0.9 crit. at −20°C.

(d) The amplitude ratio is constant to within ±2.5% from 0 to 9 c.p.s. (Fig.9).

(e) From the design of the accelerometer, pressure changes should have negligible effect on the damping.

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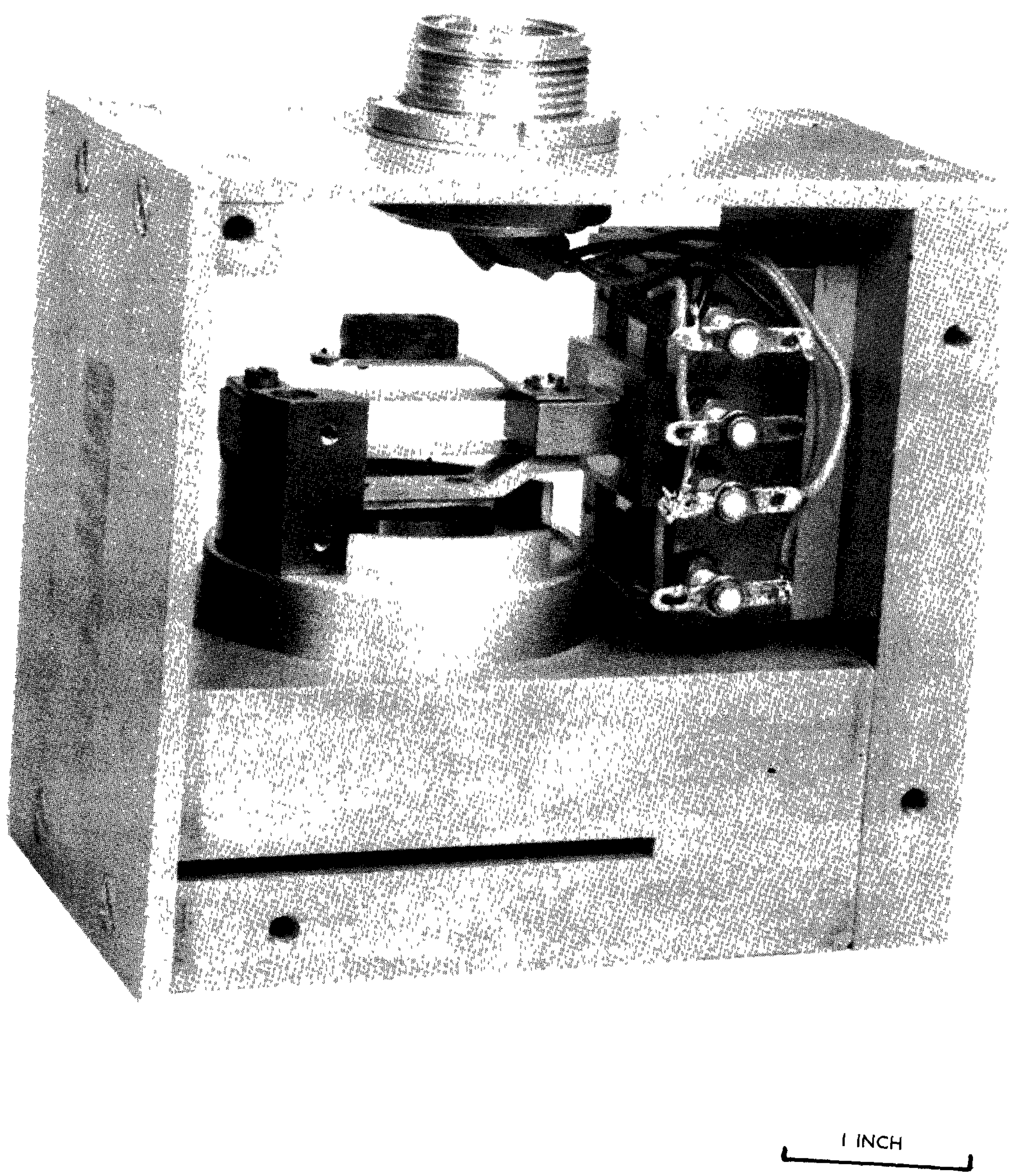


FIG.1. ACCELEROMETER

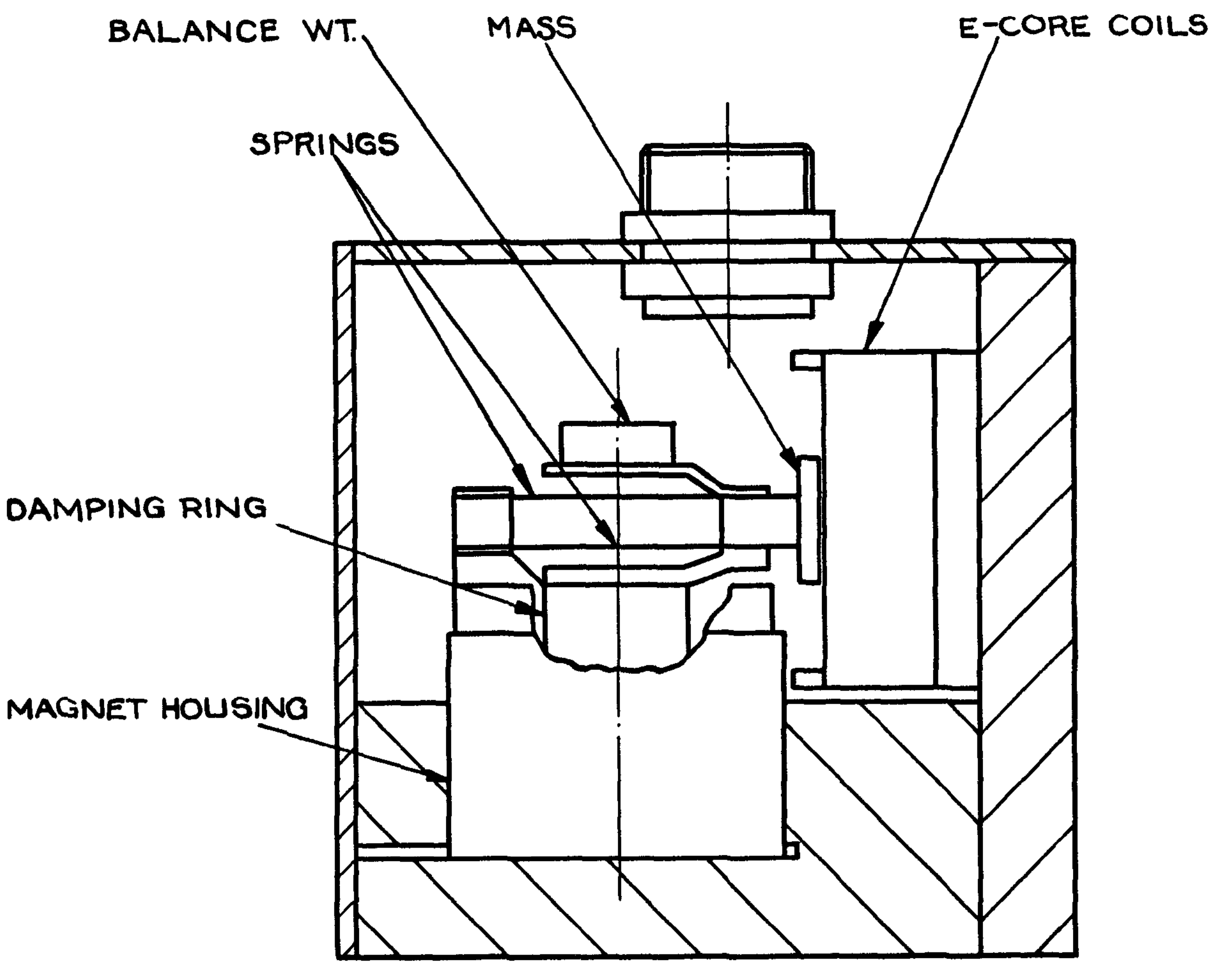
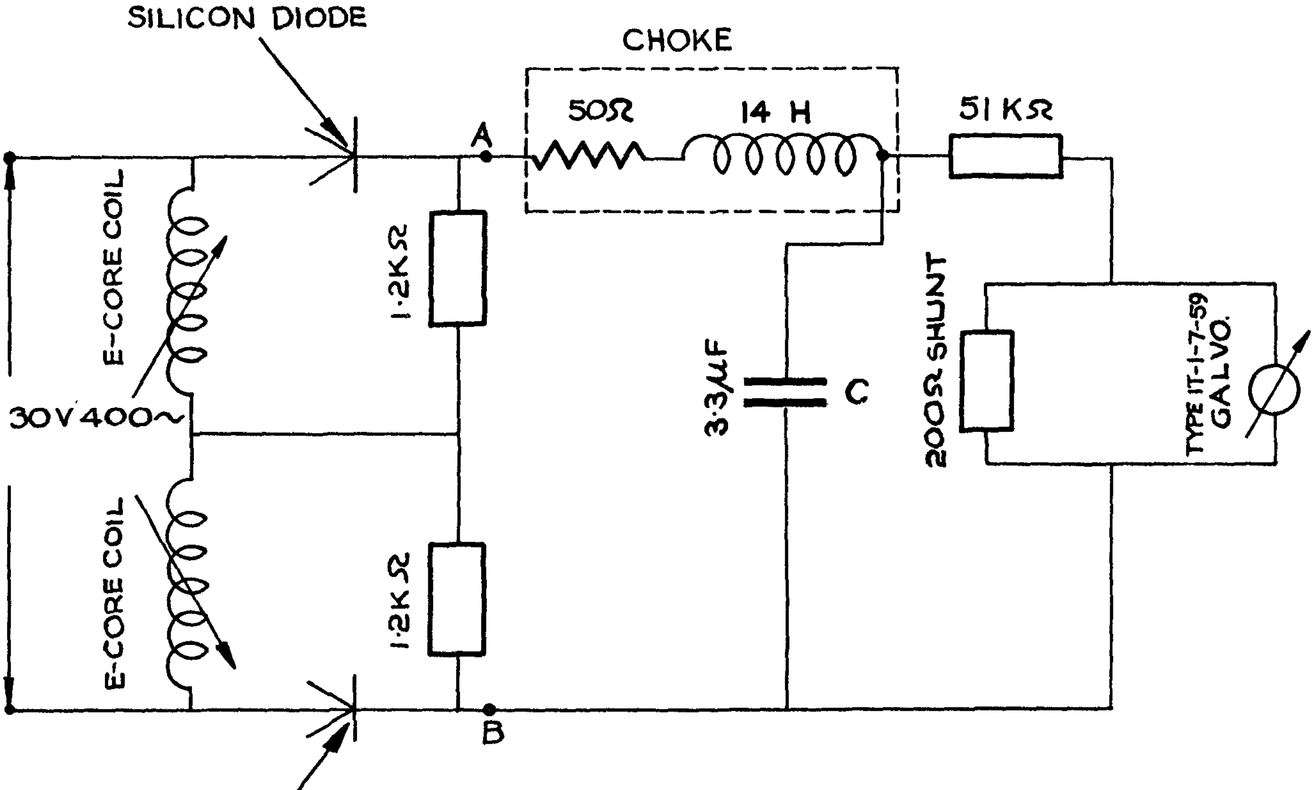
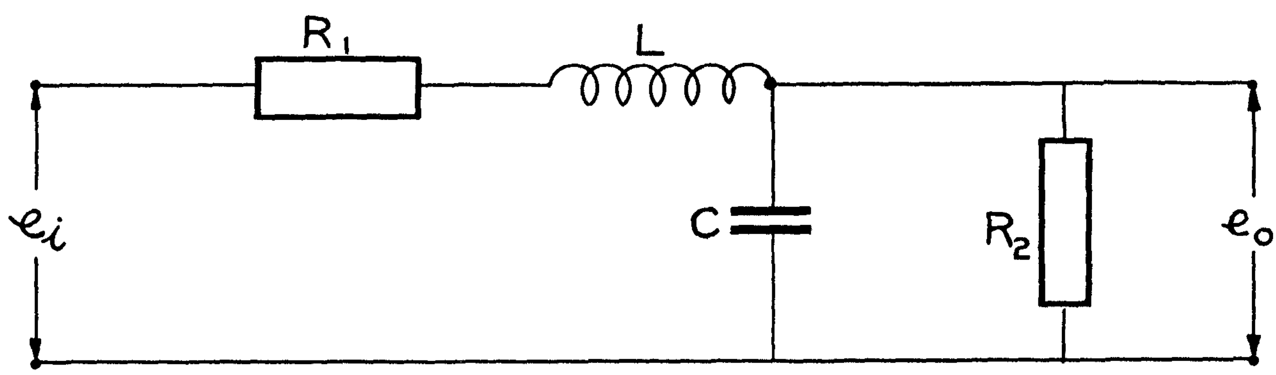


FIG. 2. CROSS-SECTIONAL VIEW OF ACCELEROMETER.



OUTPUT CIRCUIT

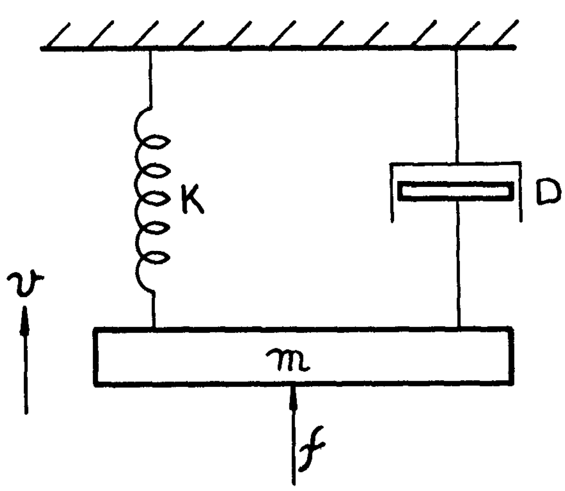


EQUIVALENT NETWORK

FIG. 3. OUTPUT CIRCUIT-EQUIVALENT NETWORK.

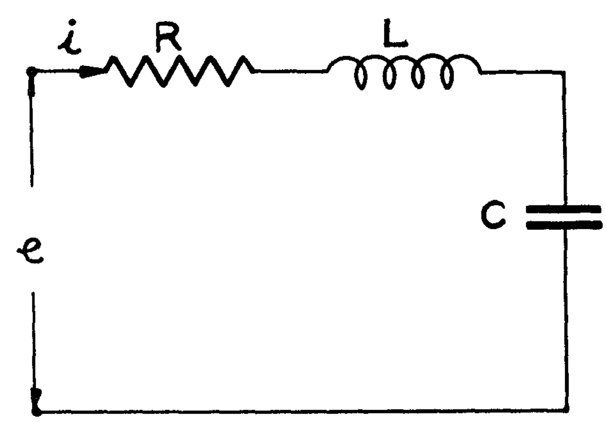
TABLE OF ANALOGOUS QUANTITIES

	MECHANICAL	ELECTRICAL	
m	MASS	INDUCTANCE	L
K	STIFFNESS	CAPACITANCE^{-1}	C^{-1}
D	DAMPING	RESISTANCE	R
x	DISPLACEMENT	CHARGE	q
v	VELOCITY	CURRENT	i
f	FORCE	VOLTAGE	e



$$f = m\frac{dv}{dt} + Dv + K\int v\,dt$$

MECHANICAL SYSTEM
(DIAGRAMMATIC)



$$e = L\frac{di}{dt} + Ri + \frac{1}{C}\int i\,dt$$

EQUIVALENT ELECTRICAL SYSTEM

FIG. 4. TABLE OF ANALOGOUS QUANTITIES.

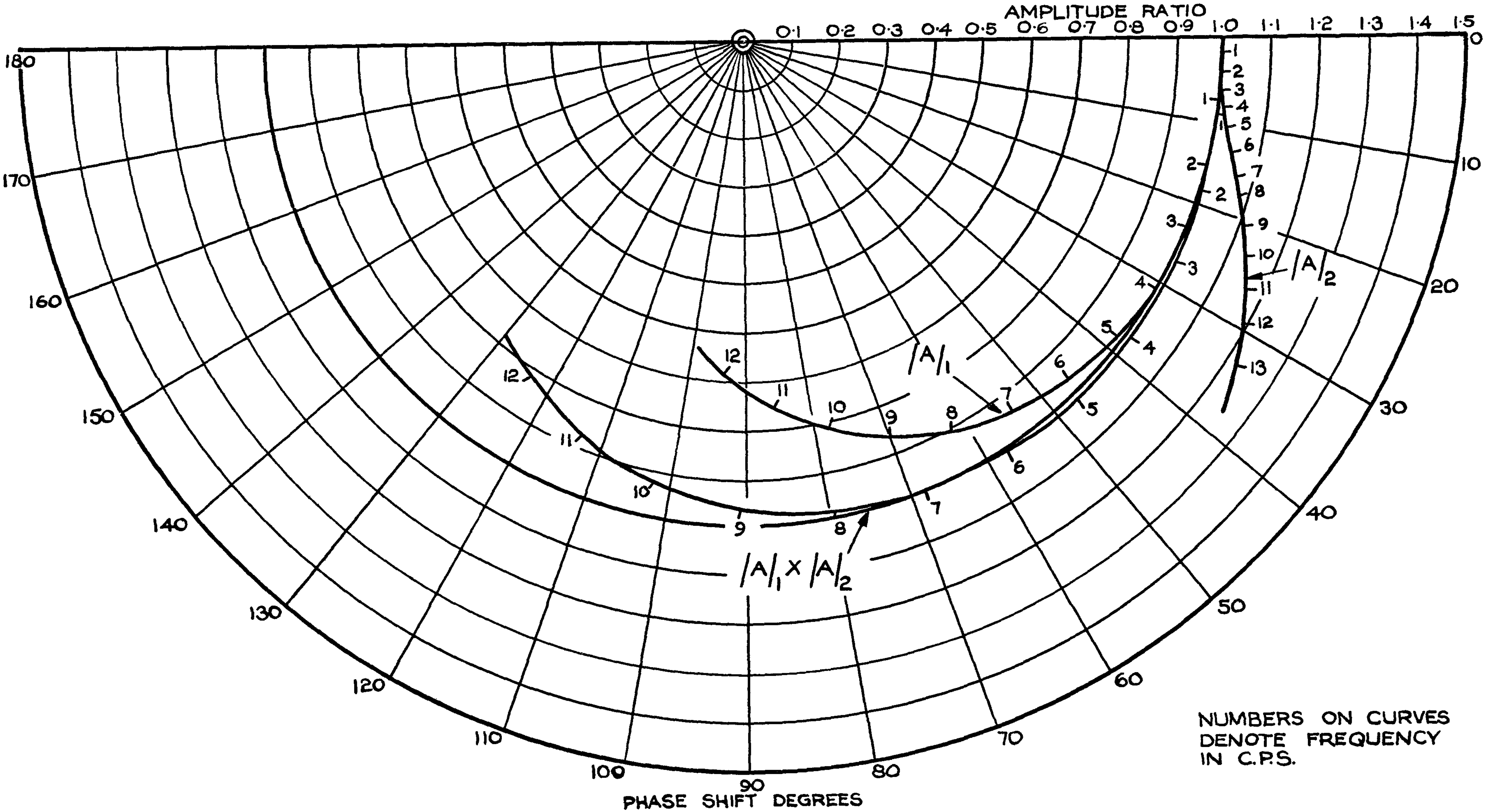


FIG. 5. CALCULATED FREQUENCY CHARACTERISTICS.

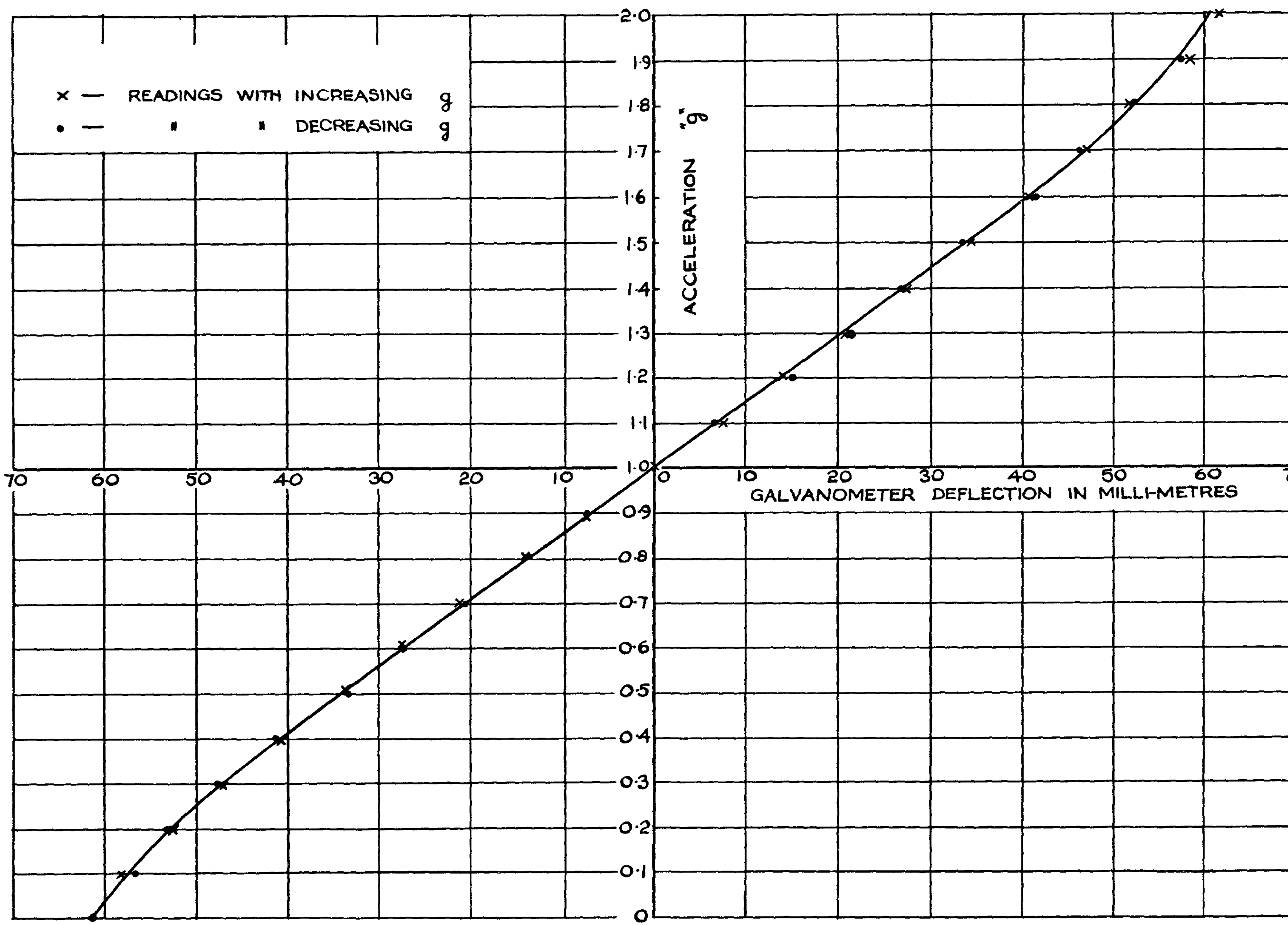


FIG. 6. STATIC CALIBRATION.

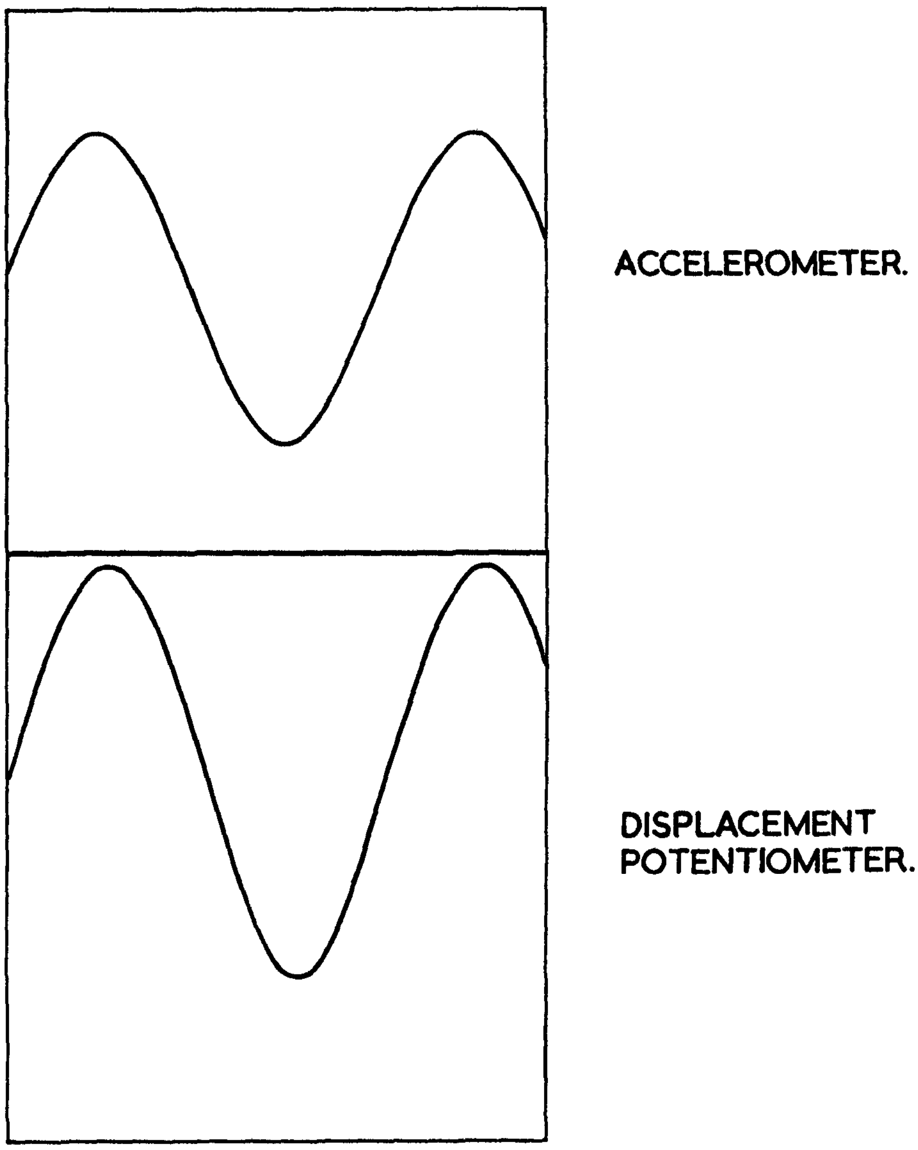


FIG. 7 SPECIMEN OF RECORDED TRACES
TAKEN DURING TESTS.

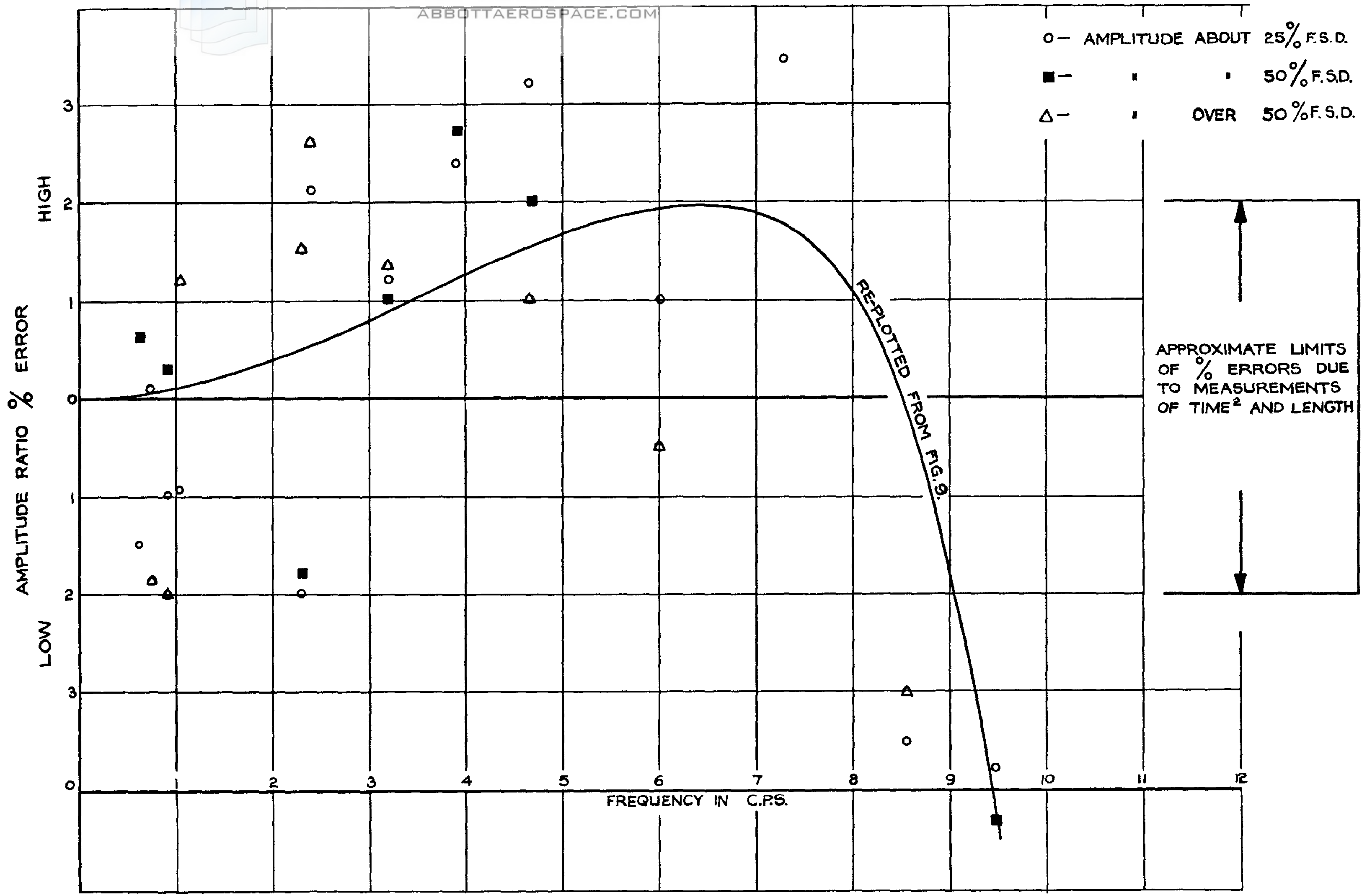


FIG. 8. AMPLITUDE RATIO EXPRESSED AS % ERROR AT VARIOUS AMPLITUDES.

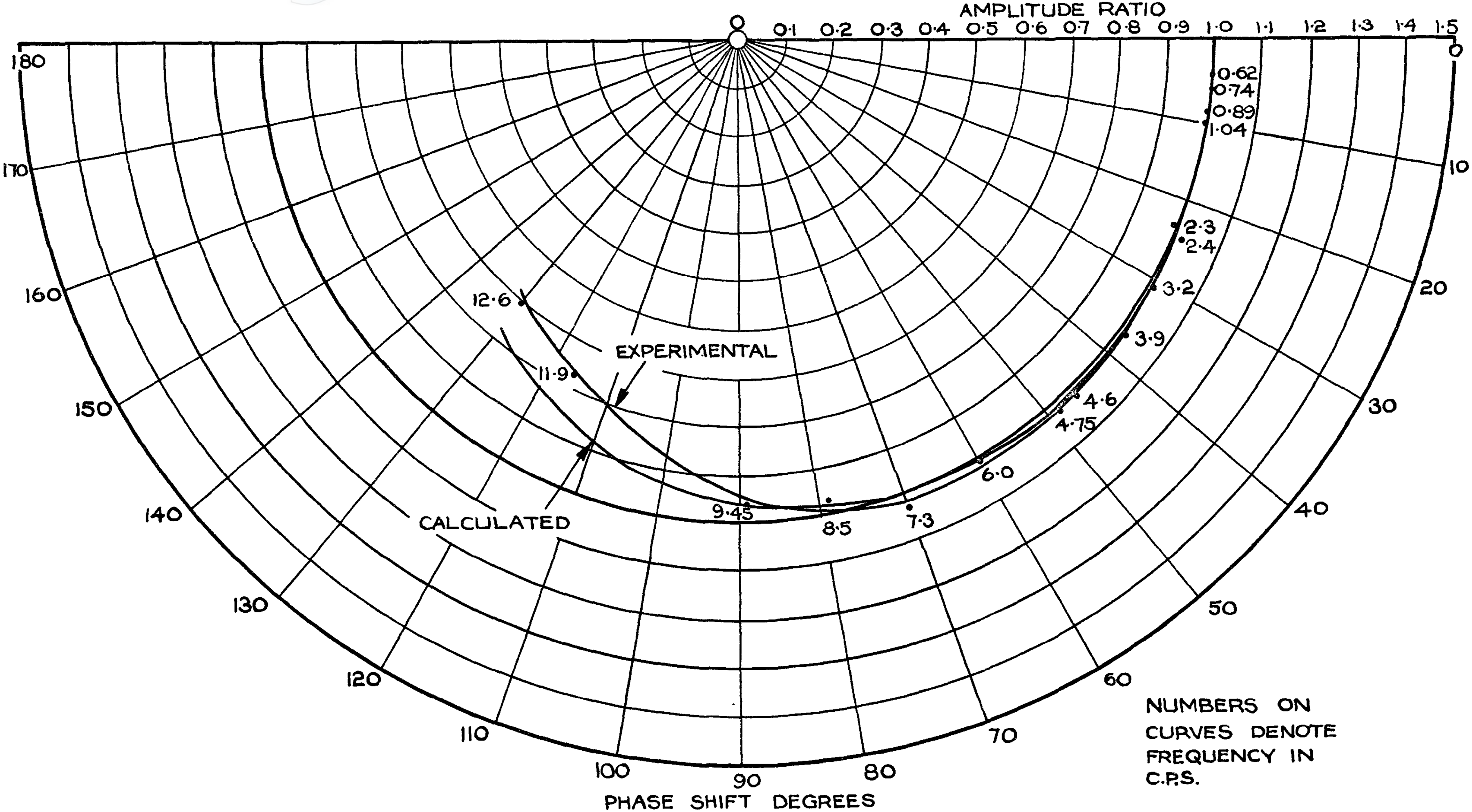


FIG. 9. FREQUENCY CHARACTERISTIC OF COMPLETE SYSTEM

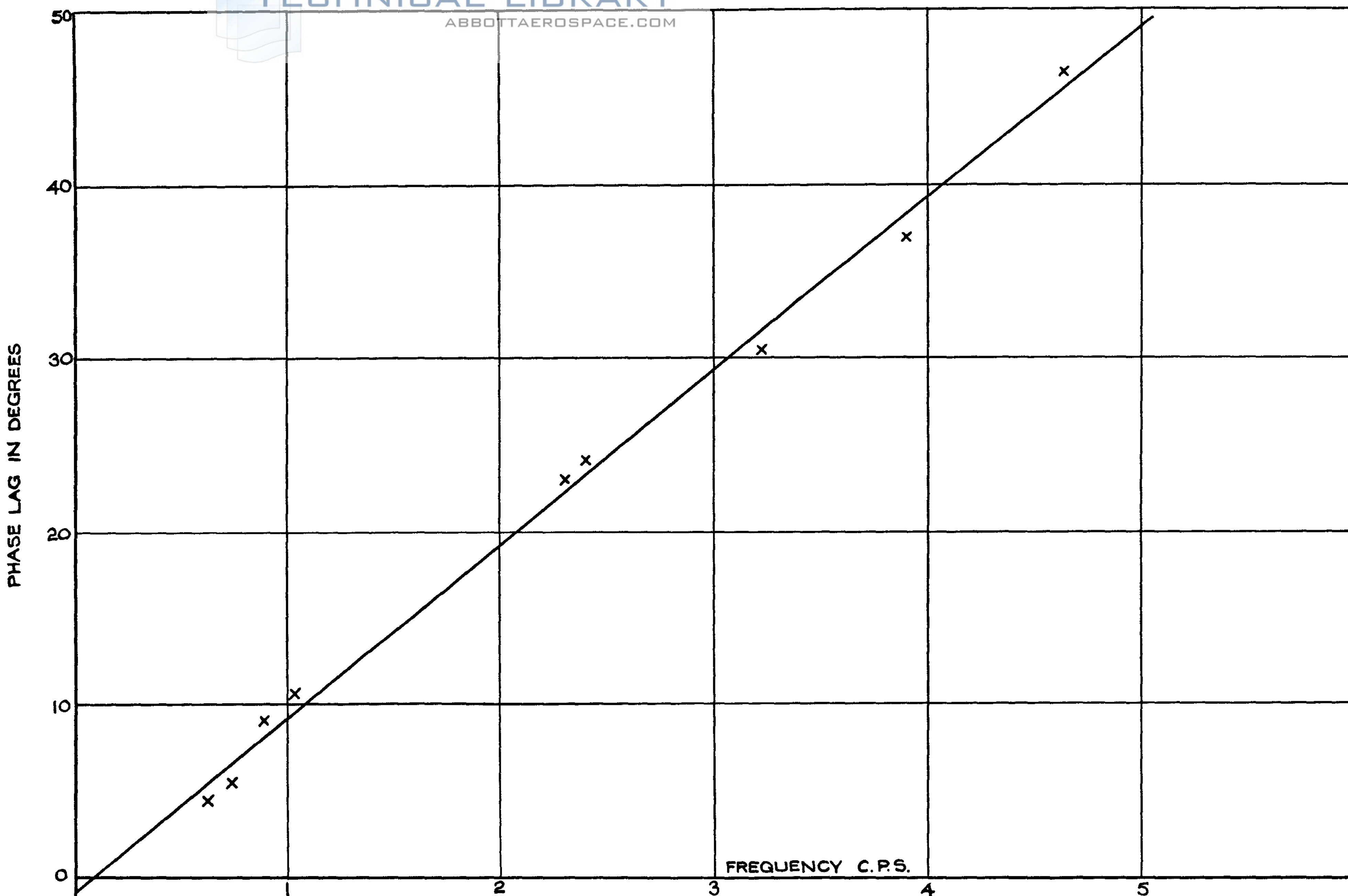


FIG. 10. A.C. ACCELEROMETER PHASE ANGLE V. FREQUENCY.

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