NRL Memorandum Report 1800

Experimental Technique for Determining Fixed-Base Natural Frequencies of Structures on Single Nonrigid Attachment Points

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NAVAL RESEARCH LABORATORY
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Experimental Technique for Determining Fixed-Base Natural Frequencies of Structures on Single Nonrigid Attachment Points

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ABSTRACT

Experimental verification of a technique for determining fixed-base natural frequencies of structures mounted on a single non-rigid support is provided. The feasibility of this technique of frequency determination was explored by measurements on individual double cantilever beams while each was attached to a frame designed to simulate characteristics of shipboard mounting conditions. The beams' fundamental frequencies were distributed throughout the frequency range normally considered in shock design and thus each beam simulated a different lightweight equipment-foundation system. Response was measured at the required locations with high sensitivity (acceleration) transducers and circuitry. Instrumentation normally employed for mechanical mobility measurements proved suitable in these vibration tests after modification to improve frequency stability and resolution. The scope of this experimental work is limited to the one-foundation problem. The two-foundation problem is expected to be the subject of future experimentation.

PROBLEM STATUS

This is an interim report on one phase of the problem; work is continuing.

AUTHORIZATION

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INTRODUCTION

Full scale shock tests of ships and submarines are conducted to study vulnerability to underwater explosion attack. Effects of shock at different levels of severity are measured by motion transducers and, if these transducers are located in appropriate places, the motions detected supply valuable information for improving the design of new equipment and systems by the Navy's Dynamic Design Analysis Method (DDAM) (1). Processing the measured shock response motions into design input values for the DDAM requires that two characteristics of the equipment-foundation systems in place during shock tests be known. These characteristics are the modal masses and the fixed-base natural frequencies (1). Since it is desirable to be able to measure these characteristics, their determination by experimental techniques has been under study. This report describes experiments which resulted in the successful measurement of the fixed-base natural frequencies of test structures while attached to a single non-rigid support.

The DDAM effects shock design by analyzing the contemplated equipment and its foundation as a base motion problem, solved by application of normal mode theory (2). With this formulation the equipment-foundation system is characterized by mode shape, modal mass, and mode frequency. Calculation of forces, deflections, stresses, etc. in each mode can be made when the base motion corresponding to a specified shock severity is applied to the base of the normal mode model. Identical results are obtained, however, if only the simple oscillator responses to the base motion at the normal mode frequencies are used with the model. Base motion data is measured during shock tests of existing ships and equipment. The processing of this shock test data into shock design values, i.e. simple oscillator peak responses, requires (a) that the normal mode frequencies of the tested equipment-foundation system be determined and (b) that the modal masses acting at these normal mode frequencies be determined. These specific input values are applicable when designing contemplated equipment with similar modal mass and normal mode frequency characteristics.

For the systems of concern, note that normal mode frequencies are those natural frequencies of maximum structural vibration that would exist if the equipment-foundation system under study had its base fixed to an infinitely massive and rigid object. Thus, normal mode frequencies as used in the DDAM are called fixed-base natural frequencies (FBNF).

One should not confuse FBNFs with the natural frequencies of maximum structural vibration that occur when the equipment-foundation system is mounted in-place on a ship (i.e. a non-rigid base). These are known as complete system frequencies and are rarely the same as the FBNFs.

Experimental measurements of FBNF are possible by conducting forced-vibration tests and measuring response at suitable locations. One means of determining these frequencies is to mount the subject equipment-foundation system on a sufficiently massive and rigid base, apply a force excitation, and measure the system's resonances. These resonances would correspond to the FBNFs or normal mode frequencies of the equipment-foundation system. Obviously the above approach is limited by the fact that almost all mounting locations on a ship are non-rigid; also, removal of the equipment-foundation systems to a more ideal fixed-base is usually impractical, if not impossible. Therefore this report explores the feasibility of a method which determines FBNFs of systems while they remain mounted in-place at relatively non-rigid locations aboard ship.

METHOD

From consideration of the pertinent theory, a technique for the determination of fixed base natural frequencies of equipment on a single non-rigid support has been developed by Petak and Kaplan (3). Its use is illustrated on the dynamic chain model shown in Fig. 1. The upper two masses and springs model an equipment-foundation system while the lower masses and springs represent the remainder of the ship, water, etc. The base is defined as the place where the equipment-foundation system is attached to the supporting structure of the ship.

The technique is this: Drive at a point on or below the base of of the equipment-foundation system with a sinusoidal force swept through a frequency range and record the response motions of the base and of a point on the equipment. Note the locations of prominent valleys in the response vs. frequency plot of the base motion; these valleys occur at the fixed base frequencies of both the system above the base, and the system below the driving point. Next, plot the ratio of equipment response to base response as a function of frequency. The fixed-base frequencies of interest are the frequencies at which prominent peaks appear in the ratio plot. These peaks should coincide with those valleys in the base response plot caused by the system above the base.

TEST STRUCTURES AND APPARATUS

Twelve double-cantilever beams were built so that each in turn would represent a different equipment-foundation system with several modes of vibration. The dimensions of these steel test structures were such that each beam weighed approximately 200 lb. while the calculated fundamental frequencies varied between 32 Hz and 949 Hz. Thus the frequency range usually considered in shock design was covered while each beam represented a "lightweight" shipboard structure, the type most likely to be mounted on a single foundation. The beam lengths varied from 28 1/4 inches to 92 inches while the thickness varied from 6 inches to 2 inches. All beams were 4 inches wide to allow for a standard mounting arrangement. Each beam was machined from rectangular bar stock and had bolt holes

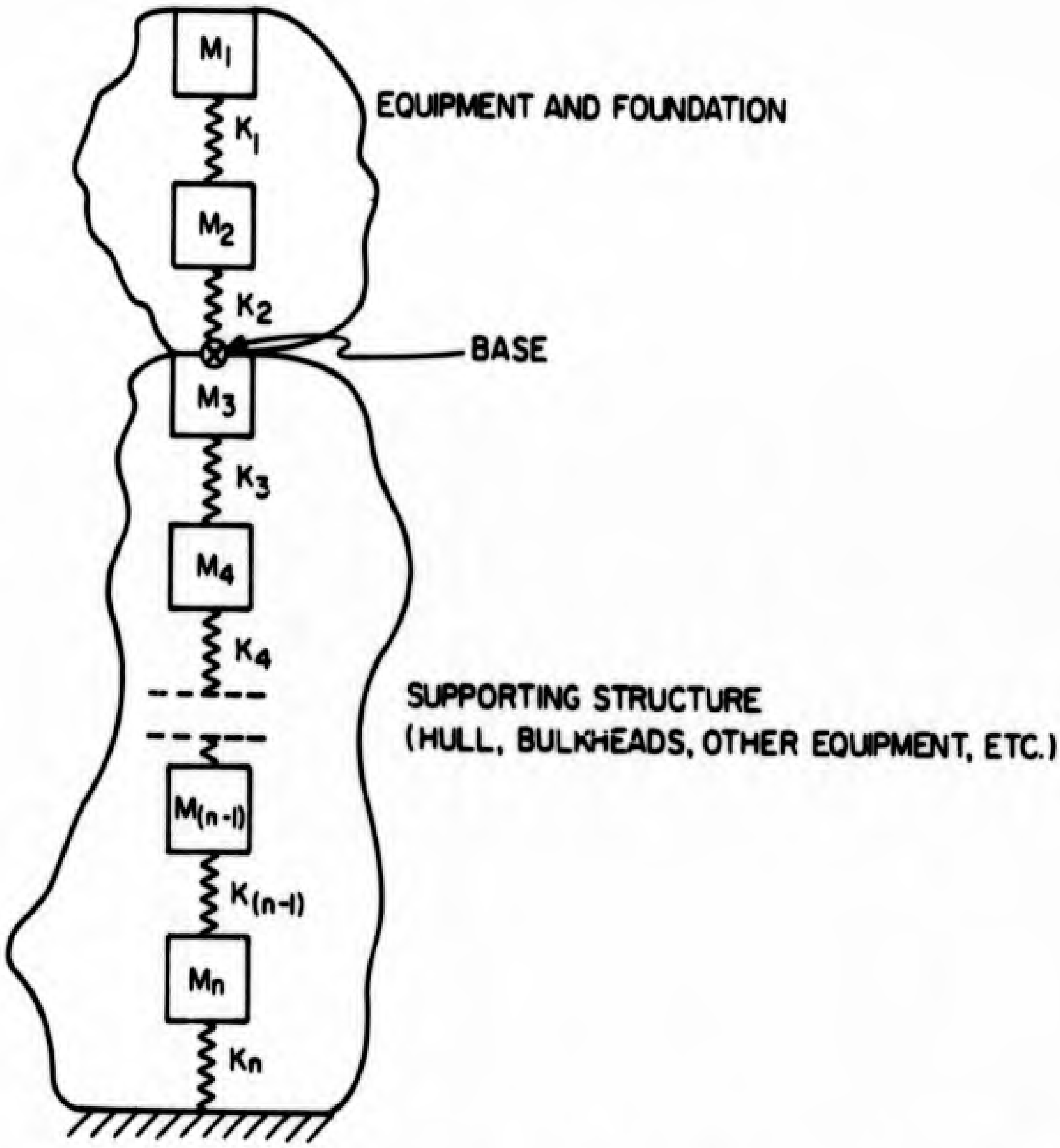

Fig. 1 - Representation of an equipment and foundation mounted on a ship

drilled in the center of the length and 4 inch width. Dimensions and weights of the beams, which are numbered 1 through 12, are given in Table 1.

Each beam in turn was attached to a mounting frame by means of clamping blocks, which also provided a convenient attachment point for an accelerometer that monitored the motion of the system at the center of the beam. The arrangement of the six 1/2-inch diameter bolts, beam, clamping blocks, and the pedestal on the mounting frame is shown in Fig. 2.

A flexible base for the beams and a simulation of other shipboard measurement conditions were provided by the mounting frame which is shown in Fig. 3. Note that because of the mounting frame's flexibility the system resonant frequencies of a beam-frame combination are different from the resonant frequencies of the same beam mounted on an infinitely massive and rigid pedestal. So with a beam attached to the frame the combination had several system resonant frequencies in the measurement range; some were close to the fixed-base natural frequencies being sought. The four legs of the mounting frame were arranged for easy access to the under side of the pedestal in order to install a shaker. Each pair of legs was attached to a pair of channels that were bolted to a concrete floor.

Sinusoidal force was applied to the under side of the pedestal (near the beam-frame connection) in a vertical direction with an electrodynamic shaker. A Wilcoxon Model F-4 shaker was used for nine of the beams, and a Goodmans Type 790 shaker was used for Beams 3, 4, and 5. The exciting force from either shaker was transmitted to the structure through a Wilcoxon Model 820 mobility head so the force waveform could be monitored by the head's force transducer.

Acceleration characteristics of motions at the cantilever tips were measured with an Endevco Model 2218 accelerometer, and acceleration characteristics of motions at the base points were measured with a Wilcoxon Model 720 accelerometer mounted on one of the clamping blocks. Only 3 oz. of weight was added by the tip accelerometer. The base accelerometer's balanced electronic output facilitated use of a low noise amplifier circuit for improved null sensing ability. Base motion was not measured with the accelerometer contained in the mobility head, partly because of local stiffness effects and partly because the head was not attached directly to the surface chosen as the base.

General requirements of the electronic apparatus were to generate sinusoidal current to drive the shaker over the required frequency range (10-5000 Hz,) detect wide variations in acceleration signal levels--especially low signal levels in the base acceleration where the nulls are significant, filter each signal through a narrow band-pass because closely-spaced peaks and dips existed in the system response, and compute the ratio of the two signal levels. It was desirable that all these

BEAM NO.	WEIGHT (LB)	TOTAL LENGTH (IN)	WIDTH (IN)	HEIGHT (IN)
1	209.75	92	4	2
2	207.5	77.5	4	2.375
3	206.5	66.25	4	2.75
4	207.5	61	4	3
5	209	56.75	4	3.25
6	202	49.5	4	3.625
7	199	44.25	4	4
8	198	39	4	4.5
9	197	37	4	4.75
10	196	35	4	5
11	195.5	31.5	4	5.5
12	190.5	28.25	4	6
Upper Clamping Block	29.75	9.5	4	3
Lower Clamping Block	29.75	9.5	4	3

Table 1

Nominal Beam and Clamping Block Specifications

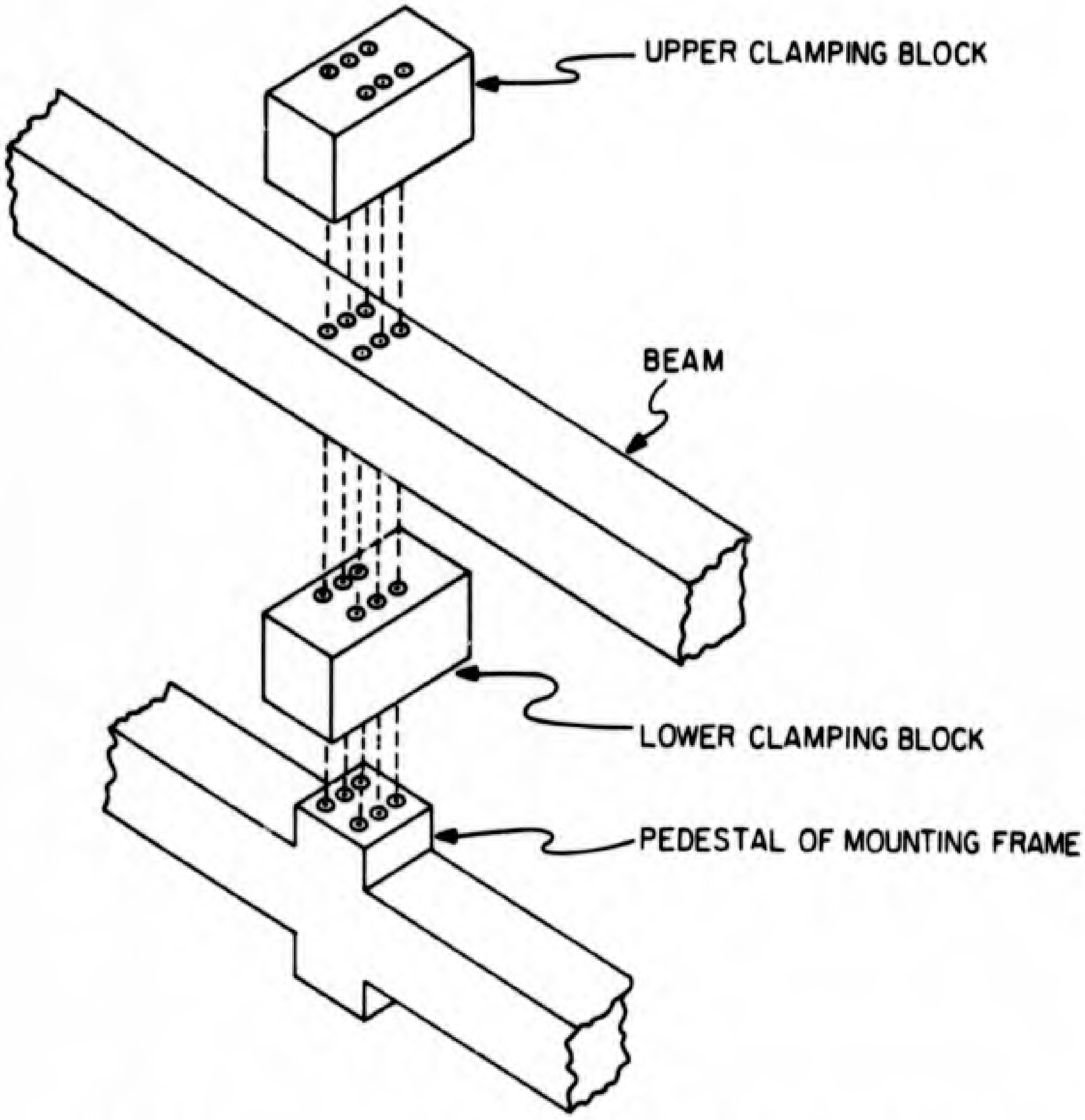

Fig. 2 - Isometric view of clamping blocks, beam, and pedstal

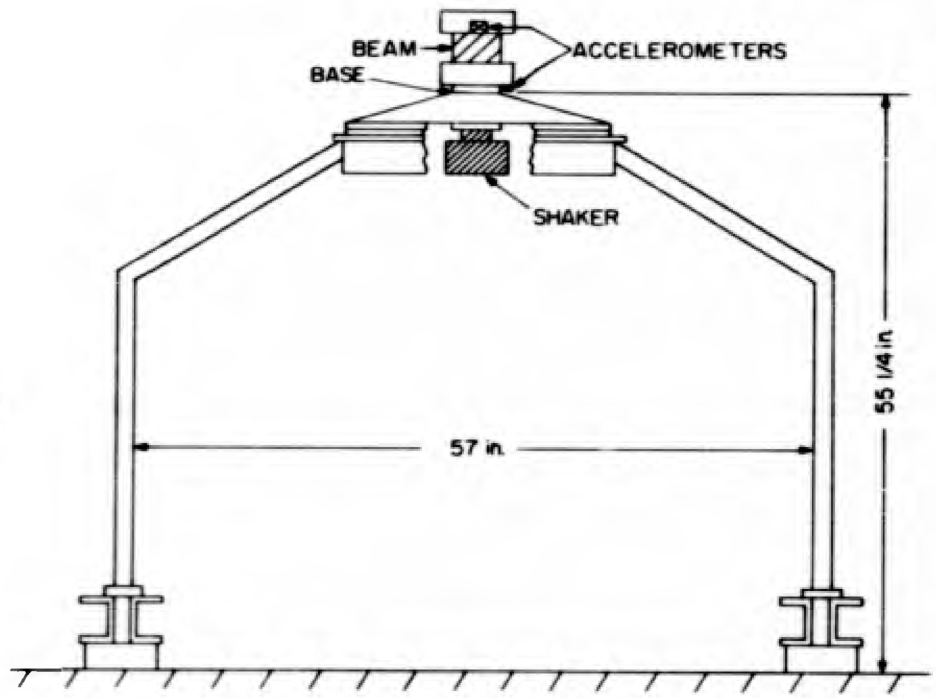

Fig. 3a - Mounting frame and beam side view

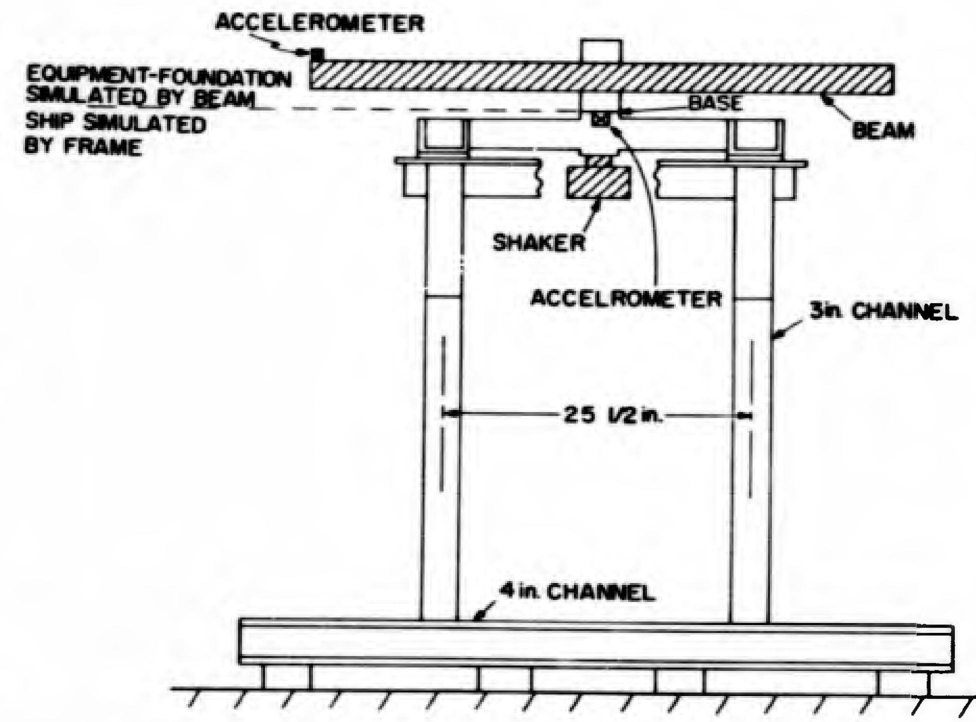

Fig. 3b - Mounting frame and beam front view functions be performed synchronously while under the control of a single local oscillator. Commercially available dual channel wave analyzing and recording systems made for mobility/impedance measurements meet most of these requirements.

In these experiments an early model Ad-Yu Type 1010 mechanical mobility recording system was used to perform the above mentioned electronic functions. The circuit block diagram is shown in Fig. 4. An essential feature of the basic recording system (not unique to the Ad-Yu) is the placement of a mixer and filter in the servo loop of each channel's recorder. Consider the E_1 acceleration signal. After E_1 passes through the preamplifier it is attenuated by the logarithmic potentiometer. Next the carrier mixer converts the E_1 signal from the shaker driving frequency to a 20.5 kHz signal carrying the same information. The mixer output is passed through a narrow-band (10Hz) crystal filter centered on 20.5 kHz. The filter output signal is fed into a detector circuit where its voltage is compared to a reference voltage. Any difference between the detected filter output voltage and the reference voltage actuates the servo which adjusts the potentiometer setting in a manner which eliminates this difference. Thus the adjusted pot setting is proportional to the magnitude of a chosen component of the accelerometer signal. This chosen component has a bandwidth of 10Hz centered on the shaker driving frequency. The E_2 acceleration signal is handled similarly. High quality filter performance is provided by this arrangement since the signal voltage at the crystal filter remains relatively constant even while the acceleration magnitude varies over a wide range. The mixer maintains a constant frequency at the filters, hence any change in filter characteristics with amplitude or frequency variations of the signal being passed are avoided. Thus accurate measurement of the E_1:E_2 ratio precisely at the shaker frequency can be made even in the presence of large amounts of noise on the acceleration signal.

Several modifications to the recording system were made to perform these tests. One noteworthy modification was the addition of a frequency synthesizer for use as an optional local oscillator for searching out the frequencies at the ratio peaks. It not only allowed the frequency to be advanced in steps of 0.1Hz, but its stable characteristics allowed a more accurate mixer balance adjustment, provided a more stable driving frequency, and more accurate tracking by the filters. Hence a better separation of nearby frequency components was achieved.

The original devices which produced a dc-voltage proportional to the logarithm of each signal magnitude were replaced with devices of NRL design. A summing circuit was added so the ratio of signal magnitudes could be plotted on the phase recorder. Thus with a digital voltmeter in parallel with the ratio magnitude plotter, the recording system used for obtaining the following results was capable of either continuous sweep or intermittent digital operation.

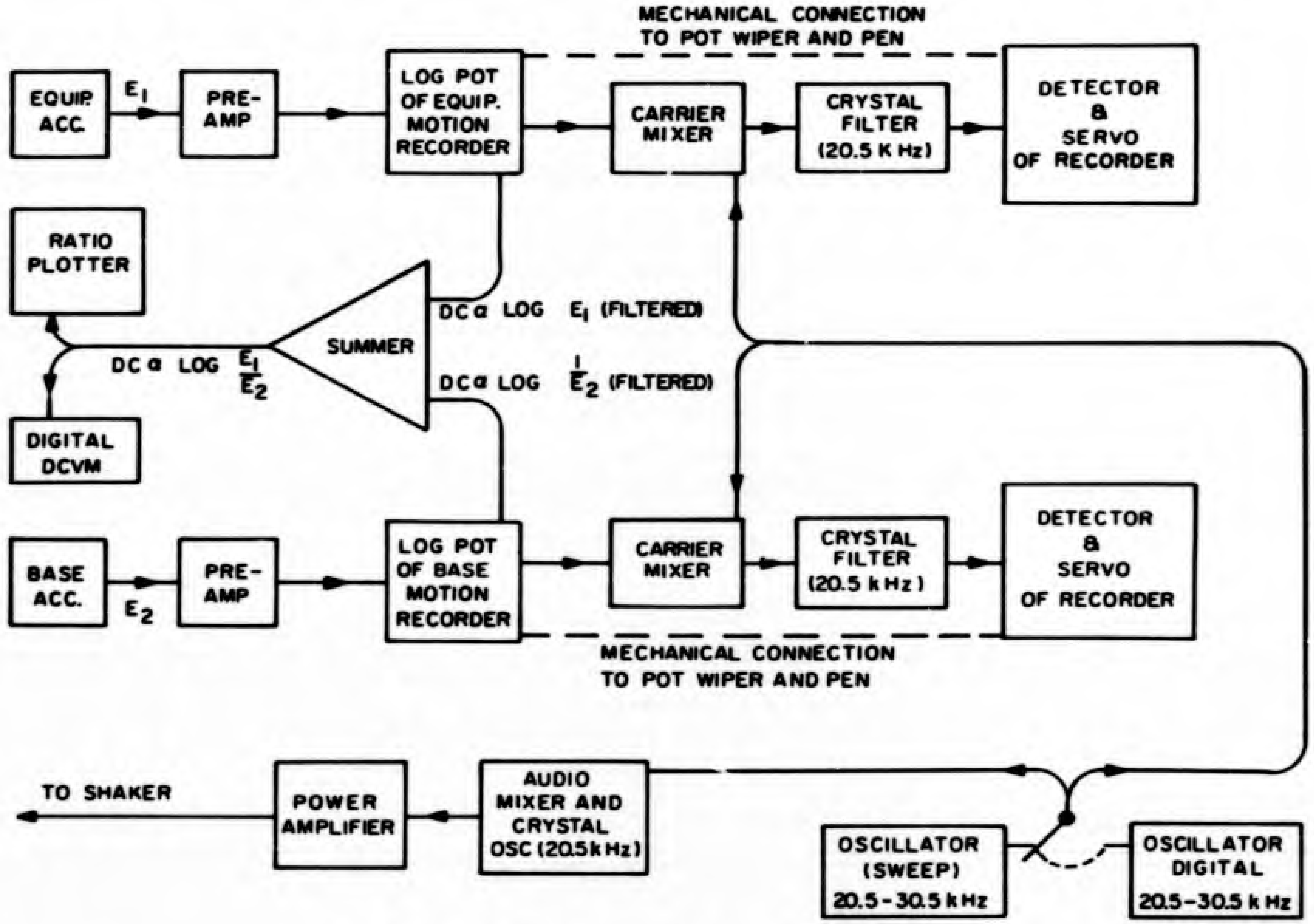

Fig. 4 - Block diagram of recording system

PROCEDURE

After aligning the beam and clamping blocks to form a symmetric double-cantilever configuration, the beam of interest was bolted to the mounting frame. The 1/2" diameter bolts were torqued to 40 lb-ft except for the middle bolt in each row of three. They were torqued to only 35 lb-ft because parts of their heads were cut away to make room for the mobility head. The equipment motion accelerometer was attached to the beam's tip by dental cement, and the base motion accelerometer was attached to the lower clamping block by a 3/8-16 stud. The mobility head was attached inside the array of clamping bolt heads on the underside of the frame member forming the beam's pedestal. The Wilcoxon F-4 shaker requires no separate suspension apart from the mobility head mounting stud. The Goodmans type 790 shaker required a separate suspension arrangement which was carefully aligned to properly mate the shaker to the mobility head.

Electronic components were adjusted during preliminary sweeps of the frequency spectrum. A plot of the ratio of tip motion to base motion was made simultaneously with plots of the individual motion levels during a logarithmic sweep from 20 Hz to 2000 Hz or higher. The fixed base natural frequencies indicated by peaks on the ratio plot were noted and checked against the base motion plot to assure that the ratio peaks were associated with base motion nulls. Next the frequency synthesizer was connected in place of the sweep oscillator and the ratio peaks were searched out by hand in 0.1 Hz steps while the magnitude changes were observed on a digital voltmeter. Finally the shaker and both accelerometers were affixed to a suspended solid weight and the frequency spectrum swept again. With both accelerometers responding to the same motion, a ratio of unity was generated and it appears on the ratio plot as a horizontal line.

RESULTS

Curves plotted from data for Beam 1 are the first example of the results of these fixed-base natural frequency experiments. Numerous peaks and dips with wide variations in amplitude characterize the separate acceleration responses measured at the tip and at the base locations. These curves are plotted on a log-log scale in Fig. 5. On first glance it is not obvious that a clearly defined ratio plot can result from the division of these spectra, however, the ratio plot in Fig. 6 clearly shows five peaks which indicate the first five fixed-base natural frequencies. Tip motion is nominally 100 times larger than the base motion at these ratio peaks. Other ratio peaks occur, for example, at 165 Hz and 350 Hz, but they are discounted because of being 10 times smaller in height and having little area under them. Note that the FBNFs indicated by the peaks in the ratio plot (Fig. 6) are different than the complete system frequencies indicated by the peaks in the tip acceleration plot (Fig. 5a). The zero db reference line or calibration line is

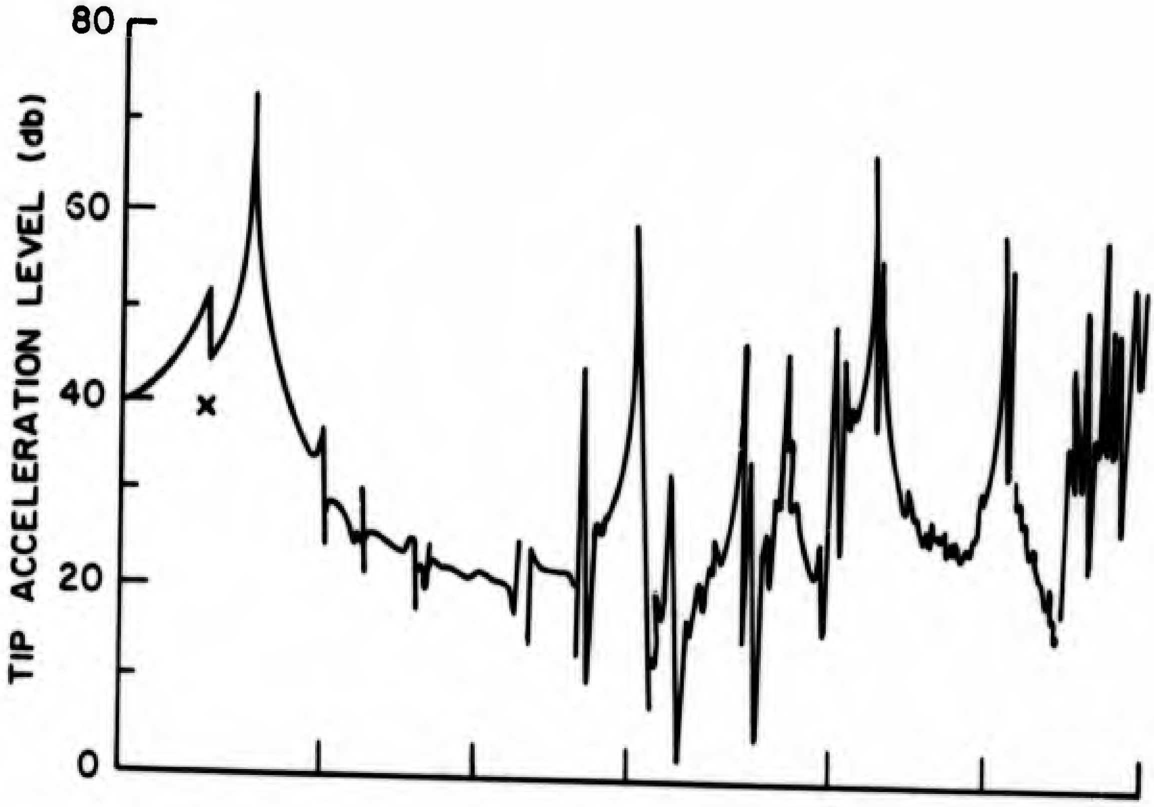

Fig. 5a - Tip motion for Beam 1 on the frame

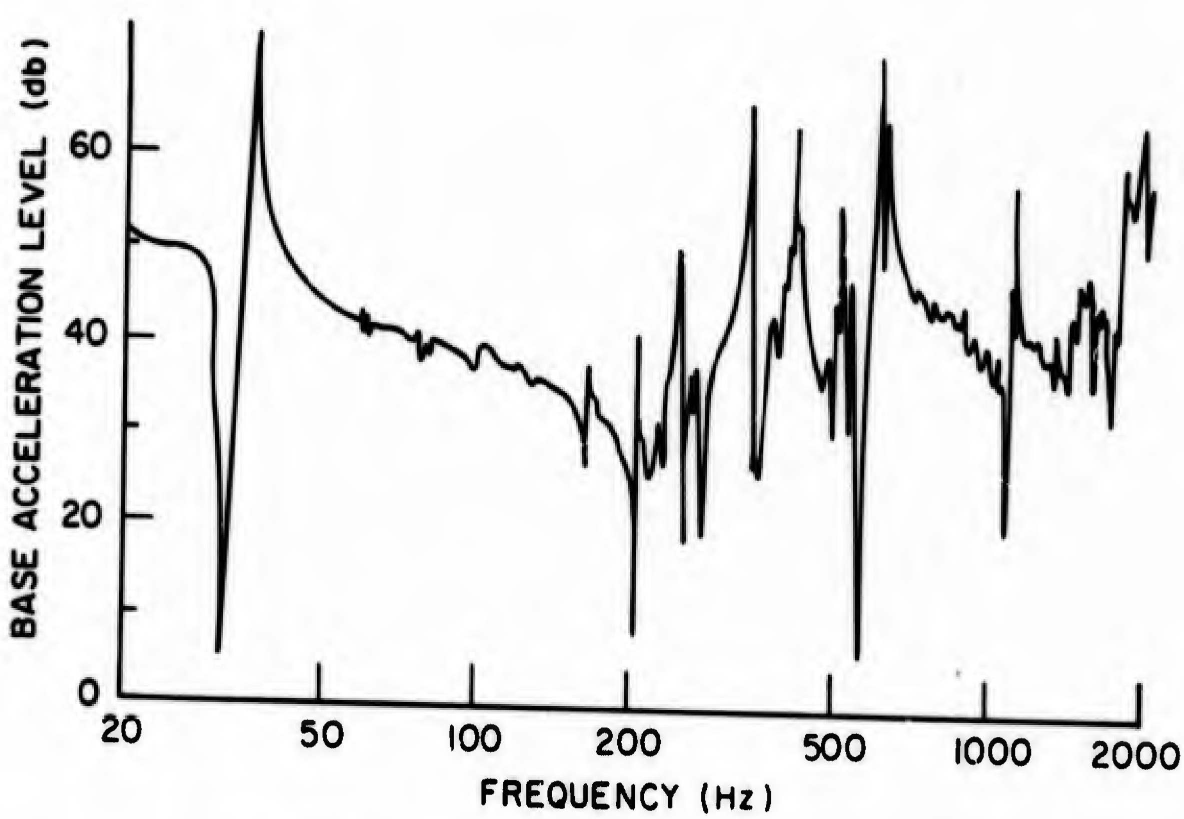

Fig. 5b - Base motion for Beam 1 on the frame

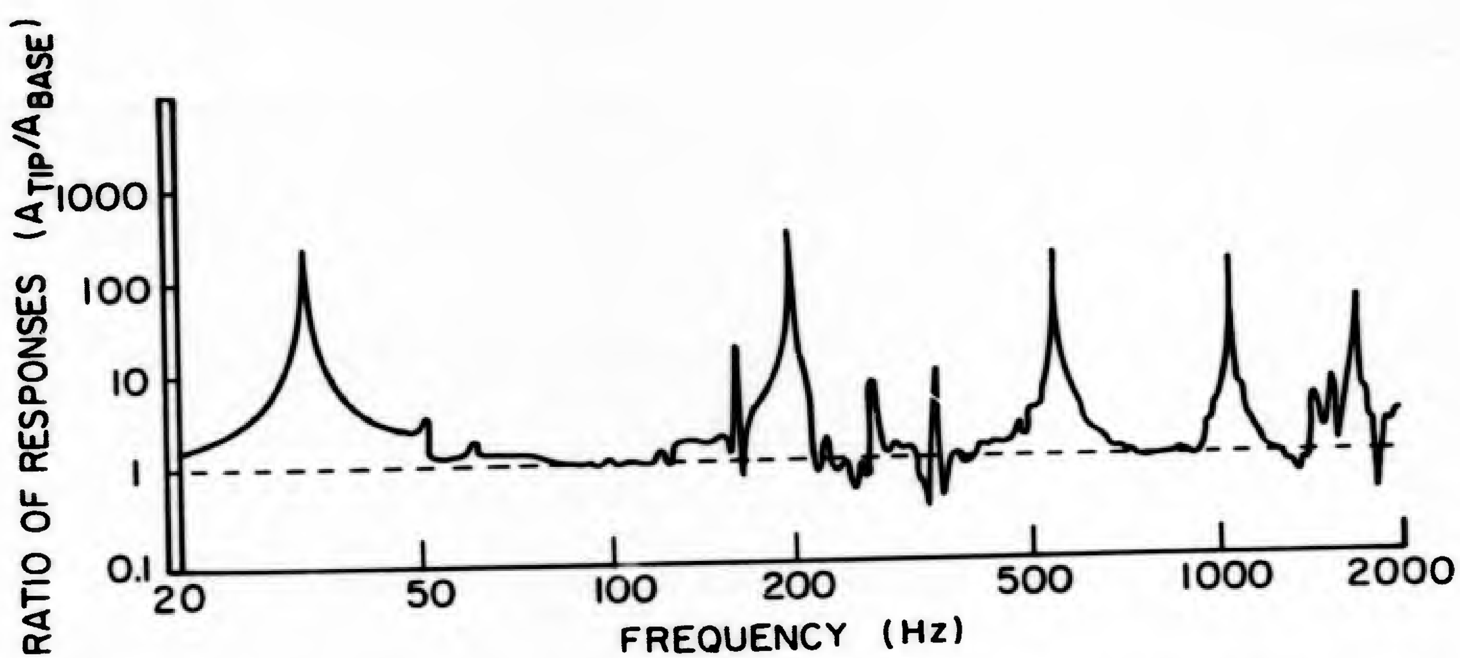

Fig. 6 - Ratio of tip motion to base motion for Beam 1 on the frame

the dashed line at the unity ratio level. The base motion plot (Fig. 5b) shows definite nulls at the frequencies of the ratio peaks, thus there is reassurance that the fixed base natural frequencies of the double cantilever portion of the system were measured. The relatively sharp downward change in tip acceleration level at the frequency marked by an X in Fig. 5a is due to a manual change in shaker current at that point of the frequency sweep. A less obvious but corresponding change is seen in the base motion level. The change was gradual, although it appears sharp because of the slow sweep rate. While Beam 1 ratio data is representative of the ratio plots for Beams 1-6, Beam 7 ratio data shown in Fig. 7 is representative of the ratio plots for Beams 7 - 12 where only one prominent peak appears below 2000 Hz. A definite first mode frequency can be seen at 291 Hz. The second mode peak is expected slightly above 1500 Hz but no definite approach to a single peak appears. The base accelerometer location was switched to the center of the top clamping block for beams 8 through 12 which brought out the second mode frequencies for beams 8 and 11. Apparently the protruding portion of the lower clamping block became less representative of the base motion for high modes as the beam thickness increased. Except for beam 7 this difficulty appeared above 2000 Hz which is well above the frequency range normally considered for shipboard shock design.

Table 2 lists the frequencies measured for each beam and compares them with calculated values for lower modes. Overall agreement is good. The calculations neglect damping and use classic beam theory with a correction factor for shear and rotary inertia effects (4); this correction factor was available for only the first three modes. The end conditions imposed by the clamping blocks on the beam were not considered to be those of a perfectly rigid clamp. The calculations account for this non-ideal end condition by using an effective length which is slightly longer than the cantilevered length. Calculated frequencies in Table 2 were based on the arbitrary assumption that the effective root of the cantilever was inside the clamped region a distance of 1/3 the total length of the clamping contact. The true effective root is expected to vary slightly from beam to beam as well as from mode to mode. The second mode frequency was strongly hinted in measurements for beams 7, 9 and 10 but it was not entirely clear. See Fig. 7 for an example. The frequencies in Table 2 were chosen without any reference to the calculated values, otherwise frequencies might have been chosen for the second modes marked by an asterisk in Table 2. The asterisk or "strongly hinted" designation was determined after comparing measured and calculated values

Plots for the measurements on other beams are not shown, but the observations were made for frequencies up to 3 kHz for beams 2, 3, and 8, up to 4 kHz for beams 4, 5, 9, and 10, and up to 5 kHz for beams 11 and 12.

Higher mode fixed base frequencies were detected from data obtained at frequencies up to the 3 kHz region. This was about as high as one could hope to go because at frequencies above 3 kHz the local effects in the

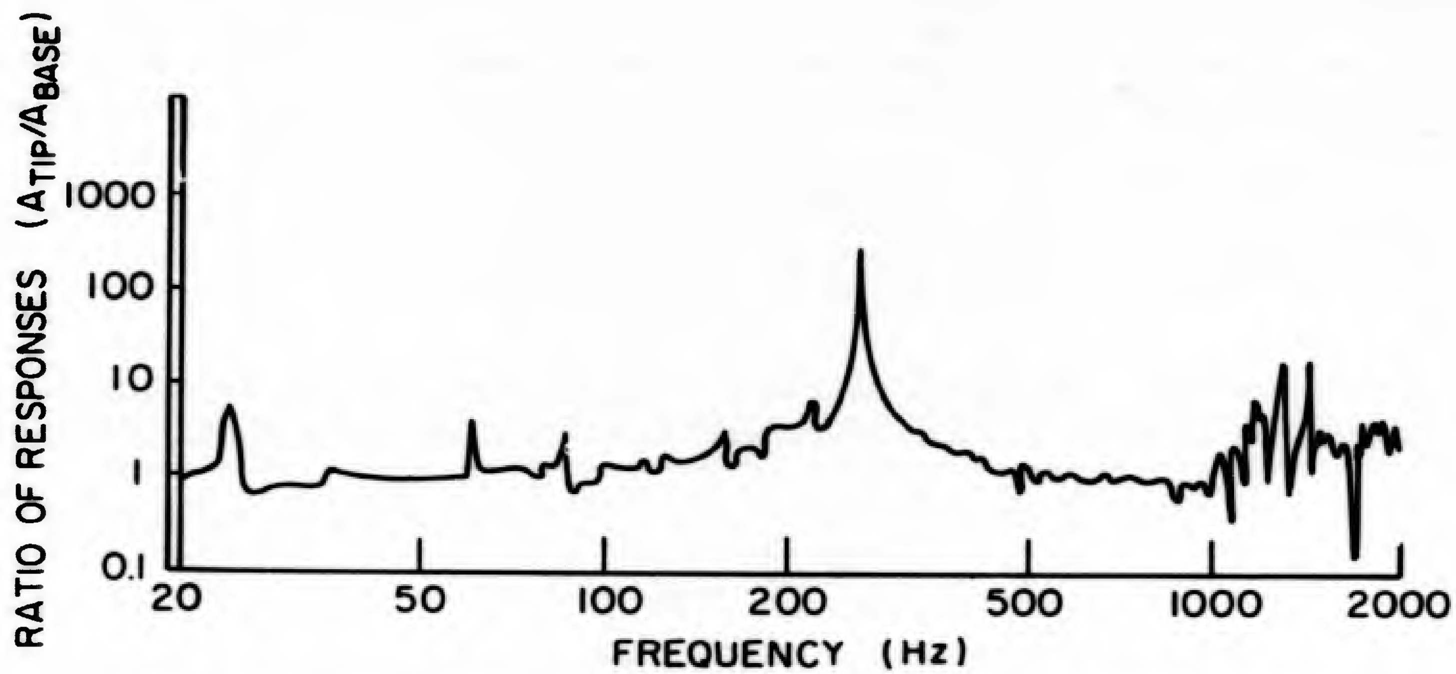

Fig. 7 - Ratio of tip motion to base motion for Beam 7 on the frame

clamping blocks became noticeable. They eventually dominated as the frequency was swept higher. The modal effective mass acting in the higher beam frequencies is a very small percentage of the total beam weight and thus these fixed-base natural frequencies became harder to detect and perhaps less significant. The lower limit of the useful frequency range was between 10 and 20 Hz because of shaker limitations and transducer sensitivity.

Changing the tip accelerometer location from one tip to the other tip of the beam made no detectable change in the frequencies measured. Ratio peaks are primarily caused by nulls in the base motion, so the base accelerometer location was the more sensitive. Three possible positions were provided for the accelerometer on the clamping blocks. Differences in fixed-base frequ ncies were small but detectable when the locations were changed for the longer beams, but the shorter beams required the use of the top center location as mentioned earlier.

Data for beams 3, 4, and 5 were measured twice; in each instance a different means of external support for the Goodmans shaker was used. The first time the usual soft suspension was employed while the second time the shaker was supported rigidly from the floor. This allowed the applied force to reach the system through the legs of the mounting frame as well as the driving point under the pedestal. Frequencies measured were the same in both cases indicating that the idea of applying a force either at or below the point chosen as the base is valid.

General features of the ratio plots are the sharp peaks and broad flat valleys where the ratio rarely passes below unity. These two characteristics appear because the "equipment" accelerometer was on tip of the beam, (end mass of the dynamic chain). See Table A1, page 15, Ref. 3. If there had been portions of the system above the equipment accelerometer the ratio plot would have contained sharp dips with ratio magnitudes much less than unity. These sharp dips would have frequencies corresponding to the fixed-base frequencies of the part of the system above the "equipment" accelerometer and to the part of the system below the base accelerometer. Hence, when dips do appear in the ratio plot of an actual equipment one knows that the "equipment" accelerometer is not on the end mass (or at the tip of the beam in this case).

CONCLUDING REMARKS

The foregoing results provide experimental verification of the technique of FBNF determination proposed by Petak and Kaplan. The feasibility of measurements on equipment-foundation systems while mounted on a flexible support structure is demonstrated with instrumentation of the mechanical mobility measuring type. But actual shipboard measurements must be attempted before one can be entirely sure this technique is practical for such purpose. In these experiments accurate measurements were made

Beam	1st Frequency		2nd Frequency		3rd Frequency		4th Frequency		5th Frequency	
	Meas.	Calc.	Meas.	Calc.	Meas.	Calc.	Meas.	Calc.	Meas.	Calc
1	32.1	31.5	199.4	195	552.2	541	1062.9	-	1710	-
2	52.4	52.8	327.0	326	904.0	889	1700.0	-	2686	-
3	84.3	84.0	517.7	512	1380	1349	2555	-	-	-
4	107.6	108	651	651	1726	1687	3091	-	-	-
5	134.8	135	805	802	2103	2046	-	-	-	-
6	194.2	199	1132	1133	-	-	-	-	-	-
7	261.2	274	*	1504	-	-	-	-	-	-
8	389.6	392	2139	2024	-	-	-	-	-	-
9	449	457	*	2278	-	-	-	-	-	-
10	529	533	*	2556	-	-	-	-	-	-
11	692.0	712	3288	3127	-	-	-	-	-	-
12	932.0	949	-	3756	-	-	-	-	-	-

*Strongly hinted (see text)

TABLE 2

Measured and Calculated Fixed-Base Natural Frequencies throughout the frequency range usually considered in shock design with simulated equipment-foundation systems mounted on a laboratory fixture which represented field measurement conditions. The mounting frame caused the base point to exhibit complicated modes of vibration which imposed difficulties similar to those expected during field tests. Even though these measurements were made in the vertical direction, the method and techniques apply to the other orthogonal directions as well. However, attaching the shaker to the base of shipboard equipment will probably be more difficult than attaching it to the underside of the frame member forming the base pedestal for the beams used in these experiments.

The primary importance of FBNF measurements is their use in the development of new shock design values for use as inputs during the design of contemplated equipment. Design input data presently in use (5) was developed while using indirect measurements or calculations of FBNF. Adequacy of the present data has been substantiated by tested designs (6) as well as a separate check of input values by another laboratory (7), but new ship types and new applications of the DDAM will require additional input data. The method of FBNF measurement described herein is intended to permit efficient and accurate determination of frequencies needed to develop these new inputs. Furthermore, direct measurement of FBNF adds strength to the DDAM by making it possible to verify assumptions and to evaluate the importance of other factors and thus guide efforts to further develop shock design theory.

The scope of this experimental work was limited to the one-support or one-foundation problem. The two-foundation problem is dealt with theoretically by Petak and O'Hara in Ref. (8), where they develop a procedure for measuring the FBNFs of in-place two-support structures capable of rotation and translation while excited in one translational direction.

This experimental work, performed on structures attached to a single support, has relevance to a large number of Navy shock problems because this technique can lead to broader applications of the DDAM. Moreover, this experiment's success encourages experiments in measuring the FBNFs of in-place structures on two supports.

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13. ABSTRACT

Experimental verification of a technique for determining fixed-base natural frequencies of structures mounted on a single non-rigid support is provided. The feasibility of this technique of frequency determination was explored by measurements on individual double cantilever beams while each was attached to a frame designed to simulate characteristics of shipboard mounting conditions. The beams' fundamental frequencies were distributed throughout the frequency range normally considered in shock design and thus each beam simulated a different lightweight equipment-foundation system. Response was measured at the required locations with high sensitivity (acceleration) transducers and circuitry. Instrumentation normally employed for mechanical mobility measurements proved suitable in these vibration tests after modification to improve frequency stability and resolution. The scope of this experimental work is limited to the one-foundation problem. The two-foundation problem is expected to be the subject of future experimentation.

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