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Resonance Testing in the Determination of Fixed Base Natural Frequencies of Shipboard Equipment

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ABSTRACT

Examination shows that the results of past shake tests to determine the fixed base natural frequencies of in-place shipboard equipment-foundation systems are not correct. A review of the equations of motion for a five-degree-of-freedom system revealed that choosing the driving point on the equipment could not give the desired frequencies. The review showed that the fixed base frequencies could be obtained if the driving point were on or below the base of the equipment-foundation system. Prominent valleys occur in the response of the base at the fixed base frequencies of the system above the base and of the supporting structure below the driving point. Comparison of these valleys with prominent peaks in a plot of the ratio of equipment response to base response will eliminate the extraneous frequencies of the structure below the driving point. Although this method functioned well when applied to systems simulated with an analog computer, it will be difficult in the field to place a shaker having sufficient force below the equipment. A situation involving rotation as well as translation magnifies the difficulties, since additional shakers are necessary which are capable of phase control and a range of force ratios at each frequency.

PROBLEM STATUS

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

AUTHORIZATION

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RESONANCE TESTING IN THE DETERMINATION OF FIXED BASE NATURAL FREQUENCIES OF SHIPBOARD EQUIPMENT

INTRODUCTION

A shock design-analysis method developed at the U.S. Naval Research Laboratory (1) for shipboard equipment requires the use of design shock spectra in the calculation of stresses and deflections of a contemplated equipment for which no measured shock spectrum exists. These shock design curves are developed either by an inverse application of the design-analysis method itself (using measured responses obtained from realistic shock tests on in-place equipment-foundation systems) or from actual shock spectra obtained from realistic shock tests. Either method requires that the fixed base natural frequencies* of the in-place system be known along with information on mode shapes and modal masses. These frequencies can be determined experimentally or by calculations. Calculations using normal mode theory can become cumbersome for complicated structures. A convenient experimental method which can be implemented directly upon the in-place system seems desirable.

At present, the method in popular use is that of shake tests. This involves exciting the structure with a sinusoidal force at various frequencies and recording the response at significant locations. If the structure were attached to a fixed base, peaks due to resonance would occur in the responses at the fixed base natural frequencies of the structure. Shipboard equipment, however, is not attached to a fixed base; its base is more in fact a boundary between the equipment-foundation system and a supporting structure. The peaks which occur in the responses of shipboard equipment are at the natural frequencies of the complete system (equipment-foundation, base, and supporting structure). The ratio of the amplitudes of equipment motion and base motion becomes the quantity which must be considered in determining the frequencies of the shipboard system.

The techniques used in conducting shake tests have not given correct frequencies. A study of the theory of resonance testing was made to determine why the results were incorrect and what method should be used to obtain correct results. An analog computer study was then performed to illustrate that the previous method was incorrect and to prove that the proposed new method would provide the correct frequencies. A five-mass system with conveniently chosen parameters (Fig. 1) was simulated on the computer. Uniform viscous damping was introduced into the model in order to assure a stable computer solution. Since the magnitude of this type of damping increases with frequency, the higher frequency motions were suppressed. However, the lower modes were prominent enough to sufficiently illustrate the problem. The new technique was also applied to several simulated special cases, where subsystems had matched frequencies, to determine its validity in those circumstances.

* The frequencies of the system as if it were mounted on a base of infinite mass and stiffness.

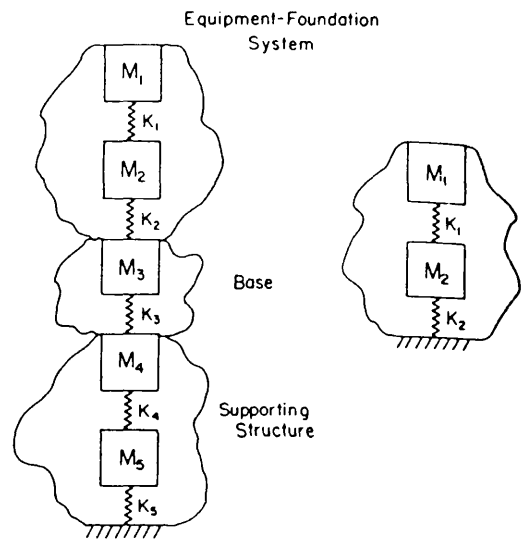


Fig. 1 - Simulation of an equipment-foundation system not mounted on a fixed base

PAST METHOD

Previous techniques used during shake tests on ships were largely based on expediency. The massiveness of the equipment system to be measured influenced the placement of the driving point. The crowded shipboard conditions and limited handling techniques placed a severe restriction on the size of shaker and, hence, the available force which could be applied. A driving point was chosen on the equipment such that significant response could be obtained. With the exciting force placed at a convenient point on the equipment, the responses of the driving point and base were recorded. An attempt was then made to select the desired frequencies by noting prominent peaks in a plot of the ratio of equipment driving point motion to base motion.

The frequencies experimentally chosen by such attempts have not agreed with calculated frequencies, in some cases being too absurd to be even a reasonable estimate. It is now realized that this method of conducting shake tests and analyzing the data cannot give correct results. The past method was applied to the analog simulated system (Fig. 1) to illustrate its error.

The five-mass system was driven at M_1, and the motions of M_1 and M_3 recorded. The ratio of the driving point motion (M_1) to base motion (M_3) is shown in Fig. 2. Peaks are expected at the two fixed base natural frequencies of the simulated equipment-foundation system (Fig. 1). A prominent peak occurs at 58 cps; it is difficult to select the second frequency. (The ratio increases at higher frequencies; this happens naturally since M_3, at a distance from the driving force, has its motion suppressed by damping while the driven mass still has some measurable motion.) By this method, then, the first fixed base frequency of the equipment-foundation system is 58 cps. The calculated frequencies of the two-mass system in the simulated problem are 37.1 cps and 107.1 cps. There is an error of 21 cps in choosing the first frequency by the previous technique.

An inspection of the equations of motion for a linear, undamped, five-degree-of-freedom system was made to explain the discrepancy. The study is outlined and results summarized in Appendix A. The review revealed that if the five-mass system is excited at the top mass, M_3 nulls at the fixed base frequencies of the two-mass system below itself. (The two-mass system is fixed at both ends: M_3 and the base of the five-mass system.) The plotted ratio of the response of M_1 to the response of M_3 will show peaks at the frequencies of this two-mass system. Frequencies of 58.7 cps and 113.0 cps were

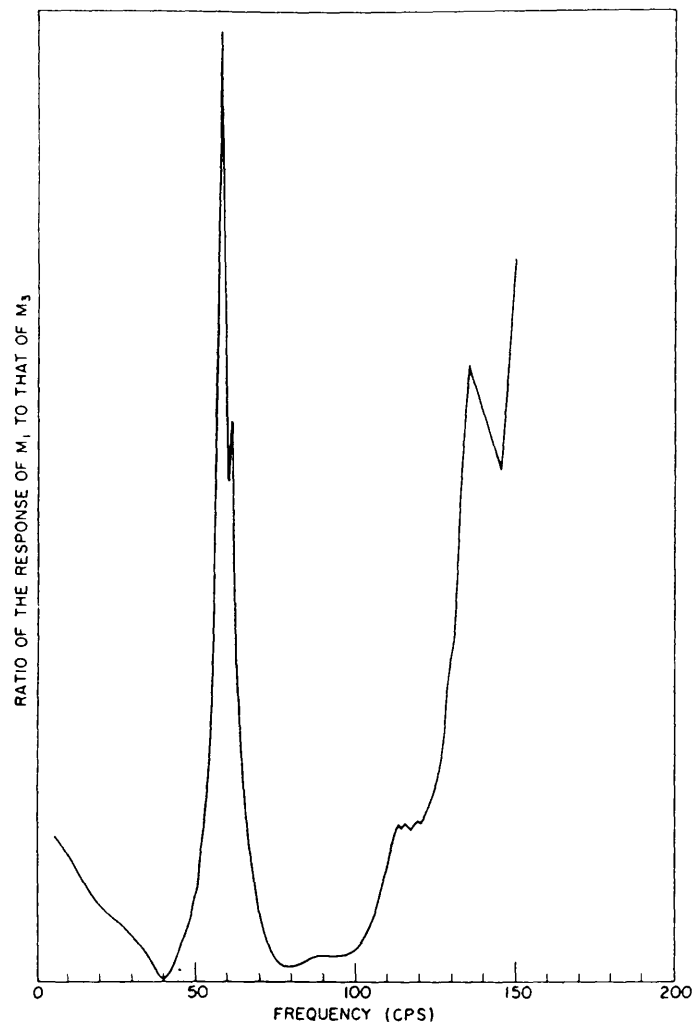


Fig. 2 - The ratio of the response of M_1 to that of M_3 when the driving force is at M_1

calculated for this system in the simulated problem. The former frequency compares favorably with that yielded by the ratio plot, which had been thought to be a fixed base frequency of the equipment-foundation system. The equation study and computer example showed that the previous shake test techniques can only reveal the fixed base natural frequencies of a system below the base of the equipment-foundation system of interest.

PROPOSED METHOD

Table A1 in Appendix A shows that if a mass of a dynamic chain is driven with a sinusoidal force and response recorded at any mass, the response will approach zero at the fixed base frequencies of the system on the side of the response mass distant from the driven mass, and at the fixed base frequencies of the system on the side of the driven mass distant from the response mass. For example, when driving at M_3 of the five-mass system, the response at M_2 nulls at the frequency of the one-mass system above M_2 as at the left in Fig. 3 and at the frequencies of the two-mass system below mass three as at the right in Fig. 3; the response of M_4 nulls at the frequency of the one-mass system

Fig. 3 - Systems at whose frequencies minima occur in the response of M_2 when the exciting force is at M_3

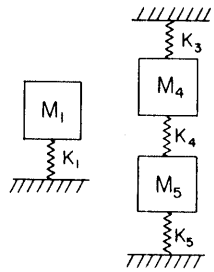
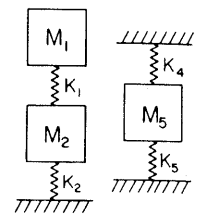


Fig. 4 - Systems at whose frequencies minima occur in the response of M_4 when the exciting force is at M_3



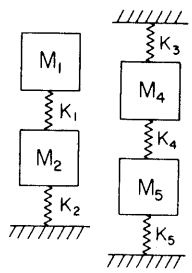
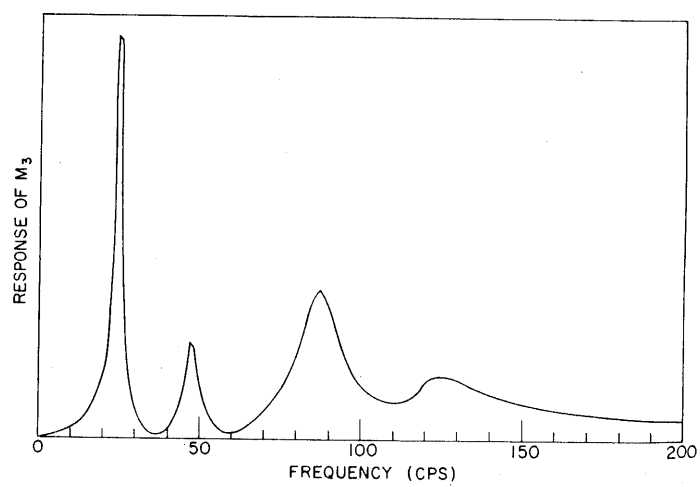


Fig. 5 - Systems at whose frequencies minima occur in the response of M_3 when the exciting force is at M_3

Fig. 6a - The response of M_3 to the driving force at M_3



below M_4 as at the right in Fig. 4 and at the frequencies of the two-mass system above M_3 as at the left in Fig. 4. If the response of the driven mass is recorded, it will approach zero at the fixed base frequencies of the systems on either side of the mass. For example, when driving at M_3 of the five-mass system, the response of M_3 nulls at the frequencies of the two-mass system above M_3 as at the left in Fig. 5 and at the frequencies of the two-mass system below M_3 as at the right in Fig. 5.

These results suggest a scheme for obtaining the fixed base frequencies of an in-place equipment-foundation system. The system is excited at its base, or some point below the base. Then the record of base response will exhibit valleys at the fixed base frequencies of the equipment-foundation system above the base and at the fixed base frequencies of the structural system below the driving point. The recorded response of a point on the equipment-foundation system will display valleys at the extraneous frequencies of the system below the driving point and appreciable response at the fixed base

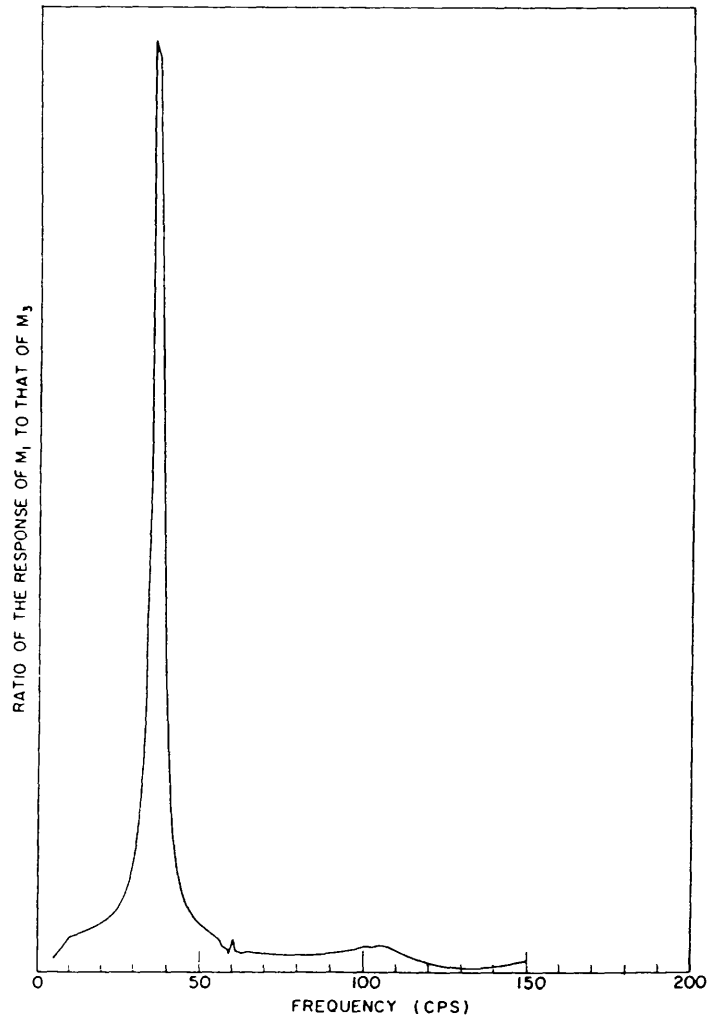


Fig. 6b - The ratio of the response of M_1 to that of M_3 when the driving force is at M_3

frequencies of the equipment-foundation system. Therefore, a ratio plot of equipment motion to base motion will cause the fixed base frequencies of interest to show as dominant peaks, and will eliminate the extraneous frequencies.

This proposed technique was applied to the analog simulated problem (Fig. 1). The system was driven at M_3 and the motions of M_1, M_2, and M_3 recorded. The response of M_3 is shown in Fig. 6a. Valleys occur at 37 cps, 60 cps, and 110 cps; the motion at higher frequencies was suppressed by damping. The ratio of the response of M_1 to the response of M_3 is given in Fig. 6b and the ratio of the response of M_2 to the response of M_3 in Fig. 6c. Prominent peaks occur in Figs. 6b and 6c at 36 cps and in the region of 105 cps to 107 cps. Since these peaks coincide approximately with the valleys of Fig. 6a at 37 cps and 110 cps, frequencies of 36 cps and 107 cps were chosen as fixed base natural frequencies of the simulated equipment-foundation system (Fig. 1). The calculated frequencies for this system, as stated before, are 37.1 cps and 107.1 cps. The new method consisting of matching the valleys in the base response with peaks in the ratio of equipment-foundation response to base response when driving at or below the base, resulted in selection of correct frequencies.

To show that the frequencies can be determined by driving at a point below the base of the equipment-foundation system the five-mass system was excited at M_4. Figure 7a shows the response of M_3 when shaking at M_4, and Figs. 7b and 7c the ratios of the response of M_1 and M_2 to that of M_3 respectively. Once again frequencies about 37 cps and 107 cps would be chosen as fixed base frequencies by this method.

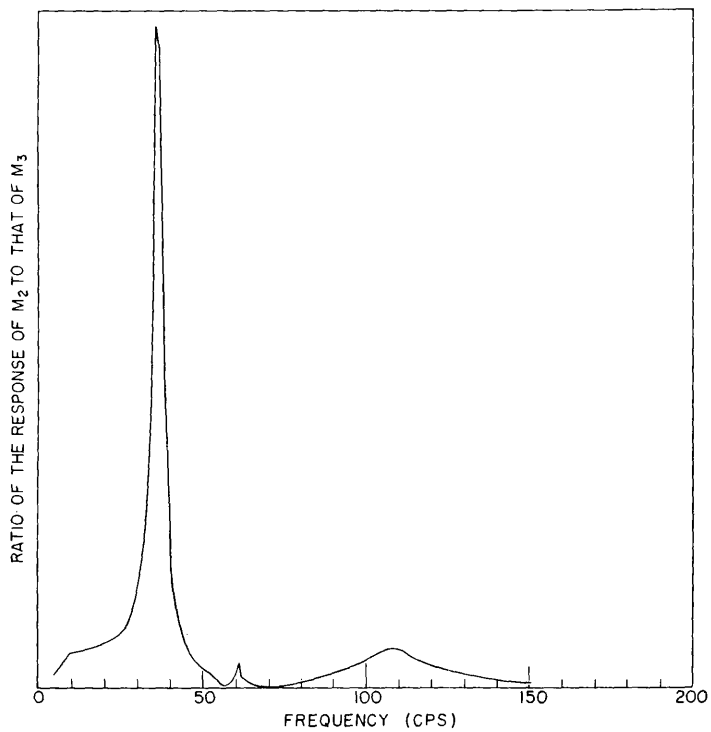


Fig. 6c - The ratio of the response of M_2 to that of M_3 when the driving force is at M_3

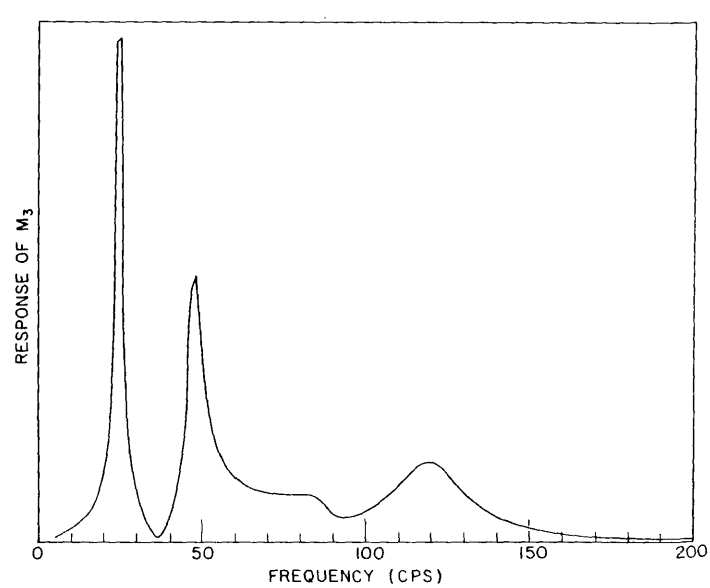


Fig. 7a - The response of M_3 to the driving force at M_4

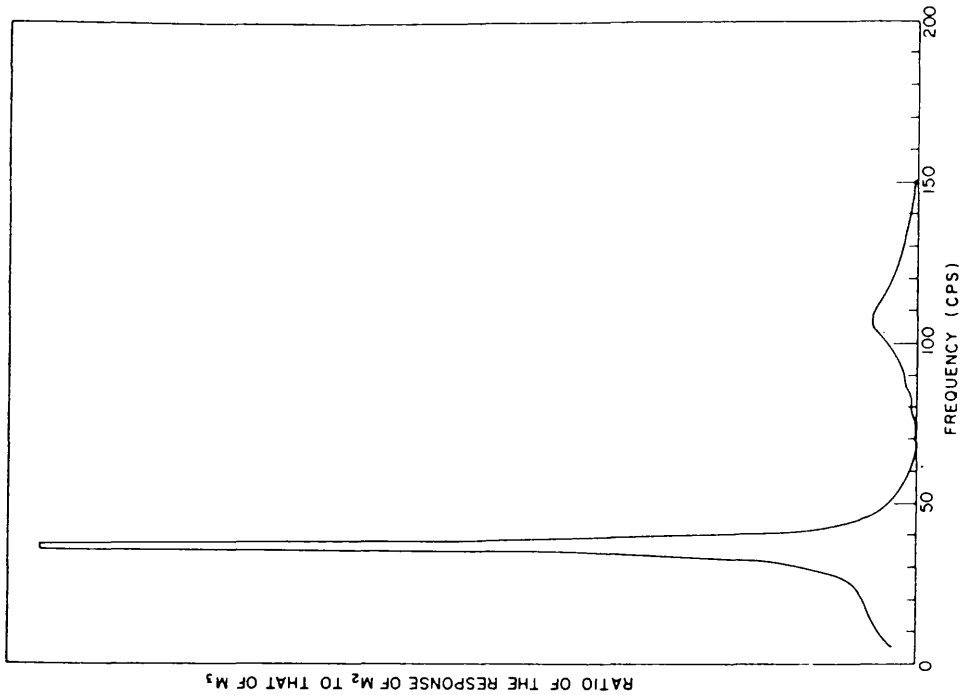


Fig. 7c - The ratio of the response of M_2 to that of M_3 when the driving force is at M_4

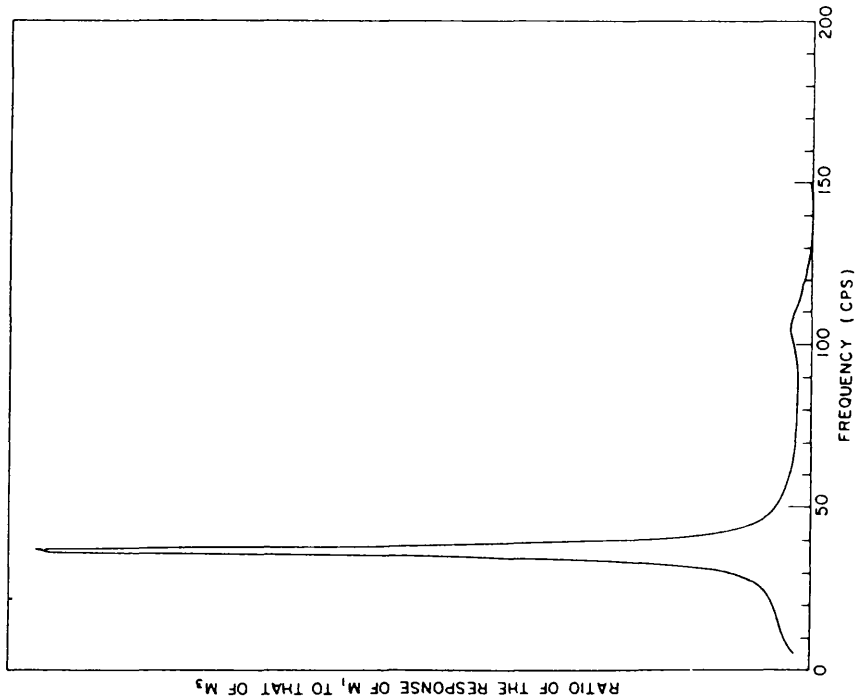


Fig. 7b - The ratio of the response of M_1 to that of M_3 when the driving force is at M_4

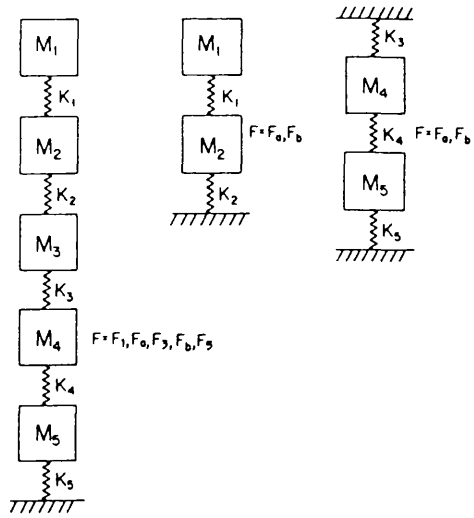


Fig. 8 - Special case in which the fixed base subsystems have identical natural frequencies which are also frequencies of the complete system (F_a and F_b are the frequencies of the two-mass systems above and below M_3)

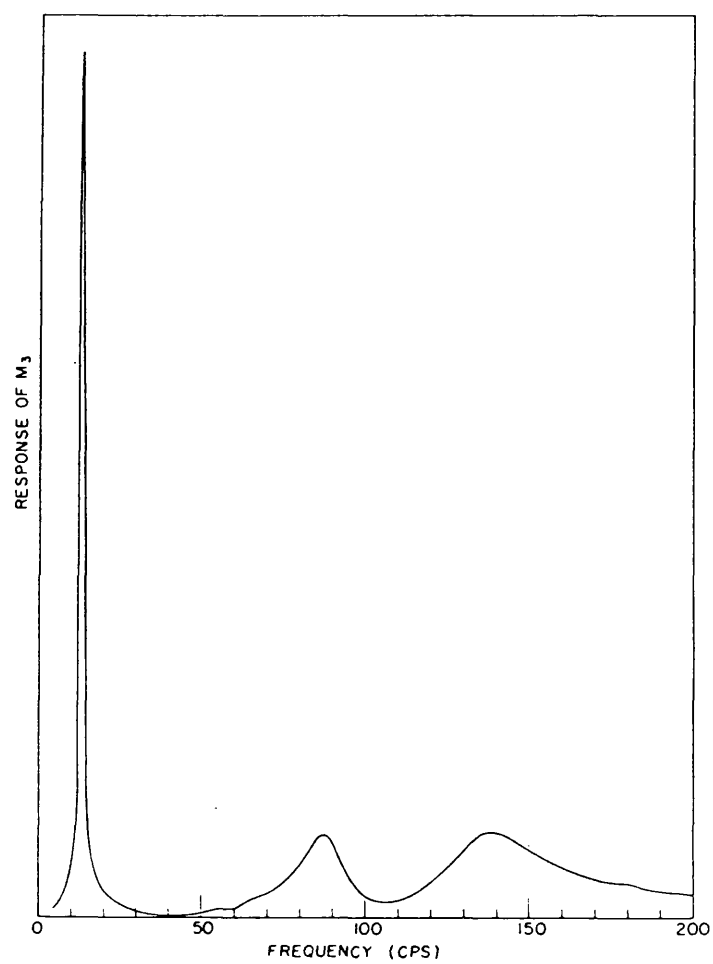


Fig. 9a - The response of M_3 to the driving force at M_3 in the special case of Fig. 8

SPECIAL CASES

Several special cases were investigated on the analog computer to determine if the proposed technique could be employed. Two are presented here.

In one special case a five-mass system was programmed (Fig. 8) so that the two frequencies of the two-mass system above M_3 were equal to the two frequencies of the two-mass system below M_3. These frequencies then become two of the five frequencies of the five-mass system. The two frequencies for this special case were 40.5 cps and 106.4 cps. The response of interest was that of M_3 when driving at M_3. Figure 9a shows this response. Valleys are seen at 40 cps and 106 cps; they occur although these two frequencies are general system frequencies where peaks would normally be anticipated. The usual ratio plots are given in Figs. 9b and 9c; peaks occur at 38 cps and 105 cps. In this case, the new technique gave correct results.

In another special case the five-mass system was programmed (Fig. 10) so that the frequency of the one-mass system above M_2 was equal to one of the three frequencies of the three-mass system below M_2. This frequency, 43.6 cps, was then also one of the five frequencies of the complete system. The response of interest in this special case

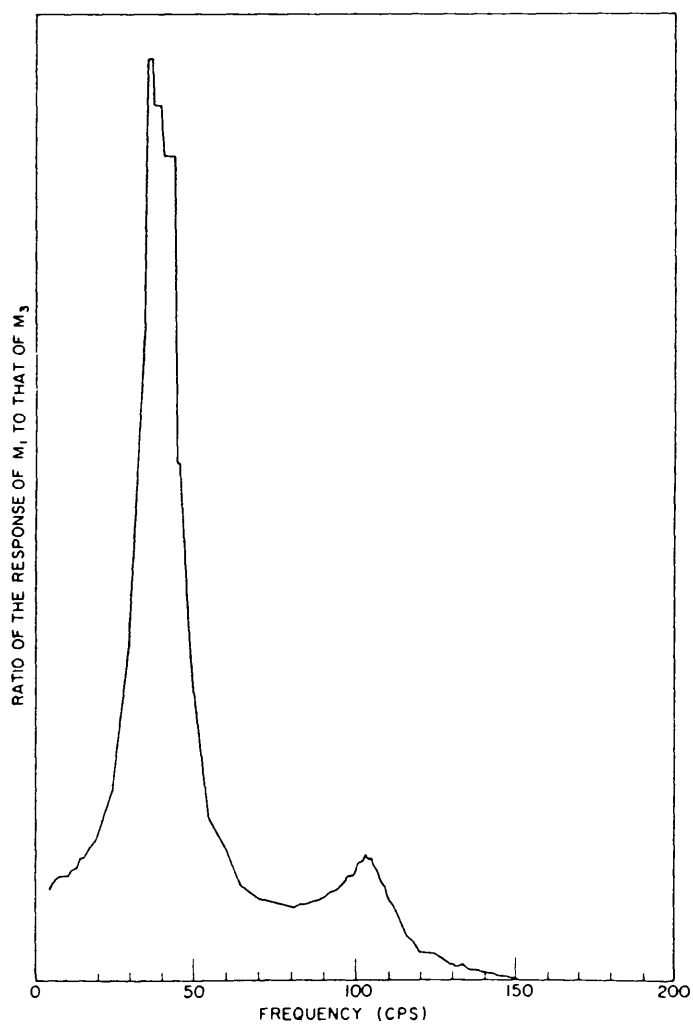


Fig. 9b - The ratio of the response of M_1 to that of M_3 when the driving force is at M_3 in the special case of Fig. 8

was that of M_2 when the system was excited at M_2. As seen in Fig. 11a, there is a prominent dip at 43 cps. Again the dip occurs in place of the peak which would normally be expected. The ratio plot of the response of M_1 to that of M_2 (Fig. 11b) displays a peak at 43 cps. The new method again produced correct results.

Several more distinctive cases involving subsystems which had frequencies matched or nearly matched were simulated on the computer. In each case the shake test technique enabled the desired fixed base natural frequencies to be chosen.

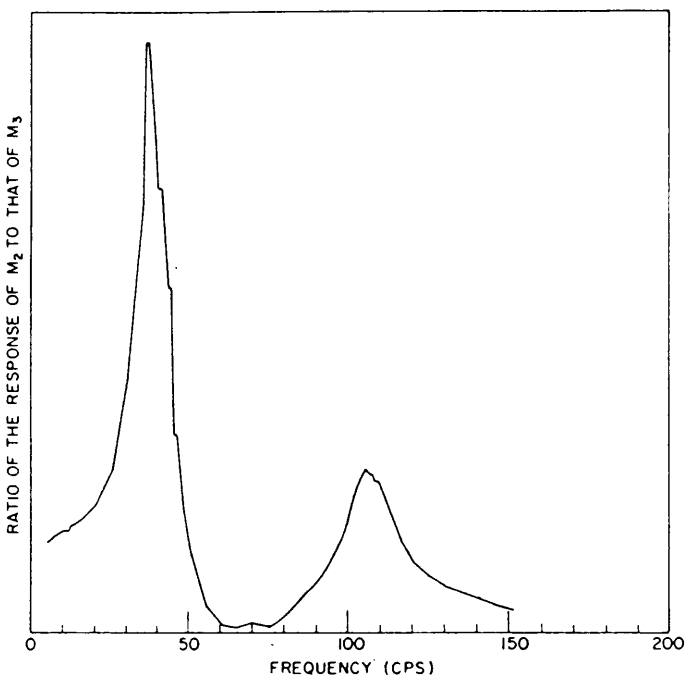


Fig. 9c - The ratio of the response of M_2 to that of M_3 when the driving force is at M_3 in the special case of Fig. 8

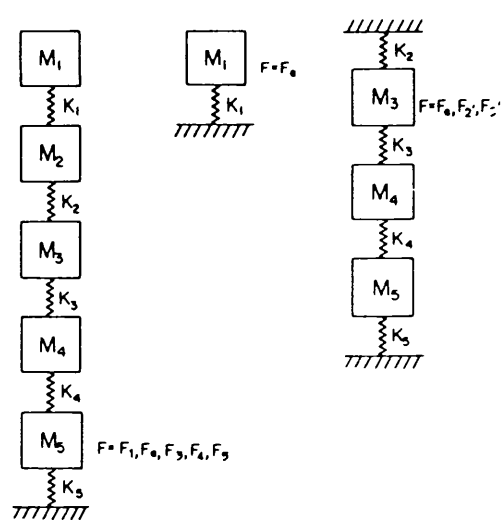


Fig. 10 - Special case in which the fixed base subsystems have a common natural freuqency which is also a frequency of the complete system

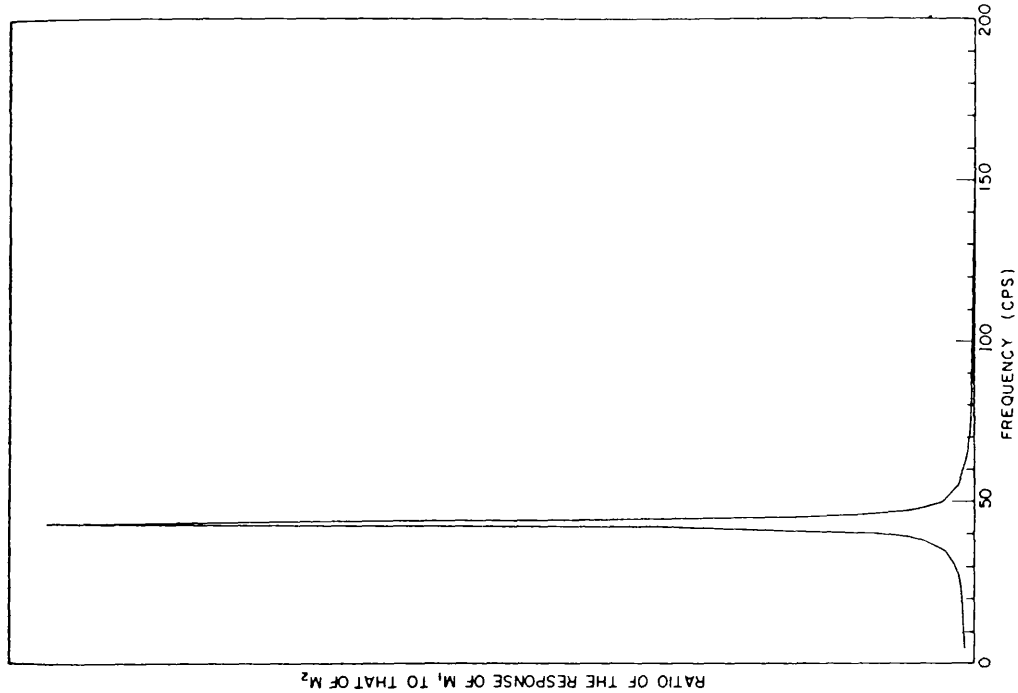


Fig. 11b - The ratio of the response of M_1 to that of M_2 when the driving force is at M_2 in the special case of Fig. 10

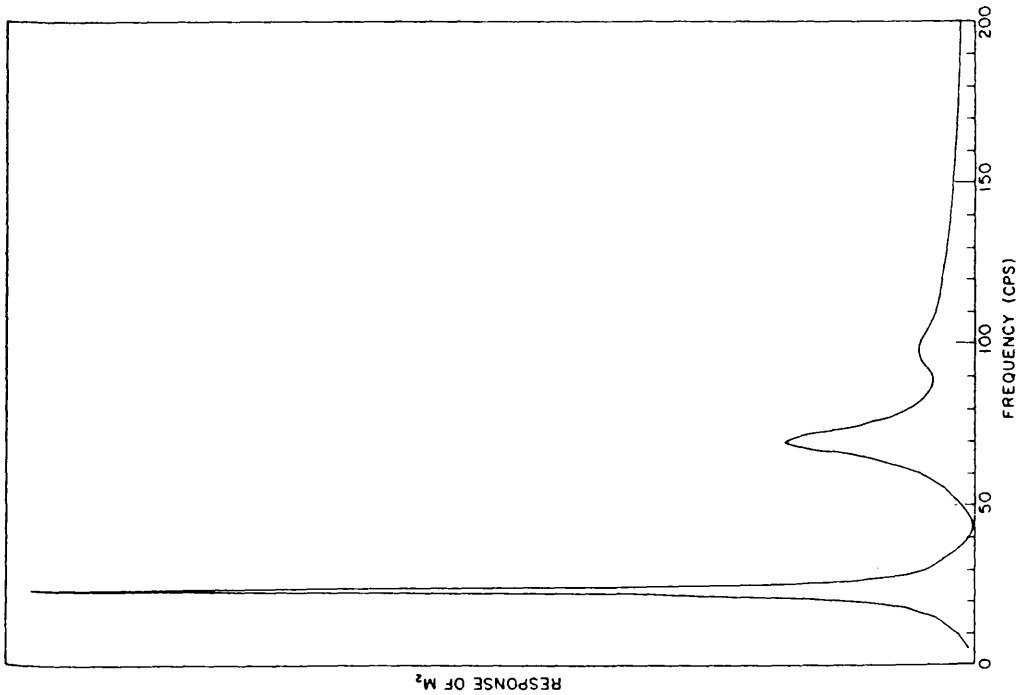


Fig. 11a - The response of M_2 to the driving force at M_2 in the special case of Fig. 10

DISCUSSION

The preceding analysis was based upon a dynamic chain model which had its motion limited to unidirectional translation. Another model of a shipboard system may be that of Fig. 12. This configuration with unequal parameters allows for rotation as well as translation. The upper massless bar with lumped masses represents the equipment-foundation system, the lower represents the supporting structure. All points are considered pinned in this analysis with constraint provided only for stability. Can the proposed technique provide the fixed base natural frequencies of the equipment-foundation system of Fig. 13?

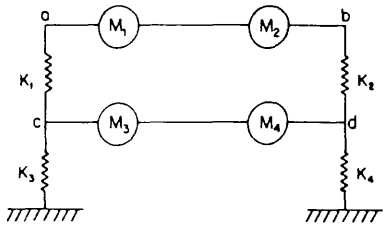


Fig. 12 - A model including rotation and translation

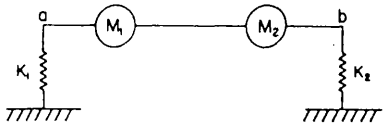


Fig. 13 - The fixed base equipment-foundation system capable of rotation and translation

The equipment-foundation system has two natural frequencies and two associated mode shapes. These mode shapes cause displacements across springs K_1 and K_2 and resultant spring forces. If the entire system (Fig. 12) is excited at points c and d with forces having magnitudes proportional to the respective spring forces, the upper system will act as a fixed base system when the driving frequency equals the modal frequency. The motion of points c and d will approach zero as this point is approached (2).

The new method would yield correct results, but difficulties in application are apparent. For the case illustrated two shakers are required, one at c and one at d. The shakers must not only sweep through the frequency range of interest, but at each frequency they must also sweep through a range of force ratios. They must operate in phase to obtain one frequency and out of phase to obtain the other. The system of interest will act as if on a fixed base when the driving frequency, force ratio, and phase are simultaneously coincident with the respective quantities for a fixed base natural mode of vibration of the system.

SUMMARY AND CONCLUSIONS

Design shock spectra are based on measured responses from realistic shock tests on in-place equipment-foundation systems. The method used to derive these design curves requires that the fixed base natural frequencies of the in-place system be known. The most convenient method to date of obtaining these frequencies has been through shake tests; however, the results have not been correct. A review of the equations of motion for a multidegree-of-freedom system showed that the method used, choosing the driving point on the equipment, could not give the desired frequencies. The review showed that the fixed base frequencies could be obtained if the driving point were on or below the system base. A new technique was proposed, which was applied to an analog simulated dynamic chain to prove its validity. The frequencies chosen by the application of this method agreed well with computed frequencies.

The new technique is this: Drive at a point on or below the base of the equipment-foundation system and record the responses of the base mass and a mass point on the equipment. Note prominent valleys in the response of the base; these points occur at the fixed base frequencies of the system above the base and of the system below the driving point. Plot the ratio of equipment response to base response. Select the fixed base

frequencies of interest as the points where prominent peaks in the ratio plot coincide with valleys in the base response plot; this eliminates the extraneous frequencies of the system below the driving point.

Although the method functioned well when applied to the simulated situation, it will be difficult to implement in the field. The equipment driving point used in the past was chosen to obtain significant response with a shaker of limited size; this point was also convenient to placing of the shaker. The correct method requires that the shaker be placed below the equipment, a location which is often inconvenient; and that the shaker must have sufficient force to excite the desired modes, a specification difficult to satisfy because of limited size. A situation involving rotation as well as translation of the system magnifies these difficulties. Additional shakers are necessary, with the same placement problem as in the case of translation only; and not only must the frequency be varied, but the force ratio and phase must be varied for each mode.

The field problems which now become associated with shake tests may prove difficult to surmount. However, once they are overcome, the tests can be utilized to obtain the fixed base natural frequencies of an in-place equipment-foundation system.

ACKNOWLEDGMENTS

The authors would like to express their thanks to Mr. G. O'Hara and Dr. P. Cunniff of the Structures Branch, NRL, for their investigations of the application of the new shake test method to a system which involves small rotation and translation.

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2. Cunniff, P.F., and O'Hara, G.J., "Shipboard Shock Fundamentals," Report of NRL Progress, Aug. 1964, p. 15

APPENDIX A

NULL POINT ANALYSIS OF A MULTIDEGREE-OF-FREEDOM SYSTEM

The equations of motion for free vibration of the system shown in Fig. A1 are

$$
\begin{aligned}
&M_1\ddot{Y}_1 + K_1(Y_1 - Y_2) = 0\\
&M_2\ddot{Y}_2 + K_1(Y_2 - Y_1) + K_2(Y_2 - Y_3) = 0\\
&M_3\ddot{Y}_3 + K_2(Y_3 - Y_2) + K_3(Y_3 - Y_4) = 0\\
&M_4\ddot{Y}_4 + K_3(Y_4 - Y_3) + K_4(Y_4 - Y_5) = 0\\
&M_5\ddot{Y}_5 + K_4(Y_5 - Y_4) + K_5(Y_5) = 0.
\end{aligned}
$$

The motion of the system in forced vibration can be described by including the force term in the equation of the driven mass. Thus if a force is applied at M_1, the equations become

$$
\begin{aligned}
&M_1\ddot{Y}_1 + K_1(Y_1 - Y_2) = F\\
&M_2\ddot{Y}_2 + K_1(Y_2 - Y_1) + K_2(Y_2 - Y_3) = 0\\
&\text{Etc.}
\end{aligned}
$$

If $F = F \sin \omega t$ and the solution $Y_i = A_i \sin \omega t$, is assumed, the set of differential equations will change to a set of simultaneous algebraic equations upon substitution of these expressions. The solution for any A_i will be F_i/A, where A is the determinant of the coefficients of the A_i's and F_i is this determinant with appropriate coefficient column replaced with the force column. By setting F_i equal to zero, the frequencies where A_i nulls can be found. Table A1 shows where nulls occur when driving each mass of the five-mass system of Fig. A1. (The system below any given mass is fixed at both ends: the base and the given mass; for instance, the three-mass system below M_2 is that system shown in Fig. A2.)

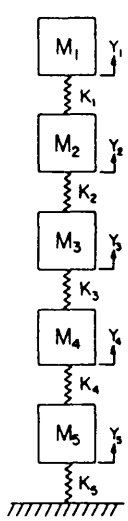

Fig. A1 - A five-degree-of-freedom system

Table A1
Frequencies of Nulls When Driving Each Mass of Fig. A1

Mass Driven	Frequencies at Which Each of the Five Masses Will Null				
	M_1	M_2	M_3	M_4	M_5
M_1	Freq. of the Four-Mass System Below M_1	Freq. of the Three-Mass System Below M_2	Freq. of the Two-Mass System Below M_3	Freq. of the One-Mass System Below M_4	Does not null
M_2	Freq. of the Three-Mass System Below M_2	Freq. of the One-Mass System Above M_1 and the Three-Mass System Below M_2	Freq. of the One-Mass System Above M_2 and the Two-Mass System Below M_3	Freq. of the One-Mass System Above M_2 and the One-Mass System Below M_4	Freq. of the One-Mass System Above M_2
M_3	Freq. of the Two-Mass System Below M_3	Freq. of the One-Mass System Above M_2 and the Two-Mass System Below M_3	Freq. of the Two-Mass System Above M_3 and the Two-Mass System Below M_3	Freq. of the Two-Mass System Above M_3 and the One-Mass System Below M_4	Freq. of the Two-Mass System Above M_3
M_4	Freq. of the One-Mass System Below M_4	Freq. of the One-Mass System Above M_2 and the One-Mass System Below M_4	Freq. of the Two-Mass System Above M_3 and the One-Mass System Below M_4	Freq. of the Three-Mass System Above M_4 and the One-Mass System Below M_4	Freq. of the Three-Mass System Above M_4
M_5	Does not null	Freq. of the One-Mass System Above M_2	Freq. of the Two-Mass System Above M_3	Freq. of the Three-Mass System Above M_4	Freq. of the Four-Mass System Above M_5

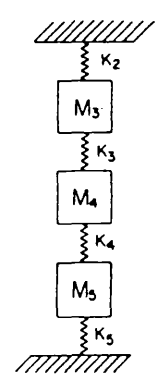

Fig. A2 - The three-mass system below M_2

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Security Classification

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1. ORIGINATING ACTIVITY *(Corporate author)* U.S. Naval Research Laboratory Washington, D.C.	2a. REPORT SECURITY CLASSIFICATION Unclassified 2b GROUP

3. REPORT TITLE
RESONANCE TESTING IN THE DETERMINATION OF FIXED BASE NATURAL FREQUENCIES OF SHIPBOARD EQUIPMENT

4. DESCRIPTIVE NOTES *(Type of report and inclusive dates)*
Interim report on one phase of the problem

5. AUTHOR(S) *(Last name, first name, initial)*
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11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY Dep't of the Navy (Office of Naval Research and Bureau of Ships)

13. ABSTRACT

Examination shows that the results of past shake tests to determine the fixed base natural frequencies of in-place shipboard equipment-foundation systems are not correct. A review of the equations of motion for a five-degree-of-freedom system revealed that choosing the driving point on the equipment could not give the desired frequencies. The review showed that the fixed base frequencies could be obtained if the driving point were on or below the base of the equipment-foundation system. Prominent valleys occur in the response of the base at the fixed base frequencies of the system above the base and of the supporting structure below the driving point. Comparison of these valleys with prominent peaks in a plot of the ratio of equipment response to base response will eliminate the extraneous frequencies of the structure below the driving point. Although this method functioned well when applied to systems simulated with an analog computer, it will be difficult in the field to place a shaker having sufficient force below the equipment. A situation involving rotation as well as translation magnifies the difficulties, since additional shakers are necessary which are capable of phase control and a range of force ratios at each frequency.

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Security Classification

14. KEY WORDS	LINK A ROLE	LINK A WT	LINK B ROLE	LINK B WT	LINK C ROLE	LINK C WT
In-place equipment						
Shipboard equipment						
Shock spectra						
Fixed base natural frequencies						
Resonance tests						
Shake tests						
Analysis of resonance tests						
New resonance test technique						
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