NRL REPORT 3597

FR-3597

PROPOSED SHOCK AND VIBRATION REQUIREMENTS OF SHIPBOARD MOUNTS

Ralph E. Blake and J. Paul Walsh

January 6, 1950

Approved by:

Dr. H. M. Trent, Head, Applied Mathematics Branch
Dr. G. R. Irwin, Superintendent, Mechanics Division

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CONTENTS

ABSTRACT

The performance under shock and steady state vibration which should be required of mounts in order that they be regarded as satisfactory for general use on naval vessels is proposed. The performance under shock is judged on the basis of the shock spectrum of the motion of the load supported by the mount. Maximum allowable values of the shock spectra are proposed in terms of the load rating, the travel, and initial height of the mount. The performance under steady state vibration is judged on the basis of either the natural frequencies or transmissibilities of a uniform-density, cubical load of the appropriate mass supported by four mounts.

PROBLEM STATUS

This is an interim report; work is continuing.

AUTHORIZATION

NRL Problem F03-05R
NS 711-019
NS 711-020

PROPOSED SHOCK AND VIBRATION REQUIREMENTS OF SHIPBOARD MOUNTS

INTRODUCTION

A study was made of the performance, under conditions of shock and vibration, which should be required of shipboard shock protection mounts and noise isolation mounts in order that they be considered satisfactory for naval use. At the present time, there is no test to determine the performance of mounts under shock except in combination with equipment. In the event that the equipment fails to pass the specified tests, it is often not clear whether the mount or the equipment design is at fault. Unless the mount itself fails, the equipment is usually assumed to have been too weak and is redesigned. Because of the lack of information on the performance characteristics of shock mounts, many varieties of mounts (including vibration isolators not designed for shock) have found their way into naval use.

A specification for mounts is desirable since it would screen out all but the best mounts. Furthermore, a mount specification would stimulate mount manufacturers to improve their products, provide mount designers with a detailed statement of the requirements of an acceptable mount, and provide equipment designers with dynamic characteristics of mounts thus removing the handicaps of designing equipment to compensate for allegedly beneficial mounts.

Some of the requirements for mounts proposed in this report had to be selected somewhat arbitrarily. Limits set in this report represent the performance which can be reasonably expected as a result of studying several mount designs. Depending upon the number of mounts which pass the tests in the next few years, some of the requirements can be so adjusted that a reasonable number of the best available designs will be accepted. It should be borne in mind that mounts which satisfy these requirements have been judged worthy of widespread use on naval ships on the basis of an estimate of the average requirements of the great majority of shock mount applications.

Some mounts may be satisfactory or even superior for specific applications though not be qualified for general use. Use of such mounts should be held to a minimum for the sake of standardization and should be used only when their superiority for a specific application has been clearly demonstrated. It is difficult to avoid the use of mounts which will not satisfy requirements proposed in this report, but have already passed a mount-equipment combination shock test. However, attempts to obtain use of inferior mounts by getting them through a shock test with

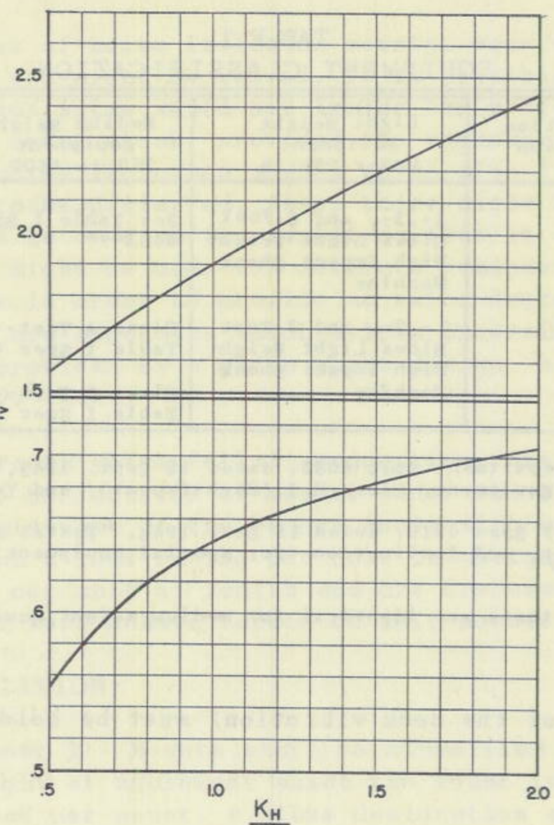

Figure 1 - Resonant frequencies of a uniform density cubical box

vibration of the specified amplitude, and finding the resonant frequency, f_n. The stiffness, K, computed from formula (1) is the dynamic stiffness in the direction of the vibration.

$$K = (2\pi f_n)^2 \frac{W}{g}. \qquad (1)$$

Vibration Transmissibilities

There are cases when a mount will not meet Requirement 2 and yet will be satisfactory so far as vibration characteristics are concerned. Such a case is a mount with a large amount of damping - the second method of maintaining low transmissibilities. In this case the requirement on the resonant frequencies should be ignored and the following performance required. A mount must comply with either Requirement 2 or Requirement 2A.

Requirement 2A - The transmissibility in any mode of vibration of a uniform density cubical box supported by mounts shall not exceed 3 when the vibration table to which the mounts are attached executes a steady state vibration of .008" ± .002" and the frequency of vibration is varied between 5 and 23 cps. The mounts shall be attached to the lower corners of the box.

Vibration Endurance

To insure that normal steady state shipboard vibrations, continuing for times of the order to two years, will not cause failure of the mount, it is necessary to examine the vibration endurance of the mount by means of an accelerated test, as specified in Requirement 3.

Requirement 3 - At the completion of the following vibration endurance tests, the mount shall not, upon disassembly and inspection, show any deterioration of the resilient element, metal parts or fastenings.

The mount shall support a load equal to the maximum rated load, W, of the mount. The base of the mount shall be subjected to a steady state vibration having an amplitude of .030" ± .006". The frequency shall be 23 cps or less and shall be that frequency which produces the maximum acceleration of the load. At least two directions of vibrations will be investigated, one parallel with the equipment-attaching bolt and one perpendicular to this bolt. (At the discretion of the engineers, other directions may be investigated.) The total duration of this test will be four hours (normally two hours in each of two directions).

Nominal Vibration Load Rating

The largest load on the mount which satisfies the natural frequency requirements, (Requirement 2) or the transmissibility requirements (Requirement 2A) and the vibration endurance requirement (Requirement 3) is defined as the "Nominal Vibration Load Rating."

It should be emphasized that if a mount is assigned a nominal vibration load rating of W lb, these tests do not assure that all equipments (or even all equipments having outer cases which are cubes) which load the mount with a force of W lb statically, will be satisfactory under the vibration requirements of equipment specifications. It can be said that if a mount has a nominal vibration load rating of W lb, in a large proportion of practical problems, 4 mounts will support an equipment weighing 4 W lb and having a cubical shape and the combination will meet the vibration requirements of an equipment specification such as 40T9.[2]

SHOCK TESTS AND REQUIREMENTS

Light Weight Mounts[3] shall be tested on the Light-Weight-High-Impact Shock Machine using the apparatus shown on Naval Research Laboratory Drawing F1371 and described in a concurrent NRL Report.[4]

Medium Weight Mounts[5] shall be tested on the Medium-Weight-High-Impact Shock Machine using the apparatus shown on Naval Research Laboratory Drawing D1372 and described in the NRL Report previously mentioned.[6]

Shock Tests

Light weight mounts are tested one at a time but medium weight mounts are tested four at a time. The various tests to be conducted are outlined in Table II and the mounting adaptor arrangements, number of bolts in the "Figure 4A plate" stiffening channels in the case of the light weight machine and the supporting channel selection in the case of the medium weight machine, are given in Table III.

The same mount (or mounts) is used for Tests 1, 2, and 3 and another is used for tests 4, 5, and 6. Separate tests (1 through 6) should be conducted for each mount design, first at the upper load rating and then the lower load rating. The following data will be recorded for each test:

(a) The maximum deflection across the mount.

(b) The reed gage record.

(c) The force exerted by the mount on the load as a function of time.

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2 *See Table I.*

3 *Ibid.*

4 *Walsh and Blake, op. cit.*

5 *See Table I.*

6 *Walsh and Blake, op. cit.*

TABLE II

Shock Tests

<table>
<tr><th>Mount Classification</th><th>Test No.</th><th>Deflection Direction</th><th>Mounting Adapter Arrangement</th><th>Schedule of Blows for Each Test.</th></tr>
<tr><td rowspan="6">66S3
Lt. Wt.</td><td>1</td><td>Axial</td><td>A</td><td rowspan="6">No. Ht.
1 1'
1 3'
3 5'</td></tr>
<tr><td>2</td><td>Axial</td><td>B</td></tr>
<tr><td>3</td><td>Axial</td><td>C</td></tr>
<tr><td>4</td><td>Radial</td><td>A</td></tr>
<tr><td>5</td><td>Radial</td><td>B</td></tr>
<tr><td>6</td><td>Radial</td><td>C</td></tr>
<tr><td rowspan="6">40T9
Lt. Wt.</td><td>1</td><td>Axial</td><td>A</td><td rowspan="6">No. Ht.
1 1'
1 2'
3 3'</td></tr>
<tr><td>2</td><td>Axial</td><td>B</td></tr>
<tr><td>3</td><td>Axial</td><td>C</td></tr>
<tr><td>4</td><td>Radial</td><td>A</td></tr>
<tr><td>5</td><td>Radial</td><td>B</td></tr>
<tr><td>6</td><td>Radial</td><td>C</td></tr>
<tr><td rowspan="6">66S3
Med. Wt.</td><td>1</td><td>Axial</td><td>A</td><td rowspan="6">No. Group
1 I
2 II
2 III
See Table I of Spec. 66S3</td></tr>
<tr><td>2</td><td>Axial</td><td>B</td></tr>
<tr><td>3</td><td>Axial</td><td>C</td></tr>
<tr><td>4</td><td>Radial</td><td>A</td></tr>
<tr><td>5</td><td>Radial</td><td>B</td></tr>
<tr><td>6</td><td>Radial</td><td>C</td></tr>
<tr><td rowspan="6">40T9
Med. Wt.
Class A
or
Class B</td><td>1</td><td>Axial</td><td>A</td><td rowspan="6">No. Group
1 I
2 II
2 III
See Table I of Spec. 40T9</td></tr>
<tr><td>2</td><td>Axial</td><td>B</td></tr>
<tr><td>3</td><td>Axial</td><td>C</td></tr>
<tr><td>4</td><td>Radial</td><td>A</td></tr>
<tr><td>5</td><td>Radial</td><td>B</td></tr>
<tr><td>6</td><td>Radial</td><td>C</td></tr>
</table>

TABLE III

Mounting Adaptor Arrangement

Shock Machine	Arrangement		
	A	B	C
Light Weight *	2 Bolts per Beam.	4 Bolts per Beam.	6 Bolts per Beam.
Medium Weight **	Test Load Center distance = 16 in. Supporting channels as for 24 in. center distance	Test Load Center distance = 24 in. Supporting channels for 24 in. center distance.	Test Load Center distance = 24 in. Supporting channels for 24 in. center distance plus two-car building channels

* Walsh, J. P. and Blake, R. E. "The Determination of Shock Isolator Performance." NRL Report 3596, January 6, 1950.

** See Table 1, Specification 66S3.

Computation of Results

Three shock spectra[7] are computed for each test - one for each height of hammer drop. The spectrum computed for those heights at which multiple drops are made will be the "envelope" of the two or more spectra resulting from the separate drops.

Composite spectra are then computed for each hammer drop and for each direction of deflection, as follows: The three spectra for a given height of hammer drop and for the same direction of deflection of the mount are averaged. For example, the spectra resulting from the 1-foot hammer drop for tests 1, 2, and 3 are averaged. The average is referred to as the composite spectrum for 1-foot hammer drop for the axial direction. This is done for each height of hammer drop. This results in six composite spectra for each mount design and each load rating.

The maximum deflection across the mount occurring during any test is reported as the mount travel, d.

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[7] *For a discussion of the shock spectrum, the reader is referred to: Walsh, J. P. and Blake, R. E. "The Equivalent Static Accelerations of Shock Motions," NRL Report F-3302, June 21, 1948.*

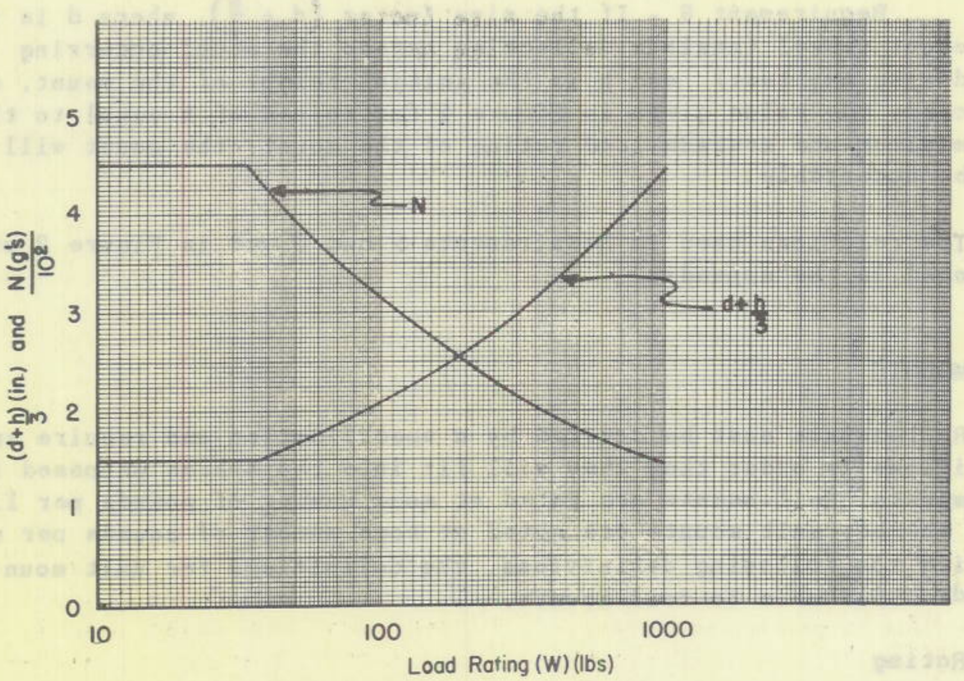

Figure 2 - Shock isolation and size limits

The maximum peak force occurring during any test is reported as the peak force of the mount.

Shock Endurance Requirement

Requirement 4 - Upon completion of the tests in one direction of deflection (tests 1, 2 and 3 or 4, 5 and 6), the mount (or mounts) will be examined. If parting of any part of the mount has occurred, the mount will not be acceptable; yielding of the parts of the mount will be acceptable.

Shock Isolation and Size Requirements

Requirement 5 - If the maximum value of any of the six composite shock spectra plotted in the frequency range of zero to 450 cps exceeds the value given on the plot of Figure 2 for values of W equal to the minimum & maximum load ratings of the mount, the mount will not be acceptable.

Requirement 6 - If the size factor $\left(d + \frac{h}{3}\right)$, where d is the mount travel (maximum deflection across the mount occurring during any test) and h is the initial height of the mount, exceeds the value given in Figure 2 for values of W equal to the minimum and maximum load rating of the mount, the mount will not be acceptable.

The relations used in Requirements 5 and 6 and in Figure 2 are developed in the appendix.

RAIL-MOUNTS

Rail-mounts must be covered by a specification and require special definitions in order that they will fit into the scheme proposed for unit mounts. Rail-mounts are rated at some number of pounds per linear foot, whereas unit mounts are rated at some number of pounds per mount. By using the following definitions, the definitions for unit mounts can be made applicable to rail-mounts.

Load Rating

The rail-mount will be classified in the same way as unit mounts except that the load rating will be given in pounds per linear foot, plus or minus some load tolerance. In addition, a maximum and minimum number of feet of the mount to be used will be given. This corresponds to an upper and lower total equipment weight to be used with the design.

Shock Tests

Two complete sets of shock tests will be conducted on the rail-mount-load system. The first will be conducted with a total load on the mount equal to the upper-total-equipment-weight load rating; the second with a load equal to the lower-total-equipment-weight load rating. The tests will be identical with those given for unit mounts. The same data will be taken and the same computations will be made.

Requirement 7 - The $\left(d + \frac{h}{3}\right)$ factor and the peak values of the composite shock spectra for rail-mounts must meet the requirements of Figure 2 where the load rating of the rail-mount, W, is considered to be one-quarter of the total equipment weight.

Vibration Requirements

Requirement 8 - The vibration requirements of rail-mounts are the same as the unit mount requirements.

In this case, however, the natural frequencies are computed on the following basis. The frequencies are computed for the maximum equipment weight allowed by the load tolerance supported by a given length of rail-mount. The load is considered to be a uniform density cube having sides of lengths equal to one-quarter of the minimum length of a rail-mount. The rail-mount is considered to act at and along the bottom edges of the cube. In order to make these computations, the vibration stiffnesses in the vertical direction, along the length of the mount, and in the direction of the width of the mount must be known.

DISCUSSION

It is impossible to anticipate all the effects of the above requirements if they are incorporated into a specification on mount designs. Some of the requirements may conflict; others which may at first appear to conflict, in fact, do not; and some will have marked effects on future mount and equipment designs.

There is general agreement among those working on the problem of equipment design and equipment protection that the quality of shock mounts on naval vessels must be improved. A performance specification for shock mounts is perhaps the most direct way to accomplish this. The requirements proposed here are intended to eliminate all but the best designs. In fact, no mount studied to date meets all the requirements.

The larger load rating mounts will probably require some new device to meet the vibration requirement. If a mount of low load rating which fulfills the requirements is proportionally scaled up in size (the easy way to obtain a large mount), it will most likely pass the shock test but will fail to pass the vibration tests.

In order to have a reasonable load tolerance for a mount design, it will be necessary to design a mount that is more efficient at some load than the curves of Figure 2 indicate. For example, suppose that a mount design at some load W, produces values of N and has a size which falls exactly on the curves. If the load on the mount is reduced, the mount is then too large for the load, that is the $\left(d + \frac{h}{3}\right)$ exceeds the allowable value of this reduced load rating. If the load on the mount is increased the requirement on N will probably not be met since the value of N must decrease as load increases.

Conditions similar to the above are true in the case of rail-mounts. Heretofore, it has generally been considered that a given rail-mount design was satisfactory for equipment of any weight. Under the requirements proposed in this report, this would not be satisfactory because of the interrelation of size and shock protection.

Since it is important to provide all the shock protection to an equipment that can be afforded, it is believed that the adoption of a specification requiring that all mounts used on naval vessels meet the performance requirements contained in this report would be a large step in this direction.

* * *

APPENDIX

MOUNT SIZE, SPACE LOSS AND SHOCK PROTECTION RELATIONS

Several relations which have been used in the paragraph on "Shock Tests and Requirements" are developed in this appendix.

SIZE OF MOUNT AND SPACE LOSS

If mounts were not used with a given equipment, it would be rigidly attached to the ship and occupy some volume, say V. When mounts are used, the volume required by this equipment is greater than V. Let us compute this increased volume. Assume that the mounts have an allowable deflection, d, in any direction, and that the initial height of the mount, under load, is h.

There are two cases which will be considered: (a) equipment with bottom mounts and bulkhead mounts; and (b) equipment with bottom mounts only. Assume that the motion perpendicular to one face of the equipment is of no importance. This assumption is realistic because one face of an equipment is usually on a passageway, and motion of the equipment into it does not require an increase in the width of the passageway.

It can be shown for the bulkhead mount case (Figure 3), that the ratio of the volume required with mounts to volume without mounts (considering one face on a passageway) depends upon both the travel of the mount, d, and the height of the mount, h, and is proportional to

$$B\ (d + ch) \qquad (2)$$

where c depends upon the shape of the equipment and B depends upon its size.

In the case of an equipment with bottom mounts only (Figure 4) the ratio of volume required with mounts to initial volume depends upon B (d + ch) also, where c depends upon the shape of the equipment and B depends upon its size.

In both cases, the volume ratio depends upon a size factor, (d + ch). The values of c have been computed (which express this ratio) for various shaped boxes. This shape is expressed by the factor n, the ratio of the height of the box to one dimension of the square base. The values of c for corresponding values of n in each of the two cases are given in Table IV.

From Table IV it can be seen that the increase in volume used by a given piece of equipment because of the use of mounts, depends upon the mount travel, the height of the mount, the shape factor of the equipment and the mounting pattern. Because of this, it is not possible to establish the relative importance of travel and height in the volume loss due to the addition of mounts, in the general case. When a mount is being evaluated, it is not possible to know the shape of the equipment it might support or if it will be used in a bulkhead mounting pattern or not. Since it is desirable to evaluate a mount for any possible use, a compromise must be made. This can be a reasonably important one because the relative importances assigned to d and h, that is the value of c, will influence the mounts designed to meet the requirements of a specification.

After an examination of Table IV and a consideration of the shapes of equipment and the possible influence of the value of c on future mount designs, a value of c = 1/3 was selected as the best to be used in stating the relative importance of travel and initial height of mount. Therefore, the measure of mount

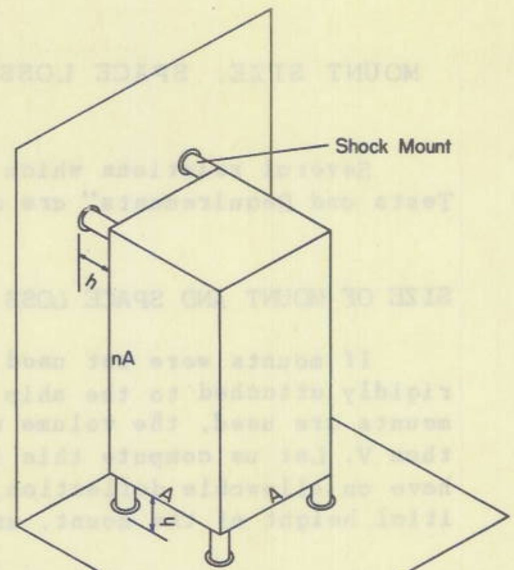

Figure 3 - Equipment with bottom mounts and bulkhead mounts

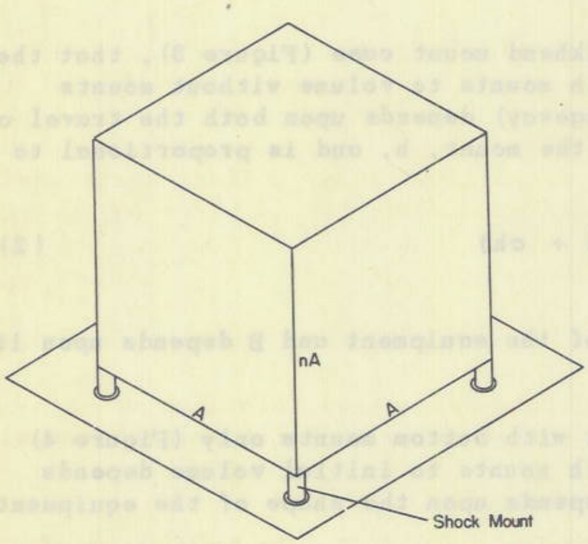

Figure 4 - Equipment with bottom mounts

TABLE IV

Bulkhead Mounts		Bottom Mounts	
n	c	n	c
∞	.5	1	.14
3	.63	.66	.27
2	.67	.50	.40

size to be used in subsequent calculations is

$$d + \frac{h}{3}. \tag{3}$$

SIZE OF MOUNT AND SHOCK PROTECTION

As the load rating of a mount increases, the allowable size of the mount increases because the resilient element, the fastenings and structural parts of the mount must become larger to absorb more energy and withstand higher forces. The larger equipments can accommodate larger mounts without an increase in the percent of the total volume devoted to mounting.

It can be shown that for mounts having similar load-deflection curves, the shock spectra, N, are inversely proportional to d.[8] This relation[9] is given by equation (4)

$$\frac{N_1}{N_2} = \frac{d_2}{d_1}. \tag{4}$$

It can be shown that if all the dimensions of a given mount design are scaled by a factor S the shock load rating, W, is given by

$$\frac{W_2}{W_1} = S^3. \tag{5}$$

In proportional scaling, the load-deflection relations are geometrically similar, which means, for example, that the initial travel of d, is now scaled to $d_2 = Sd_1$.

---

[8] *To be described in forthcoming NRL Report: Blake, R. E. "The Modification of Shocks to Minimize their Damaging Effect."*

[9] *For the remainder of the report subscript 1 refers to the initial mount and subscript 2 refers to the scaled mount.*

From equation (4) the ratio of the maximum values of the shock spectra for the original mount and the scaled mount is

$$\frac{(N_1)\ max}{(N_2)\ max} = S, \tag{6}$$

The ratio of the peak forces F_1 and F_2 exerted by the mounts can be computed by equation (7):

$$\frac{F_1}{F_2} = \frac{W_1N_1}{W_2N_2} = \frac{1}{S^2}. \tag{7}$$

Equation (7) expresses the peak force relation in scaling; examination will now be made of the strength relations existing when structural parts of mounts are scaled. A simple, but representative case of a mount structural member is a circular simply supported, uniform plate.

The maximum stress in a plate resulting from a uniform distributed load perpendicular to the plane of the plate is given by[10]

$$\sigma = \frac{CWr^2}{t^2}. \tag{8}$$

Where W = intensity of load

r = radius of plate

t = thickness of plate.

With proportional scaling, the ratio of stresses in the plate-like members of the original mount to the scaled mount can be expressed approximately by

$$\frac{\sigma_1}{\sigma_2} = \frac{W_1 {r_1}^2}{{t_1}^2} \times \frac{{t_2}^2}{W_2 {r_2}^2} \tag{9}$$

but $W_1 = \dfrac{F_1}{\pi {r_1}^2}$, $r_2 = Sr_1$, and $t_2 = St_1$.

Therefore, $\dfrac{\sigma_1}{\sigma_2} = 1.$

A consideration of the strength of fastenings in the scaled mount compared with those in the original, similar to that given above for

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[10] *Timoshenko, S. "Strength of Materials Part II," p. 495, New York; D Van Nostrand, 1945*

plates indicates that if all dimensions of the fastening are scaled by S, the fastening in the scaled mount will be subjected to the same stress as those in the original mount.

This is not surprising since stress has the dimensions F/L^2 where F is the force on the mount and L is derived from dimensions of the mount. Since all dimensions, and L, are scaled in proportion to S and F is proportional to S^2. S is eliminated and stress should thus be independent of S.

The following conclusions can be drawn from these considerations. If a given mount design is proportionally scaled by a factor S the load rating scales as S^3, the shock isolation, measured by the shock spectra, scales as S and the stresses in the structural members of the two mounts will be equal.

SIZE FACTOR AND LOAD RATING

The relation between the size of a mount, expressed by the size factor, and load rating will now be examined. From the definition of a proportional scale factor there is obtained

$$\frac{d_2 + \frac{h_2}{3}}{d_1 + \frac{h_1}{3}} = S \quad . \tag{10}$$

But equation (6) states that

$$S = \left(\frac{W_2}{W_1}\right)^{\frac{1}{3}}$$

therefore (10) becomes

$$\frac{d_2 + \frac{h_2}{3}}{d_1 + \frac{h_1}{3}} = \left(\frac{W_2}{W_1}\right)^{\frac{1}{3}} . \tag{11}$$

Equation (11) expresses the relation between the load rating and size factor for two mounts which are geometrically similar. This allows an estimate to be made of the size of a mount of load rating W_2 knowing the size of a mount having load rating W_1.

LOAD RATING AND SHOCK PROTECTION

The relation between the shock protection provided by a mount of a given size and load rating will now be developed.

Equation (6) states that

$$\frac{(N_1)\ max}{(N_2)\ max} = S$$

and equation (10) states that

$$\frac{d_2 + \frac{h_2}{3}}{d_1 + \frac{h_1}{3}} = S$$

therefore

$$\frac{N_2}{N_1} = \left(\frac{W_1}{W_2}\right)^{\frac{1}{3}} . \qquad (12)$$

Equation (12) allows the maximum value of the shock spectrum of a new scaled mount of load rating W_2 to be computed if the maximum value of the shock spectrum for a moa nt of load rating W_1 is known. Thus, a given mount design can be built in a variety of sizes and the properties of the scaled mount can be predicted. Of course, the better the original design, the better all the scaled mounts will be. It is possible to compare the value of the basic ideas of various mounts even though they may be of different sizes and load ratings. The results of tests can all be scaled to the values which would be obtained if the mounts all had the same load rating.

SIZE AND SHOCK PROTECTION AS A FUNCTION OF LOAD RATING

The relations which have been stated express the ratio of mount parameters but do not yield values of these parameters. To obtain these values, the values for one size of a given design must be known. As the result of the examination of several mount designs, the authors have estimated that a mount having a load rating of 60 pounds, a size factor, $\left(d + \frac{h}{3}\right)$, of 1.75 in., and a shock spectrum, having a maximum value of N equal to 375 g in the frequency range of 0 to 450 cps, can be designed. Using these estimates, expected values of $(d_2 + h_2/3)$ and N_2 have been computed for values of W_2 and are plotted on Figure 2. It is proposed that these curves be used to define the maximum allowable values of N and $\left(d + \frac{h}{3}\right)$ for a mount having a load rating W. It will be noticed that these curves have been modified in the low weight range. This has been done with the final use of the curves in mind, for reasons which will now be discussed.

It is possible to reduce the size of a mount design until it reaches a size which provides so little shock protection that the mount is not acceptable. The value of N_{max} that is not acceptable is not and cannot be rationally established. A general idea seems to have become established with designers that small equipments only require small shock mounts. There is only a little truth in this notion; small equipments and large equipments require the same travel for the same shock protection under the same shock, and furthermore, it has not been established that small equipments are stronger, in general, than large equipments. However, since the use of shock mounts results in a net increase in the volume required for a given equipment, it is necessary to tolerate higher accelerations on the smaller equipments. But shock mounts are not advantageous unless they produce lower spectra than rigid mounting. On this basis, the maximum allowable value of N has been selected as 450 g and the corresponding $(d + \frac{h}{3})$ is 1.5 inches. This explains the horizontal portions of the curves in the low weight range.

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