

NRL REPORT NO. S-3153

VIBRATION ISOLATOR FOR AIRCRAFT VOLTAGE REGULATOR



FR-3153



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**VIBRATION ISOLATOR
FOR AIRCRAFT VOLTAGE REGULATOR**

by

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Problem No. 40S06-04

July 1947

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ABSTRACT

The Bureau of Aeronautics requested that the Naval Research Laboratory design a vibration isolator system for a coil and pile housing assembly of a carbon pile voltage regulator. The limitations placed on the mounting were severe and of such a nature that conventional commercial-type vibration isolators were not applicable. A system conforming to the limitations requested was designed and the details of its construction and functioning are reported herein, together with a discussion of the theory and an example of typical calculations.

AUTHORIZATION

The design of a vibration isolator system for a coil and pile housing assembly of a carbon pile voltage regulator for aircraft was requested by the Aircraft Electrical Research Division, as a part of the problem outlined in BuAer letter Serial Aer-E-3124-WCB, F36-1(1) of 2 January 1946 to NRL.

PROBLEM STATUS

This is the final report on this phase of the work, but another report will be prepared dealing with other phases.

VIBRATION ISOLATOR FOR AIRCRAFT VOLTAGE REGULATOR

DESCRIPTION

The voltage regulator pictured in Figures 1 and 2 weighs approximately two pounds, and is 3-3/4 inches long and 3 inches in diameter. It is extremely sensitive to external vibrations, the effect of which is to interfere with its proper operation. Temperatures external to the regulator when installed have been recorded at 150° C; this precluded the use of rubber as a mounting medium, for, in general, rubber should not be subjected to mechanical work at temperatures greater than 82° C. *

It was stated in the problem that the mounting system was required to isolate the regulator from aircraft vibrations with no restrictions as to the axis of mounting in the aircraft, and for all attitudes of the plane in flight. The space which the regulator and its mounting system could occupy was limited to 3-1/2 by 5 by 5-1/2 inches.

GENERAL THEORY OF CENTER-OF-GRAVITY TYPE MOUNTING

The design of a vibration isolation system for a body consists essentially of reducing the transmissibility (ratio of unit displacement to exciting displacement) to some acceptable value less than one. This is done by keeping the natural frequencies of the mounted body considerably below the frequencies of the source of vibration excitation. In naval aircraft equipment specifications,† it is indicated that the upper limit of natural frequencies for resiliently mounted equipment should be approximately 15 cycles per second, and that the transmissibility for frequencies above 30 cps should not be greater than 0.35.

Most methods of mounting equipment have the mounts applied merely at convenient locations. Generally, under such conditions a simple translational exciting vibration will not only cause the equipment to vibrate in the direction of the exciting vibration, but will also cause the equipment to vibrate in rocking modes and in translational modes in other directions. The different modes of vibration are then said to be "coupled" which means that a vibration in one mode will set up a vibration in another mode because of the mechanical coupling of the mounting units.

It is possible to mount a system so that no such coupling exists.‡ One method of

*United States Rubber Company Handbook, "Physical Properties of Rubber," April 1938, p. 18

†Bureau of Ships Specification 16V1(RE) dated 1 April 1945

‡Crout, P. D., "The Elastic Properties of Shock Mounts, and the Conditions for Preventing Translatory Excitation From Causing Angular Motion and Vice Versa," Shock Mounts - Theoretical Study by the M.I.T. Radiation Laboratory Group

doing this is to employ a "center-of-gravity" arrangement of the mounts. This can most simply be done by using similar mounts in pairs spaced equidistant from the center of gravity of the unit. When this is done, there is no coupling between different modes of vibration; rocking modes do not exist except when independently excited; and clearance and vibration problems are at a minimum.

The type of mounting used for the system under discussion is shown in Figures 1 and 2. This is the "center-of-gravity" system with the support locations in a horizontal plane.

To rationally design this mounting system, the required information is: (a) The highest natural frequency permissible, or the allowable transmissibility factors for a frequency range; (b) the clearance available, (c) the weight, and weight-distribution of the equipment to be mounted, and (d) operating conditions under which the mounting must function. With the above information, it is possible to solve the required problem of determining the spring constants and support locations.

A good discussion of this entire problem, and in particular further information concerning the rotational frequencies is given by Den Hartog, Crede, Walsh, and Hull.[§]

In general, the procedure for designing a vibration-isolation mounting similar to the one here discussed would consist of determining the highest resonant frequency permissible. A high resonant frequency is desirable as this means stiff mounts can be used

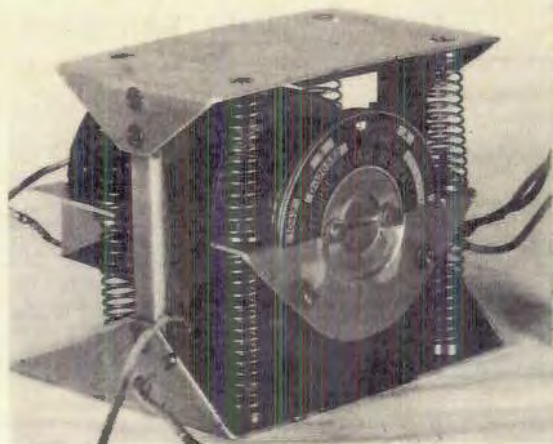


Fig. 1 - Voltage Regulator and Vibration Isolator System

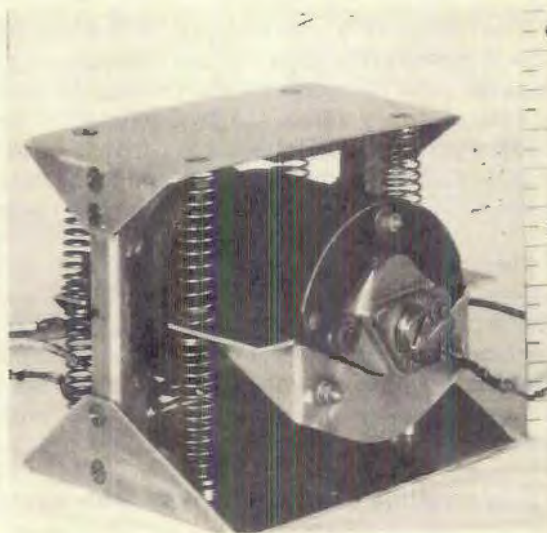


Fig. 2 - Voltage Regulator and Vibration Isolator System

[§] Den Hartog, J. P., "Mechanical Vibration," McGraw-Hill Book Company, Inc., 1940 pp. 47, 59; Crede, C. E. and Walsh, J.P., "The Design of Vibration-Isolating Bases for Machinery," *Journal of Applied Mechanics*, Vol. 14, No. 1, March 1947, p. A-10; Hull, "The Use of Rubber in Vibration Isolation," *Journal of Applied Mechanics*, Vol. 4-5, 1937-38; Crede, C.E., "Shock Mounts for Naval Shipboard Service," *NavShips* 250-600, 1

and problems of clearance are reduced. If the resonant frequency is too high, however, transmissibility of the exciting vibration into the mounted structure becomes too high.

The problem resolves itself into finding the dimensions of the springs that will provide natural frequencies of about 15 cps, and that will fit into the available space, with a center of gravity arrangement of supports. The resonant frequencies of the mounted apparatus are calculable by the methods outlined by Hull.** Wahl†† has presented further information on the characteristics of steel springs.

The vertical combined spring constant, K_{CV} , is determined from the following equation:

$$f_1 = \frac{1}{2\pi} \left(\frac{K_{CV}g}{W} \right)^{\frac{1}{2}} \quad (1)$$

The vertical static deflection is then:

$$\delta = \left(\frac{3.125}{f_1} \right)^2 \quad (2)$$

Subject to the space limitations, the height, diameter, and number of turns of the individual springs are assumed and the diameter of wire is calculated to give the proper spring constant.

$$d = \left(\frac{64 Pr^3 n}{G \delta} \right)^{\frac{1}{4}} \quad (3)$$

The horizontal spring constant per spring is given by:

$$K_H = \frac{K_V D^2}{0.390 D^2 + 0.297 h^2} \quad (4)$$

It is required that the natural frequencies in the horizontal direction be about 15 cps. For the general case, these frequencies are given by:

$$f_2 = \frac{1}{2\pi} \left\{ \left(\frac{K_{CH}}{2M} + \frac{K_{CV}b_z^2 + b_y^2 K_{CH}}{2I_x} \right) - \left[\left(\frac{K_{CH}}{2M} + \frac{K_{CV}b_z^2 + b_y^2 K_{CH}}{2I_x} \right)^2 - \left(\frac{K_{CV}K_{CH}b_z^2}{I_x M} \right) \right]^{\frac{1}{2}} \right\}^{\frac{1}{2}} \quad (5)$$

$$f_3 = \frac{1}{2\pi} \left\{ \left(\frac{K_{CH}}{2M} + \frac{K_{CV}b_x^2 + K_{CH}b_y^2}{2I_z} \right) - \left[\left(\frac{K_{CH}}{2M} + \frac{K_{CV}b_x^2 + K_{CH}b_y^2}{2I_z} \right)^2 - \left(\frac{K_{CV}K_{CH}b_x^2}{I_z M} \right) \right]^{\frac{1}{2}} \right\}^{\frac{1}{2}} \quad (6)$$

**Hull, E. H., "The Use of Rubber in Vibration Isolation," Journal of Applied Mechanics, Vol. 4-5, 1937-38

††Wahl, A. M., "Mechanical Springs," Penton Publishing Company, 1944, pp. 28-29

However, if the mounting is truly of a center-of-gravity type, equations (5) and (6) will reduce to a simple equation of the form of equation (1). If the horizontal natural frequencies do not meet the frequency requirements, different assumptions must be made as to the spring height, diameter and number of turns, until by "cut and try" methods a satisfactory horizontal spring constant is obtained. An example is given in the Appendix for the purpose of demonstrating the above technique.

RESULTS AND CONCLUSIONS

This vibration isolation system comprises a solution to the problem in conformance with the limitations requested. All natural frequencies are less than 15 cps, and the transmissibility at 30 cps is less than 0.20 and becomes progressively less as the exciting frequency becomes higher (see Figure 3). The system is stable for all mounting positions of the unit. One-half inch clearance is provided on all sides of the unit. The vibration characteristics of the mounting will not vary over the range of ordinary ambient temperatures. Construction details of the mounting system are shown in Figure 4.

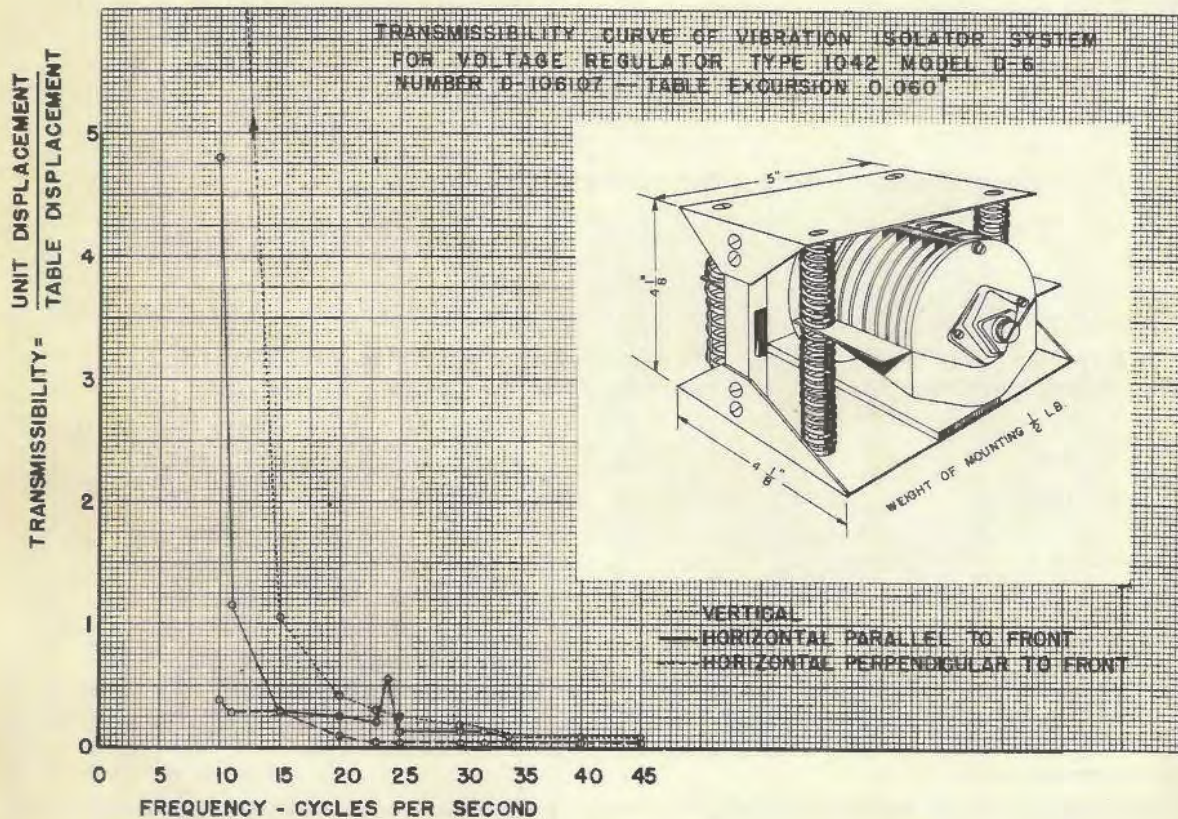
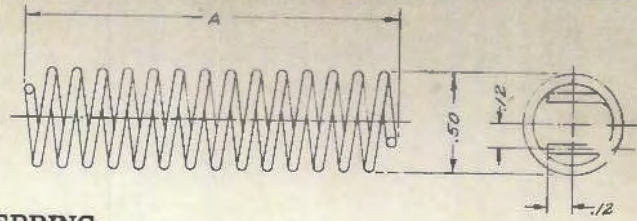
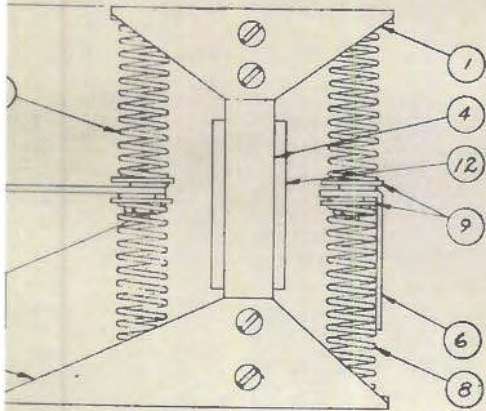


Fig. 3 - Transmissibility Curve of Voltage Regulator Mounted on Vibration Isolator System



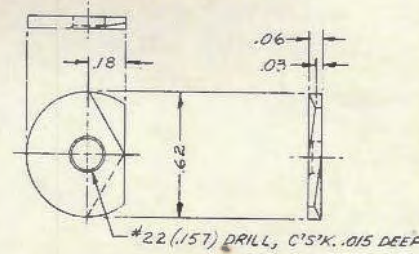
SPRING

Steel Wire No. 21(.047) Amer. S. & W. Music Wire Gage
 A = 1.812-13 Turns 4 Req'd.

⑦

A = 2.125-16 Turns 4 Req'd.

⑧

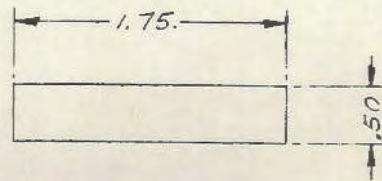


Scale: Full size
 (Approx.)

⑨

WASHER

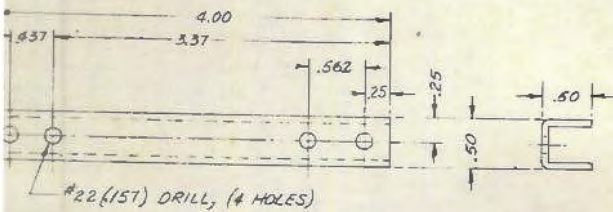
Brass (Compo B-r)
 16 Req'd.



⑫

RUBBER PAD

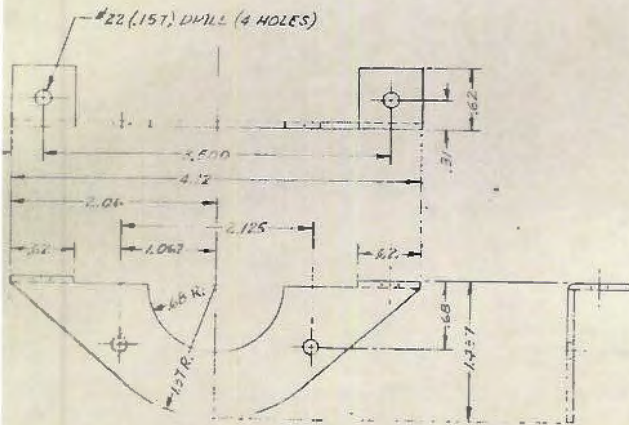
Cloth Inserted Rubber .125 Thick
 6 Req'd.



④

CHANNEL

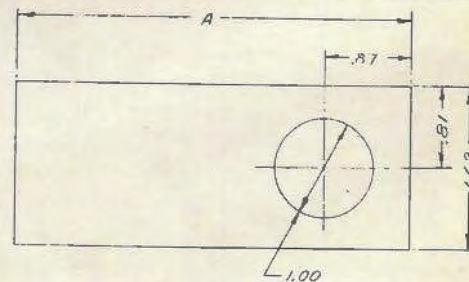
Aluminum Alloy 52s .062 Thick
 2 Req'd.



⑥

MTG. BRACKET

Aluminum Alloy 52s .062 Thick
 1 Req'd.



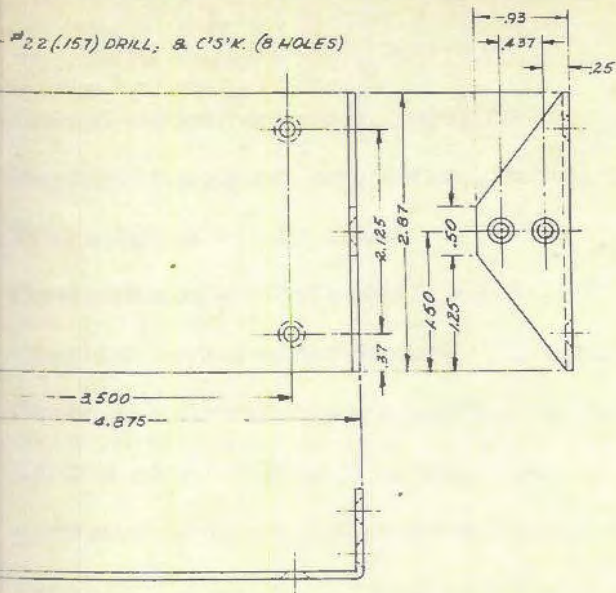
⑩

RUBBER PAD

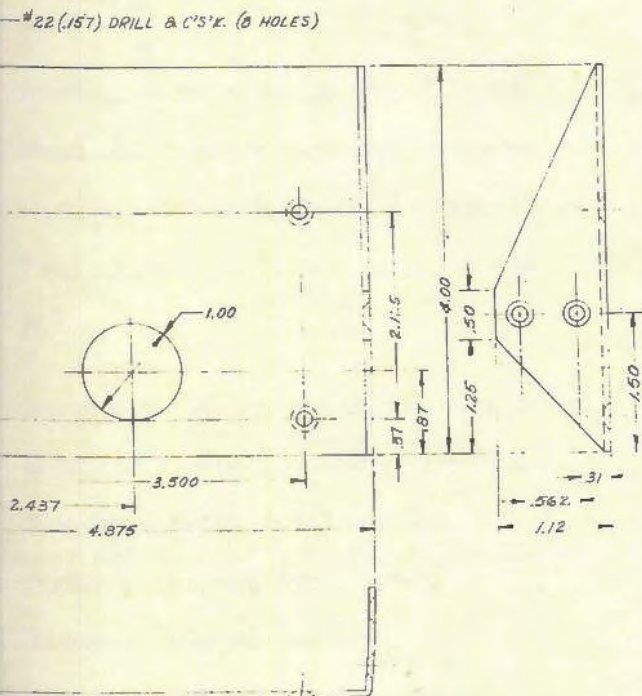
Cloth Inserted Rubber .125 Thick
 A = 4.00-1 Req'd.

⑪

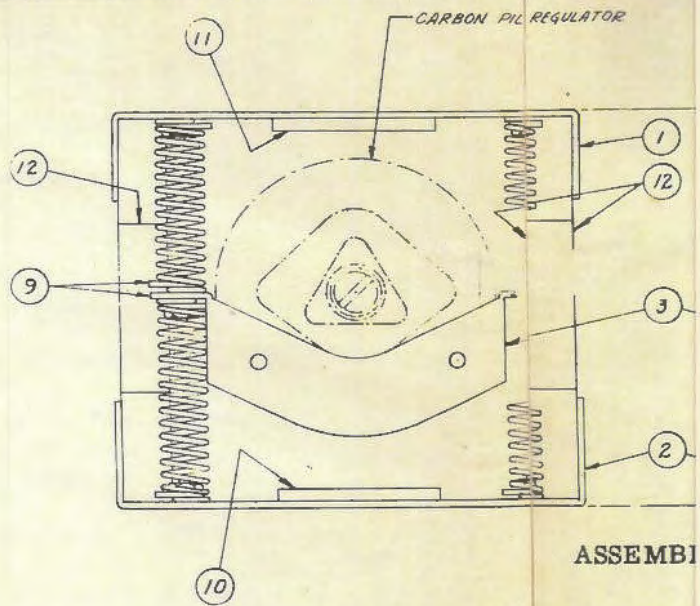
A = 2.87, Omit Hole - 1 Req'd.



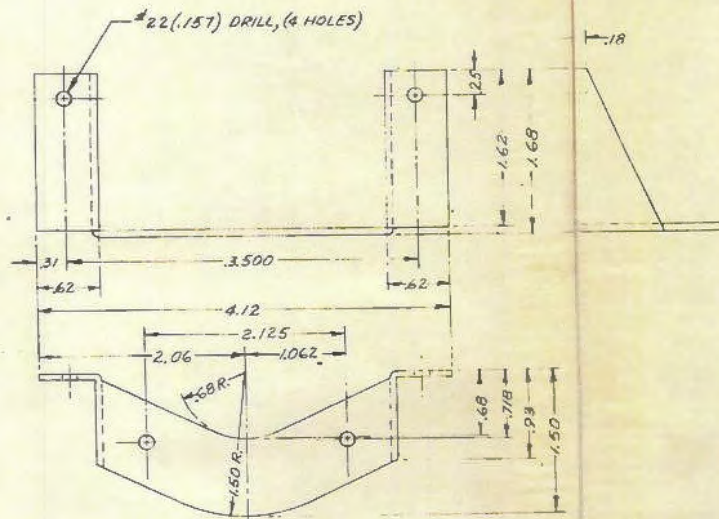
① **UPPER BRACKET**
 Aluminum Alloy 52s .062 Thick
 1 Req'd.



② **LOWER BRACKET**
 Aluminum Alloy 52s .062 Thick
 1 Req'd.



ASSEMBLY



③ **MTG. BRACKET**
 Aluminum Alloy 52s .062 Thick
 1 Req'd.

Scale: Half size
 (Approx.)

Fig. 4 - Working Drawing of Parts of

APPENDIX

Example:

Figure 5 is a diagram of the system as it is assumed to be for purposes of calculation.

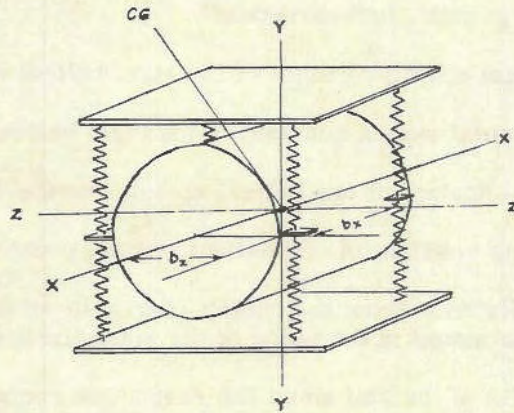


Fig. 5 - Diagram of the System Used in Making Calculations

The following example is calculated for the purpose of demonstrating the technique applying the formulas.

Assuming f_1 equal to 12 cps:

$$f_1 = \frac{1}{2\pi} \left(\frac{K_{cv}g}{W} \right)^{\frac{1}{2}}$$

$$f_1 = \frac{1}{2\pi} \left(\frac{386K_{cv}}{2.25} \right)^{\frac{1}{2}} \tag{1}$$

$$K_{cv} = 33.2 \text{ pounds/inch}$$

* * *

$$\delta = \left(\frac{3.125}{f_1} \right)^2$$

$$\delta = \left(\frac{3.125}{12} \right)^2 \tag{2}$$

$$\delta = 0.0678$$

The following assumptions are now made in order to determine the construction details of the specific springs. The space limitations are the primary factor in making these assumptions.

$$H = 2''$$

$$r = 0.25''$$

$$n = 14$$

$$G = 11.5 \times 10^6 \text{ (for music wire), pounds/inch}^2$$

$$P = \frac{W}{8} = \frac{2.25}{8} = 0.281 \text{ pounds}$$

$$d = \left(\frac{64Pr^3n}{G\delta} \right)^{\frac{1}{4}} \quad (3)$$

$$d = \left(\frac{64 \times 0.281 \times 0.25^3 \times 14}{11.5 \times 10^6 \times 0.0678} \right)^{\frac{1}{4}}$$

$$d = 0.047''$$

* * *

$$K_H = \frac{K_V D^2}{0.390 D^2 + 0.297 h^2}$$

$$K_H = \frac{16.55 \times 0.500^2}{(0.390 \times 0.500^2) + (0.297 \times 1.966^2)} \quad (4)$$

$$K_H = 3.32 \text{ pounds/inch/spring}$$

Information necessary for the calculation of the horizontal translatory resonances is now computed.

$$I_z = \frac{M}{12} (3R^2 + L^2)$$

$$= \frac{2.25}{12 \times 386} (3 \times 1.438^2 + 2.125^2)$$

$$I_z = 0.005215 \text{ inch}^4$$

$$I_x = \frac{MR^2}{2}$$

$$I_x = \frac{2.25 \times 1.438^2}{2}$$

$$I_x = 0.00605$$

This information is derived from the conditions of the mounting.

$$b_z = 1.75''$$

$$b_x = 1.0625$$

$$b_y = 0$$

As mentioned in the text, if the mounting is a true center-of-gravity type, equations (5) and (6) reduce to the form of equation (1). Since this condition is to be true of the mounting in question, the horizontal translatory resonances are calculated as follows:

$$\begin{aligned} f_2 = f_3 &= \frac{1}{2\pi} \left(\frac{K_{CHG}}{W} \right)^{\frac{1}{2}} \\ &= \frac{1}{2\pi} \left(\frac{26.6 \times 386}{2.25} \right)^{\frac{1}{2}} \end{aligned}$$

$$f_2 = f_3 = 10.75 \text{ C.P.S.}$$
