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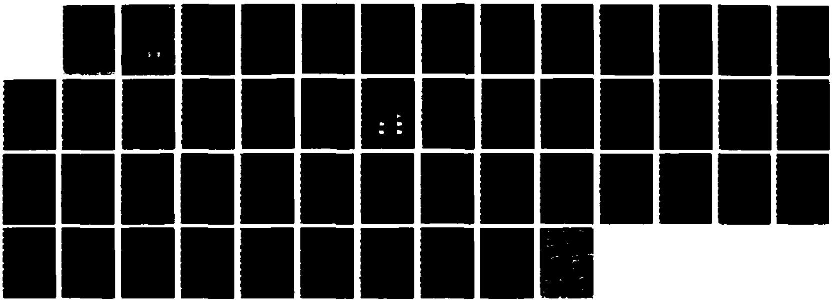

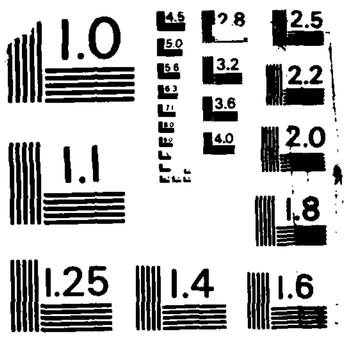

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DESIGN METHODOLOGY FOR A LIGHTWEIGHT, RESONANT-FREE PLATEN FOR VIBRATION TESTING

by Louie Jackson Lipp
PRODUCT ASSURANCE DIRECTORATE

April 1987

U.S. ARMY
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Aberdeen Proving Ground, Maryland 21010-5423

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Vibration testing, particularly when performed in reliability test chambers, is very costly. Discussed in this report are the equations for designing a resonant-free platen for holding several test specimens for simultaneous testing. The unique feature of this platen is that it is an inverted pyramid that has been truncated to mate with a shaker head. This design is less likely to have resonance problems and is significantly lighter than one of constant thickness. Specific points covered are: determination of the dimensions for a resonant-free platen, determination of the total weight of the platen, selection of a structurally damped material, and a computer program for performing all necessary calculations.

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PREFACE

The work described in this report was authorized under Project No. D601-09, ACADA Program, Increase for XM21 Contract from MICAD. This work was started in January 1984 and completed in March 1985.

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DESIGN METHODOLOGY FOR A LIGHTWEIGHT, RESONANT-FREE PLATEN FOR VIBRATION TESTING

1. INTRODUCTION

Prudent economics for reliability testing demands that as large a number of test items as possible be placed in an environmental chamber. This permits a rapid accumulation of test hours, thereby gaining high statistical confidence for a given test chamber time. However, serious difficulties can arise when vibration-induced environmental stressing is required. If frequencies are induced that excite the natural frequency of portions of the platen on which the specimens are attached, a resonant condition results. Those test specimens in the resonating area of the platen experience a much higher vibrational stress than either the test plan requires or those items attached to a nonresonating portion of the platen receive. In addition, the location of the resonating portion of the platen changes as different frequency inputs excite different spring-mass systems at their individual natural frequencies.

Naturally, if at a certain frequency, one test specimen is subjected to greater vibration stress than another, the test may provide erroneous failure data for test and reliability analysis. A problem such as this can defeat the entire purpose of the test and cause confusion as to the correct classification of a vibration-caused failure.[1]

A previous technical report[2] discussed two methods of solving or minimizing the resonance problem. One was to design the platen thick enough so that the natural frequency of a loaded corner of a platen would exceed the highest frequency excited by the shaker. If this was impossible because of a weight-displacement limitation of the shaker, then alternatives in the sandwiching of damping materials were offered as an alternate solution. After reanalyzing the contents and causes of platen resonances[3] and beam deflection parameters,[3] it appears that the design can be refined to get higher natural frequencies at highly significant reductions of weight.

This improvement can take place by removing material from outer portions of the platen, where it is a dead mass, and placing it more in the center, where it adds strength. In reality, the platen takes the form of an inverted, truncated pyramid (ITP) with a square-rectangular-parallelepiped base and top, as illustrated in Figure 1. This report will: (a) present the equations and methods for utilizing this type of design; (b) compare its mass efficiency with that of a constant thickness platen; and (c) discuss the advantages of using magnesium as the platen material. Appendix A contains the derivation of the natural frequency equation used. Appendix B contains a computer program written in Applesoft Basic to ease the burden of repetitive calculations.

2. BENDING MODES OF A RESONATING PLATEN

The Spring-Mass System.

When any structural system vibrates such that the mass for inertial force is identical to the spring (of stiffness) force, the system vibrates at its natural frequency. On a complex structure this can occur at any number of

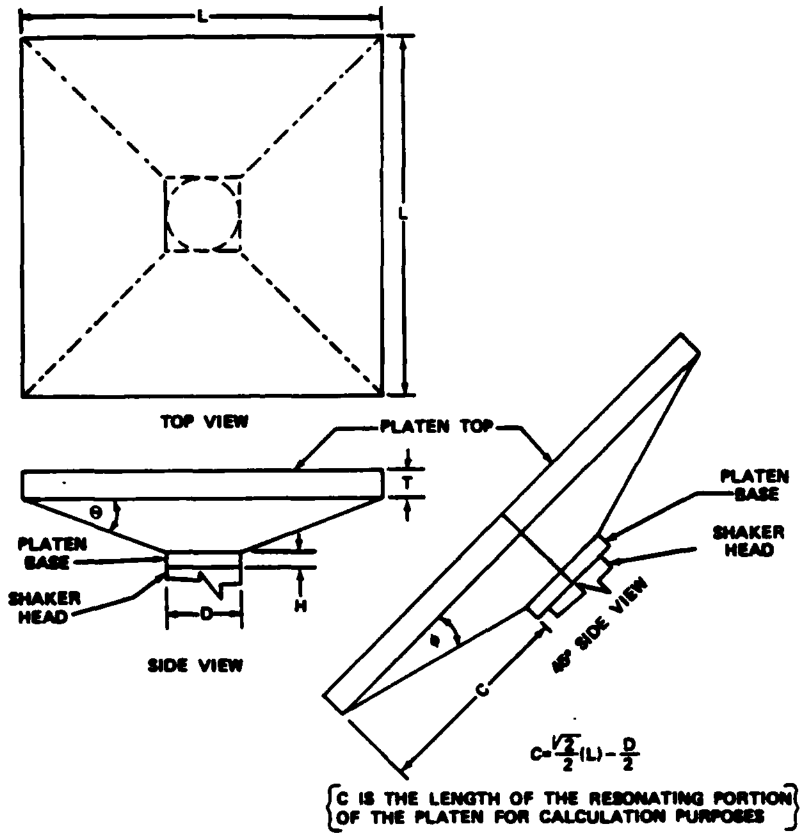


Figure 1. Inverted Truncated Pyramid Type of Platen with Square-Rectangular-Parallelpiped Base and Top and Attached to Cylindrical Shaker Head

frequencies when different components combine or separate into different subsystems, each with its own natural frequency. Experience has shown, however, that when a test platen is loaded with many items to be vibrated on a hydraulic or electrodynamic shaker, the corners will go into resonance starting at a line of bending. These lines of bending occur where there is a relatively sudden and significant increase in the platen's area moment of inertia.

3. SUDDEN STIFFNESS CAUSED BY CHANGE IN AREA MOMENT OF INERTIA

Usually test platens are fabricated as a constant thickness slab of a lightweight metal, as illustrated in Figure 2. There is a rapid increase in the section modulus where the platen bolts onto the circular shaker head, caused by both the additional thickness and higher modulus of elasticity. Although the edge of the shaker head is not a true fixed clamp, in that it fixes an entire line of bending in a vise-like grip to form a cantilevered triangular beam, the thickness of the platen forces a straight-line vibration behavior. This has been confirmed by experimental results.[2]

The ITP type of platen, as shown in Figure 1, will behave the same as the constant thickness platen when it experiences a sudden increase in area moment of inertia. However, two important physical limitations must be taken into account:

- Prudent fabrication techniques require that the base of the platen have a square cross section instead of a round cross section to mate with the shaker head.

- The shaker head is usually about an inch thick. If the slopes of the sides of the inverted pyramid were to be projected to an apex instead of truncated, it would likely project deeper than the thickness of the shaker head.

This indicates that the truncation of the ITP should then have a square base with the same thickness as the removed tip of the pyramid (the precise solid geometric name for this shape would be square-rectangular parallel piped). Also, since the base's square corner does not present a sudden or rapid increase in the area moment of inertia, the line of bending is less precise. Therefore, if the distance "C" from the platen's corner to the line of bending is assumed to be to the shaker-head's circular edge, the most conservative assumption would be used in the calculations. In other words, the actual natural frequency experienced in testing should not exceed that used in any calculations.

Platen Design Methodology

Appendix A contains the derivation of the equation for determining the natural frequency of a corner of an ITP type of platen. It is Equation A-27 on page 31:

$$f_N = 48.47\sqrt{\frac{E\tan^3\phi}{C}\left(\frac{C\rho\tan\phi + 12\dfrac{W}{A}}{54\,C^2\rho^2\tan^2\phi + 1380\,C\dfrac{W}{A}\rho\tan\phi + 8960\dfrac{W^2}{A^2}}\right)} \qquad (1)$$

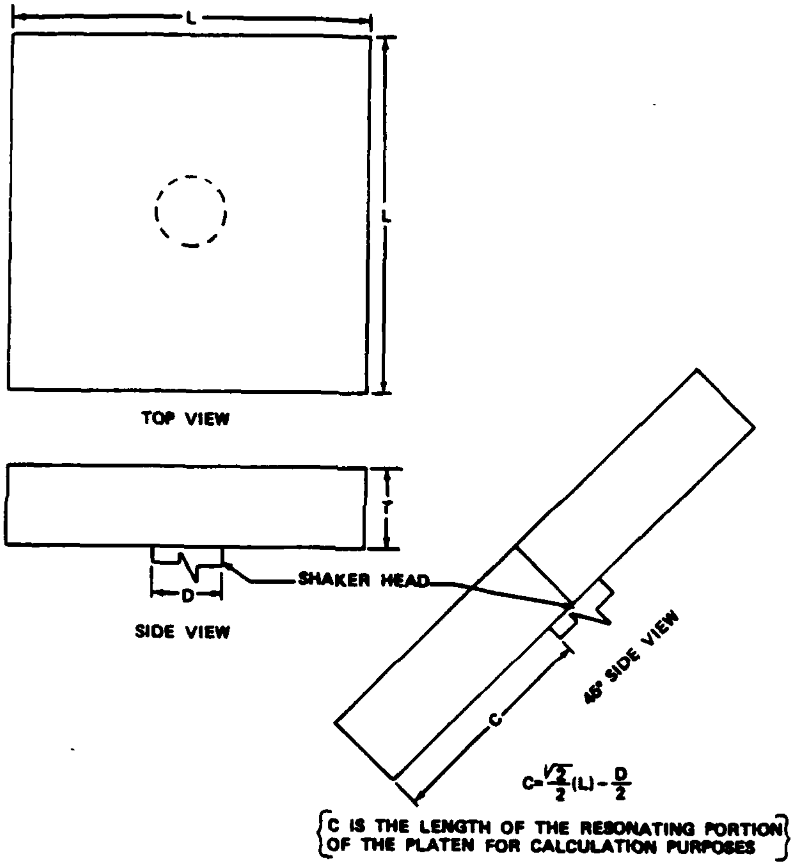


Figure 2. Illustration of Typical Flat Plate Platen Attached to Cylindrical Shaker Head

where

f_N = Natural frequency in Hertz

E = Young's Modulus of Elasticity of Platen Material, lb/sq in.

ϕ = Angle of the slope of the pyramid measured from the base along the edge formed by any two sides that come together up to the apex in degrees (Figure 1).

W = Weight of the corner-most test specimen, fixture/adapters, and bolt heads in pounds.

A = Platen area displaced by the corner-most test specimen and its fixturing in square inches.

ρ = Density of platen material in lb/cu in.

C = Distance from platen corner to shaker head in inches.

This equation will determine the natural frequency of a platen corner for various angles ϕ. (Test fixture engineers should already have determined all other inputs to the equation.) However, several very practical considerations are not taken into account by following this equation:

- The edges around the perimeter of the platen form a wedge. There will probably be insufficient material to bolt the test specimens and their respective fixtures/adaptors to the platen.

- As was discussed previously, the shaker head may not be of sufficient thickness to provide a significant enough increase in moment of inertia. If it is not, the platen will have to compensate by having its truncated portion take on a geometric shape that allows bolting onto the shaker head.

- The angle ϕ was convenient for formula derivation, but should not be used on fabrication drawings. The slope will have to be restated in terms of the angle formed by any one of the pyramid's sides with respect to the base. This conversion is simply:

$$\theta = \text{ARCTAN} \quad (\sqrt{2}\tan\phi) \qquad (2)$$

The problem of the corners of the platen forming a wedge that is too thin can be solved by adding a mass of constant thickness to the top of the platen. This was not taken into account during the original derivation, because there is a transition of the neutral axis from the top constant thickness portion of the platen to the variable thickness portion when the line of bending moves from the corner towards the shaker head. The differential equation becomes extremely unwieldy, and the final equation would be several magnitudes larger than Equation (1). It was decided that a short iteration process would perfom the job faster and easier, be almost as accurate (any errors would be on the conservative side, i.e., a natural frequency slightly higher than the calculations indicate), yet much lighter than a platen of constant cross section. The computer program in Appendix B performs all of the calculations necessary to design the platen. The program performs in the following order.

a. It steps through increasingly larger angles of ϕ using Equation (1) until the natural frequency is equal to, or just exceeds, the maximum frequency required by the test plan (Figure 3A).

b. The program then adds a constant thickness mass to the top of the platen to ensure sufficient thread depth for mounting test specimens. This will lower the natural frequency below that desired because of the added mass (Figure 3B).

c. Next, the program calculates a new value for the distance from the corner of the platen to the shaker head by projecting the angle ϕ to form a new wedge with the added mass (Figure 3C). This will drop the natural frequency again.

d. Using Equation (1) again, the program steps through increasingly greater values of ϕ until the natural frequency is equal to or slightly greater than that of the test plan (Figure 3D).

e. The program now moves the distance from the corner of the platen back to where it was (paragraph b above) with the top constant thickness mass, but with the value of ϕ as newly calculated in paragraph d. This will raise the natural frequency slightly above that required by the test plan.

f. The program now computes a new angle, Θ for fabrication. It is the angle the sides of the pyramid make with the base (Figure 3E).

g. The minimum thickness of the mating surface of the platen to the shaker head is calculated by projecting the angle Θ to the center of the platen. The distance from the platen's top surface to this projected point is the recommended thickness of the platen at the mating surface (Figure 3F). This will assure that the line of bending will occur where the calculations assume it will, at the point where the platen's diagonal intersects the shaker head.

h. Finally, it calculates the estimated platen weight. If the platen weight, when added to the specimen and fixture weight, exceeds that which the shaker armature can support for the maximum displacement required, then either a new layout with less test specimens should be considered or a platen material of high damping should be chosen. If the latter route is taken, the calculations should be rerun with a lower natural frequency until an acceptable weight is reached.

If the computer program is used for the above set of calculations, the display on the computer screen will show the side and bottom views of the platen with all critical dimensions. At the bottom of the screen is a window showing all of the critical dimensions required to fabricate the item, with the exception of the specimen mounting holes. These should have been determined in advance before the computer program was run.

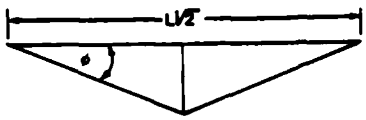


Figure 3A. Side View, 45°, Showing the Calculated Value of φ for a Given Natural Frequency for the First Iteration*

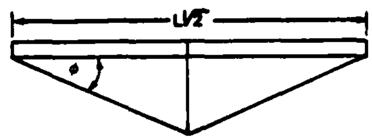


Figure 3B. Side View, 45°, Showing the Addition of Material to the Top Surface for Thread Depth for Bolting Test Specimens to Platen*

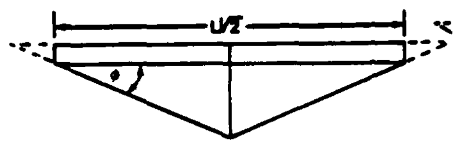


Figure 3C. Side View, 45°, Showing the Projection of the Angle φ Up to the Plane of the Top Surface to Form a New Inverted Pyramid*

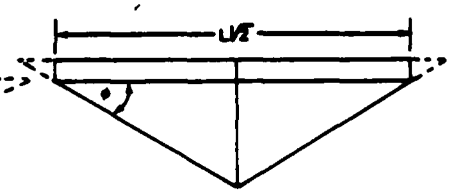


Figure 3D. Side View, 45°, Showing the New Larger Calculated Value of φ for the Given Natural Frequency for the Second Iteration*

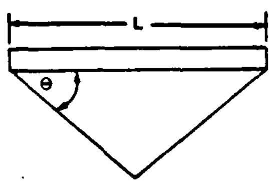


Figure 3E. Side View Showing the True Angle θ Formed by the Inverted Pyramid's Sides and Its Base

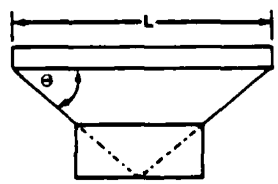


Figure 3F. Side View Showing the Formation of a Base to Mate with the Shaker Head with a Thickness Equal to the Projected Pyramid Tip

*NOTE: Refer to Figure 1 for a detailed illustration and explanation of the 45° side view.

4. SPECIMEN DAMPING CAPACITY

Background.

Damping is the term given for the conversion of strain energy to heat energy when the strain energy results from the kinetic energy bending the mechanical system. Damping results from the intermolecular friction that is experienced when there is alternating compression/tension strain from vibration. Without damping, every mechanical device that is excited at its resonant frequency would fail from metal fatigue. This is because the spring force is equal in magnitude and in the opposite direction from the mass force, thereby putting the two forces in dynamic equilibrium. The only structural force available to oppose the vibrating input forces (which are neither a spring force or a mass force, but are external to the mechanical system) would be the damping force, which is the product of a damping constant multiplied by the sinusoidal velocity of the vibration input.[4].

Although the actual damping constant, or the damping force, may possibly be of interest in specific applications, the specific damping capacity (SDC) is of greater interest to the designer of test fixtures. When it is not possible to design a platen that will not experience resonance somewhere within the vibration spectrum of the test plan, it becomes necessary to select a material that has a reasonable amount of damping and yet not compromise other important qualities such as density, modulus of elasticity, or machinability. Therefore, when selecting material, there is a compromise among relative damping, relative density, the modulus of elasticity/density ratio, and relative machinability.

5. SPECIFIC DAMPING CAPACITY

The specific damping capacity (SDC) has units of percent per cycle (%/cyc). The value of the SDC increases as the vibration-induced stress in the material increases. Therefore, anytime an SDC number is quoted, a stress level is quoted also. Since the sole objective of a vibration platen is to simply hold bolted test specimens while in a vibration environment, you can reasonably expect the lowest stress level values of the SDC to be acceptable.

Close examination of Figure 4 shows that a <u>rough estimate</u> of the impact of the SDC on the vibration amplitude can be <u>calculated by using an inverse proportion</u>. An example of this would be Equation 3:

$$X_1\ (SDC_1) = X_2\ (SDC_2) \tag{3}$$

where

X_1 = The relative amplitude at resonance of a platen mode of material #1

X_2 = The relative amplitude at resonance of a platen made of material #2

SDC_1 = Specific damping capacity of material #1

SDC_2 = Specific damping capacity of material #2

REFRENCE: Stephen C. Erickson, Magnesium's High Damping Capacity for Automotive Noise and Vibration Attenuation, May, 1975. By Permission.

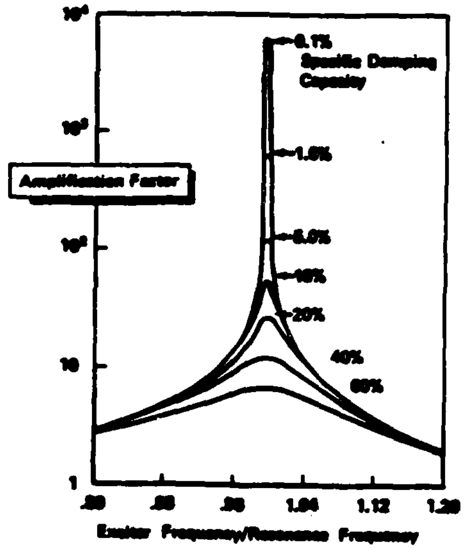


Figure 4. Influence of Specific Damping Capacity on Theoretical Resonance Amplification Factor

If you were to pick a reference material from a past test run with a logged vibrational amplitude at resonance, you could easily substitute the SDC of any other material to determine its probable vibrational amplitude. However, a ratio of X_2 to X_1, where X_1 would be the reference material, would suffice just as well.

6. MAGNESIUM AS A PLATEN MATERIAL

Background.

When the test program requires vibration frequencies higher than test economics or physical constraints will permit a resonance-free platen design, then the selection of fabrication material with a high SDC becomes important. Two structural materials that offer SDC's sufficiently high for a designer's consideration are usually considered, iron and magnesium. Equation (1) illustrates the importance of a high elastic modulus to density ratio in any material selection. The higher this ratio, the higher the resonant frequency. Also, since the total weight suspended on the armature of the electrodynamic shaker greatly affects the system's displacement (this is most important at very low frequencies), platen density becomes critical. In other words, the lighter the material, the higher the displacement of oscillation.

Table 1 compares the densities and the elastic moduli to density ratios of two classes of cast iron to that of cast magnesium. Iron is about four times more dense than magnesium, and the elastic modulus to density ratio can be as low as less than half that of magnesium. Both of these qualities make cast iron ideal for stationary machinery or vehicles where traction and vibration damping are important considerations. However, a platen for vibration testing requires the opposite qualities, and magnesium is the wiser choice.

7. WHICH MAGNESIUM ALLOY SHOULD BE USED?

The fixture designer can choose the magnesium for a platen from the alloys available, the fabrication method (such as sand cast or wrought), and heat treat methods. Extensive research into the effects of these metallurgical factors in relation to the SDC has provided these significant findings:

- The SDC has an inverse relationship to alloy content; i.e., the lower the alloy content, the higher the SDC.

- Cast magnesium, which provides no grain orientation, has the highest SDC of any fabrication method. Any working of the magnesium will have adverse effects on its use as a good vibration damping material.

- Heat treating to increase the tensile yield strength will lower the SDC.[5]

Figure 5 illustrates the alloying effect on the SDC of magnesium and other structural materials. It also shows how changing the bending stress levels changes the SDC. Particularly notice:

- How poor a material aluminum is. Because of its cost and availability, aluminum has long been used as a platen material, thereby overstressing test components greatly at resonance.

- How high the SDC is for both unalloyed magnesium and K1A magnesium, but the latter is able to withstand much higher stress levels.[4]

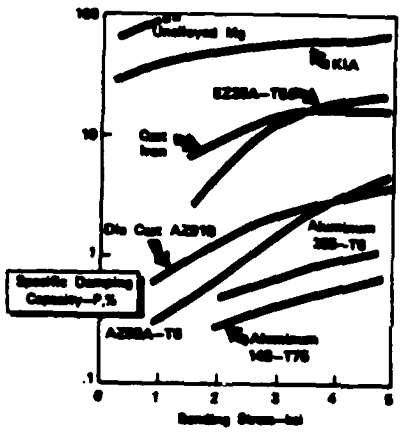
Figure 6 shows oscilloscope traces of several sand-cast magnesium and aluminum alloys that have been allowed to decay in their vibration levels after receiving the same initial excitation. Magnesium K1A has a dramatic effect when compared with the other materials.[5]

The K1A Magnesium Alloy

As may be suspected, K1A is a magnesium with only one alloy. Its chemical composition is shown in Table 2. The very low amount of the single alloy, zirconium, is sufficient to bring the yield tensile stress up from a little over 1,000 psi for unalloyed magnesium to a minimum of 6,000 psi for the K1A-type of magnesium (Table 3). This yield stress level can be expected to more than adequately handle any anticipated streses from a vibration test. If, however, there is any reason to suspect the bending stress level in the platen, a simple stress calculation at the highest programmed "g" loading can be performed. If the stresses approach or exceed 5,000 psi, the computer program found in Appendix B should be rerun with a greater tread depth of the top portion of the platen.

Table 1. Comparison of Density and Elastic Modulus for Iron and Magnesium[3,6]

Material	Density	Tensile elastic modulus of elasticity	Modulus of elasticity/density
	lb/in.^{3}	X10^6 psi	X10^6 in.
Grey iron	0.278	13	46.76
Malleable iron	0.278	25	89.93
Cast magnesium	0.0650 to 0.0665	6.5	100.00 to 97.74



REFRENCE: Stephen C. Erickson, Magnesium's High Damping Capacity for Automotive Noise and Vibration Attenuation, May, 1976. By Permission.

Figure 5. Damping Capacity of Some Sand Cast Alloys

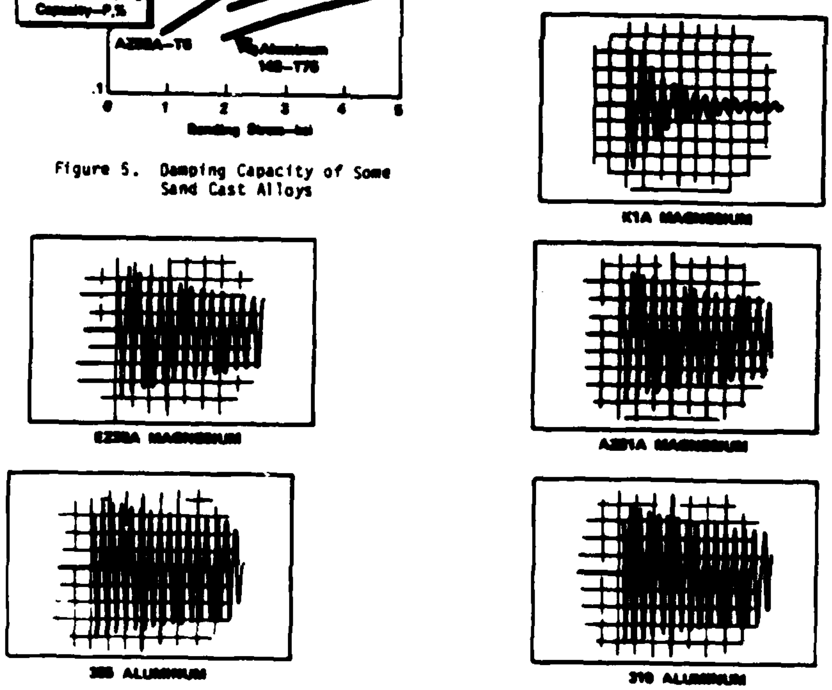


Figure 6. Oscillograms Showing Damping Capacity of Several Sand Cast Magnesium and Aluminum Alloys—Test Conditions Constant

Table 2. Chemical Composition of K1A Magnesium[7]

Element	Percent
Zirconium	0.40 - 1.0
Total, other elements	0.30 Max
Magnesium	Remainder

Table 3. Mechanical Property Requirements for Separately Cast Specimens[7]

Alloy and temper	Minimum tensile strength	Yield strength at 0.2% offset or at extension under load indicated		Elongation in 2 inches minimum
	psi	Minimum psi	Extension under load	Percent
K1A-F	24,000	6,000	0.0029	14

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APPENDIX A

A DERIVATION OF THE NATURAL FREQUENCY EQUATION FOR VARIABLE THICKNESS PLATENS

1. INTRODUCTION

When it resonates, each corner of a shaker's inverted, truncated pyramid behaves as a cantilevered, variable thickness, right triangular plate as illustrated in Figure 1 of the text. If loading is equal on all four corners of the test specimen, they will resonate at the same frequency. If the loading is unequal, the corners will not only resonate at different frequencies, but the cantilevered line from which the bending originates may shift as well. Therefore, any generalized equation must be in terms of (a) the distance from the corner of the platen to the line where bending begins; (b) equivalent mass loading; (c) area moment of inertia and; (d) Young's modulus of elasticity.

2. RALEIGH METHOD OF NATURAL FREQUENCY CALCULATION

The Raleigh method of natural* frequency calculation has been found to produce values that agree closely with experimental findings if accurate assumptions are made of the physical characteristics of the vibrating system. The equation for this method is:

$$\omega = \sqrt{g \frac{\int Y \, dx}{\int Y^2 \, dx}} \tag{A-1}$$

where

ω = Natural frequency, radians/sec

Y = Equation for the deflection of the vibrating system, inches

g = Acceleration due to gravity, which is a constant of 386.4 in./sec^2

The deflection is found by solving the fourth-order differential equation:**

$$\frac{d^4 Y}{dX^4} = \frac{W_T}{EI} \tag{A-2}$$

*Hansen, H.M., and Citenea, Paul F. Mechanics of Vibration, John Wiley and Sons, New York, NY. 1952.

**Popov, E.P. Mechanics of Materials, Printice-Hall, Inc., New York, NY. 1952.

where:

W_T = total loading on the vibrating system, lb/in.

E = Young's Modulus of Elasticity, lb/in.^{2}

I = Area moment of inertia, in.^{4}

The primary problem in solving Equation A-2 is determining the generalized expressions of W_T and I for a specimen loaded, inverted truncated pyramid, right triangular, cantilevered plate. The moment of inertia will be calculated first.

Equations for Moment of Inertia and Axis of Bending

Following is the equation for the moment of inertia of a right-triangular plate with an isosceles triangular cross section where the two angles angles are and the thickness changes "x tan" as shown in Figures A-1A and A-1B.

a. Moment of Inertia.

$$I = \frac{(\text{width})(\text{thickness})^3}{36}$$

where:

$$\text{Width} = 2x$$

$$\text{Thickness} = x \tan \phi$$

$$I = \frac{(2x)(x \tan \phi)^3}{36}$$

$$I = \frac{x^4 \tan^3 \phi}{18} \qquad \text{(A-3)}$$

To simplify for future calculation

$$\text{Let } D = \frac{\tan^3 \phi}{18} \qquad \text{(A-4)}$$

$$\text{then } I = Dx^4 \qquad \text{(A-5)}$$

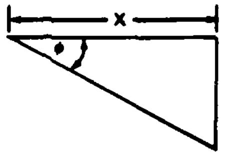

Figure A-1A. Geometry of the Triangle Formed by the Intersection of Two Planes of a Pyramid

The distance X is measured from the inverted pyramid's corner towards its center. The angle ϕ is measured from the base to the line of intersection.

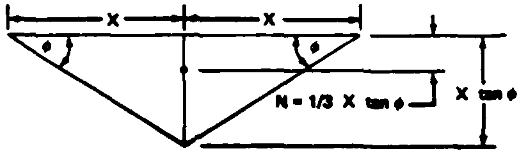

Figure A-1B. Geometry of the Cross-Section of the Inverted Pyramid at Distance X from the Corner and Perpendicular to Figure A-1B

b. Neutral Axis of Bending. The centroid is located halfway between the left and right edges of the platen (along the diagonal) and one-third of the way down from the top surface; therefore,

$$N = (1/3)\ x \tan \phi \qquad \text{(A-6)}$$

Calculation of Load

a. Platen Load. The cross-sectional area varies as the distance "x" from the corner increases. The change in cross-sectional area is one-half the function of both the width and thickness:

$$d\ \text{area} = 1/2\ (d\ \text{width}) \times (d\ \text{thickness})$$

$$\iint da\ \text{area} = 1/2 \iint (2dx)(\tan \phi\ dx)$$

$$\iint da\ \text{area} = \tan \phi \iint dx2$$

At $x = 0$, all constants of integration $= 0$

then $a = 1/2\ x\ ^2\tan \phi$ (A-7)

This, then is the cross-sectional area at any distance "x" from the platen corner. To get the distributed platen load, simply multiply the area by the density ρ and obtain:

$$Wp = 1/2\ \rho\ x^2 \tan\phi \qquad \text{(A-8)}$$

b. Specimen Load. The natural frequency of a cantilever is most affected by masses that are furthest from the edge of bending. The greater the mass and the further it is from this edge, the lower the natural frequency. This is because the natural frequency is related to the deflection; therefore, to simplify the deflection equation, one can assume that the weight per displaced unit area of the corner-most test item is the same for the entire triangular plate being considered. (Test prudence would dictate, wherever possible, that the lightest items per square inch go at the extreme corners of the platen to get the highest natural frequency). Include the weight of the test item, the adapters (if any), and the bolt heads to obtain the most accurate estimate of the cantilever's loading. This total specimen weight shall be designated as "W_x." Next, calculate the area displaced on the platen by the test specimen and all associated hardware and designate this area as "A." This provides a load pressure of W/A in pounds per square inch for the corner-most position on the test platen. What is now needed is the equivalent distributed load as the right triangular area expands from the corner to the line of bending. This load increases at a rate of "2x" since the width expands at twice the rate as the diagonal distance from the platen corner increases. The specimen load becomes:

$$Wp = 2\frac{W}{A}x \qquad \text{(A-9)}$$

c. Total Platen Load. The total platen load is the addition of the specimen load, Equation A-9, to the platen load, Equation A-8. It now becomes:

$$W_T = W_x + Wp$$

$$W_T = 2\frac{W}{A}x + 1/2\ \rho\ x^2 \tan\phi \qquad \text{(A-10)}$$

To simplify for future number manipulation:

$$\text{let } B = 2\frac{W}{A} \qquad \text{(A-11)}$$

$$\text{and } G = 1/2\ \rho \tan \phi \qquad \text{(A-12)}$$

$$\text{then } W_t = Bx + Gx^2 \qquad \text{(A-13)}$$

With both the load and moment of inertia equations developed, the deflection can now be derived.

Calculation of Deflection

From Equation A-2 we get the equation for deflection:

$$\frac{d^4Y}{dx^4} = \frac{WT}{EI}$$

The solution to this fourth order differential equation, fortunately, is four easy steps of integration:

$$S = \int W_T\,dx + S_0 \qquad \text{(A-14)}$$

$$M = \int S dx + M_0 \qquad \text{(A-15)}$$

$$\Psi = \int \frac{M}{EI}\,dx + \Psi_0 \qquad \text{(A-16)}$$

$$Y = \int \Psi dx + Y_0 \qquad \text{(A-17)}$$

where:

S = shear load, lb

M = moment load, in.-lb

Ψ = slope, rad

Y = deflection, in.

The mathematical work will follow these steps with integration starting from the free end and going to the fixed end, as shown in Figure A-1.

Shear Calculation

From Equation A-14

$$S = \int W_T \, dx + S_o$$

From Equation A-13

$$W_T = Bx + Gx^2$$

then

$$S = \int (Bx - Gx^2)dx + S_o$$

$$S = 1/2\ Bx^2 + (1/3)GX^3 + S_o$$

$$\text{at } X = 0,\ S_o = 0$$

therefore

$$S = 1/2Bx^2 + (1/3)Gx^3 \qquad \text{(A-18)}$$

Moment Calculation

From Equation A-15

$$M = \int Sdx = M_o$$

Substitute the value of S from Equation A-18, into the above equation:

$$M = \int (1/2Bx^2 + (1/3)GX^3)dx + M_o$$

$$M = (1/6)Bx^3 + (1/12)GX^4 + M_o$$

$$\text{at } X = 0,\ M_o = 0$$

Therefore:

$$M = (1/6)Bx^3 + (1/12)GX^4 \qquad \text{(A-19)}$$

Slope Calculation

From Equation A-16

$$\Psi = \int \frac{M}{EI} dx + \Psi_0$$

From substituting Equations A-5 and A-19 into Equation A-16, we get:

$$\Psi = \int \frac{1/6Bx^3 + 1/12Gx^4}{ED\ x^4} dx + \Psi_0$$

$$\Psi = \frac{B}{6\ ED} \ln x + \frac{G}{12\ ED} x + \Psi_0$$

At $X = C$ (see Figures 2 and 3), $\Psi = 0$

Then

$$\Psi_0 = -\left(\frac{B}{6\ ED} \ln C + \frac{GC}{12\ ED}\right)$$

$$\Psi = \frac{B}{6\ ED} \ln x + \frac{G}{12\ ED} x - \left(\frac{B}{6\ ED} \ln C + \frac{GC}{12\ ED}\right) \qquad \text{(A-20)}$$

Deflection Calculation

From Equation A-17:

$$Y = \Psi\ dx + Y_0$$

Substitute in the value from Equation A-20 and we get

$$Y = \int \left(\frac{B}{6ED} \ln x + \frac{G}{12ED} x - \frac{B}{6ED} \ln C - \frac{GC}{12ED}\right) dx + Y_0$$

$$Y = \frac{xB}{6ED} \ln x - \frac{B}{6ED} x + \frac{G}{24ED} x^2 - \frac{B \ln C}{6ED} x - \frac{GC}{12ED} x + Y_o$$

$A_T \quad x = C, \; Y = 0$

then

$$Y_o = -\frac{CB}{6ED} \ln C + \frac{BC}{6ED} - \frac{GC^2}{24ED} + \frac{BC}{6ED} \ln C + \frac{GC^2}{12ED}$$

$$Y_o = \frac{BC}{6ED} + \frac{GC^2}{24ED}$$

$$Y_o = \frac{BC}{6ED} \ln x - \frac{B}{6ED} x + \frac{G}{24ED} x^2 - \frac{B \ln C}{6ED} x - \frac{GC}{12ED} x + \frac{BC}{6ED} + \frac{GC^2}{24ED}$$

$$Y = \frac{1}{24ED} \left(4 \; x \; B \ln x - 4Bx + Gx^2 - 4Bx \ln C - 2G \, Cx + 4BC + GC^2 \right) \quad \text{(A-21)}$$

This completes the derivation of the deflection equation for a right triangular plate with an isosceles triangular cross section.

Calculation of Natural Frequency

Equation A-1 provides the Raleigh equation for determining the natural frequency of a vibrating system. It is:

$$\omega = \sqrt{Q \frac{\int Y dx}{\int Y^2 dx}}$$

This requires an additional integration step of the numerator and an additional integration step after squaring in the denominator. To prevent confusion in the calculations, let's set:

$$k = \int Y dx \quad \text{(A-22)}$$

and

$$j = \int Y^2 dx \quad \text{(A-23)}$$

First, let's solve for "k" by substituting Equation A-21 into Equation A-22:

$$k = \frac{1}{24ED}\int_o^c \left(4\ Bx\ \ln x - 4Bx + Gx^2 - 4Bx\ \ln C - 2GCx + 4BC + GC^2\right) dx$$

$$k = \frac{1}{24ED}\left[4B\left(x^2\left(\frac{\ln x}{1+1} - \frac{1}{(1+1)^2}\right)\right) - 2Bx^2 + 1/3Gx^3 - 2Bx^2\ \ln C - GC_x{}^2 + 4BCx + GC^2x\right]_o^c$$

$$k = \frac{1}{24ED}\left[2Bx^2\ \ln x - 2Bx^2 + 1/3Gx^3 - 2Bx^2\ \ln C - GCx^2 + 4BCx + GC^2x\right]_o^c$$

$$k = \frac{1}{24ED}\left(2BC^2\ \ln C - 3BC^3 + 1/3GC^3 - 2BC^2 \ln C - GC^3 + 4BC^2 + GC^3\right)$$

$$k = \frac{1}{24ED}\left(1/3GC^3 + BC^2\right)$$

$$K = \frac{C^2}{24ED}\left(1/3\ GC + B\right) \qquad \text{(A-24)}$$

Now, solve for "j" by substituting Equation A-21 into A-23:

$$j = \left(\frac{1}{24ED}\right)^2 \int \left(4\ x\ B\ \ln x - 4Bx + Gx^2 - 4Bx \ln C - 2GCx + 4BC + GC^2\right)^2 dx$$

$$j = \left(\frac{1}{24ED}\right)^2 \int \Big(16\ B^2x^2(\ln x)^2 - 32B^2x^2 \ln x + 8BGx^3 \ln x - 32B^2x^2\ \ln C$$

$$\ln x - 16BCGx^2\ \ln x + 32B^2\ C\ x\ \ln x + 8BC^2Gx\ \ln x + 16B^2x^2 - 8BGx^3$$

$$+ 32B^2x^2\ \ln C + 24BCGx^2 - 32B^2Cx - 24BC^2Gx + G^2x^4 - 8BGx^3\ \ln C$$

$$- 4CG^2x^3 + 6C^2G^2x^2 + 16B^2x^2(\ln C)^2 + 16BCGx^2\ \ln C - 32B^2Cx\ \ln C$$

$$-8BC^2Gx \ln C - 4G^2C^3x + 16B^2C^2 + 8\ BC^3G + G^2C^4\Big)dx$$

$$j = \left(\frac{1}{24ED}\right)^2 \left[\frac{56}{27} B^2C^3 + \frac{23}{18} BC^4G + \frac{1}{5} C^5G^2\right]$$

$$j = \left(\frac{1}{24ED}\right)^2 C^3\left(\frac{56}{27}B^2 + \frac{23}{18} BCG + \frac{1}{5} C^2G^2\right) \qquad \text{(A-25)}$$

By substituting Equation A-24 into A-22 and A-25 into A-23 and both of these into Equation A-1, we get:

$$\omega = \sqrt{g \frac{\left(\frac{1}{24ED}\right)C^2\left(\frac{1}{3} CG + B\right)}{\left(\frac{1}{24ED}\right)^2 C^3\left(\frac{1}{5} C^2G^2 + \frac{23}{18} BCG + \frac{56}{27} B^2\right)}}$$

$$\omega = \sqrt{\left(\frac{24EDg}{C}\right) \frac{\frac{1}{3} CG + B}{\frac{1}{5} C^2G^2 + \frac{23}{18} BCG + \frac{56}{27} B^2}}$$

The natural frequency is usually expressed as:

$$f_N = \frac{1}{2\pi} \omega$$

Simplifying, we get:

$$f_N = 15.3265 \sqrt{\frac{ED}{C}\left(\frac{\frac{1}{3} CG + B}{\frac{1}{5} C^2G^2 + \frac{23}{18} BCG + \frac{56}{27} B^2}\right)} \qquad \text{(A-26)}$$

Substituting Equations A-4, A-11, and A-12 into Equation A-26, we get:

$$f_N = 15.3265\sqrt{\frac{E\tan^3\phi}{18\,C}\left(\frac{\frac{1}{6}C\rho\tan\phi + 2\frac{W}{A}}{\frac{1}{20}C^2\rho^2\tan^2\phi + \frac{23}{18}C\frac{W}{A}\rho\tan\phi + \frac{224}{27}\frac{W^2}{A^2}}\right)} \qquad \text{(A-26)}$$

$$f_N = 3.61249\sqrt{\frac{E\tan^3\phi}{C}\left[\frac{\frac{1}{6}C\rho\tan\phi + 12\frac{W}{A}}{\frac{1}{1080}\;54\,C^2\rho^2\,\phi + 1380\,C\frac{W}{A}\rho\tan\phi + 8460\frac{W^2}{A^2}}\right]}$$

$$f_N = 48.47\sqrt{\frac{E\tan^3\phi}{C}\left(\frac{C\rho\tan\phi + 12\frac{W}{A}}{54\,C^2\rho^2\tan^2 + 1380\,C\frac{W}{A}\rho\tan\phi + 8960\frac{W^2}{A^2}}\right)} \qquad \text{(A-27)}$$

This, then, is the final equation for determining the natural frequency of a corner of a truncated inverted pyramid. This equation is found in the main text of this report as Equation 1.

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APPENDIX B

COMPUTER PROGRAM FOR DETERMINATION OF OPTIMUM DESIGN PARAMETERS

This program determines the optimum design parameters for an inverted truncated pyramid type of platen for holding test specimens during vibration testing.

L.J. LIPP

USAAMCCOM
Product Assurance Directorate
For Chemical Systems
1985

THE PROGRAM REQUIRES THAT THE FIXTURE DESIGNER COMPLETE A LAYOUT OF THE TOP OF A SQUARE VIBRATION PLATEN BEFORE GETTING ON THE COMPUTER. THE DESIGNER IS EXPECTED TO HAVE:

1. ARRANGED ALL OF THE TEST SPECIMENS SO THAT THOSE WITH THE LEAST WEIGHT (INCLUDING FIXTURE ADAPTERS AND BOLTS) PER DISPLACED PLATEN AREA ARE AT THE FOUR CORNERS;

2. THE TEST SPECIMENS AS CLOSE TOGETHER AS POSSIBLE, YET WITH ROOM FOR WRENCHES, POWER AND SIGNAL CABLES, ETC. TO KEEP THE PLATEN'S SQUARE DIMENSIONS AS SHORT AS POSSIBLE;

3. SELECTED THE PLATEN MATERIAL AND TO KNOW ITS YOUNG'S MODULUS OF ELASTICITY AND DENSITY;

4. DETERMINED THE MINIMUM DEPTH REQUIRED FOR THE BOLT THREADS TO FASTEN THE TEST SPECIMENS AND FIXTURES DOWN TO THE PLATEN.

HAVE YOU COMPLETED THESE PRELIMINARY DESIGN DETAILS? <Y OR N>
IMPORTANT NOTE: ALL RESPONSES TO THE FOLLOWING QUESTIONS ARE TO BE IN DECIMAL FORM, NOT FRACTIONS.

```
WHAT IS THE LENGTH OF A SIDE OF YOUR
SQUARE PLATEN IN INCHES?
?42

WHAT IS THE DIAMETER OF THE SHAKER HEAD
IN INCHES?
?18

WHAT IS THE MINIMUM MATERIAL THICKNESS
REQUIRED TO BE TAPPED TO BOLT THE TEST
SPECIMENS TO THE PLATEN IN INCHES?
?.5

WHAT IS THE MAXIMUM FREQUENCY REQUIRED
IN THE TEST PROGRAM IN HERTZ?
?500

WHAT MATERIAL HAVE YOU CHOSEN FOR THE
PLATEN?
?MAGNESIUM KIA

WHAT IS MAGNESIUM KIA'S YOUNG'S MODULUS
OF ELASTICITY IN POUNDS PER SQUARE INCH?
?7000000

WHAT IS MAGNESIUM KIA'S DENSITY IN
POUNDS PER CUBIC INCH?
?0.07

ASSIGN EACH CORNER OF THE PLATEN A
NUMBER SO THE WEIGHTS OF THE AREAS DIS-
PLACED BY THE TEST SPECIMENS AND THEIR
FIXTURING CAN BE IDENTIFIED BY THEIR
LOCATION ASSIGNMENT.


WHAT IS THE WEIGHT OF THE TEST
SPECIMEN, FIXTURING, BOLTS, WASHERS,
ETC. IN CORNER 1?
?11

WHAT IS THE AREA DISPLACED ON THE
PLATEN BY THE TEST SPECIMEN, FIXTUR-
ING, ETC. IN CORNER 1?
?11

THE CORNER'S LOAD IS 1 PSI
```

```
WHAT IS THE WEIGHT OF THE TEST
SPECIMEN, FIXTURING, BOLTS, WASHERS,
ETC. IN CORNER 2?
?11

WHAT IS THE AREA DISPLACED ON THE
PLATEN BY THE TEST SPECIMEN, FIXTUR-
ING, ETC. IN CORNER 2?
?11

THE CORNER'S LOAD IS 1 PSI

WHAT IS THE WEIGHT OF THE TEST
SPECIMEN, FIXTURING, BOLTS, WASHERS,
ETC. IN CORNER 3?
?11

WHAT IS THE AREA DISPLACED ON THE
PLATEN BY THE TEST SPECIMEN, FIXTUR-
ING, ETC. IN CORNER 3?
?11

THE CORNER'S LOAD IS 1 PSI

WHAT IS THE WEIGHT OF THE TEST
SPECIMEN, FIXTURING, BOLTS, WASHERS,
ETC. IN CORNER 4?
?11

WHAT IS THE AREA DISPLACED ON THE
PLATEN BY THE TEST SPECIMEN, FIXTUR-
ING, ETC. IN CORNER 4?
?11

THE CORNER'S LOAD IS 1 PSI
CORNER NUMBER 1 HAS THE GREATEST
LOAD OF 1 POUNDS PER SQUARE INCH.

PRESS <ANY KEY> TO CONTINUE.

LENGTHS:  L=42 IN.   D=18 IN.
TO CONTINUE PRINTING DESIGN INFORMATION
PRESS <ANY KEY>.
TO CONTINUE PRINTING DESIGN INFORMATION
PRESS <ANY KEY>.
T=.5 IN.   P=10.39 IN.    B=7.79 IN.
TO CONTINUE PRINTING DESIGN INFORMATION
PRESS <ANY KEY>.
```

```
ANGLE THETA=40 DEG.  PLATEN WT.=810 LBS.
WHEN FINISHED,PRESS <ANY KEY>.
DO YOU WISH TO TRY ANOTHER DESIGN?
IF SO, PRESS KEY <C>; IF NOT, PRESS
ANY KEY.
                GOODBY

]PR#0
```

```
1  IF  PEEK (104) <  > 64 THEN  POKE
     103,1: POKE 104,64: POKE 163
     84,0: PRINT  CHR$ (4)"RUN PL
     ATEN"
5  SPEED= 150
10  REM  THIS PROGRAM DETERMINES
     THE OPTIMUM DESIGN PARAMETER
     S FOR A TEST PLATEN
20  REM  FOR VIBRATION TESTING OF
      TEST SPECIMENS WITHOUT REAC
     HING A RESONANT
30  REM  CONDITION WITH A MINIMUM
      DEAD   WEIGHT LOAD ON THE S
     HAKERS ARMATURE.
50  HOME
60  VTAB (6)
70  PRINT "      THIS  PROGRAM DET
     ERMINES  THE"
80  PRINT "      OPTIMUM DESIGN PA
     RAMETERS FOR"
90  PRINT "      AN INVERTED TRUNC
     ATED PYRAMID"
100  PRINT "      TYPE  OF  PLATEN
       FOR HOLDING"
110  PRINT "      TEST SPECIMENS D
     URING VIBRAT-"
115  PRINT "      ION TESTING"
117  PRINT
120  PRINT  TAB( 14);"BY L.J.LIPP
     "
125  PRINT
130  PRINT  TAB( 14);"USAAMCCOM"
140  PRINT "      PRODUCT ASSURANC
     E DIRECTORATE"
150  PRINT "           FOR CHEMICA
     L SYSTEMS"
160  PRINT  TAB( 17);"1985"
170  FOR X = 10 TO 4000: NEXT X
200  HOME
210  VTAB (5)
220  PRINT "THIS PROGRAM REQUIRES
      THAT THE FIXTURE"
230  PRINT "DESIGNER COMPLETE A L
     AYOUT OF THE TOP"
240  PRINT "OF A SQUARE VIBRATION
      PLATEN BEFORE"
250  PRINT "GETTING ON THE COMPUT
     ER. THE DESIGNER"
260  PRINT "IS EXPECTED TO HAVE:"
```

```
270  PRINT
280  PRINT  TAB( 5);"1. ARRANGED
     ALL OF THE TEST SPECI-"
290  PRINT  TAB( 8);"MENS SO THAT
      THOSE WITH THE"
300  PRINT  TAB( 8);"LEAST WEIGHT
      (INCLUDING FIXTURE"
310  PRINT  TAB( 8);"ADAPTERS AND
      BOLTS) PER DISPLAC-"
320  PRINT  TAB( 8);"ED PLATEN AR
     EA ARE AT THE FOUR"
324  PRINT  TAB( 8);"CORNERS;"
330  PRINT
340  PRINT  TAB( 5);"2. THE TEST
     SPECIMENS ARE AS CLOSE"
350  PRINT  TAB( 8);"TOGETHER AS
     POSSIBLE, YET LEAVE"
360  PRINT  TAB( 8);"ROOM FOR WRE
     NCHES, POWER AND"
370  PRINT  TAB( 8);"SIGNAL CABLE
     S, ETC. TO KEEP THE"
380  PRINT  TAB( 8);"PLATEN'S SQU
     ARE DIMENSIONS AS"
390  PRINT  TAB( 8);"SHORT AS POS
     SIBLE;"
400  PRINT
410  PRINT  TAB( 5);"3. SELECTED
     THE PLATEN MATERIAL AND"
420  PRINT  TAB( 8);"KNOWS ITS YO
     UNGS MODULUS OF"
430  PRINT  TAB( 8);"ELASTICITY A
     ND DENSITY;"
440  PRINT
450  PRINT  TAB( 5);"4. DETERMINE
     D THE MINIMUM DEPTH"
460  PRINT  TAB( 8);"REQUIRED FOR
      THE BOLT THREADS"
470  PRINT  TAB( 8);"TO FASTEN TH
     E TEST SPECIMENS"
480  PRINT  TAB( 8);"AND FIXTURES
      DOWN TO THE PLATEN."
490  PRINT
500  PRINT "HAVE YOU COMPLETED TH
     ESE PRELIMINARY"
510  PRINT "DESIGN DETAILS? <Y OR
      N>"
515  GET A$
520  IF A$ <  > "Y" AND A$ <  > "
     y" AND A$ <  > "N" AND A$ <
      > "n" THEN 2000
530  IF A$ = "N" OR A$ = "n" THEN
     2100
```

```
540   HOME
550   PRINT "IMPORTANT NOTE: ALL R
      ESPONSES TO THE"
560   PRINT "FOLLOWING QUESTIONS A
      RE TO BE IN"
570   PRINT "DECIMAL FORM, NOT FRA
      CTIONS."
580   PRINT
590   PRINT "WHAT IS THE LENGTH OF
       A SIDE OF YOUR"
600   PRINT "SQUARE PLATEN IN INCH
      ES?": INPUT L
610   PRINT
620   PRINT "WHAT IS THE DIAMETER
      OF THE SHAKER HEAD"
630   PRINT "IN INCHES?": INPUT DI
      A
640   PRINT
650   PRINT "WHAT IS THE MINIMUM M
      ATERIAL THICKNESS"
660   PRINT "REQUIRED TO BE TAPPED
       TO BOLT THE TEST"
670   PRINT "SPECIMENS TO THE PLAT
      EN IN INCHES?"
680   INPUT TK
681   PRINT
682   PRINT "WHAT IS THE MAXIMUM F
      REQUENCY REQUIRED"
685   PRINT "IN THE TEST PROGRAM I
      N HERTZ?"
687   INPUT MF
690   PRINT
700   PRINT "WHAT MATERIAL HAVE YO
      U CHOSEN FOR THE"
710   PRINT "PLATEN?": INPUT M$
720   PRINT
730   PRINT "WHAT IS ";M$;"'S YOUN
      GS MODULUS"
740   PRINT "OF ELASTICITY IN POUN
      DS PER SQUARE INCH?"
750   INPUT E
760   PRINT
770   PRINT "WHAT IS ";M$;"'S DENS
      ITY IN"
780   PRINT "POUNDS PER CUBIC INCH
      ?": INPUT DEN
784   REM  THE FIRST SUBROUTINE SE
      LECTS THE PLATEN CORNER WITH
       THE HIGHEST SPECIMEN LOAD.
787   GOSUB 2500
800   REM  DETERMINE THE ANGLE PHI
       FOR THE ANGLE OF THE PYRAMI
```

```
     D'S EDGES.
805  SPEED= 255
810  GOSUB 3000
820  REM  THE THIRD SUBROUTINE AD
     DS THE MATERIAL THICKNESS TO
      THE PLATEN,
830  REM  AND CALCULATES A NEW TE
     MPORARY PLATEN LENGTH SO THE
      EDGES WILL BE WEDGE
840  REM  SHAPED AGAIN.
850  GOSUB 4000
860  REM  THE 4TH SUBROUTINE RECA
     LCULATES THE ANGLE AGAIN, BU
     T CALLING IT PHI2
870  REM  WHICH IS THE FINAL ANGL
     E OF THE PYRAMIDS EDGES.
880  GOSUB 5000
890  REM  THE 5TH SUBROUTINE CALC
     ULATES THE ANGLE THETA WHICH
      IS THE ANGLE
900  REM  FORMED BETWEEN THE INTE
     RSECTING PLANES OF THE PLATE
     N'S TOP AND SLOPING
910  REM  SIDES.
920  GOSUB 6000
930  REM  THE 6TH SUBROUTINE CALC
     ULATES THE THICKNESS OF THE
     [LATEN'S BASE. ITS
940  REM  SQUARE DIMENSIONS WILL
     BE IDENTICAL TO THE DIAMETER
      OF THE SHAKER HEAD.
950  GOSUB 7000
960  REM  THE 7TH SUBROUTINE CALC
     ULATES THE PLATEN WEIGHT.
970  GOSUB 8000
980  REM  THE 8TH SUBROUTINE DISP
     LAYS A PLATEN IN HIGH RESOLU
     TION GRAPHICS
990  REM  AND LABELS ALL DIMENSIO
     NS NECESSARY FOR FABRICATION
      EXCEPT THE TAPPED
1000  REM  HOLE LOCATIONS FOR TES
     T SPECIMEN MOUNTING.
1010  GOSUB 9000
1015  HOME
1020  PRINT "DO YOU WISH TO TRY A
     NOTHER DESIGN?"
1030  PRINT "IF SO, PRESS KEY <C>
     ; IF NOT, PRESS"
1035  PRINT "ANY KEY.": GET C$
1040  IF C$ = "C" OR C$ = "c" THEN
     550
```

```
1045  HOME
1047  VTAB (15)
1048  HTAB (17)
1049  PRINT "GOODBY": GOTO 2140
2000  PRINT "YOU HAVE ENTERED A R
     ESPONSE THAT THE"
2010  PRINT "COMPUTER DOES NOT RE
     COGNIZE.
2020  PRINT "LETS TRY AGAIN."
2030  PRINT
2040  PRINT "HAVE YOU COMPLETED T
     HESE PRELIMINARY"
2050  PRINT "DESIGN DETAILS? <Y O
     R N>": GET A$
2060  IF A$ = "Y" OR A$ = "y" THEN
     540
2070  IF A$ = "N" OR A$ = "n" THEN
     2100
2080  IF A$ <  > "Y" AND A$ <  >
     "y" AND A$ <  > "N" AND AS$ <
      > "n" THEN 2100
2100  PRINT "PLEASE PERFORM THE P
     RELIMINARY LAYOUT"
2110  PRINT "STUDIES REQUIRED TO
     USE THIS PROGRAM."
2115  FOR X = 1 TO 2500: NEXT X
2121  HOME : VTAB (15): HTAB (17)
      : PRINT "GOODBY"
2140  END
2500  REM  THIS SUB-ROUTINE DETER
     MINES   WHICH CORNER HAS THE
      GREATEST SPECIMEN
2510  REM  LOAD BY DIVIDING THE W
     EIGHT IN EACH CORNER BY THE
     AREA IT DISPLACES.
2520  HOME : PRINT
2530  PRINT "ASSIGN EACH CORNER O
     F THE PLATEN A"
2540  PRINT "NUMBER SO THE WEIGHT
     S OF THE AREAS DIS-"
2550  PRINT "PLACED BY THE TEST S
     PECIMENS AND THEIR"
2560  PRINT "FIXTURING CAN BE IDE
     NTIFIED BY THEIR"
2570  PRINT "LOCATION ASSIGNMENT.
     "
2580  PRINT
2590  FOR X = 1 TO 4
2595  PRINT
2600  PRINT "WHAT IS THE WEIGHT O
     F THE TEST"
```

```
2610  PRINT "SPECIMEN, FIXTURING,
       BOLTS, WASHERS,
2620  PRINT "ETC. IN CORNER ";X;"
     ?": INPUT W(X)
2630  PRINT
2640  PRINT "WHAT IS THE AREA DIS
     PLACED ON THE"
2650  PRINT "PLATEN BY THE TEST S
     PECIMEN, FIXTUR-"
2660  PRINT "ING, ETC. IN CORNER
     ";X;"?": INPUT A(X)
2670 P(X) = W(X) / A(X)
2680  PRINT
2685  PRINT "THE CORNER'S LOAD IS
       ";P(X);" PSI"
2690  NEXT X
2700 S = P1:Y = 1
2705  IF P(1) < P(2) THEN P(1) =
     P(2):Y = 2
2710  IF P(1) < P(3) THEN P(1) =
     P(3):Y = 3
2720  IF P(1) < P(4) THEN P(1) =
     P(4):Y = 4
2730  PRINT "CORNER NUMBER ";Y;"
     HAS THE GREATEST"
2740  PRINT "LOAD OF ";P(1);" POU
     NDS PER SQUARE INCH."
2750  PRINT
2770  PRINT "PRESS <ANY KEY> TO C
     ONTINUE.": GET Z$
2780  IF Z$ =  >  CHR$ (0) THEN 2
     790
2790  HOME
2800  RETURN
3000  PRINT
3010  REM  THIS SUBROUTINE IS THE
       FIRST ITERATION TO DETERMIN
     E WHAT ANGLE PHI
3020  REM  SATIFIES THE EQUATION
     THAT DETERMINES THE MINIMUM
     NATURAL FREQUENCY.
3030 C = (L /  SQR (2)) - (DIA /
     2)
3040  FOR PHI = 5 TO 45 STEP .5
3050 RAD = PHI * 3.14159265 / 180

3060 T =  TAN (RAD)
3080 NU = (E * T ^ 3) * ((C * DEN
      * T) + (12 * P(1)))
3090 DM = C * ((54 * C * C * DEN *
     DEN * T * T) + (8960 * (P(1)
      ^ 2)))
```

```
3100  IF DM <  = 0 THEN 3130
3105 CF = 48.47 * ((NU / DM) ^ .5
     )
3110  IF CF >  = MF THEN 3700
3120  NEXT PHI
3125  SPEED= 150
3130  PRINT "THERE IS EITHER AN E
     RROR IN YOUR INPUT"
3140  PRINT "DATA OR THE MAXIMUM
     NATURAL FREQUENCY"
3150  PRINT "OF YOUR PROGRAM IS T
     OO HIGH FOR"
3260  PRINT "PRACTICALITY. THE HI
     GHEST FREQUENCY"
3270  PRINT "POSSIBLE FOR THE DES
     IGN INFORMATION"
3280  PRINT "PROVIDED IS ";CF;" C
     YCLES PER SECOND."
3290  PRINT
3300  PRINT "IT IS SUGGESTED THAT
      YOU PERFORM THESE"
3310  PRINT "CHECKS:"
3320  PRINT
3330  PRINT  TAB( 5);"1. RECHECK
     YOUR DATA INPUTS TO"
3340  PRINT  TAB( 8);"SEE IF THEY
      ARE CORRECT."
3350  PRINT
3360  PRINT  TAB( 5);"2. IF THE D
     ATA IS CORRECT IN #1"
3370  PRINT  TAB( 8);"ABOVE, THEN
      RECHECK THE LAYOUT"
3380  PRINT  TAB( 8);"TO DETERMIN
     E IF THE SPECIMENS"
3390  PRINT  TAB( 8);"CAN BE SET
     CLOSER TOGETHER. THE"
3400  PRINT  TAB( 8);"NATURAL FRE
     QUENCY INCREASES VERY"
3410  PRINT  TAB( 8);"RAPIDLY AS
     THE LENGTH OF THE"
3420  PRINT  TAB( 8);"PLATEN'S SI
     DES DECREASES."
3430  PRINT
3440  PRINT  TAB( 5);"3. IF NONE
     OF THE ABOVE CAN BE"
3450  PRINT  TAB( 8);"CORRECTED,
     YOU WILL HAVE TO"
3460  PRINT  TAB( 8);"ACCEPT A RE
     SONANT CONDITION IN"
3465  PRINT  TAB( 8);"THE PLATEN.
     "
```

```
3468  SPEED= 255
3470  PRINT
3480  PRINT "WHEN YOU PRESS <ANY
     KEY>"
3485  PRINT "THE PROGRAM WILL TER
     MINATE."
3490  GET Z$
3600  IF Z$ =  >  CHR$ (0) THEN 1
     045
3700  HOME
3710  RETURN
4000  REM  THIS SUBROUTINE ADDS T
     HE NECESSARY ADDITIONAL THIC
     KNESS FOR TAPPING
4010  REM  THREADS INTO THE PLATE
     N FOR THE BOLTING ON OF TEST
      SPECIMENS. IT THEN
4020  REM  CALCULATES A NEW PLATE
     N LENGTH TO FORM WEDGE EDGES
     .
4030 NT = C * T + TK
4040  REM  THE ABOVE CALCULATION
     DETERMINES THE NEW TOTAL THI
     CKNESS OF THE PLATEN
4050  REM  AT DISTANCE 'C' ALONG
     THE PLATENS DIAGONAL.
4060  REM  THE NEXT CALCULATION D
     ETERMINES A NEW "C" BASED UP
     ON THIS THICKNESS
4070  REM  CALLED "NC".
4080 NC = NT / T
4100  RETURN
5000  REM  THIS 4TH SUBROUTINE RE
     CALCULATES THE ANGLE PHI BAS
     ED UPON "NC" AND "NT"
5010  REM  EXCEPT IT IS CALLED PH
     I2 TO KEEP IT SEPARATE FROM
     PHI. THE EQUATION IS
5020  REM  THEN REARRANGED AGAIN
     TO UTILIZE LINES 3050-3710 O
     F THE 2ND SUBROUTINE.
5040  FOR PHI2 = PHI TO 55 STEP .
     1
5050 RAD2 = PHI2 * 3.14159265 / 1
     80
5060 T2 =  TAN (RAD2)
5080 NU2 = (E * T2 ^ 3) * ((NC *
     DEN * T2) + (12 * P(1)))
5090 DM2 = NC * ((54 * C * C * DE
     N * DEN * T2 * T2) + (8960 *
     (P(1) ^ 2)))
```

```
5100  IF DM2 <  = 0 THEN 3130
5110 CF2 = 48.47 * ((NU2 / DM2) •
     .5)
5125  IF CF2 >  = MF THEN 5200
5130  NEXT PHI2
5200  HOME
5210  RETURN
6000  REM  THIS SUBROUTINE CALCUL
     ATES THE ANGLE "THETA" WHICH
      IS THE ANGLE
6010  REM  FORMED BETWEEN THE INT
     ERSECTING PLANES OF THE PLAT
     EN'S TOP AND ITS
6020  REM  SLOPING SIDES.
6030 BETA = 45 * 3.1415926 / 180
6035 NG = (NC + (DIA / 2)) *  SIN
     (BETA)
6040 NT2 = (NC + (DIA / 2)) * T2
6045  IF NG = 0 THEN  HOME : PRINT
     "A DIVISION BY ZERO ERROR, R
     ECHECK YOUR WORK."
6050 TETA =  ATN (NT2 / NG)
6060  HOME
6070  RETURN
7000  REM  THS SUBROUTINE CALCULA
     TES THE THICKNESS OF THE PLA
     TEN'S BASE. IT DOES
7010  REM  SO BY MULTIPLYING THE
     ANGLE THETA BY HALF THE DIAM
     ETER OF THE SHAKER
7020  REM  HEAD. IT ALSO CALCULAT
     ES THE TOTAL THICKNESS OF TH
     E PYRAMID FROM ITS
7030  REM  BASE (ITS TOP SINCE IT
     IS INVERTED) DOWN TO WHERE I
     TIS TRUNCATED.
7040 BASEHEIT =  TAN (TETA) * (DI
     A) / 2
7050 PYRAHEIT =  TAN (TETA) * L /
     2 - BASEHEIT
7060  HOME
7070  RETURN
8000  REM  THIS SUBROUTINE CALCUL
     ATES THE PLATEN'S TOTAL WEIG
     HT.
8010  REM  IT STARTS WITH THE TOT
     AL VOLUME OF THE TOP SQUARE-
     RETANGULAR PARALLEL-
8020  REM  PIPED.
8030 TPVOL = TK * L • 2
8040  REM  NEXT IS THE CALCULATIO
     N OF THE TOTAL VOLUME OF THE
      PYRAMID PORTION OF
```

```
8050  REM  THE PLATEN
8060 P1 = (( TAN (TETA) * L / 2) *
     L * 2) / 3
8063 P2 = (BASEHEIT * D * 2) / 3
8065 PVOL = P1 - P2
8070  REM  NEXT IS THE CALCULATIO
     N OF THE BASE VOLUME
8080 BVOL = BASEHEIT * D * 2
8090  REM  NEXT IS THE TOTAL VOLU
     ME
8100 TTVOL = TPVOL + PVOL + BVOL
8110  REM  FINALLY, THE TOTAL PLA
     TEN WEIGHT.
8120 PLTWT = TTVOL * DEN
8130 PYRAHEIT = ( INT (PYRAHEIT *
     100)) / 100
8140 BASEHEIT = ( INT (BASEHEIT *
     100)) / 100
8150 TETA = ( INT ((TETA) * (180 /
     3.14159)) * 100) / 100
8160 PLTWT =  INT (PLTWT)
8200  RETURN
9000  REM  THIS SUBROUTINE PRINTS
      A DISPLAY OF THE SIDE AND B
     OTTOM VIEWS OF THE
9010  REM  PLATEN AND PRINTS OUT
     THE FINAL INFORMATION REQUIR
     ED TO DESIGN THE
9020  REM  PLATEN.
9040  HGR
9045  HCOLOR= 3
9050  PRINT  CHR$ (4)"BLOAD GRAPH
     "
10000  VTAB (22)
10010  PRINT "LENGTHS:"; TAB( 11)
     ;"L=";L;" IN."; TAB( 22);"D=
     ";DIA;" IN."
10015  PRINT "TO CONTINUE PRINTIN
     G DESIGN INFORMATION"
10016  PRINT "PRESS <ANY KEY>.": GET
     C$
10017  IF C$ =  >  CHR$ (0) THEN
     10020
10018  GOTO 10010
10020  VTAB (22)
10025  PRINT "TO CONTINUE PRINTIN
     G DESIGN INFORMATION"
10026  PRINT "PRESS <ANY KEY>."
10027  IF C$ =  >  CHR$ (0) THEN
     10030
10028  GOTO 10010
```

```
10030  PRINT "T=";TK;" IN."; TAB(
     12);"P=";PYRAHEIT;" IN."; TAB(
     28);"B=";BASEHEIT;" IN."
10035  PRINT "TO CONTINUE PRINTIN
     G DESIGN INFORMATION"
10036  PRINT "PRESS <ANY KEY>."
10037  GET C$
10038  IF C$ =  >  CHR$ (0) THEN
     10040
10039  GOTO 10010
10040  VTAB (22)
10050  PRINT "ANGLE THETA=";TETA;
     " DEG."; TAB( 22);"PLATEN WT
     .=";PLTWT;" LBS."
10060  VTAB (24)
10260  PRINT "WHEN FINISHED,PRESS
      <ANY KEY>."
10270  GET C$
10280  IF C$ =  >  CHR$ (0) THEN
     10300
10290  GOTO 10000
10300  TEXT
10310  RETURN
```

END
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DTIC