CALIFORNIA INSTITUTE OF TECHNOLOGY

EARTHQUAKE ENGINEERING RESEARCH LABORATORY

A NEW VIBRATION EXCITER FOR DYNAMIC TEST OF FULL-SCALE STRUCTURES

by

D. E. Hudson

A report of research carried out by the Earthquake Engineering Research Institute under State of California Standard Agreement No. 2163, for the State of California, Department of Public Works, Division of Architecture, Anson Boyd, State Architect.

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Prepared for

"Primeras Jornadas de Ingenieria Antisismica"
April 16-21, 1962
University of Cuyo
Facultad de Ingenieria
Parque de Mayo
San Juan, Argentina

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Abstract

Design criteria are established for a vibration exciter unit suitable for steady-state resonance tests of full-scale building structures, and several design configurations are compared. These ideas are embodied in the design of a new unit, which produces a horizontal, uni-directional sinusoidally varying force of about 1000 lb magnitude at 1 cycle per second, with a maximum force limit of 5000 lb. The total weight of the machine with its maximum load of eccentric weights is about 1500 lb, which can be broken down into units not exceeding 580 lb for handling and transportation. The general features of a servo-controlled electronic amplidyne speed control and D. C. drive for the vibration exciter are given. A field test of the new vibration exciter system is described, and typical results of the determination of the dynamic properties of a concrete intake tower of a dam by means of the new machine are given.

A New Vibration Exciter for Dynamic Tests of Full-Scale Structures

Introduction. The only way in which the dynamic properties of structures can be accurately determined is by tests of actual full-scale structures under relatively high loading conditions. Such important parameters as the natural frequencies and mode shapes of vibration, and the energy dissipation characteristics, can only be approximated by a theoretical analysis. There is no alternative to field testing to establish an adequate knowledge of such dynamic properties.

Dynamic tests on full-scale structures are of two main types. From free oscillation tests the natural frequencies and damping over a range of loading conditions can be determined. Such free oscillations may be excited by giving the system an initial displacement, and suddenly releasing the system from rest. This is commonly done for such structures as water towers by pulling on a cable attached to the tower, and then suddenly releasing the cable.[1] Another method recently used with success was to set up initial velocities in a tall chimney by firing a group of small rockets from the top.[2] The short-duration rocket thrust acts like a single impulse to impart an initial velocity to the chimney.

One difficulty with free vibration tests is that it is usually not easy to separate the effects of the individual modes of vibration in a complicated structure. It is also usually difficult to accurately control the test conditions for repeated tests.

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[1] Carder, D.S. and Jacobsen, L. S., "Vibration Observations", Earthquake Investigations in California, Spec. Pub. No. 201, U. S. Department of Commerce, Coast and Geodetic Survey, Washington, D. C., 1936.

[2] Scruton, C., and Harding, D. A., "Measurement of the Structural Damping of a Reinforced Concrete Chimney Stack at Ferrybridge 'B' Power Station", National Physical Laboratory, NPL/Aero/323/1957.

The second class of dynamic tests are steady-state forced oscillation tests. These require the application to the structure of sinusoidally varying forces of adjustable frequency. By plotting resonance curves corresponding to the various modes of vibration, relatively complete information on the dynamic properties of the structure can be determined.

An intermediate type of testing makes use of very small vibrations caused by the action of the wind, traffic, or installed machinery in the structure. Such vibrations should perhaps be considered as a series of random free vibration tests rather than as a forced vibration test. Such wind-excited tests are usually limited to a determination of a few of the lowest frequencies of vibration at very low stress levels.

The most complete and accurate dynamic investigations are those made with forced vibration tests. Such tests have seldom been possible for full-scale structures because of (1) the difficulty of applying dynamic loads of the required type and magnitude; (2) the difficulty of making the measurements of dynamic structural response with the required accuracy; and, (3) the unavailability of suitable test structures which could be loaded to the point of significant damage, with the attendant risk of expensive failures.

Because of the importance of dynamic testing for the development of earthquake resistant design, the California State Division of Architecture is sponsoring through the Earthquake Engineering Research Institute the development of a large scale vibration exciter system suitable for tests of full-size structures. The design and development of this system is being carried out at the California Institute of Technology, and it is the first unit built under this program which will be described in the present paper.

Basic Design Concepts for the Vibration Exciter. After a survey of the potential use of vibration test equipment for studies in the field of earthquake resistant design, the following basic design criteria were established.

(1) The vibration exciter unit should produce a uni-directional, sinusoidally varying force magnitude of 1000 lb at 1 cycle per second, and a maximum force of 5000 lb. A horizontal force only would be satisfactory, but if no serious compromises in design would be involved, it would be advantageous if the unit could be used for vertical forces as well.

(2) The total weight of the unit should be of the order of 1000 lb, and this total weight should break down into sub-assemblies for ease of handling, transportation, and installation.

(3) The final complete vibration exciter system should consist of four of the above units, with suitable means of combining the force outputs in various phase relationships. By adding the force outputs in phase a force output of 4000 lb at 1 cycle per second could be produced, with a maximum force of 20,000 lb. Considering the fact that the force can be applied under resonance conditions, these force magnitudes should reach damaging levels for even large structures.

The most important feature in the above specifications is the decision to build four relatively small units that can either be used separately, or synchronized for multiple operation. Two very important advantages result: (1) problems of handling, transportation, and installation are very much simplified; and (2) the forces can be applied in various configurations to most effectively excite different modes of vibration, such as torsional modes, or higher modes of bending or shear vibration.

With the above specifications in mind, various methods of producing the required alternating forces were explored. Rotating unbalanced weights,

Table I

Some Rotating Weight Vibration Exciters Suitable for Structural Dynamics Tests

Description	Alternating Force Amplitude at 1 cyc/sec, pounds	Total Weight Approx.	Remarks
U.S.C.G.S. No. 1[3]	12	100	Counter-rotating weights, uni-directional force
U.S.C.G.S. No. 2[4]	150	350	"
U.S.C.G.S. No. 3[5]	600	800	"
TMB 5000[6]	600	5600	"
Losenhausen[7]	14,400	49,000	"
New Unit	970	1480	"
Japanese Building Research Institute[8]	3080	3320	Single arm-rotating force, horizontal only

[3] Picture in Macelwane, J. B., "When the Earth Quakes", Bruce Pub. Co., Milwaukee, 1947.

[4] Blume, J. A., and L. S. Jacobsen, "The Building and Ground Vibrator", Earthquake Investigations in California, U. S. Department of Commerce, Coast and Geodetic Survey, Washington, D. C., 1936.

[5] Patterson, W. D., "Determination of Ground Periods", Bull. Seis. Soc. Amer., vol. 30, No. 2, April 1940.

[6] Berdahl, E.O., "Construction and Operation of the Taylor Model Basin 5000-Pound Vibration Generator", David Taylor Model Basin Report No. 524, April, 1944.

[7] Berdahl, E. O., "Construction and Operation of the Losenhauser 44000-Pound Vibration Generator", David Taylor Model Basin Report No. 554, April, 1947.

[8] Hisada, T., and Nakagawa, K., "Vibration Tests of Various Types of Building Structures up to Failure", Proc. World Conference on Earthquake Engineering, Earthquake Engineering Research Institute and the University of California, Berkeley, 1956.

reciprocating masses driven by compressed air or by internal combustion processes, and various crank and connecting rod systems were examined. The most common method used in the past has been to employ the inertia force produced by an eccentric rotating weight, and this was finally settled on as the most feasible system.

A number of such rotating-weight vibration exciters have been built in the past, and have been used with reasonable satisfaction for many structural dynamics problems. Table I summarizes the characteristics of several typical machines.

The three U. S. Coast and Geodetic Survey exciters were specially designed for earthquake engineering studies, and gave satisfactory results for the force levels involved. The designs used were not well adapted to the increased size required for the present application without considerable modification. The Losenhausen machine is an example of a very large device suitable for ship vibration studies, whose great weight would preclude its use for most structural applications. The large vibration exciter designed by the Japanese Building Research Institute is an ingenious machine having the advantages of simplicity and light weight. It does not produce a uni-directional force, however, and is limited to relatively low frequencies and horizontal forces.

Rotating Weight Vibration Exciters. Most of the rotating weight vibration exciters of the past have been horizontal shaft machines using some kind of multiple weight system to cancel inertia torques as far as possible.[9] Typical of these machines is the U. S. C. G. S. No. 3 of the table, which has an arrangement of three weights rotating about one horizontal shaft as

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[9] Späth, W., Theorie und Praxis der Schwingungsprüfmachinen, Springer, 1934, English translation "Theory and Practice of Vibration Testing Machines", issued as David Taylor Model Basin Translation No. 51, March, 1938.

indicated in Fig. 1. In Fig. 1 (a), (b) the phase relationships between the three unbalanced weights is adjusted to generate a vertical inertia force. The middle disk of eccentric weight 2W is rotated by a gear system in the opposite direction to the two outside disks, each of which carries an eccentric weight W. In Fig. 1(a) the weights are shown in a position to balance out any horizontal force, and in Fig. 1(b), one-quarter revolution later, the inertia forces add to give the maximum vertical force. The complete forces and moments are seen to be:

Fig. 1(a) $F_x = 0; \quad F_y = -4W; \quad F_z = 0$

$M_x = 0; \quad M_y = 0; \quad M_z = 0$

Fig. 1(b) $F_x = 0; \quad F_y = \frac{4W}{g} R\omega^2 - 4W; \quad F_z = 0$

$M_x = 0; \quad M_y = 0; \quad M_z = 0$

There are thus no inertia torques involved in this arrangement, and a uni-directional sinusoidally varying force of magnitude $\frac{4W}{g} R\omega^2$ is produced.

For horizontal excitation, the situation is not quite so simple, since it will not be possible to entirely eliminate all alternating inertia couples. In Fig. 1 (c) the horizontal forces are zero, and the alternating vertical forces are balanced out. In Fig. 1 (d), one-quarter revolution later, the masses are in the position of maximum horizontal force. The total forces and moments are:

Fig. 1 (c) $F_x = 0; \quad F_y = -4W; \quad F_z = 0$

$M_x = 0; \quad M_y = 0; \quad M_z = 0$

Fig. 1 (d) $F_x = \frac{4W}{g} R\omega^2; \quad F_y = -4W; \quad F_z = 0$

$M_x = 0; \quad M_y = 0; \quad M_z = -4WR - \frac{4W}{g} R\omega^2 h$

It will be noted that in addition to the desired horizontal varying force F_x of magnitude $\frac{4W}{g} R\omega^2$, there is a sinusoidally varying torque about the horizontal z axis of magnitude

$$M_z = 4WR + \frac{4W}{g} R\omega^2 h = F_x \left(\frac{g}{\omega^2} + h \right)$$

At frequencies of the order of 1 cycle per second, and with typical mounting distances h of one to two feet, the two terms in the expression for M_z would be of about the same magnitude. Even if h could be made equal to zero, which is usually not possible considering the way in which the machine is to be attached to the structure, the alternating torque M_z would be of the same order of magnitude as that produced on the mounting by the horizontal force F_x. The component of M_z of magnitude 4WR is a consequence of the fact that the center of mass of the machine is periodically being shifted horizontally, and such an alternating torque appears to be inevitable for any configuration of an inertia type machine for horizontal excitation.

The alternating torque M_z does not in practice cause difficulties, since in effect it is simply combined with the horizontal force to give a generalized exciting force acting on the structure. This generalized force can usually be so directed that the desired motions of the structure are excited without appreciable interference from undesired motions in other directions.

An advantage of the horizontal shaft machine is that the same machine can be adjusted for horizontal or vertical excitations without repositioning the mounting system.

A basic difficulty with horizontal shaft machines with large eccentricities is the fact that there is a large alternating gravity torque acting about the drive shaft or in the drive system, which complicates the mechanical design. In vertical force machines having large eccentric masses, this gravity torque may be so large as to make it difficult to start the machine.

The New Vibration Exciter. A primary requirement of the new vibration exciter was a large force at low frequencies. This means that relatively large eccentric weights would be involved, and the above-mentioned problem of large alternating loads in the drive system would be troublesome if horizontal shafts were to be used. For this reason it was decided to go to a vertical shaft machine of the type shown in Fig. 2. This decision limited the machine to horizontal excitations, but this was not felt to be a serious matter in view of the intended applications of the exciter. In fact, the limitation to horizontal excitations permitted such a great simplification in design and mounting that it was felt to be justified even if other designs for vertical excitation might later need to be developed and constructed.

The forces and moments involved in the two weight system of Fig. 2 are as follows:

Fig. 2 (a) $F_x = \frac{2W}{g} R\omega^2; \quad F_y = -2W; \quad F_z = 0$

$$M_x = 0; \quad M_y = 0; \quad M_z = -2WR - \frac{W}{g} R\omega^2(h_1 + h_2)$$

Fig. 2 (b) $F_x = 0; \quad F_y = -2W; \quad F_z = 0$

$$M_x = \frac{W}{g} R\omega^2(h_2 - h_1); \quad M_y = 0; \quad M_z = 0$$

Comparing these equations with the similar relationship of Fig. 1, it will be noted first that a similar alternating torque M_z exists, and second that an additional transverse alternating M_x torque is also present. This transverse torque could be eliminated by going to a three-mass vertical shaft arrangement, but this could be done only at the expense of increasing the M_z torque, since the three-mass system could not be designed quite as compactly, or located as close to the floor. The transverse M_x torque is in any event small, since the term (h_2-h_1) can be kept small by the mechanical design of the system. In fact, by rotating the two masses in the same plane, as is done in the U. S. C. G. S. No. 1 machine, the transverse M_x torque could be entirely eliminated. This design complication did not seem to be justified for the contemplated applications.

The advantages of simplicity and compact design of the two-mass system were considered to outweigh the small disadvantage of the transverse torque, which at the frequencies involved in this relatively low-speed system would not significantly alter the generalized exciting forces.

The final form of the two-mass vertical shaft machine can be seen in the photograph of Fig. 3 and the assembly drawings of Figs. 4 and 5. A significant feature of the design is the chain drive. All of the previously designed rotating-weight machines have obtained counter-rotation by a system of gears. A consequence of such gear systems has been the generation of significant amounts of high-frequency vibration, which has been in some instances so serious as to interfere with the required structural motion measurements. This difficulty is particularly troublesome if the motion measurements are made with acceleration-type transducers, since the accelerations corresponding to the high-frequency gear vibrations may be significantly large. There are a number of reasons for preferring acceleration-type transducers for building response measurements, and thus the absence of disturbing high-frequency vibrations in the exciter system is an important consideration. The use of a roller chain drive eliminates most of these difficulties, and at the same time affords a very simple means of providing counter-rotation by driving off the back of the chain. The natural flexibility of the roller chain also provides an internal shock absorber for the drive system.

An important consideration in the design of the unit was to keep the weight of the device down for ease in transportation and handling. It will be noted in Fig. 4 that to this end the rotating masses were made in the form of sectors of a circle. Since the center of mass of a circular sector moves towards the center as the sector angle increases, there is no significant gain in increasing the sector angle past a certain point. For simplicity in design and manufacture, the eccentric weights have been made approximately in the form of circular sectors with a cut-out hub, as shown in Fig. 4. In Fig. 6 is shown the product (area) (c. g. distance) for the eccentric weights, which is a measure of the amount of inertia force produced at a given frequency, plotted versus the sector angle for two different hub radii. It will be observed that there is little gain in making the sector angle greater than 120°. The constant weight contours shown in Fig. 6 also indicate that in the region of optimum sector angle the hub size is not a critical parameter.

In the final design the 120° sector was divided into three sections, and each section was divided into four layers of lead weights. This resulted in lead weights which could easily be handled by one man. The "square" center sections weigh 22. 7 lb each, and the other sections weigh 38. 0 lb each. In the foreground of Fig. 3 stacks of the standard lead weights can be seen. Each weight has cast into its upper surface a recessed thread which will take a special tool for ease in handling. The cast aluminum receptacles for the lead weights have recesses which are machined to fit the milled edges of the weights, so that the weights are securely fitted without the necessity of attachment bolts.

The WR eccentricity corresponding to the smaller "square" weights is about 38.0 ft lb for two counterrotating weights, while that corresponding to the larger sector shaped weights is 75.0 ft lb. With the maximum load of 391 lb of lead weights in each receptacle, the total combined WR eccentricity is about 750 ft lb, which gives a total horizontal exciting inertia force at 1 cyc/sec of 920 lb.

It will be noted in Fig. 3 that counterweights have been added to the rotating receptacles so that high speed tests can be made. The receptacles themselves have a total WR eccentricity for the two of 41 ft lb, corresponding to a total horizontal exciting force of 50 lb at 1 cyc/sec. At the top design speed of 10 cyc/sec the total horizontal force generated by the receptacles themselves would thus be 5000 lb, just at the design limit of strength.

With the maximum load of lead weights plus the receptacles without the counterweight, the total horizontal inertia force at 1 cyc/sec is 970 lb, which approximates the original design value of 1000 lb. With this maximum eccentric weight the design limit strength of the frame of 5000 lb is reached at a speed of 2.27 cyc/sec.

It will be noted that no effort has been made to reduce the air resistance drag forces on the rotating members. Because of the relatively low speeds involved, these air resistance forces are not ordinarily important. At the top speed of 10 cyc/sec, however, the air resistance forces absorb a significant amount of the drive power. These drag forces, in fact, act as an automatic safety device to prevent excessive speeds at low eccentricity, since the power absorbed by these air resistance forces rapidly becomes greater than can be supplied by the motor above 10 cyc/sec.

The limiting total horizontal force parallel to the long back plate on which the motor drive assembly is attached is 5000 lb (Fig. 7). Above this load, significant overstresses occur in some elements. If the eccentrics are oriented so that the horizontal force is at right angles to the motor attachment plate (Fig. 7(b)), the maximum load limit is reduced.

Figure 8 shows the total horizontal force output for various lead weight loads and speeds for the counterbalanced exciter. It is easily possible to exceed the 5000 lb load limit with the input drive power available, and if the speed should run away there is a real danger that the machine might disintegrate. It is essential that a trained operator be stationed at all times at the control console within reach of the shut-down push-button, with an eye always on the speed indicator meter. The system should be immediately shut down upon any evidence of an abnormal speed rise.

Considerable thought has been given to the possibility of incorporating in the system some kind of strain switch or safety interlock which would ensure against overloading. The complexity of any such device carries with it additional dangers of mal-function, and it has been concluded that the best insurance against failure would be to rely on carefully selected and trained operators who understand thoroughly the dangers and limitations of the machine.

Each rotating element is supported by two 2-1/2" bore pre-loaded tapered roller bearings. The dead shaft arrangement makes it possible to avoid concentric shafts with their additional design complications. Figures 4 and 5 which reproduce the complete assembly drawing of the machines may be consulted for additional details of the structure and drive system.

An important feature of the design is the way in which the unit can be disassembled for ease in handling, transportation, and installation. The drive motor and control tachometer assembly can be removed as a unit, which weighs about 100 lb. With the lead weights removed, the basic unit weighs about 580 lb, and this is the largest single weight which must be handled.

For ease in mounting, a special baseplate has been designed, which can be seen under the machine in Fig. 3. This baseplate serves as a template for locating mounting bolt holes in the concrete floor slab, and provides a simple way of securely mounting the machine. A typical way of fastening the baseplate to a concrete floor is to bolt it down with eight 7/8" bolts in expandable lead mountings in drilled holes in the concrete.

Electric Drive and Speed Control System. The requirements of the electric drive for the vibration exciter system are particularly stringent because of the variable torques imposed on the drive and the necessity to ensure stability when operating near resonance of lightly damped structures. The ability to hold an accurate speed control at and near the resonance peak requires that the speed-torque curve of the drive system be unusually flat, with essentially a constant speed maintained under relatively large torque variations. It is also required that the speed be variable over a very wide range.

To meet these requirements, a 1-1/2 h.p. D.C. drive motor is used, along with a servo controlled electronic amplidyne system. The general features of this control system are illustrated in the schematic diagram of Fig. 9. A tachometer driven directly from the drive motor supplies a speed signal which is compared with a standard set voltage. The difference between these voltages provides an input signal to the amplidyne amplifier, which then acts on the drive motor to adjust its speed to correspond to the set speed. Figure 10 shows the portable speed control console which contains this amplidyne apparatus. The single large black knob between the two meters gives a smooth uniform stable speed adjustment over a 40 to 1 speed range. For field applications, the necessary three-phase 220 volts A. C. electric power supply can easily be obtained from a portable gas-engine driven generator.

An accurate measure of the vibration exciter speed is read from the digital electronic counter seen on top of the control console in Fig. 10. This counter is triggered by a 100 pulse per revolution permanent magnet tachometer generator mounted on the drive motor shaft. This digital counter contains its own frequency standard accurate to 0.1%.

Field Tests of the Vibration Exciter. The problem of testing the vibration exciter presents some difficulties, since a test arrangement which would satisfactorily simulate the environment of an actual structure would be as complex and expensive as an actual building. Fortunately just at the time that the first unit was ready for the final test runs, an opportunity occurred to make tests on an actual structure which was in many respects ideal for the purpose.

When the City of Los Angeles recently undertook a program of increasing the storage capacity of one of their water supply reservoirs by raising the height of an earth dam, it became necessary to replace the existing concrete intake tower by a taller structure in a different location. The old tower was made available for test in the empty reservoir, and since it was to be demolished, the tests could be carried out at force levels which were potentially damaging.

A sketch of the intake tower is shown in Fig. 11. The tower is of a sufficiently simple design so that calculations can be made of its dynamic properties, such as its natural frequencies of vibration, with some expectation of success.

The field conditions under which the tower tests were conducted were considerably more severe than had been contemplated for typical applications of the vibration exciter, and hence constituted a very useful check on the operation of the equipment.

Figure 12 shows the way in which the disassembled vibration exciter was hoisted to the top of the intake tower. All equipment and instruments had to be taken up a 100 ft high iron ladder running up the outside of the tower.

Figure 13 shows the installation location of the vibration exciter at the top of the tower. The outlet pipe at the base of the tower was a source of assymetry in the elastic properties of the tower, and the major and minor principal axes of elasticity for bending are indicated in Fig. 13. Because of the convenience of mounting motion measuring transducers near the ladder along the height of the tower, the vibration exciter was not aligned with these principal axes. This resulted in the simultaneous excitation of two modes of vibration, which considerably complicated the analysis of the data, as will be discussed in a later section.

Motion Transducer and Recording System. The transducer used for the motion measurements was a variable reluctance accelerometer having a natural frequency of about 100 cyc/sec and 0.7 critical damping. The accelerometer was energized by a 2000 cyc/sec carrier voltage, and the output was amplified, demodulated, and recorded on a direct ink-writing oscillograph. The overall frequency response of the accelerometer-recording system was essentially constant from zero to 80 cyc/sec.

An important advantage of an accelerometer type transducer system with zero-frequency response is that an accurate absolute calibration can easily be obtained. By rotating the accelerometer through 90° in the earth's gravitational field a 1 g static acceleration is applied. Such overall system calibrations were made in the laboratory before the field tests, in the field before and after the test runs, and subsequently in the laboratory again.

The accelerometers were mounted to measure horizontal motions. Two accelerometers were attached to the top of the tower, and one was mounted on the side at mid height. These three instruments were simultaneously recorded during the tests.

Figure 14 shows a typical acceleration-time record as obtained during the tests. It will be noted that a relatively pure single frequency sine wave results, and that no disturbing high-frequency vibrations are present. This indicates that the drive system adopted for the new vibration exciter has successfully met the initial requirements of smoothness of operation.

Analysis of Tower Test Data. The object of the field tests was to obtain a family of steady-state resonance curves for various exciting force magnitudes. From such resonance curves most of the structural dynamic properties of importance can be derived.

Figure 15 shows computed resonance curves for a single degree of freedom viscous damped oscillator, in which the acceleration of the mass is plotted versus frequency for a frequency-squared type of inertia force excitation. Although the intake tower is inherently a much more complicated system than this, the measured resonance curves would be expected to have the same general character.

In Fig. 16 are shown measured resonance curves for three different exciting force levels. The concrete tower was unreinforced, so it was not considered wise to vibrate the tower at appreciable tension stress levels. The resonant peak at the largest exciting force level corresponds to a load which just about relieves all of the gravity load compressive stresses at the base of the tower.

It will be noted from Fig. 16 that even at these relatively low stress levels, pronounced non-linear effects are present. There is a definite shift of the resonance frequency towards the lower frequencies as the amplitude level is increased, which would not occur in a linear system. It will also be noted that the amplitudes are not exactly proportional to the exciting force levels.

The most prominent difference between Figs. 15 and 16 is the presence of the double resonance peak in the experimental curves. These two peaks represent vibrations about the two principal axes of bending, and are a consequence of the fact that the vibration exciter force was not lined up with a principal axis, as shown in Fig. 13.

One of the most important dynamic parameters of a structure is the energy dissipation or damping in the structure. An approximate value of the equivalent viscous friction of the intake tower can be obtained by comparing Figs. 15 and 16. This comparison is complicated by the double resonance peak present in the measured curve. To avoid this difficulty, the following expedient was adopted. It will be noted in Fig. 15 that as the damping changes, there is a rapid change of the ratio of the resonance peak amplitude to the amplitude of the curve when it again has a horizontal slope at the high frequency side of the resonance curve. These ratios were calculated from the theoretical curves of Fig. 15, and are plotted versus damping in Fig. 17. By applying this same curve to the experimentally determined resonance curve, an approximate value of equivalent viscous damping can be obtained. Since both resonance peaks contribute to the amplitude levels, the above technique must be applied in an iterative way, but this is easy to do with the desired accuracy.[10] The advantage of this method of calculating damping for the present experiment is that data on the high side of the resonance peak is used, which is little modified by the double resonance peak.

The values of damping obtained for the concrete intake tower by this technique from Fig. 16 are:

Force Level Relative Magnitude	Per Cent Critical Damping
4.92	2.8
2.46	2.6
1.00	2.0

As would be expected, the damping is highest at the largest exciting force, and decreases as the dynamic stresses decrease.

The experimentally determined values of the natural frequencies were checked against analytical calculations made on a digital computing machine, and a satisfactory agreement was obtained, as shown in ref. 10. A major uncertainty in calculations of this type is the value of the effective dynamic modulus of elasticity of the concrete.

It will be seen from the distribution of the experimental points on the resonance curves of Fig. 16 that the control and adjustment of the new vibration exciter were very satisfactory. The stability of the system was such that even for such a lightly damped structure, the resonant peak could be accurately determined and duplicated. The agreement between the points on the ascending and descending portions of the curves indicates a high degree of reproducibility in the data.

[10] Keightley, W. O., Housner, G. W., and Hudson, D. E., Vibration Tests of the Encino Dam Intake Tower, Earthquake Engineering Research Laboratory, California Institute of Technology, Pasadena, California, 1961.

Conclusions. The field handling and installation characteristics of the new vibration exciter were completely satisfactory under unusually severe conditions. The mechanical operation of the unit was satisfactory, and the electric drive and control system was very successful. The portability and reliability of the speed control system under adverse field conditions was satisfactory, and the stability of speed control was excellent. No major design changes were indicated as a result of the field tests, and at present work is being completed on three additional identical units.

Acknowledgements. The development of the new vibration exciter system was sponsored by the State of California, Department of Public Works, Division of Architecture, Anson Boyd, State Architect. Mr. J. F. Meehan, Research Director, Schoolhouse Section, Division of Architecture, supervises this project, which was carried out through the Earthquake Engineering Research Institute under a committee composed of Professors L. S. Jacobsen, Stanford University, R. W. Clough, University of California, Berkeley, and G. W. Housner and D. E. Hudson, California Institute of Technology. The basic mechanical design of the exciter and drive unit was carried out at the California Institute of Technology by Professor D. A. Morelli, with the assistance of Mr. R. D. dePencier, and the electrical drive and control system was developed by Professor T. K. Caughey with the assistance of Mr. R. V. Powell. The arrangements for the Intake Tower test were made by Mr. S. B. Nelson, Chief Engineer of Water Works and Assistant Manager, Department of Water and Power, City of Los Angeles, and Mr. H. B. Hemborg of the Department of Water and Power. The field tests were under the general supervision of Mr. W. O. Keightley, of the California Institute of Technology.

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Fig. 16. Acceleration Resonance Curve at Top of Tower

Fig. 17. Ratio of Peak Acceleration to Higher Frequency Horizontal Slope Amplitude.

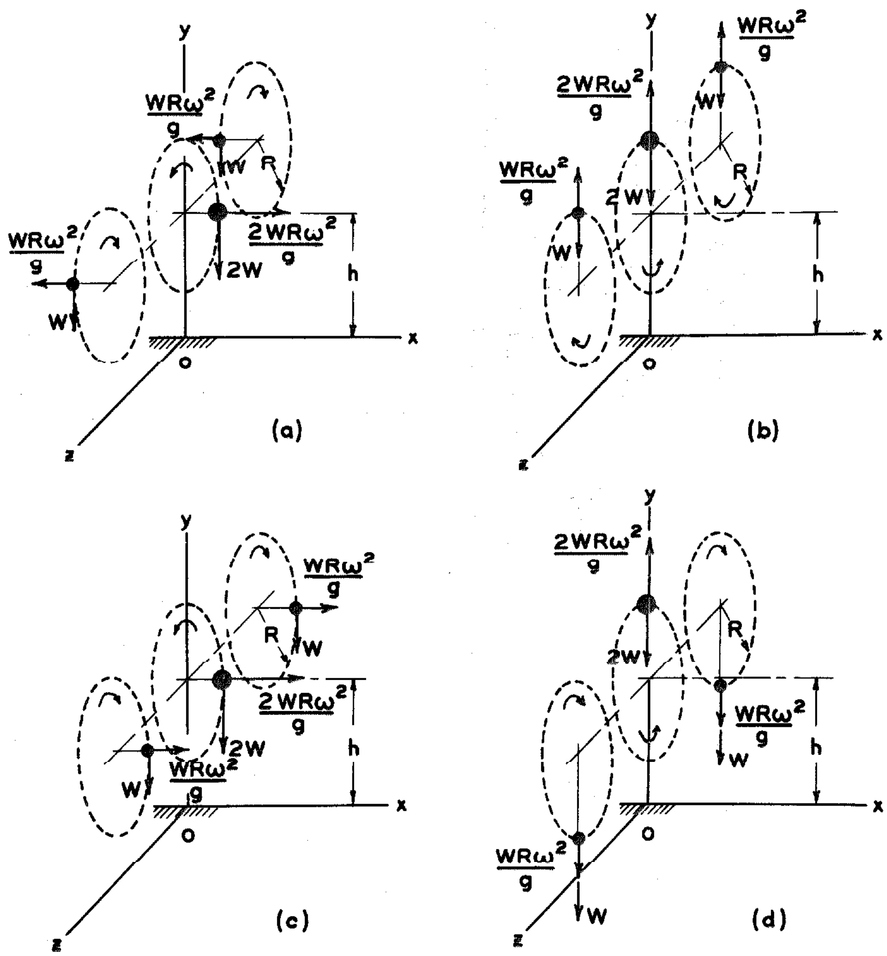


Fig. 1. Three-mass Horizontal Shaft Vibration Exciter

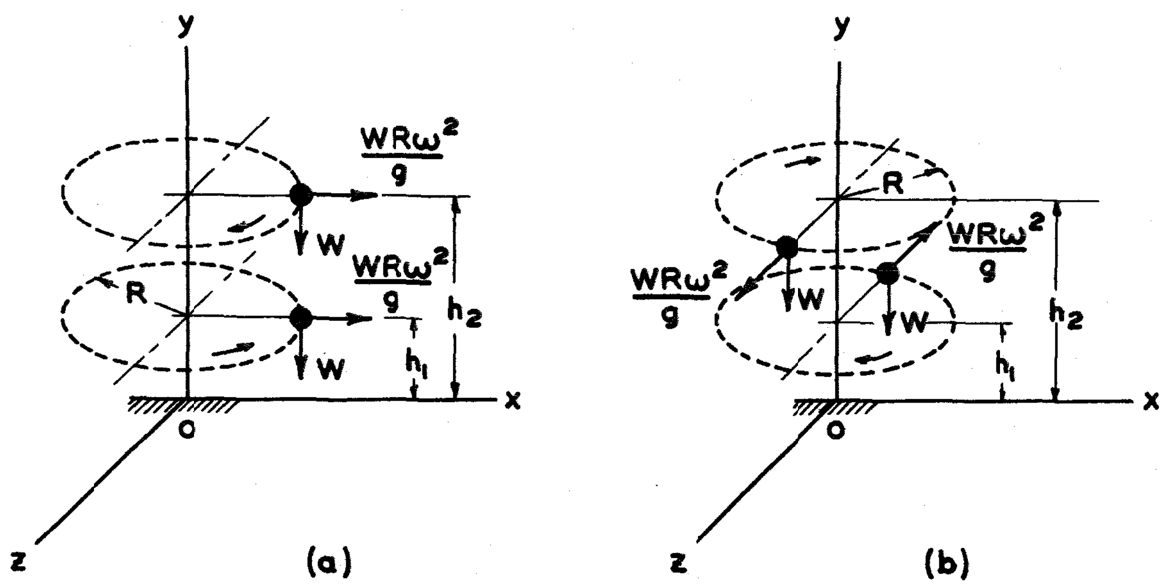


Fig. 2. Two-mass Vertical Shaft Vibration Exciter

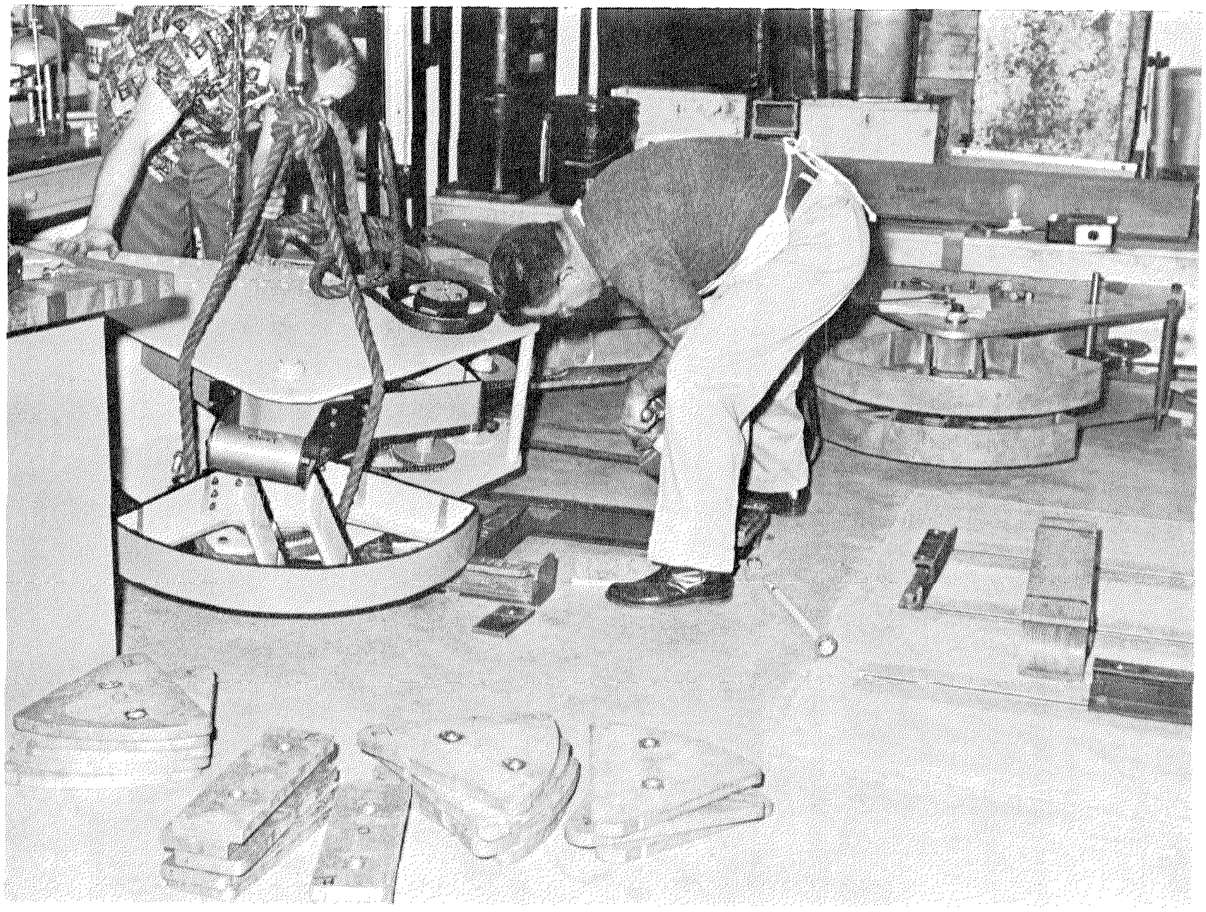


Fig. 3. Vibration Exciter Unit Being Installed in Laboratory

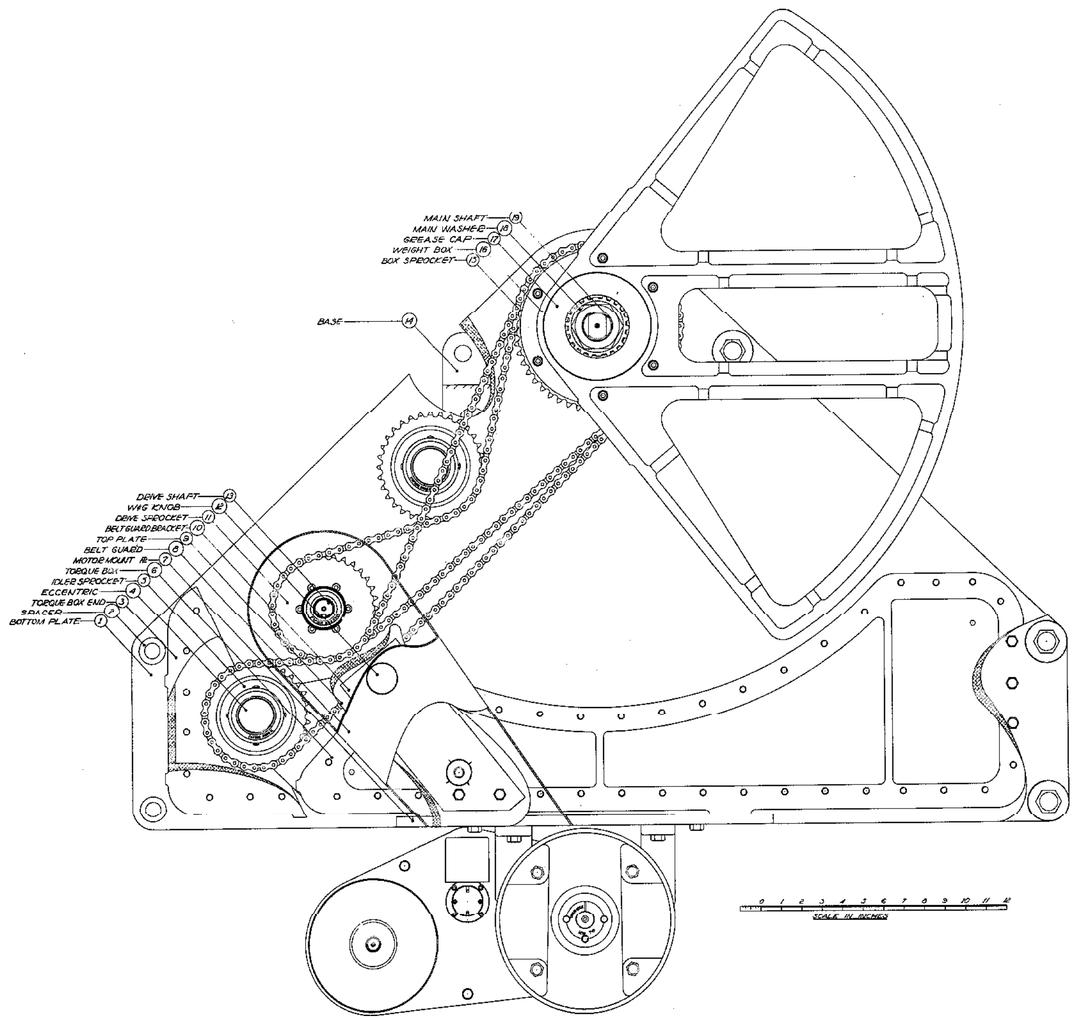


Fig. 4. Assembly Drawing of Vibration Exciter

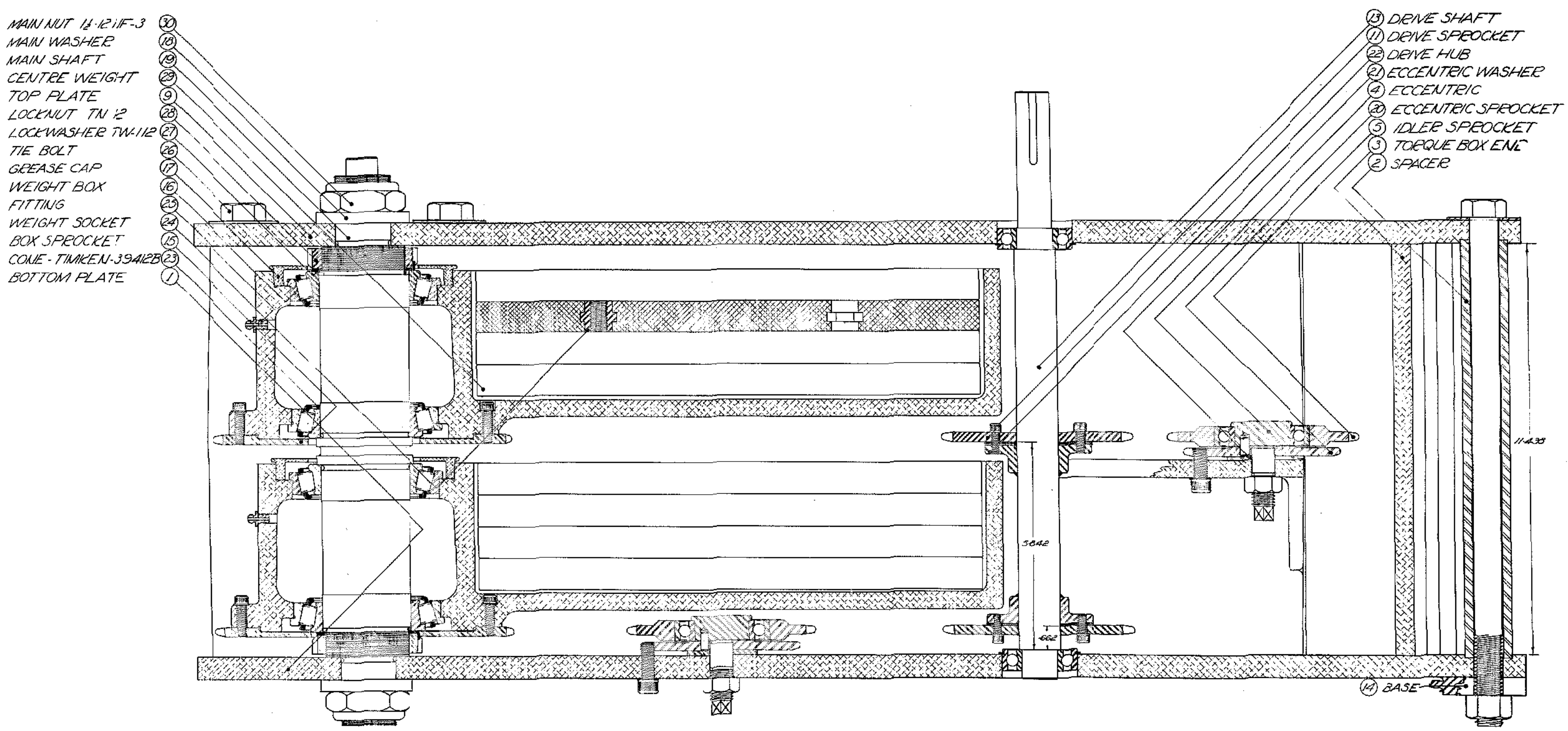


Fig. 5. Assembly Drawing of Vibration Exciter

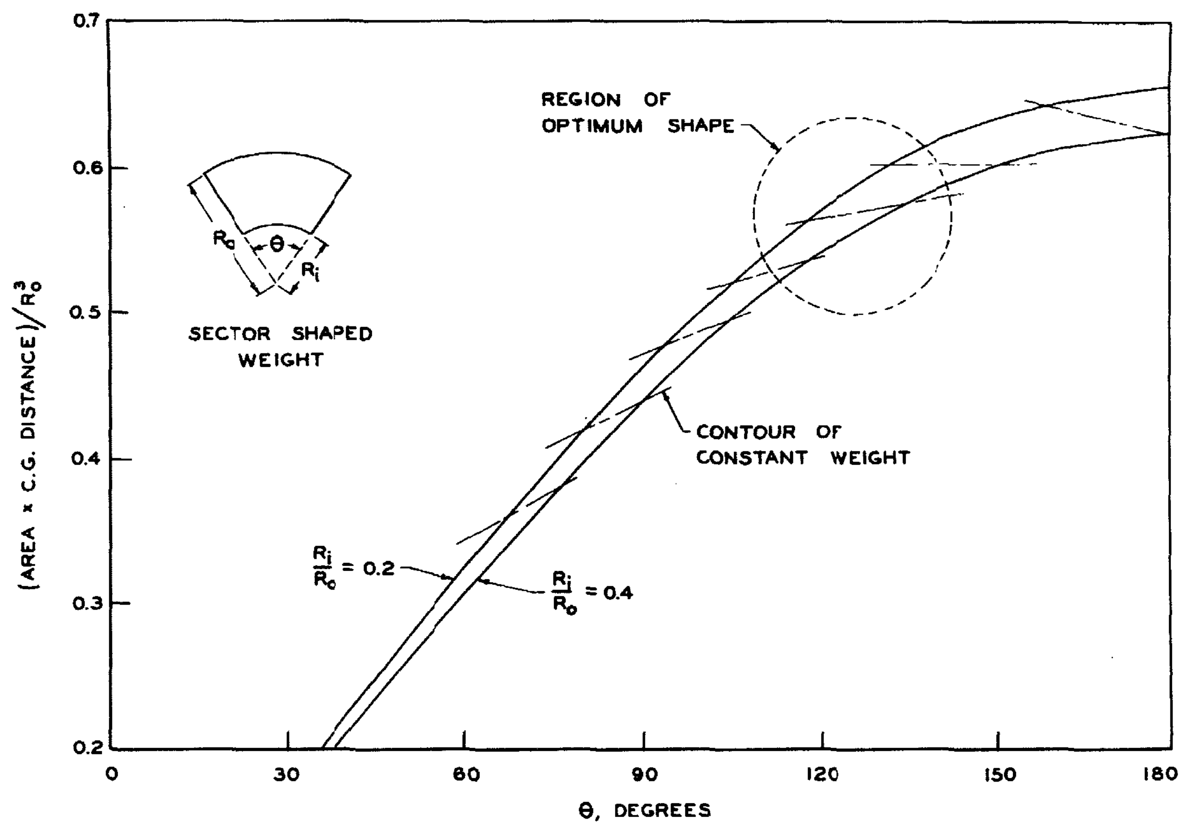


Fig. 6. Optimum Configuration of Eccentric Weight

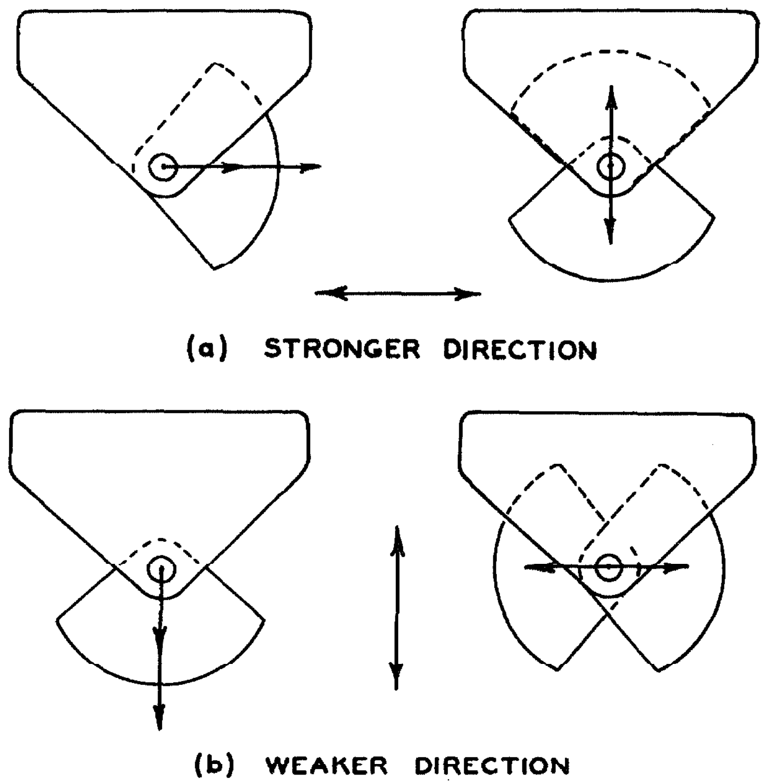


Fig. 7. Orientation of Vibration Exciter Force

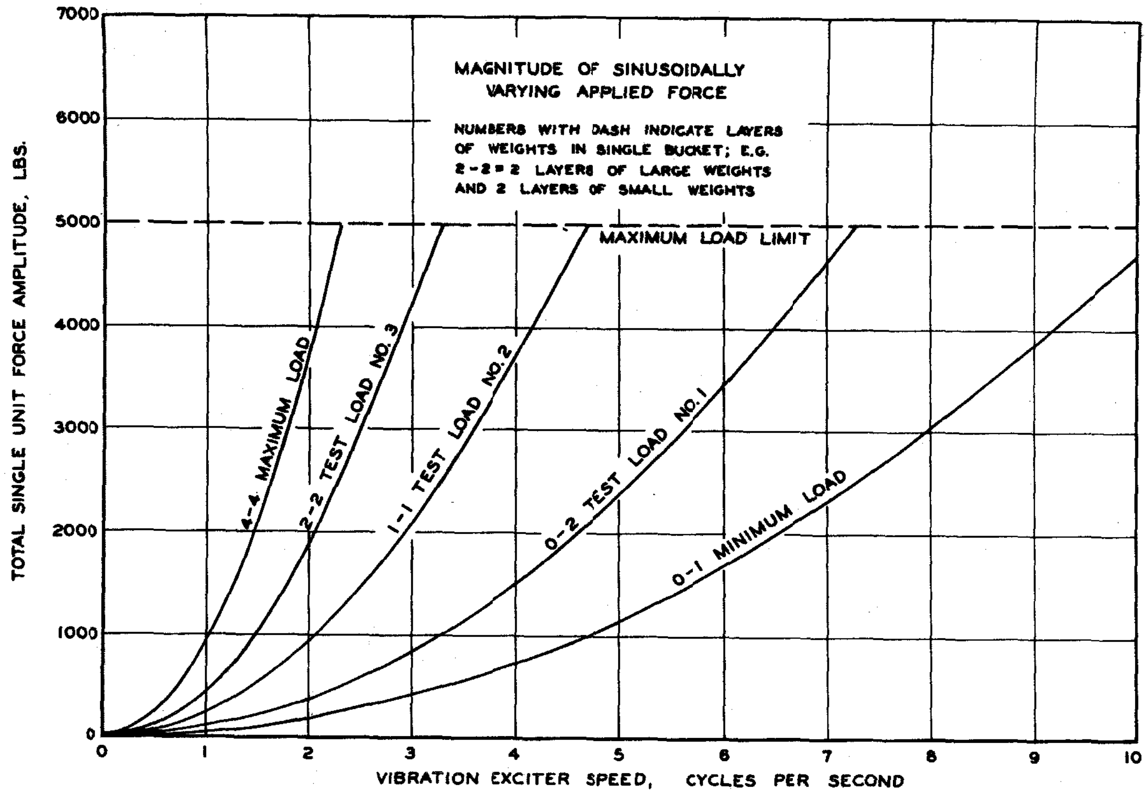


Fig. 8. Vibration Exciter Force Output for Counterbalanced Exciter

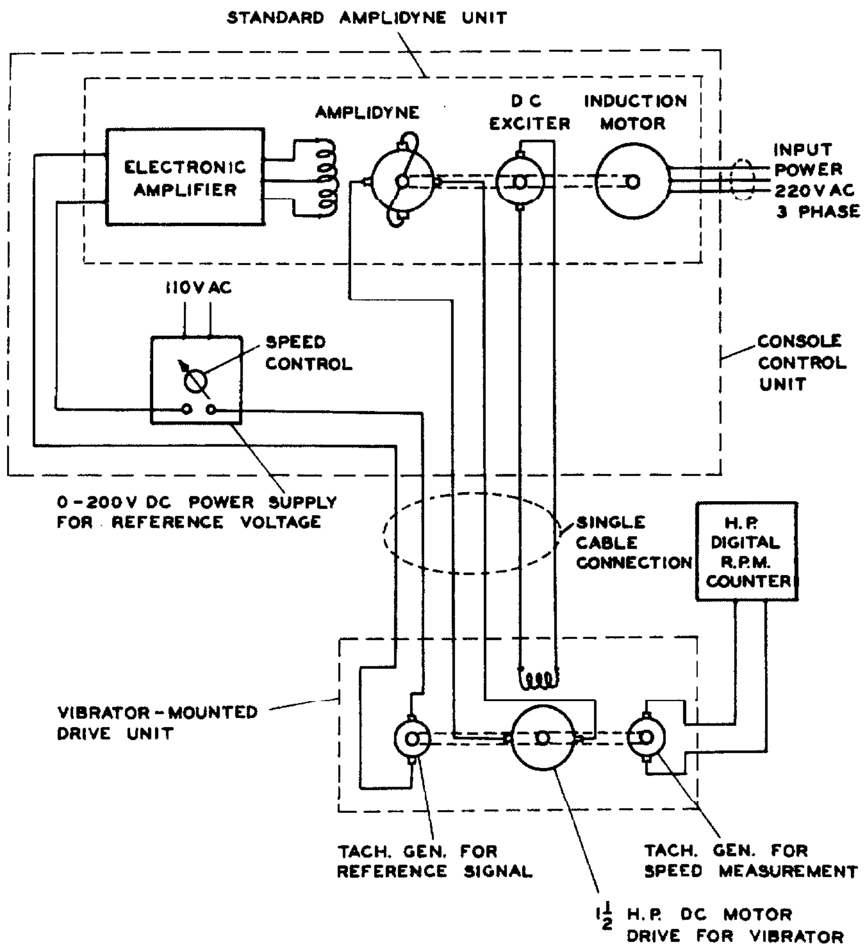


Fig. 9. Schematic Diagram of Electrical System for Vibration Exciter Speed Control

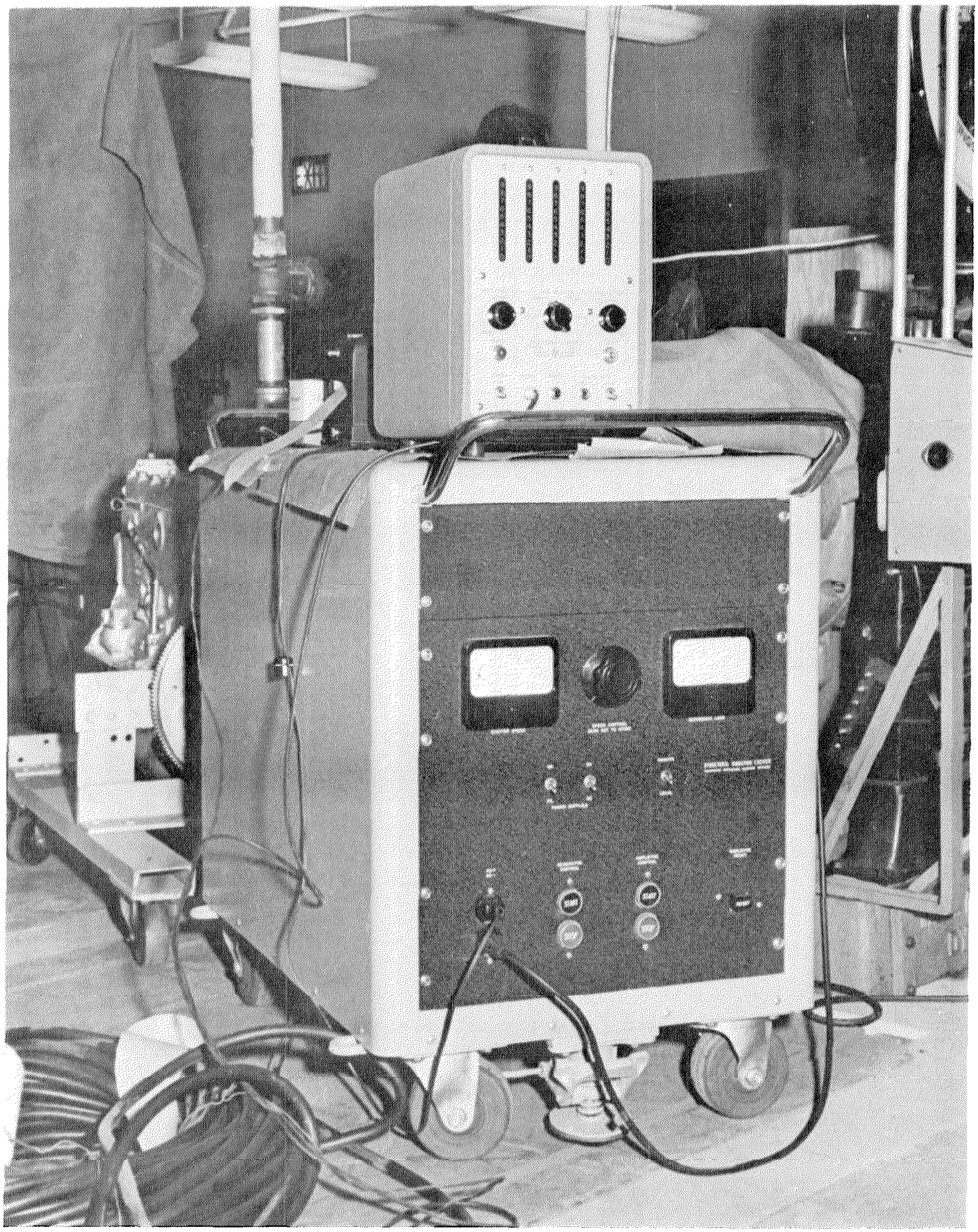


Fig. 10. Speed Control Console for Vibration Exciter

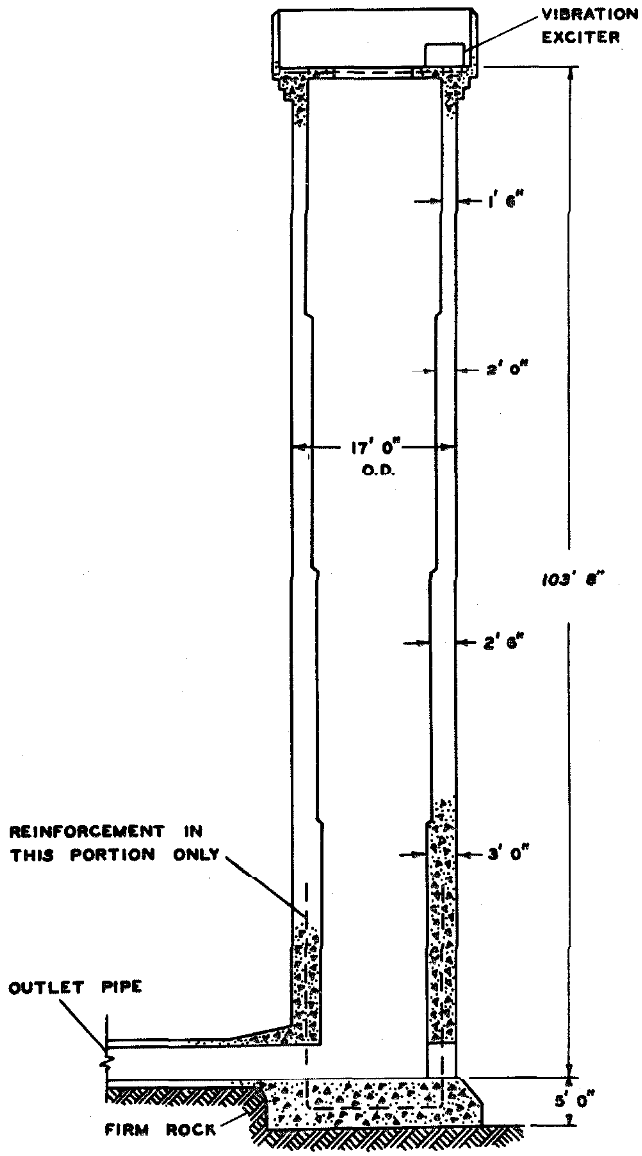


Fig. 11. Construction and Dimensions of the Encino Dam Intake Tower

Fig. 12. Vibration Exciter Unit Being Hoisted to Top of Tower

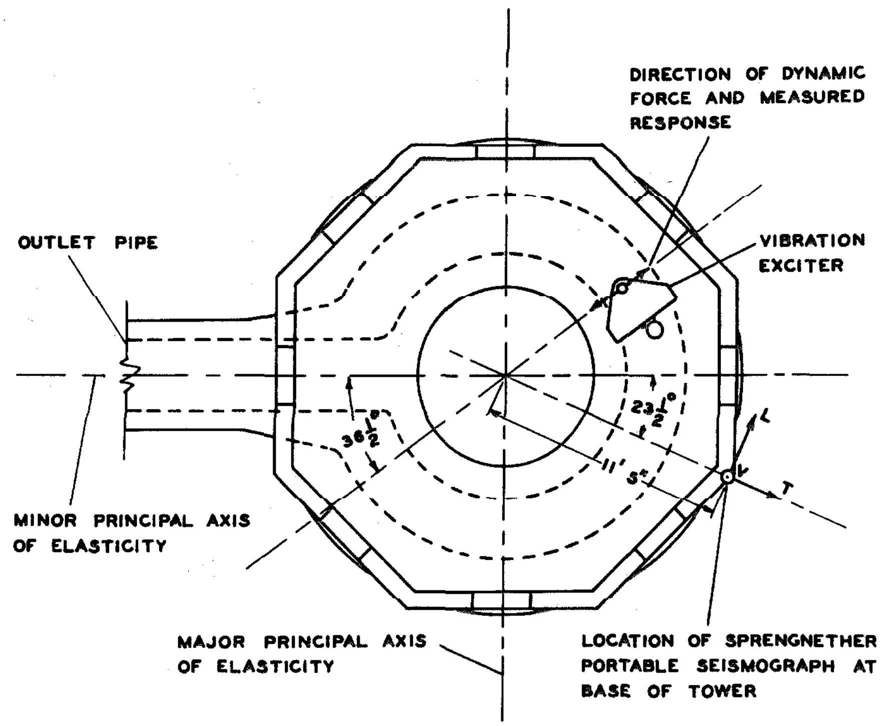
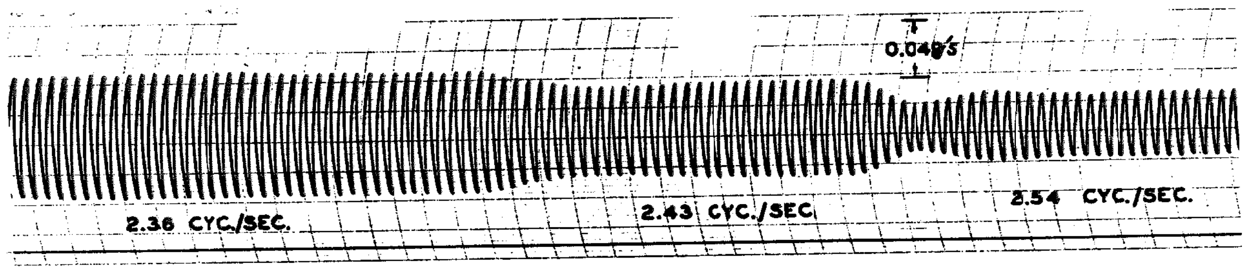


Fig. 13. Plan View of Top of Tower Showing Location of Vibration Exciter Unit



ACCELERATION-TIME CURVES, TOP OF TOWER BELOW FLOOR SLAB
LOAD NO. 3 (2-2) FREQUENCY FROM DIGITAL COUNTER

Fig. 14. Sample Record of Tower Acceleration During Forced Vibration Resonance Test. Data for Fig. 16.

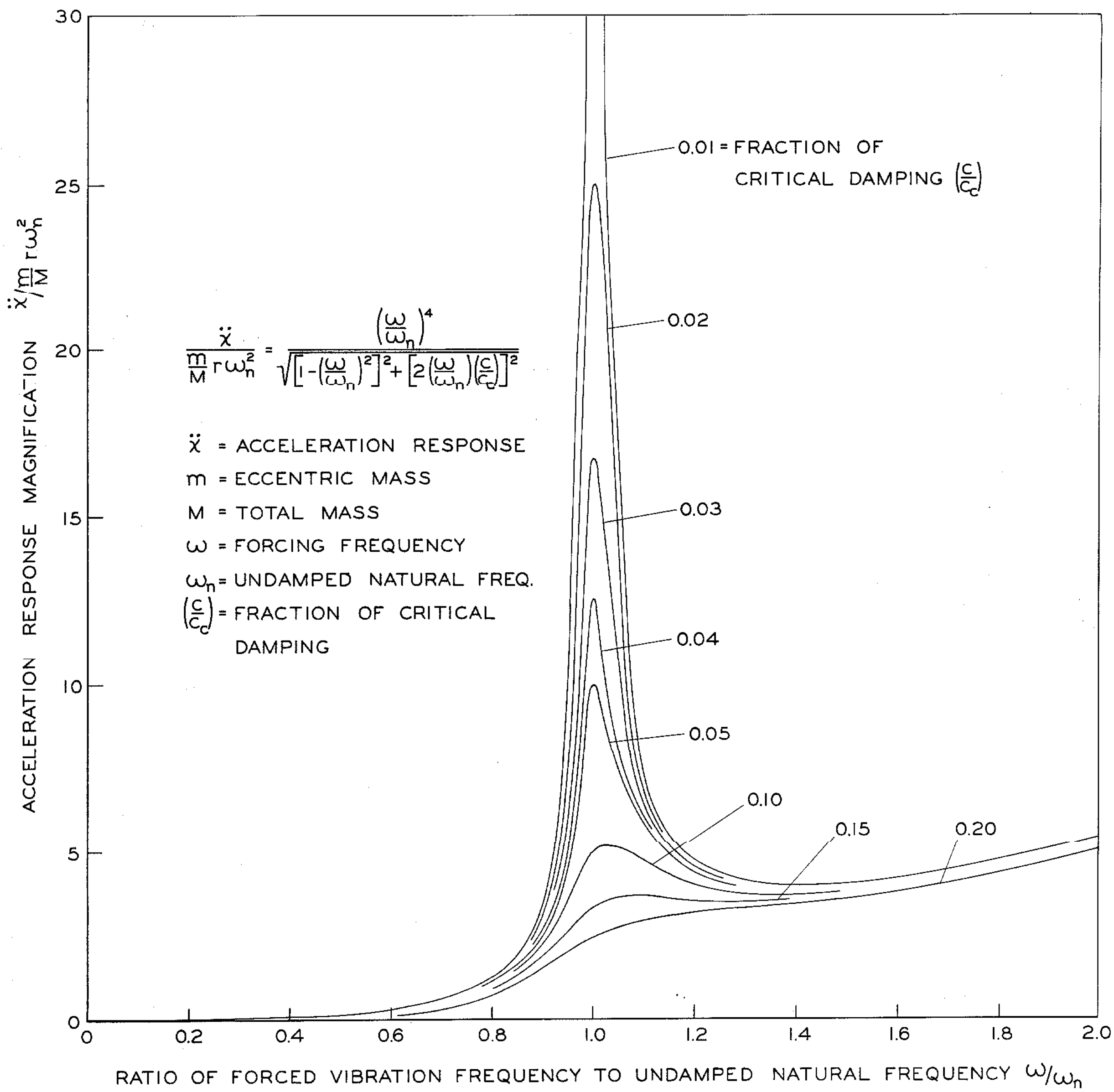


Fig. 15. Computed Acceleration Response Resonance Curve

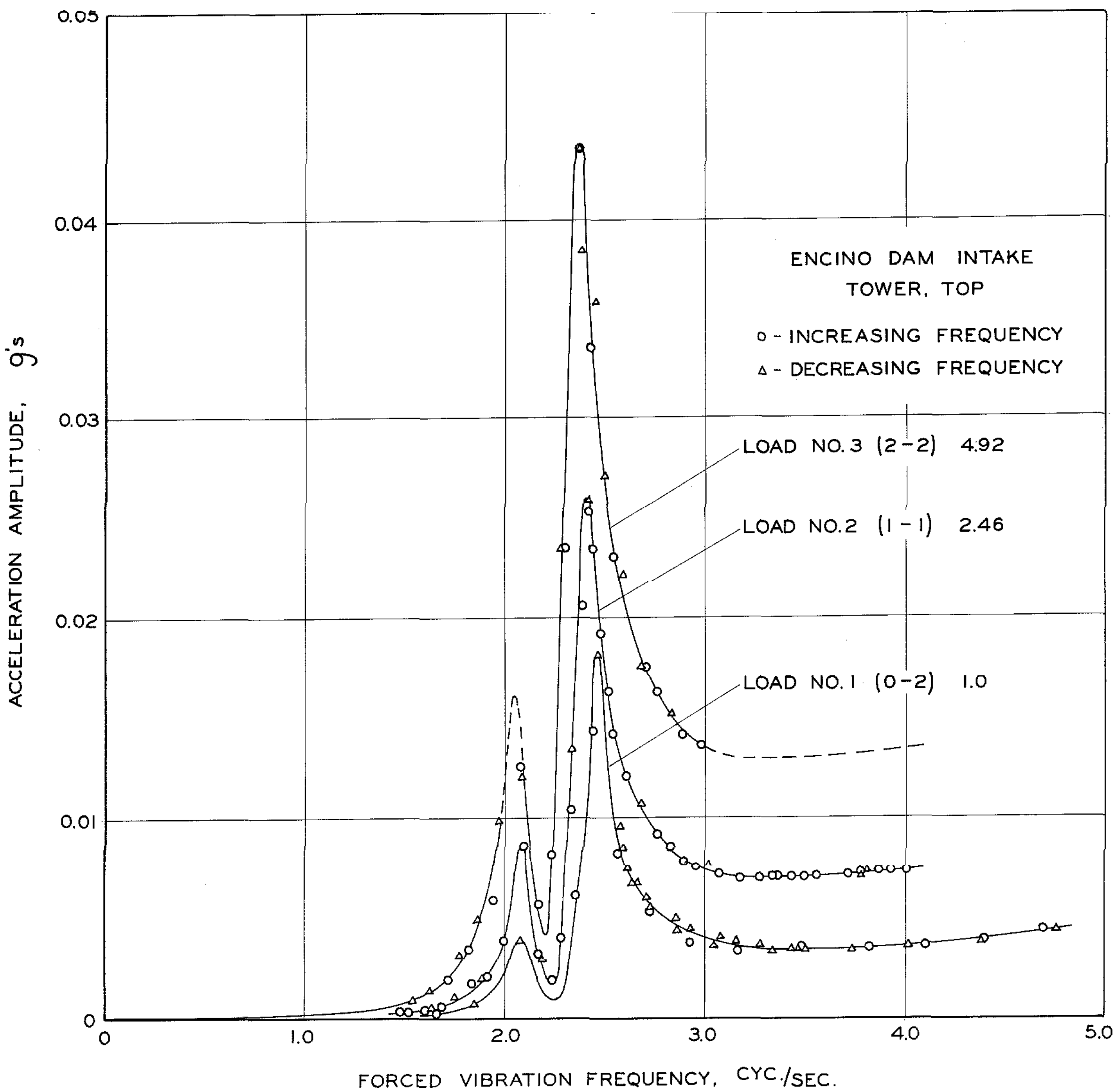


Fig. 16. Acceleration Resonance Curve at Top of Tower

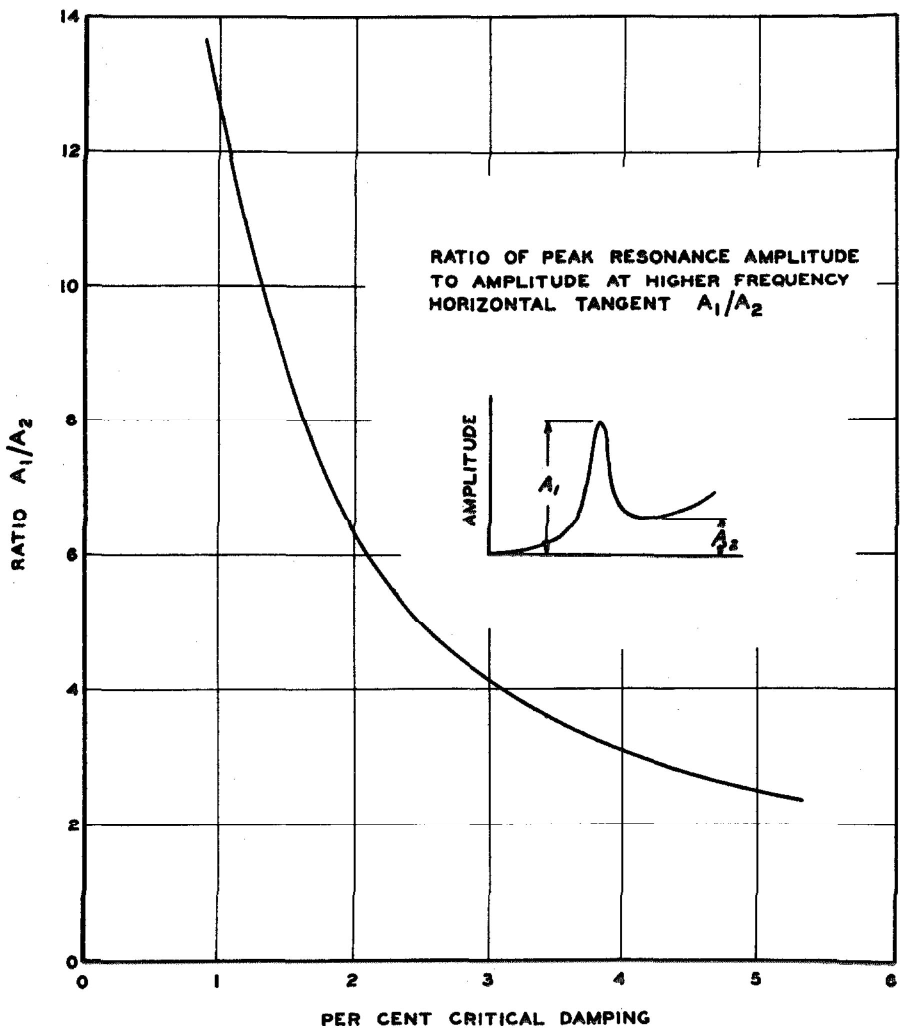


Fig. 17. Ratio of Peak Acceleration to Higher Frequency Horizontal Slope Amplitude.