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EXPERIMENTS WITH VIBRATION NEUTRALIZERS

BY R. T. McGOLDRICK

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MAY 1938

REPORT NO. 449

EXPERIMENTS WITH VIBRATION NEUTRALIZERS

by

R. T. McGoldrick

U.S. Experimental Model Basin
Navy Yard, Washington, D.C.
May 1938 Report No. 449

EXPERIMENTS WITH VIBRATION NEUTRALIZERS

Summary

This report describes experiments with various forms of vibration neutralizers or dynamic vibration absorbers. These experiments were undertaken with a view to determining the feasibility of applying such devices to the problem of eliminating general hull vibration as well as certain local vibrations on board ship. Various attempts to improve the operation of neutralizers by subjecting the inertia element to periodic impulses properly phased with respect to the forces acting on the vibrating structure are discussed. The problems arising in attempting to build large size neutralizers such as would be required on ships are also discussed.

Introduction

The principle of the vibration neutralizer has been known for many years. The term "dynamic vibration absorber" is frequently used for this device and the term "resonance damper" as used by Geislinger[6]* would be a convenient and descriptive term.

If a structure is found to vibrate violently due to resonance when subjected to a harmonic force of a definite frequency, the vibration may theoretically be entirely eliminated by attaching to the structure by spring connection a relatively small mass, free to vibrate in the direction of the exciting force, and satisfying the relation:

$$\sqrt{\frac{k}{m}} = \text{exciting frequency} \times 2\pi$$

Here m is the small mass, and k the constant of the spring by which it is attached to the structure. This relation is proved in the theory of coupled systems with two degrees of freedom, where it is shown that the mass m will vibrate at such an amplitude and in such phase as to set up periodic force reactions on the structure always equal and opposite to the exciting force. But, while vibration is eliminated at the original resonance frequency, there now arise two other frequencies or critical speeds at which violent vibration may occur, one higher and one lower than the original critical speed. Thus, if in the system illustrated in Fig 1 the secondary system (k,m) is tuned exactly to the original natural frequency of the primary system (K,M), the two new critical frequencies (assuming no damping) are given by the two positive values of n in the relation:

$$n = n_0 \sqrt{1 + \frac{\mu}{2} \pm \sqrt{\mu + \frac{\mu^2}{4}}}$$

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*Note: Numerals indicate references listed in the bibliography at the end of the report.

where μ is the ratio of the secondary mass to the primary mass, n is the exciting frequency, and n_0 is the natural frequency of the primary system alone. At these new critical frequencies the exciting force and the force reaction from the neutralizer are in phase and the amplitudes of both masses become infinite, whereas at the original resonance point the exciting force and the force reaction from the neutralizer are in opposite phase and of equal magnitude, so that they cancel one another.

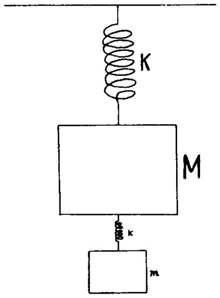


Fig. 1 Simple linear system with undamped neutralizer.

The resonance curve of the system shown in Fig 1 is given in Fig 1a. There is also plotted the resonance curve of the system M,K without the neutralizer. These curves represent the amplitude of the mass M when subjected to a harmonic force of constant amplitude P_0 and varying frequency n. It is clear from these curves that unless the frequency stays within very narrow limits in the neighborhood of the original resonance frequency n_0 the neutralizer does not improve conditions.

When damping is introduced between the primary and secondary systems the situation is considerably altered. The danger of excessive amplitudes at the upper and lower criticals is removed but the neutralizing effect at the original critical is diminished. In Fig 1b is shown the resonance curve of the system with the neutralizer when the optimum degree of damping is introduced.

In the theory of the damped neutralizer[2] it is shown that for any given value of μ and of the ratio of the natural frequency of the secondary system to that of the primary system there is an optimum damping which can be calculated. The determination of the constants in actual mechanical vibration systems, however, consisting of complex structures such as ships, is extremely difficult, so that results are best obtained experimentally.

The electrical analogies to mechanical systems are frequently helpful in predicting the behavior of the latter. One analogy to the undamped neutralizer is the wave trap used to eliminate undesired frequencies from radio receivers. In Fig 2 the natural frequency of the L,C circuit is $\frac{1}{2\pi\sqrt{LC}}$ If an alternating voltage is induced in the antenna of this same frequency, no current will flow through the receiver but an oscillation will exist in the wave trap. The wave trap corresponds to the secondary system in Fig 1, the inductance L representing the mass m, and the reciprocal of the capacitance $\frac{1}{C}$ representing the spring constant k.

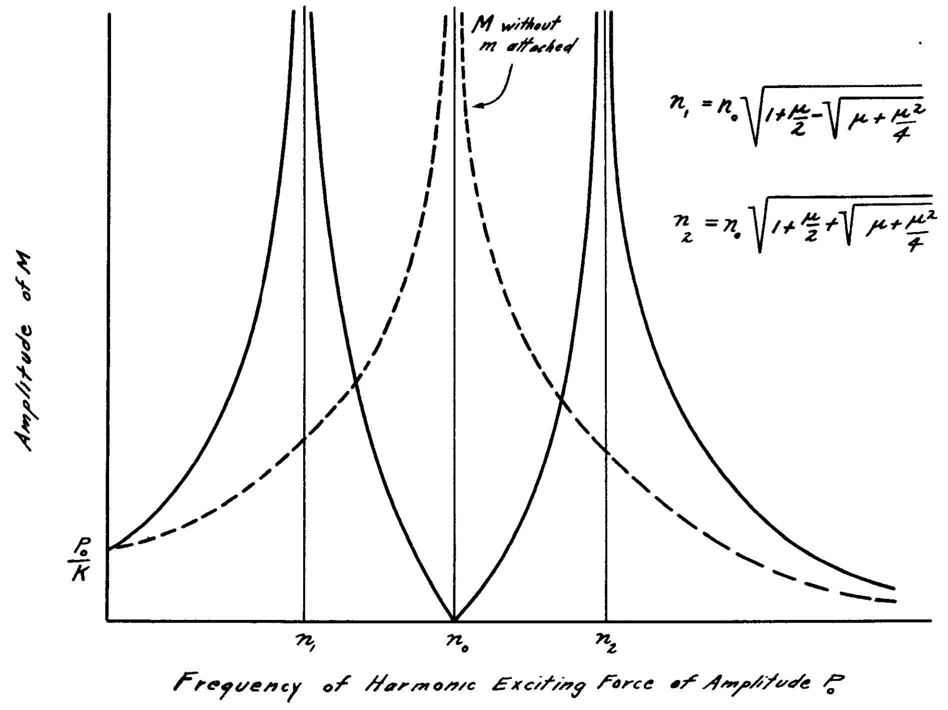


Fig. 1a Resonance curves of mass M of Fig. 1

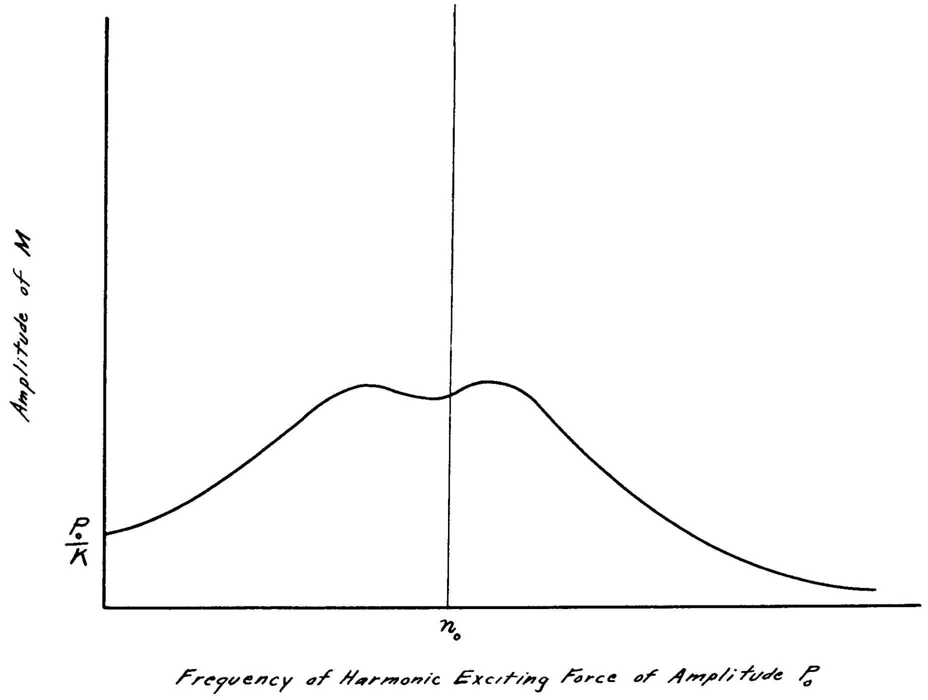


Fig. 1b General form of resonance curve of mass M of Fig. 1 with the optimum degree of damping between M and m

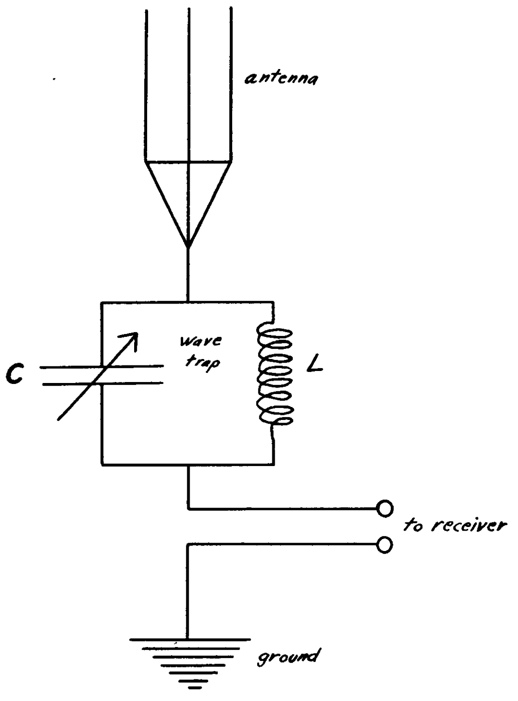


Fig. 2 Wave trap

An early application of the principle of the vibration neutralizer to ship vibration was made by Frahm who secured a patent on various types of the device in 1911 (U.S. Patent No. 989,958). Although some success was met in reducing local vibration by this means, few applications to the elimination of hull vibration are on record. An installation capable of reducing the vibration of the complete hull structure is reported in the case of the Italian motorship Maria[3]. Several features of this installation should be noted. The weight of the neutralizer is 0.1% of the displacement of the vessel (11,700 tons). The inertial element consists of a tank mounted on coil spring under compression. The tank is divided into compartments into which varying amounts of water can be admitted through valves, thus permitting tuning of the system by varying its mass. The exciting force to be neutralized in this case is due to unbalance in machinery located amidships, but the neutralizer is installed in the stern (on the opposite side of the node). The exciting frequency is about 200 per minute. It is claimed that the vibration is reduced by 94 per cent.

There is also mention in the literature[4] of the use of neutralizers by the British Navy. In these installations, which were made on small craft such as tugs, variable damping was introduced by causing a fluid to surge back and forth through a throttling valve during oscillation of the neutralizer element. These installations date back to 1902.

Experimental Set-Up

In order to obtain a primary system of low natural frequency a thirty foot length of twelve inch I-beam was selected, weighing 40.8 lb per ft. This was set on two supports resting on the edges of the flanges. In the photograph Fig. 3 one of the supports is shown. The exciting force was furnished by a small Losenhausen vibration generator previously used to vibrate sheet steel floating models and described in E.M.B. Report No. 395. Amplitudes were measured with the type B pallograph described in the same report.

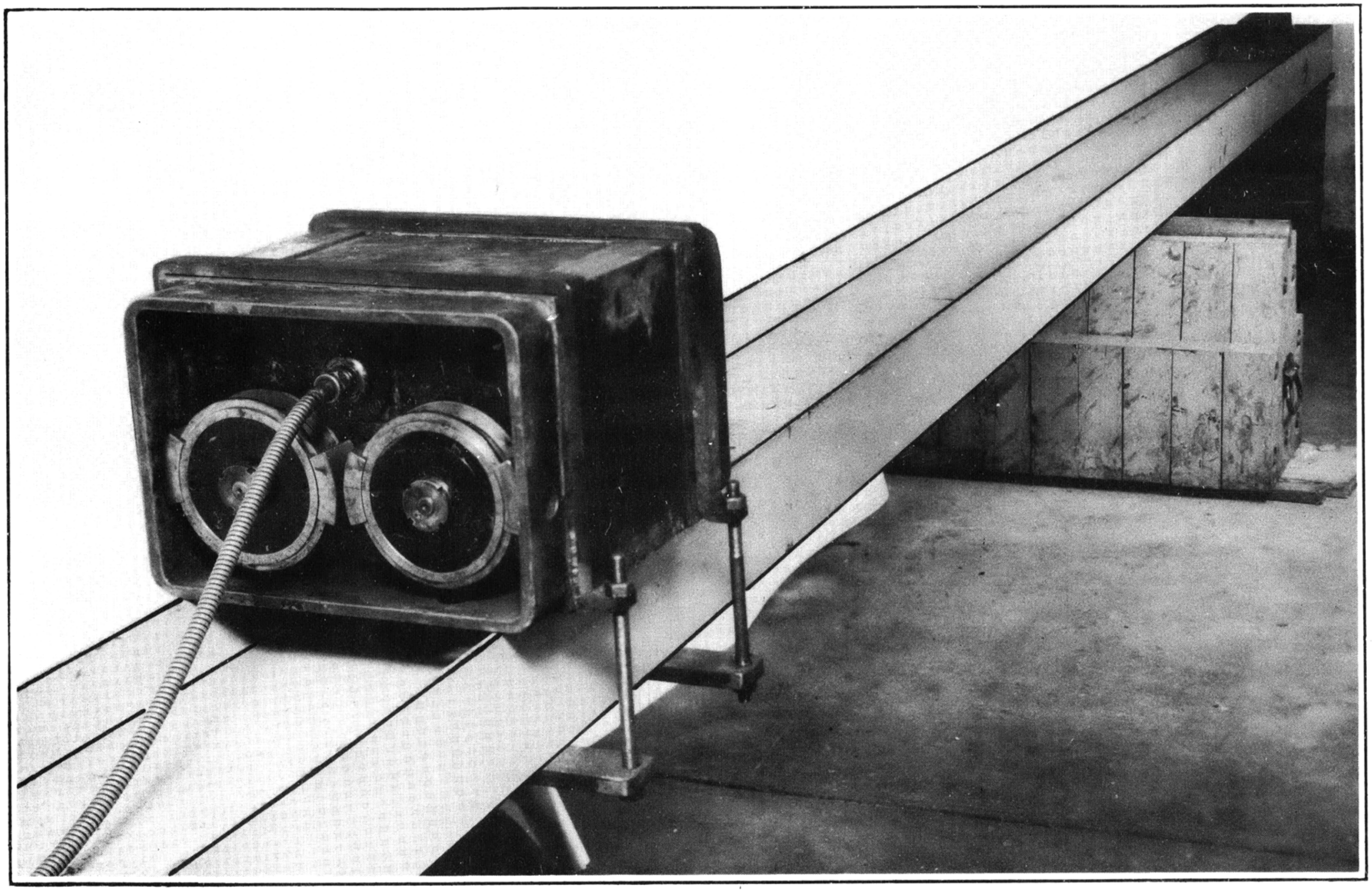
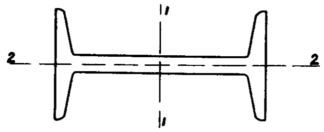


Fig. 3 I-beam and vibration generator used in experiments

The principal characteristics of the beam are as follows:

Depth: 12"
Width of flange: 5-1/4"
Weight: 40.8 lb/ft
Length: 30 ft
Total weight: 1224 lb

I_{2-2} : 13.8 in^4 = 0.0959 ft^2in^2
I_{1-1} : 268.9 in^4
$E = 30 \times 10^6$ lb/in^2 (assumed)



The two-noded natural frequency of the beam referred to axis 2-2 and computed by the free-free uniform bar formula (nodes 6'8½" from the ends) is:

$$n_o = \frac{9\pi}{8}\sqrt{\frac{EI}{ML^3}} = \frac{9\pi}{8}\sqrt{\frac{30 \times 10^6 \times 0.0959}{\frac{1224}{32.2} \times 27{,}000}} = 5.90 \text{ per sec.}$$

In this formula

n_o is the natural frequency per second
E is the modulus of elasticity in lb/in^2
I is the moment of inertia of section in ft^2in^2 units
M is the total mass of the beam in slugs
L is the total length in ft

The three-noded natural frequency is $(\frac{5}{3})^2$ times the two-noded or

$$\frac{25\pi}{8}\sqrt{\frac{EI}{ML^3}}$$

An approximate formula for the natural frequency of uniform beams supported at the free-free nodes is:

$$n_o = \frac{9000}{L^2}\sqrt{\frac{I}{w}}$$

where I is expressed in in^4 and w is the weight per unit length in pounds per foot (assuming the modulus to be 30×10^6 lb/in^2). As above the three-noded frequency would be 25/9 times the two-noded.

Actually the frequency was lowered by the addition of the mass of the vibration machine and recording instruments to the beam, and an attempt was made in the experiments to obtain a natural frequency of 5.0 per second or 300 per minute.

The logarithmic decrement of the system was found to be about 0.007, indicating very little internal damping within the range of amplitudes used. This value is of the same order of magnitude as found previously on the U.S.S. HAMILTON.

Another feature of the system in question deserves mention at this point, namely its effective mass. As previously pointed out the constants of mechanical systems are frequently difficult to determine. In the theoretical treatment mechanical systems are usually represented by lumped masses connected by linear springs, the masses being free to oscillate in one direction only. It is obvious that in the case of a ship or beam, vibrating flexurally with nodes, the effective mass is not equal to the total mass since the amplitude varies along the length. In the case of the flexural vibration of a beam the same harmonic force will produce very different amplitudes of the beam as a whole when applied at different points along the length. We may consider that at any point of the beam there is an effective mass m_e, an effective spring constant k_e, and an effective damping constant c_e, which if substituted in the standard formulas for linear oscillators will give the actual amplitudes and natural frequencies. Thus as for linear oscillators

$$n_o = \frac{1}{2\pi}\sqrt{\frac{k_e}{m_e}}$$

and

$$x_o = \frac{P_o}{\sqrt{(c_e \omega)^2 + (k_e - m_e \omega^2)^2}}$$

where x_o is the amplitude at the point where the force is applied

P_o is the amplitude of the harmonic exciting force of circular frequency ω (in the experiments in question P_o would be the vertical component of the centrifugal forces of the rotating weights of the vibration generator)

m_e is the effective mass referred to this point

c_e is the effective damping constant referred to this point

n_o is the natural frequency

If the above relations hold, it is obvious that the effective mass at any point of the beam could be estimated by measuring the change in the natural frequency due to a known increment of mass at that point. For example, if we had a simple linear system consisting of a concentrated mass m connected by a spring of constant k to infinite mass, and were required to determine m and k, we could first measure the natural frequency, then add a known mass Δm and again measure the natural frequency. If we call the two natural frequencies n_1 and n_2 we then have the two equations

$$n_1 = \frac{1}{2\pi}\sqrt{\frac{k}{m}}$$

$$n_2 = \frac{1}{2\pi}\sqrt{\frac{k}{m + \Delta m}}$$

in which the only unknowns are k and m which can thus be determined. Coming back to the beam we find that the same Δm will produce different changes in the natural frequency according to the position at which it is added, but the effective mass and effective spring constant at any point can be determined by the same method as for the simple linear system.

Thus the effective mass of the thirty foot I-beam (whose actual mass is 1224 lb) was estimated. The values tabulated below show to what extent the effective mass may vary.

Distance between supports (ft)	Position at which m_e is estimated	m_e (lb)
13	center	2600
13	end	230
28½	center	582

In attempting to apply the theory of Hahnkamm[2] these values of m_e should be used in calculating the ratio μ rather than the actual mass of 1224 lb. In connection with effective mass it should also be pointed out that in using a beam resting on two supports as a model of a ship the assumption must be made that the total mass of the ship and its mass moment of inertia about an axis through the mass center are so great in proportion to the exciting forces that the motions of the ship as a rigid body are negligible, that is, the ship is restricted to the same number of degrees of freedom as the beam supported at the nodes; otherwise the restraints applied to the beam render the two cases incomparable. In the vibration tests on the U.S.S. HAMILTON with the vibration generator installed in the bow it was observed that the motion was a combination of flexural vibration and pitching of the ship as a rigid body. Had the ship been infinitely rigid pitching alone would have occurred, except for a slight vertical oscillation of the mass center.

Another matter of considerable interest in such problems is the power required to maintain vibration. A wattmeter is supplied with the vibration generator for measuring this power and it is shown theoretically[5] that the peak of power consumption occurs at resonance. However, while the wattmeter is so connected as to measure power taken by the armature only, excluding field excitation, the measurement necessarily includes resistance losses in the armature, as well as bearing friction, brush friction, and air resistance losses. A series of power measurements showed that up to double amplitudes of one inch the power required to maintain vibration of the beam is negligible in comparison with the machine losses; whereas from one inch amplitude up, the power increases as about the 5/2 power of the amplitude. The small amount of power required to maintain vibration at ordinary amplitudes was also confirmed in tests on the U.S.S. HAMILTON with the large vibration generator. As regards the problem of neutralizing vibration, this indicates that the neutralizer is not required to dissipate considerable power. In fact, from another point of view it is clear that the ideal undamped neutralizer, by setting up a force equal and opposite to the exciting force, would prevent any power whatever from flowing from the source to the structure. Practically, sufficient power must be supplied to oscillate the neutralizing mass against its frictional resistance, but this need not equal the power which would otherwise go into vibrating the ship, and the latter is in any case small.

The Experimental Neutralizer

In the design of the experimental neutralizer the first consideration was to keep the mass as small as possible. The first type tried was a double cantilever or pair of reeds with movable weights as shown in Fig 4. This was clamped to the center of the beam above the vibration generator, the latter being slung under the beam. Due to the flexibility of the supporting structure, coupling between the two reeds existed causing a tendency to transfer the vibration back and forth from one to the other, and it was difficult to keep them vibrating in step. Nevertheless, when the system was properly tuned, the neutralizing action was quite appreciable, and the vibration of the beam was reduced by 50% under the best conditions.

Next various forms of cylindrical weight and helical spring combinations were tried culminating in the form shown in Fig 5. This neutralizer has the following features: The natural frequency may be varied either by changing the mass (removable brass discs) or by changing the spring constant (by threading the coil spring through the special nut on the upper plate). After tuning, the weight can be brought back to its original position by raising or lowering the upper plate. This neutralizer proved quite effective at both the two and three noded frequencies of the beam, the conditions being as follows:

Type of Vibration	Two-noded	Three-noded
Distance between supports	13 ft	19 ft
Location of vibration generator	center	one end
Location of pallograph	one end	end opposite to vibration generator
Location of neutralizer	same as pallograph	same as pallograph
Ballast	83 lb at end opposite neutralizer	none
Neutralizing mass	7.1 lb	6.2 lb
Number of coils of spring effective	31½	8
Frequency	5 per sec	11½ per sec
Neutralizing effect	70%	80%

It is to be noted from the above cases that the neutralizer need not be in line with the disturbing force. In fact in the two-noded case, with the vibration generator in the center, the neutralizer is much more effective when at the end than when directly over the vibration generator. This is to be expected in view of the relative values of effective mass at the two positions previously stated. In these experiments speed control is a difficult problem for the reason already mentioned that the neutralizer is not a power absorbing device. The procedure is to lock the neutralizer and adjust the speed of the vibration generator until the beam is in resonance, recording the amplitude on the pallograph. The neutralizer element is then released and the change in amplitude noted on the record. However,

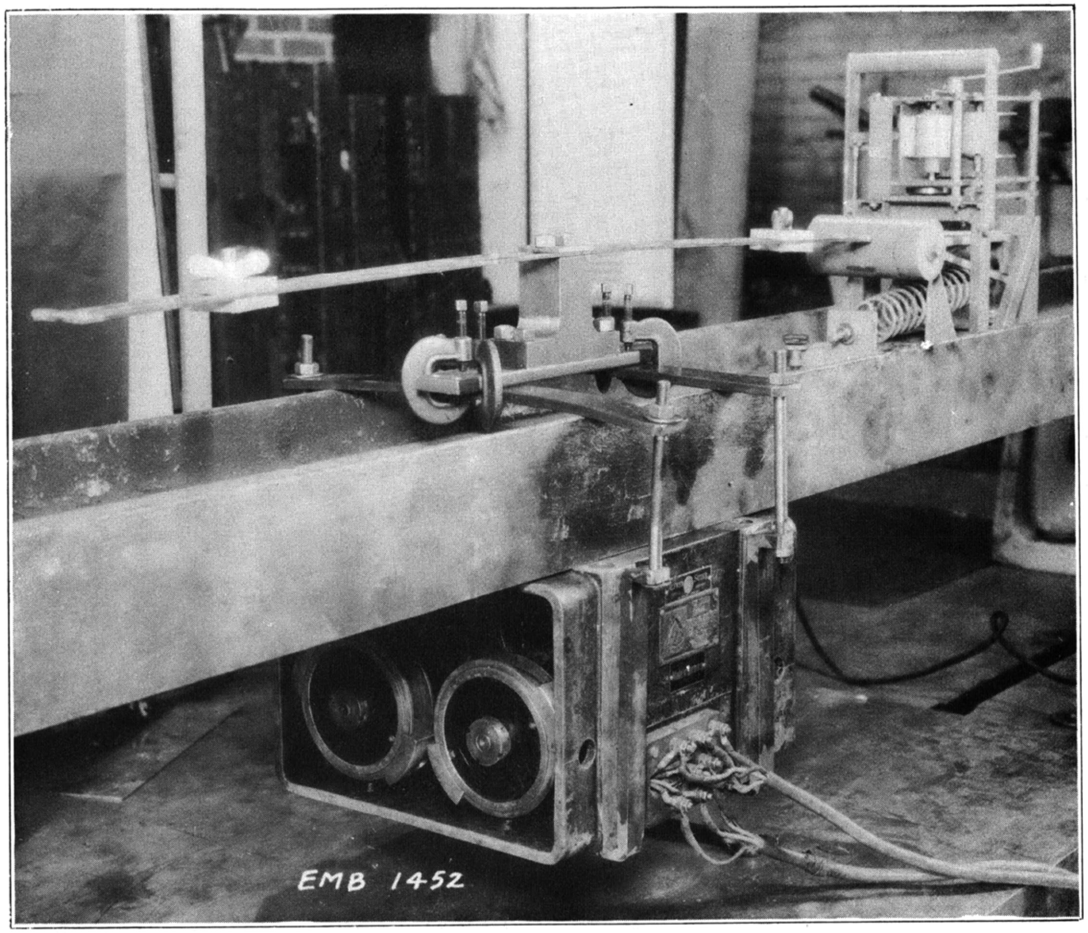


Fig. 4 Experimental set-up showing double cantilever type of neutralizer.

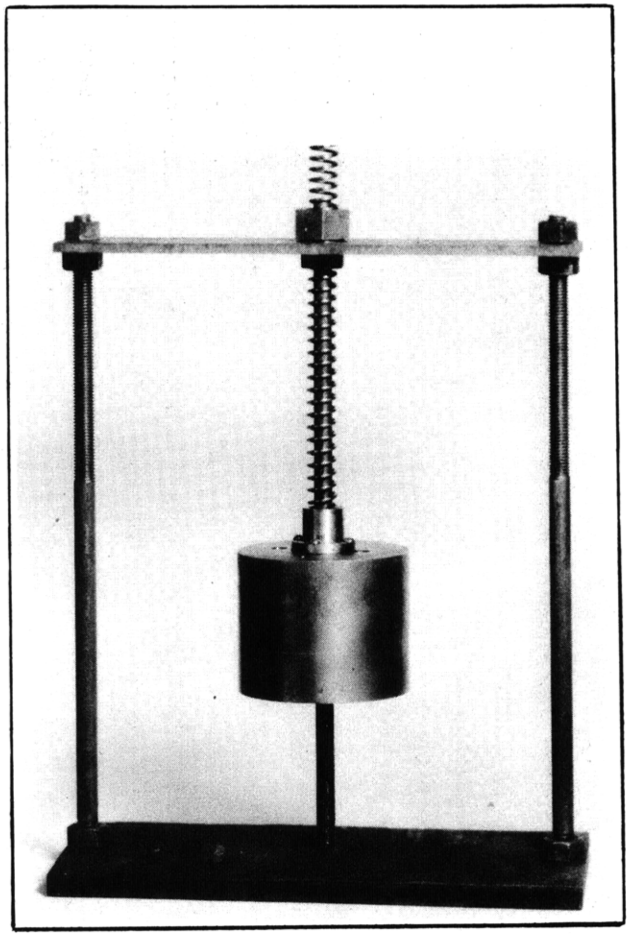


Fig. 5 Cylindrical weight type of neutralizer

the sudden decrease of load of the vibration generator due to cutting in the neutralizer, even though small, frequently causes an increase in speed and what might appear to be a neutralizing action is frequently due to being out of resonance. Only when the speed remains the same after cutting in the neutralizer is there assurance that it is effective.

Another form of neutralizer tried out is shown in Fig 6. The steel tube with cage represents a structure supported by a tower or column, vibrating horizontally. When clamped to the I-beam at the node the system had a natural frequency in the horizontal direction of 1100 per minute. When the vibration generator was run at this speed there was sufficient horizontal disturbance at the base of the column to cause a resonance vibration. The neutralizing element was a small cylindrical brass weight attached to a short length of drill rod. The rod could be moved up and down in a brass bushing and be locked in place by means of a thumb screw, after the system was tuned to the proper frequency. To arrest the motion of the element, without otherwise disturbing the system, a small electromagnet was attached to the end of a rod extending down from the top of the cage and a small piece of steel was inserted in the top of the brass cylinder. When current flowed through the magnet the element was held fixed relative to the cage, but when the circuit was broken it was free to vibrate. On breaking the circuit while the system was vibrating in resonance the vibration of the latter was reduced over 90 per cent and in both longitudinal and transverse directions simultaneously, the path of the element being elliptical. The principal dimensions of this system were; length of steel tube 24 in., outside diameter of tube 1-5/8 in., thickness of tube 7/64 in., weight of cage 15 lb, weight of neutralizing element 0.1 lb; diameter of drill rod 0.06 in., natural frequency 1100 per minute.

These results were considered sufficient to demonstrate the effectiveness of the undamped neutralizer under more or less ideal conditions.

The Neutralizer with Damping

As pointed out in the introduction, the undamped neutralizer causes severe vibration at the upper and lower criticals which it introduces, although it eliminates vibration at the original resonance frequency. The upper and lower criticals were evident when the neutralizer was added to the I-beam system. In order to reduce vibration at these speeds damping was introduced by adding two dashpots to the neutralizer as shown in Fig 7. Each dashpot consists of two concentric cylindrical shells attached to the base of the apparatus within which a third shell attached to the neutralizing element could oscillate. The third shell carried two rings to reduce the clearance between the fixed and moving shells to a minimum without their coming in contact. The purpose of this arrangement was to provide a long path for the air displaced from the inner fixed shell thus creating a maximum of air resistance with a minimum of spring action due to compression of the air.

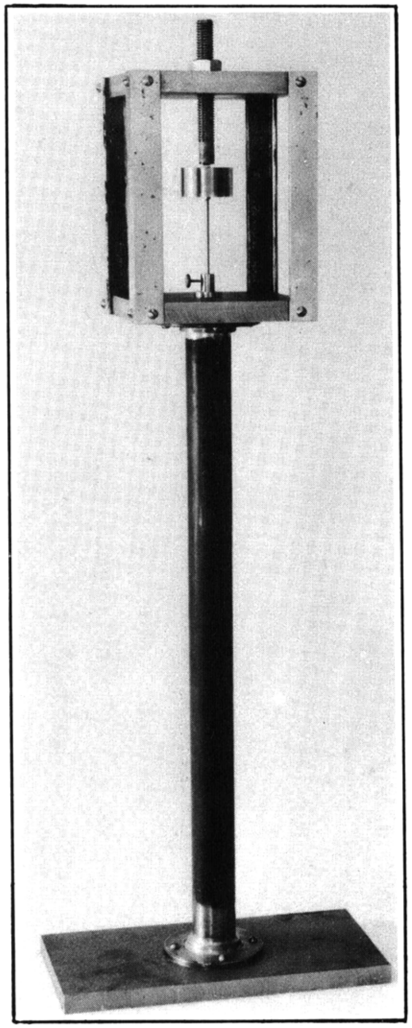


Fig. 6 "Tower" neutralizer

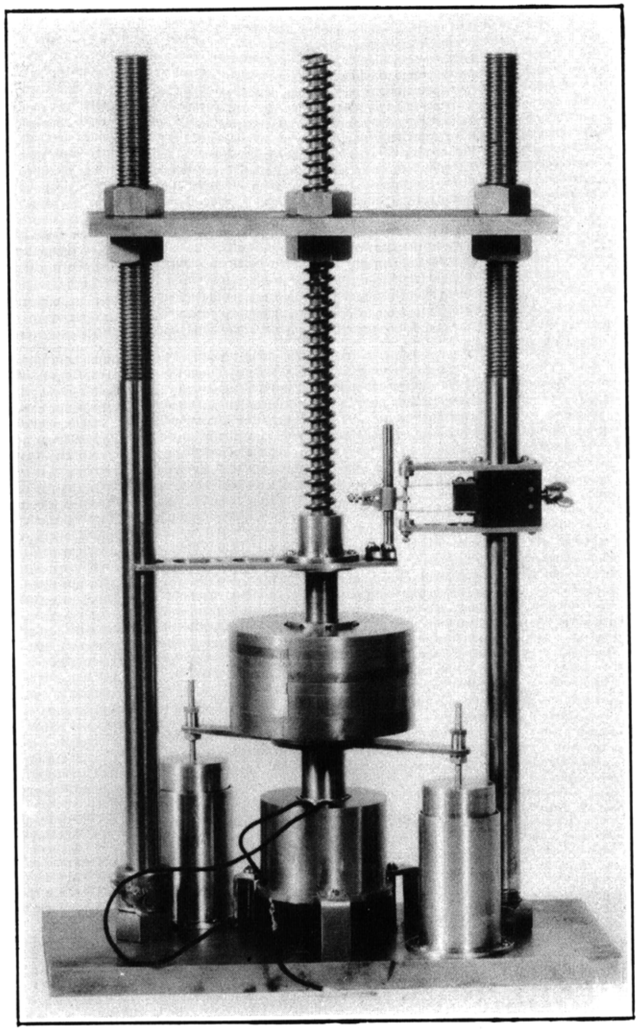


Fig. 7 Neutralizer with solenoid and plunger, also showing dashpots

The arrangement was not altogether satisfactory due to the difficulty of maintaining accurate alignment of the cylinders.

In the experiments with the dampers the vibration generator was installed in the center of the beam, the neutralizer and pallograph at one end, and ballast equivalent to the latter two at the other end. The distance between supports was 13 ft. The primary critical speed of the system was found to be 5.0 per sec. and the secondary criticals 4.7 and 5.5 per sec., respectively. When adjusted for best results, the neutralizer with dampers behaved about as predicted by theory. The vibration at the upper and lower criticals was reduced by about 50 per cent when the dampers were cut in, and the neutralizing action at the original critical was considerably diminished. Thus the neutralizer with damping prevented excessive vibration over the entire speed range, but it had the disadvantage that some vibration occurred over a much greater range of speeds than was the case without the neutralizer. In other words the system was much less selective.

The Booster

In present forms of vibration neutralizers the amplitude of the element is diminished by the presence of friction which cannot be avoided. The result is that 100 per cent neutralization is never attained. The effect of friction becomes more pronounced when an attempt is made to reduce the mass of the neutralizing element, as the amplitude required is correspondingly increased. In view of this the idea suggested itself of energizing the element periodically so as to build up its amplitude, or in other words overcome the effect of friction. This should make feasible the use of much smaller elements and in large installations this weight saving would be of great importance.

To accomplish this result it was proposed to fasten to the element an iron plunger which would move in a solenoid during oscillation of the element, the solenoid receiving periodic pulses of current of the same frequency as the vibration, and in such phase as to improve the neutralizing action. The neutralizer with these attachments is shown in Fig 7.

When the natural frequency of the beam, the natural frequency of the neutralizer, and the exciting frequency are all synchronized the element without booster will automatically oscillate in the correct phase to neutralize. In this case the problem is to cause the impulse on the plunger to occur when the element is at the middle of its stroke, as this will increase the amplitude without disturbing the phase.

The first timing device tried employed a sliding contactor attached to the neutralizing element as shown in Fig 7. With this type of contactor the make or break as the case may be occurs at the same time in the cycle regardless of the stroke, as the slider is held fixed after striking the stops and does not move again until the stroke is reversed. This is shown more clearly in the sketch Fig 7a.

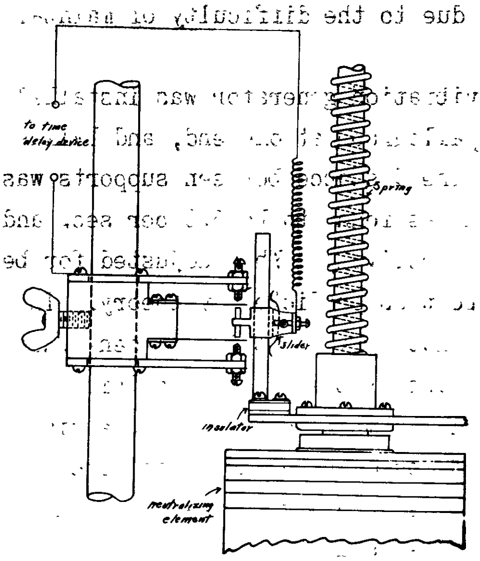


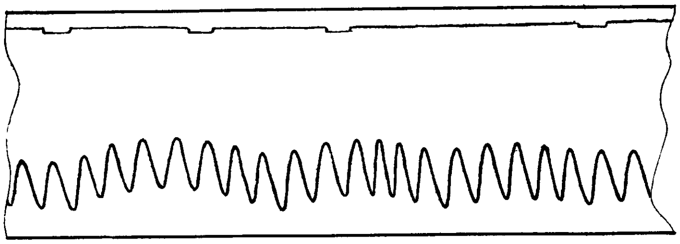
Fig. 7a Contact device for use with time delay device

The contactor was connected in series with an electro-magnet type relay which in turn closed the circuit through the solenoid. When the spring on the relay was adjusted so as to give the proper time delay there was a marked improvement in the neutralizing action as shown in the pallograph records, Fig. 8. Such results could be obtained, however, only after tedious adjustment of the relay and could not be readily duplicated. Sparking at the breaker due to the high inductance of the solenoid caused considerable trouble. As a matter of fact when the time delay was incorrect the booster made the vibration worse. When over-energized the neutralizer was capable of vibrating the beam itself, with the vibration generator not running.

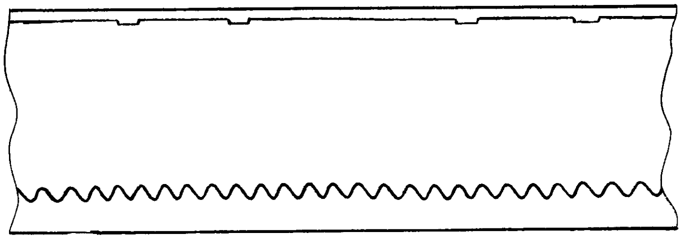
This experiment showed that the idea of boosting the neutralizer was fundamentally sound but that satisfactory operation imposed rather rigid requirements upon the timing device. It was therefore decided to substitute for the mechanical relay a vacuum tube relay in order to obtain a time delay that could be varied by turning the knob of a rheostat and which would remain constant after adjustment.

The vacuum tube circuit finally adopted is shown in Fig 9. The essential elements of this circuit are the $2\mu f$ condenser, the neon tube, the pentode type 57, and the thyratron type F G 17. The contact indicated in the diagram is made by the device in Fig 7a at the top of the stroke. Assuming the $2\mu f$ condenser to be discharged before contact the current in the 3500 ohm resistance puts a positive bias on the control grid of the pentode. This causes a flow of plate current at the same time charging the $2\mu f$ condenser. The time taken to charge the $2\mu f$ condenser will depend on the effective plate resistance of the pentode which in turn can be varied by moving the slider S. When the condenser is charged to the ionization potential of the neon tube a flash occurs and the pulse of current through the primary of the transformer T puts a positive bias on the grid of the thyratron which causes ionization. When the thyratron becomes conducting, the $8\mu f$ condenser becomes charged and then discharges through the solenoid causing attraction of the plunger. The plunger receives only a short impulse whose time delay after contact is controlled by the slider S and whose magnitude is controlled by S'.

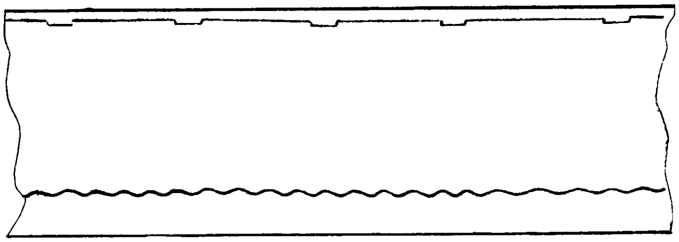
With this time delay circuit it was possible to reduce the vibration of the



a: Neutralizer locked



b: Neutralizer released
Booster not operating



c: Neutralizer released
and booster operating

Fig. 8 Pallograph records taken on vibrating beam showing effect of neutralizer with booster and mechanical relay

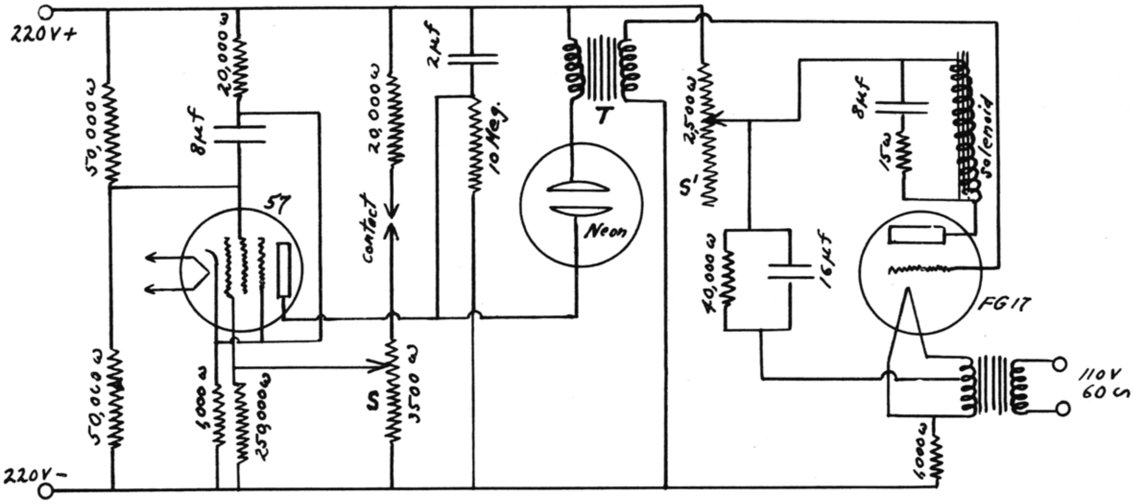


Fig. 9 Time delay circuit for vibration neutralizer for use with contactor on the neutralizer

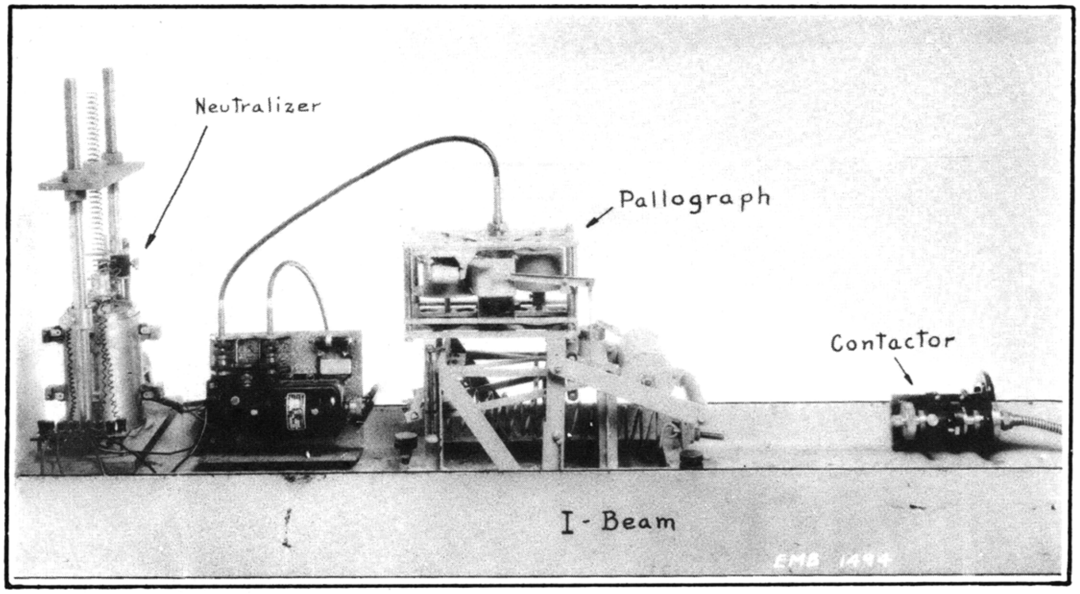


Fig. 10 Latest form of neutralizer

beam at the original critical to a negligible amount, but consistent results could not always be obtained due to voltage fluctuations. The upper and lower criticals were still present, and obviously one setting of the time delay could not give proper phasing at the three frequencies. The neutralizer in this form was thus limited to practically constant frequency. When the dampers were cut in to improve conditions at the upper and lower criticals the booster was found to exert insufficient force to make its effect noticeable.

The most recent form of neutralizer is illustrated in Fig 10. In this model no provision is made for damping. The inertia element consists of a winding enclosed in iron which sets up a magnetic flux which cuts the turns of a fixed coil attached to the base of the neutralizer. The magnetic element takes a steady direct current. As may be seen from the sketch Fig 11 the magnetic element (20) is suspended by a coil spring whose effective length can be varied by threading the coils through the special nut (12). The magnetic element is guided by the rollers (30) so that it can move only in the vertical direction. By sending pulses of current through the fixed coil (21) every cycle, the neutralizing action is increased. With this form of neutralizer the timing of the impulses is best obtained by a contactor driven by the vibration generator. This is feasible since the circuit to be broken now has a relatively low inductance. The phase is adjusted by turning a knob as in an ignition timing device. This is shown in Fig 12. The magnitude of the impulse is easily controlled by a rheostat or potentiometer.

This neutralizer was tested on the I-beam under the following conditions: supports 16 ft apart, vibration generator in the center, neutralizer and pallograph at one end, 83 lb of weights at the opposite end. The primary resonance in this case occurred at a speed of 4.5 revolutions per second, the lower and upper criticals at 4.2 and 4.9 per sec. At the primary critical the double amplitude with the neutralizer locked was 0.210 in. With the neutralizer released at the same speed this was reduced to 0.015 in., the neutralizing effect being about 93 per cent. With the booster cut in and the phase and magnitude of the impulses properly adjusted the amplitude at the same speed was reduced to about 0.002 in., the neutralizing effect being about 99 per cent. However the amplitude at the lower critical was 0.045 in. and at the upper critical 0.055 in. with the neutralizer cut in and the booster out and with the booster in they were about the same. It is obvious that one adjustment of the booster could not take care of all three critical speeds since at the primary critical the neutralizer itself is in resonance and in the proper phase so that all that is required is a slight impulse at its mid stroke to improve its action. At the upper and lower criticals, however, it tends to oscillate in the wrong phase so that the impulse would have to be great enough to shift the phase in order to improve conditions. Thus in this form the neutralizer is still limited to cases of practically constant speed.

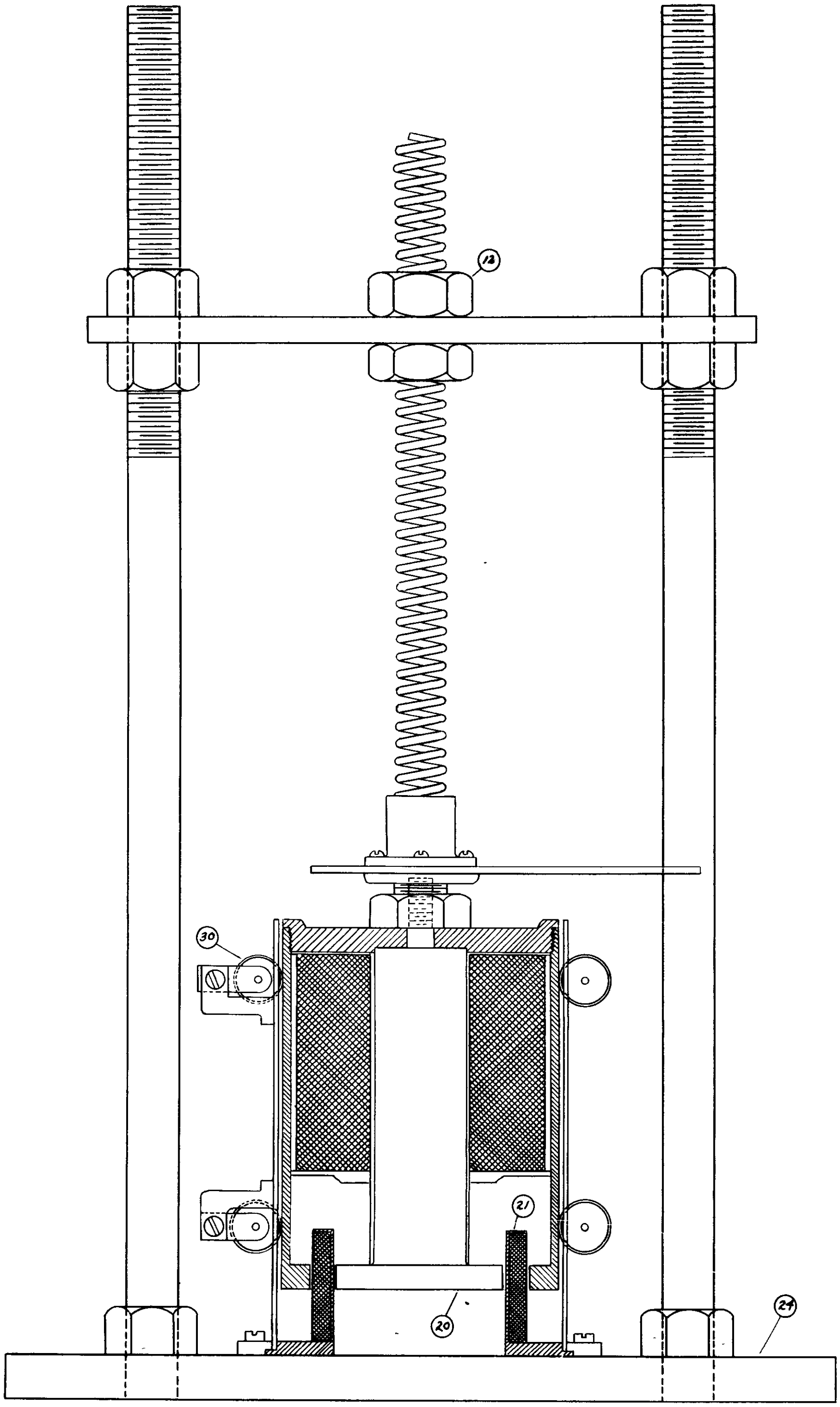


Fig. 11 Dynamic vibration neutralizer

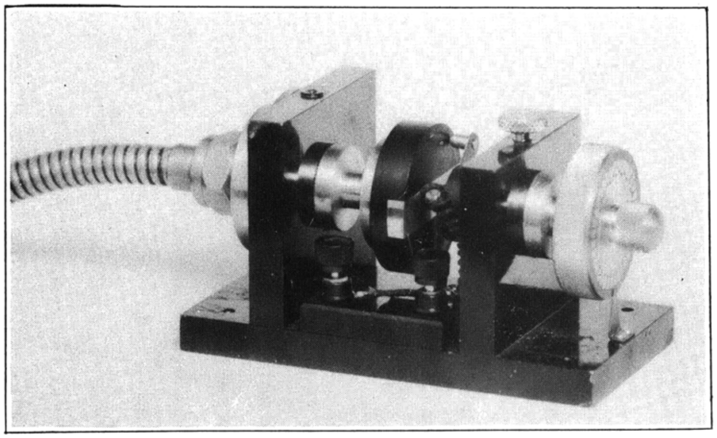
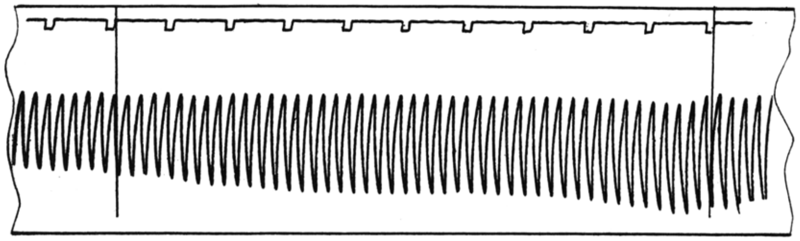
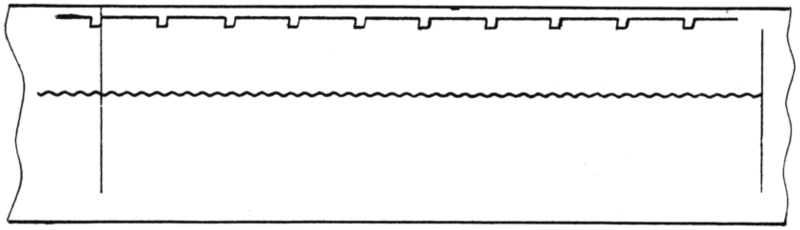


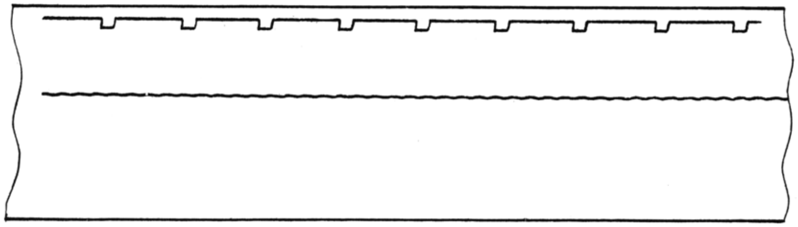
Fig. 12 Timing device for dynamic vibration neutralizer.



Neutralizer locked



Neutralizer released, no current in coil



Neutralizer released, coil functioning

Fig. 13 Pallograph records taken on I-beam showing effect of dynamic vibration neutralizer at the primary critical speed

Future Development

The experiments so far described are limited in scope and serve merely to indicate in what direction future investigation of this problem might be carried out. Obviously the design of neutralizers cannot be standardized but will differ radically according to the magnitude, frequency, and direction of the disturbing forces.

In the application of the principle of boosting or energizing the element numerous methods could obviously be used. Not only can electrical impulses be applied but in large installations air or steam acting on pistons attached to the element could conceivably be used. On ships this might be more convenient than the electrical method. The chief advantage of the electrical method over the use of air or steam is its greater susceptibility to control.

As previously indicated a problem yet to be solved is the elimination of vibration at the upper and lower criticals without the use of damping. Two possible methods of accomplishing this suggest themselves. One is the use of an automatic cut-out which will arrest the motion of the element altogether except when the exciting frequency is in the immediate vicinity of the primary critical. This could be accomplished by two centrifugal contact devices geared to the shaft one of which closes when the speed reaches a value just below the primary resonance, and the other of which opens when the speed rises just above the primary resonance. The element would be free to oscillate only when both contacts were made, which would occur only over a very narrow range in the vicinity of resonance. As the three criticals are fairly close together the centrifugal contactors would have to be capable of fine adjustment.

Another method of preventing severe vibration at the upper and lower criticals without damping would be the use of an automatic phase shifting device. This would operate as follows: a seismic element similar to the element of a pallograph or vibrometer would cause a relay to operate whenever vibration of any frequency exceeded a certain limit. The relay would in turn start a motor geared to the shaft contactor. This would rotate the contactor relative to the shaft so as to keep shifting the phase until the vibration was reduced to such a value that the relay stopped the motor. Naturally such a device could be successful only where speeds were changing slowly.

In connection with the vacuum tube time delay circuit for controlling the action of the booster, other types of contactors than the one already described are possible. Instead of the friction contactor being on the neutralizer itself it could be contained in a separate unit mounted on the structure. This unit would consist of a seismic element similar to the inertia element of a pallograph. This element would support a rod carrying the slider similar to the one shown in Fig 7. The adjustable contacts would be attached to the case. The relative motion between the seismic element and the case during vibration of the structure would cause contact to be made at either end of the stroke. This would eliminate the danger of the neutralizer going into self-maintained oscillation for, once the amplitude of the structure fell below the value necessary to make contact, the neutralizer would no longer be energized. Thus 100 per cent neutralization could not be maintained by this method, but by close setting of the contacts the vibration could be held down to a very small amplitude.

There might also be tried another principle of neutralization. If the element instead of being resonant is made seismic by giving it a very low natural frequency it will tend to remain fixed in space while the structure vibrates under it. If then periodic forces are created between the seismic element and the structure, in such phase that the force on the structure is equal and opposite to the disturbing force producing the vibration, neutralization will result. The neutralizer has the same form as before, the only difference being that the spring constant is made much lower. Thus if a structure is subjected to an exciting force whose vertical component is $F_y = mrw^2 \mathrm{Sin}\ wt$ due to a rotating unbalanced mass m, the neutralizing element suspended now by a weak spring will have a low natural frequency and will not respond to the vibration. If then forces are created between the structure and the element such that the reaction on the structure is at all times equal to $-F_y$ it is obvious that no vibration of the structure will exist. To accomplish this, however, a succession of impulses would not be sufficient since in this case the element being out of resonance would not oscillate sinusoidally. It would be necessary to send an alternating current through the coil of the neutralizer to accomplish the desired effect. Whether the element is resonant or seismic it is clear that the same amplitude would be required to give the same neutralizing action. In the resonant case the spring forces are large and the electrical forces small, whereas in the seismic case the spring forces are small and the electrical forces large. Thus there is danger of overheating the coil in the latter case and the electrical power requirements are much greater. A distinct advantage, however, is that in this case no other criticals are introduced by the neutralizer.

Seismic elements with friction have been used successfully for damping torsional vibration as in the Lanchester damper.[1] In this case the element is a heavy flywheel free to rotate relative to the shaft except for friction, there being no spring connection. Thus the natural frequency is zero. As the shaft rotates, the flywheel tends to rotate at uniform speed, and any variations in the angular velocity of the shaft due to torsional vibration cause a frictional torque to exist between the flywheel and the shaft tending to reduce the torsional vibration. In the linear case it would be impossible to keep the inertia element in position without the use of springs so that its natural frequency could never be zero. In the experiments with the I-beam it was distinctly noted that the vibration of the beam could be reduced by cutting in the dashpot of the pallograph. For this reason the pallograph was usually operated without damping which was feasible under laboratory conditions.

If the tuning of the neutralizer can be varied as the speed changes, either manually or by an automatic tuning device, so that the natural frequency of the element is always equal to the frequency of the exciting force, then no secondary critical speeds will exist. If the booster were to be used under these conditions the magnitude of the impulse would also have to be adjusted for each speed since the maximum impulse would be required at the primary resonance of the structure, little neutralizing action being necessary when the structure was out of resonance. The phase setting, however, should remain constant if the shaft contactor is used, for if the neutralizer is kept in tune the element will always have the proper phase relation to the rotating shaft and hence if the impulse comes at mid stroke for one speed it will come at mid stroke for all speeds.

A problem that frequently becomes serious on board ship is the vibration of optical instruments. In such cases the trouble is usually due to resonance either of the instrument itself or its supporting structure. In several cases frequencies in the neighborhood of 800 per minute have been measured. The design of neutralizers for high frequencies is much simpler than for low frequencies, the spring suspensions being shorter and the masses smaller. Hence in going from model to full scale neutralizers the development of such devices for instrument suspensions or other local structures would be a logical step. The information thus obtained would be of great value in attacking the problem of neutralizing the vibration of the entire hull.

Conclusions

The vibration neutralizer is sound in principle and offers the only present remedy for vibration where dynamic balance cannot be attained and structural changes are not feasible. It is most effective when the disturbing force is of constant frequency. Owing to the fact that it introduces two new critical frequencies it can seldom be used in its simplest form. Vibration at the upper and lower criticals can be reduced by introducing damping. This, however, renders the device less effective at the primary critical speed. The experiments described here indicate that the operation of the neutralizer may be greatly improved by introducing energizing devices of various forms.

Few large installations of neutralizers are on record, but the device has been applied successfully to a motorship of 11,300 tons. This installation requires the attention of an operator. Whether it can be developed into a reliable apparatus requiring no attention remains to be seen. It is clear that a special design would be required for every installation according to the magnitude, frequency, and direction of the exciting forces. The design becomes more difficult the lower the frequency, as the springs are then subjected to a greater static preloading. In multiple screw vessels the vibration frequently is characterized by beats due to slight variations in the speeds of the shafts. A single neutralizer could hardly contend with such conditions, but separate neutralizers, one for each shaft might be satisfactory. To reduce the weight of the element and still retain the same neutralizing action requires increasing its amplitude in proportion. This also imposes severe loads on the springs. While the application of the device to the reduction of hull vibration on naval vessels may be limited because of the varying speeds, these restrictions do not apply to passenger or cargo vessels.

Local vibrations of superstructures may be neutralized where it is impossible to reduce the hull vibration in the vicinity to amplitudes within the desired limits. In the case of local vibrations the disturbing frequencies are fairly high permitting the use of lighter neutralizing elements. In this case also provision must be made against vibration at the upper and lower criticals by introducing either damping or automatic locking devices.

Acknowledgment

Acknowledgment is due to Commander E. L. Gayhart and to Messrs. S. E. Dawson and G. A. DeShazer for numerous contributions to the development of this apparatus, and to Lieutenant R. D. Conrad and Mr. D. F. Windenburg for helpful advice and suggestions in the conduct of the investigation.

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