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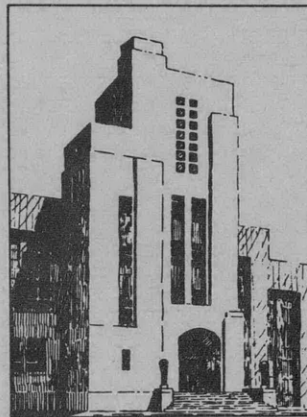
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EXPERIMENTS WITH AN IMPACT VIBRATION DAMPER

by

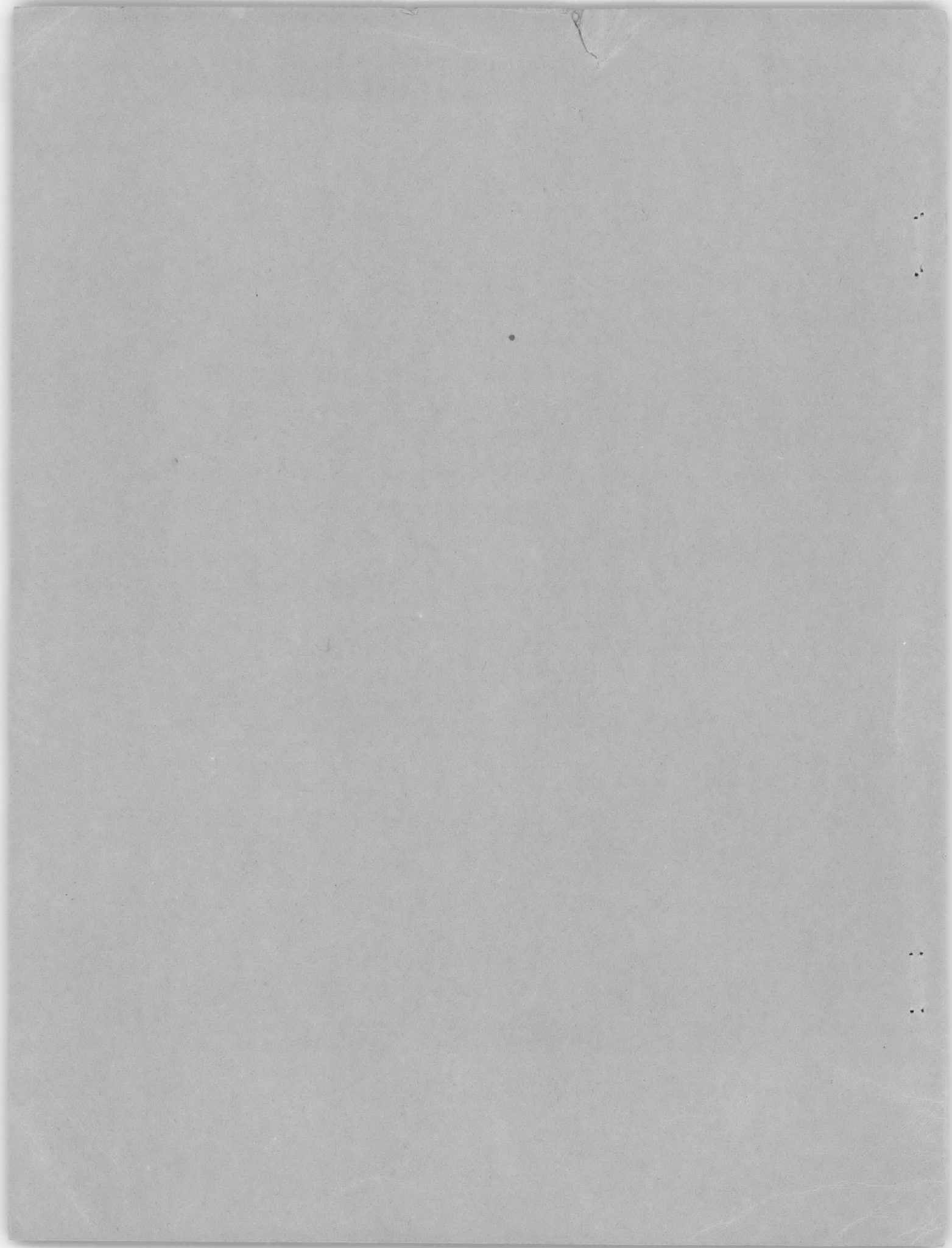
R.T. McGoldrick



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# EXPERIMENTS WITH AN IMPACT VIBRATION DAMPER

by

R.T. McGoldrick

## ABSTRACT

Laboratory experiments were conducted in which an impact damper was applied to relatively simple mass-elastic systems maintained in resonant forced vibration by means of a mechanical vibration generator. The damping element consisted of a number of hardened steel balls, such as are used in bearings, which impinged against hardened steel surfaces giving a high coefficient of restitution.

Definite but limited damping action was obtained, and the possibilities of applying this device to suppressing propeller-excited vibration of entire ship hulls are discussed. It is concluded that the difficulties of construction and maintenance would make the impact damper unsuitable for this particular application even though it might be well suited to other vibration problems.

## INTRODUCTION

Naval architects and marine engineers have frequently sought a remedy for the propeller-excited vibration of ships, but at present the chief practical remedy seems to be a change in the propeller design, which usually means increasing the number of propeller blades. Sometimes it may be possible to improve conditions by straightening out the flow to the propellers by means of vanes as discussed in Reference 1.\*

The vibration neutralizer, also frequently called the dynamic vibration absorber, has been used to some extent for the reduction of ship vibration,<sup>2,3</sup> but there are many difficulties in the way of its practical application where shaft speeds are variable.

The impact vibration damper,\*\* since it is free from one of the major design problems encountered with vibration neutralizers, namely, that of tuning, has a natural appeal. Some success has been claimed for this

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\*References are listed on page 11.

\*\*The type of vibration damper under consideration has been referred to in the literature both as an "acceleration damper" and as an "impact damper." The latter term has been retained here as it seems to be more descriptive of the action of the device.

damper in the reduction of vibration of aircraft structures. If the device has well-defined and predictable characteristics, it would be expected to have application not only to the suppression of vibration of local hull structures but also to the suppression of the vibration of the entire hull.

In such a damper one or more particles are impelled back and forth between two surfaces of a container attached rigidly to a vibrating structure, and the succession of reversals of momentum of the particles results in a reduction in the amplitude of the vibration.

The theoretical treatment of the action of such a damper is complicated by the fact that the mechanical system including the damper is not only nonlinear but possesses properties not readily evaluated. One theoretical treatment of this type of damper has been given by Lieber and Jensen.<sup>4</sup> This theory is based on the following assumptions:

1. The coefficient of restitution is zero.
2. Although the action of the damper reduces the amplitude of the structure, the motion of the latter remains approximately simple harmonic.
3. The action of the damper results in a steady-state vibration.
4. The damper causes a constant dissipation of energy per cycle.

A preliminary study of the impact damper at the David Taylor Model Basin suggested that the device might possibly be applied to ships in a design based on materials giving a high coefficient of restitution. This was encouraging since, obviously, to be successful on board ship the materials would have to withstand very many impacts without disintegrating, which would probably mean using hard materials such as steel rather than soft materials such as lead, which has been used in the experiments on aircraft.

To evaluate the device when made of materials with a high coefficient of restitution, the Bureau of Ships authorized the Taylor Model Basin to experiment with the impact damper on a small scale.<sup>5</sup> This report deals with these small-scale experiments and with possible large-scale application of this form of impact damper.

#### DESIGN OF TEST SETUP

In the design of the experimental setup at the Taylor Model Basin, simplicity was one of the chief considerations. The Model Basin has a small vibration generator<sup>6</sup> capable of producing a simple harmonic force in one direction only, and it was decided to utilize this machine to supply both the force and the necessary mass. In order to make possible a variation in the mass of the damper element as well as its clearance, the damper unit was

made of a block of steel with slots to provide for ten balls of one-inch diameter. Also, a method was provided for adjusting the clearances of the balls individually. This unit is shown in Figure 1. The damper unit was attached to the same framework in which the vibration generator was mounted, and this assembly was in turn mounted between roller guides attached to a stationary frame bolted to a heavy foundation. There was also bolted to the same foundation a strongback to which were attached four pairs of opposed compression springs inside of which were tie rods connected to the frame carrying the vibration generator. The tie rods were threaded so that by turning the special nuts at the free ends of the springs it was possible to preload the springs so that they were always in compression within the limits of amplitude allowed for the vibration generator. The assembly as shown in Figure 2 had a natural frequency of 1050 cpm. It will be observed that the damper unit is on the side opposite to the springs. In this setup the total

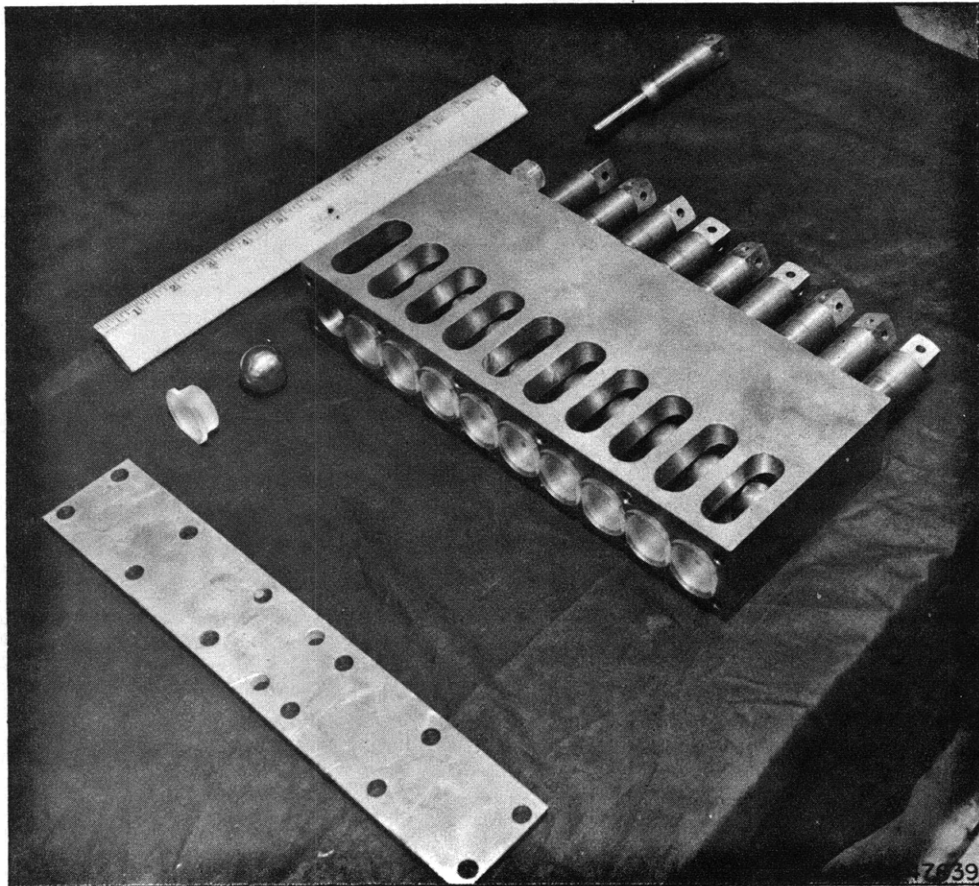


Figure 1 - The Experimental Impact Damper Unit

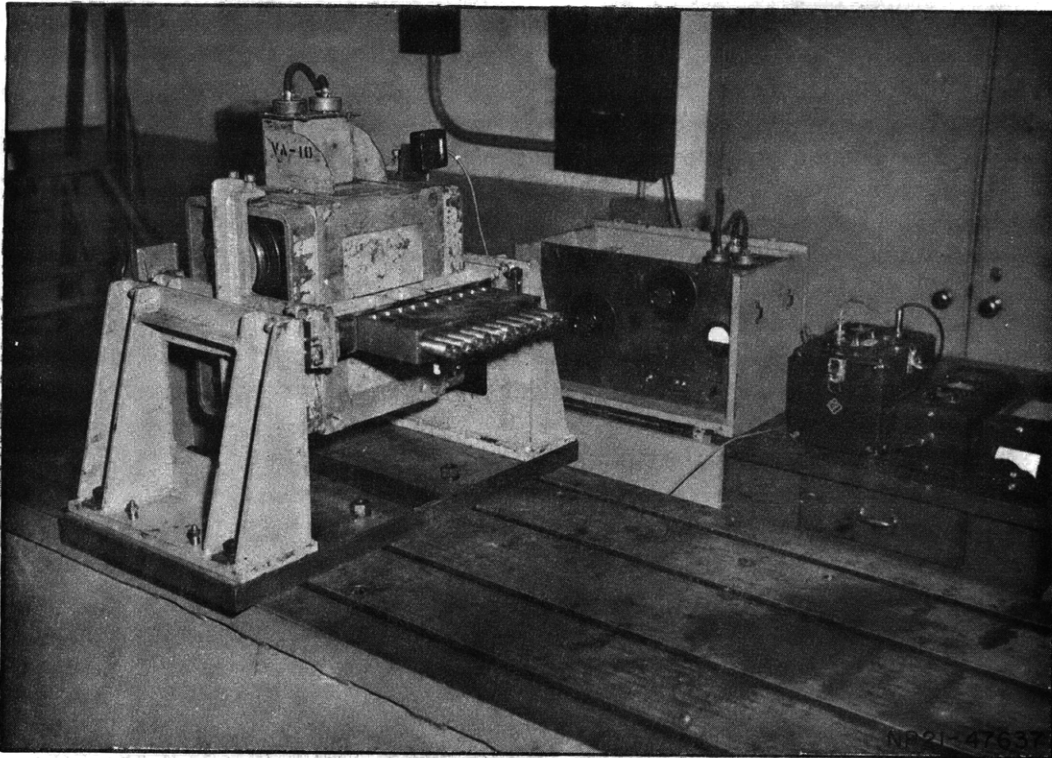
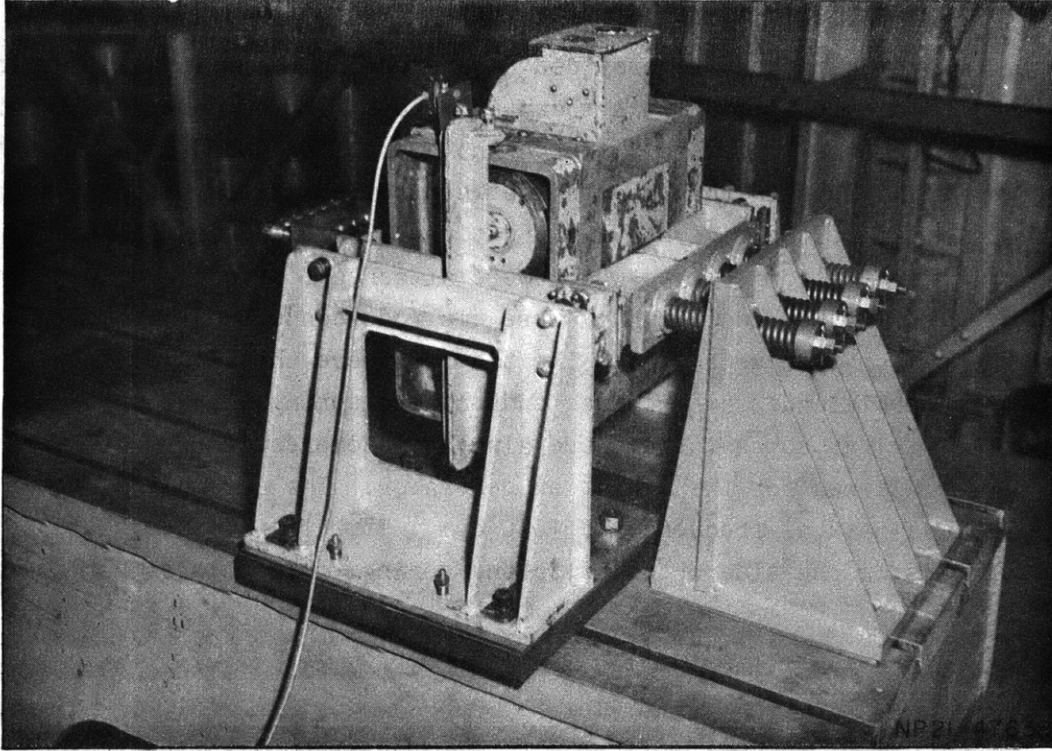


Figure 2 - Experimental Setup for Testing Impact Damper  
with Vibration Generator Supported on Rollers



vibrating mass was  $0.645 \text{ lb-sec}^2/\text{in}$  and the combined spring constant was  $7800 \text{ lb/in}$ .

An alternative experimental setup is shown in Figure 3. In this setup the vibration generator was firmly attached to a "bent" or horizontal beam supported by vertical legs at each end. The base plates at the bottom of the legs were bolted securely to the rails of the vibration test pit, which are permanently anchored to bedrock. A structure of this type has a normal mode of vibration in which the motion near the ends of the beam is practically horizontal and in the direction of its longitudinal axis, while along the beam the motion has also a vertical component which reverses phase near the middle. The resonant frequency of the system for this mode was 990 cpm, and the vertical motion at the location of the vibration generator was found to be negligible. For this setup the effective mass of the entire system was estimated to be  $4.96 \text{ lb-sec}^2/\text{in}$  at the location of the vibration generator, and the effective spring constant referred to this location was  $53,200 \text{ lb/in}$ . The method of deriving these values is discussed in the following section.

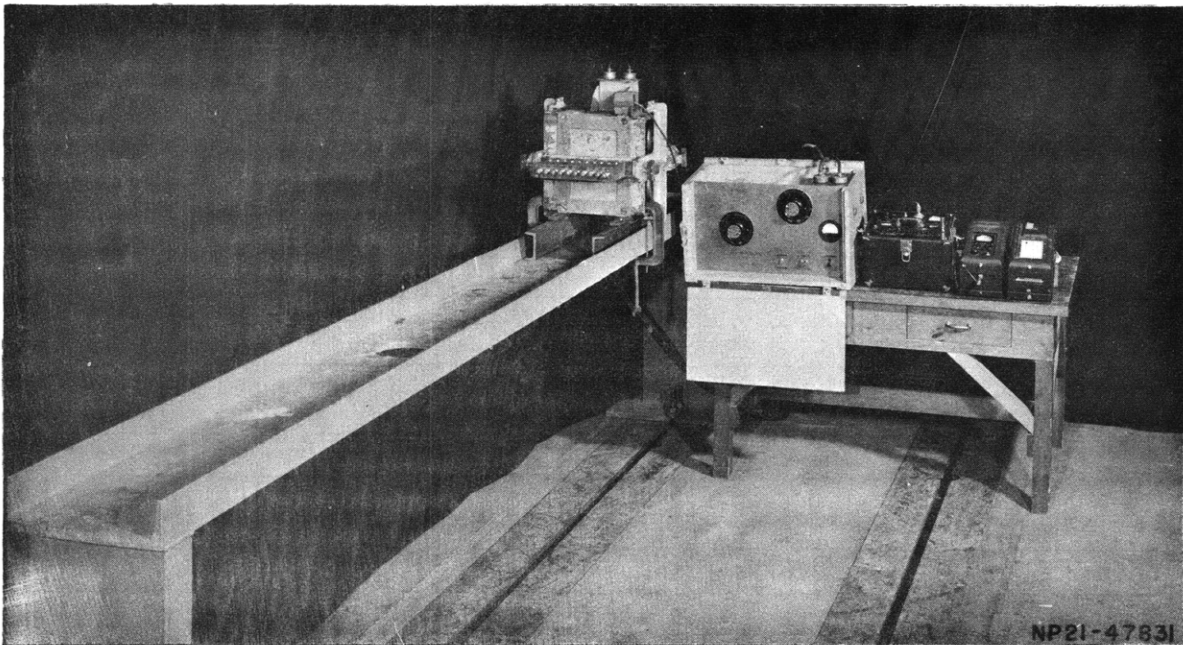


Figure 3 - Experimental Setup for Testing Impact Damper  
with Vibration Generator Mounted on "Bent"

## EXPERIMENTAL RESULTS

The experimental setup with the roller guides (Figure 2) proved to have a variable amount of inherent friction that made it difficult to obtain consistent results with the damper. It would no doubt be possible to modify this setup to improve this condition, but since the bent arrangement proved much more satisfactory in this respect no attempt has so far been made to modify the roller-guide setup. A further reason for giving preference to the bent is that it has a much greater effective mass, and one of the important points to be demonstrated is whether massive structures can be damped with relatively small damper elements.

With both experimental setups it was found that, as the clearance in the damper was increased, the amplitude of the structure decreased proportionately until a clearance was reached for which the balls ceased to bounce. The results with the roller-guide setup under the most favorable conditions are given for completeness in Table 1, even though the friction in the system varied to such an extent that the results could not be readily duplicated.

TABLE 1

Experimental Results Obtained with the Roller Guide and Bent Setups

Quantity	Value for Roller-Guide Setup	Value for Bent Setup
Resonant frequency, cpm	1050	990
Resonant circular frequency $p$ , rad/sec	110	104
Amplitude of driving force, pounds	22.3	39.5
Effective mass $M$ , lb-sec <sup>2</sup> /in	0.645	4.96
Equivalent viscous damping constant inherent in system $C$ , lb-sec/in	2.54	3.54
Ratio of inherent damping to critical damping, percent	1.8	0.34
Effective spring constant $K$ , lb/in	7800	53,200
Coefficient of restitution for steel balls on steel surfaces $\mu$	0.94	0.94
Mass of ten balls $m$ , lb-sec <sup>2</sup> /in	0.00384	0.00384
Mass ratio $M/m$	168	1290
Single amplitude with damper locked, in.	0.080	0.100
Single amplitude with optimum damper clearance, in.	0.053	0.085
Optimum damper clearance $D$ , in.	0.600	0.660
Observed reduction in amplitude, in.	0.027	0.015

For the bent setup it was necessary to determine experimentally the effective mass and the effective spring constant as well as the inherent damping of the structure. For the roller-guide setup the effective mass was assumed to be simply the mass of the moving system including an allowance of one-third of the mass of the coil springs. The justification of the process of treating an extensive elastic structure as a system with a single degree of freedom for each of its normal modes of vibration has been widely discussed in the literature. The evaluation of effective masses and spring constants representing each of the normal modes of vibration of ship hulls is discussed in Reference 7. In this report it has been found preferable to deal with effective masses referred to the driving point, but the basic theory is the same. Where an experimental resonance curve (obtained from a test with a vibration generator) is available, the effective mass and spring constant can be estimated by the formulas given in Reference 8. This process was applied to the bent. The equivalent viscous damping constant was obtained for both setups by the use of the formula for the amplitude of forced resonant vibration

$$X_0 = \frac{P_0}{Cp}$$

where  $X_0$  is the amplitude at the driving point in pounds,  
 $P_0$  is the amplitude of the driving force in inches,  
 $C$  is the effective viscous damping constant, and  
 $p$  is the resonant circular frequency in radians per second.

A set of lucite balls of the same diameter as the steel balls was made as well as a set of lucite inserts so that the two impinging surfaces would be of the same material. The coefficient of restitution for this condition was found to be 0.95, which was practically the same as that for the steel balls on a steel surface. No measurable damping action could be obtained with the lucite balls; this was assumed to be due to their low density as compared with steel (mass of ten lucite balls 0.00070 lb-sec<sup>2</sup>/in).

During the test on the bent, high-speed motion pictures were taken in order to check the phase relations. Recorded on an oscillograph were signals picked up by a microphone indicating the time of impacts. These records showed that the impacts were evenly spaced in time. The high-speed motion pictures taken at 3000 frames per second showed the relations between the motions of the damper balls and the container indicated in Figure 4.

It is to be noted from Table 1 that relatively large amplitudes were used with both experimental setups. In fact, it was only by working with such large amplitudes that any measurable reduction in amplitude due to

the action of the impact damper could be demonstrated. With the bent setup an experiment was made to determine whether the damper would function with a smaller initial amplitude. However, when the amplitude with damper locked was reduced by one-half by using a smaller eccentricity, the bouncing action of the balls could not be maintained and no appreciable damping action could be observed.

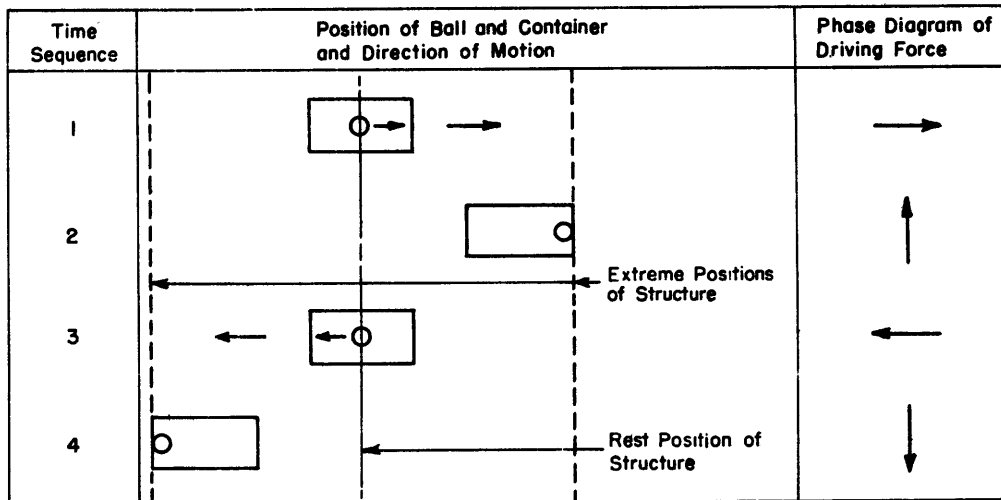


Figure 4 - Phase Relations in Impact Damper When Operating on a Structure in Forced Resonant Vibration as Shown by High-Speed Motion Pictures

#### DISCUSSION OF RESULTS AND OF POSSIBLE APPLICATION TO SHIPS

Inasmuch as it was found possible in the experiments described to produce a definite reduction in amplitude by means of the impact damper under the most favorable conditions, it is appropriate to consider the possibilities of this device in reducing ship vibration.

It is essential to consider whether, in the absence of a satisfactory theory, it is possible to predict the performance of an impact damper on a scale large enough for a ship from small-scale laboratory experiments of the type described in this report.

Both laboratory setups represented essentially systems of a single degree of freedom as far as motion in the direction of the exciting force was concerned, even though the bent itself is a continuous structure. It is only by reducing the ship to such an equivalent system in each of its normal modes that such a prediction could be made. It is essential to clarify a matter of terminology at this point of the discussion. In Reference 7 there are defined effective values of mass, damping, and spring constant of a beam or ship for each of its normal modes of vibration, without reference to any

point in the system. In many cases, as here, it is more convenient to deal with effective values at a particular point, usually the driving point. When so defined the values of M, K, and C are the values that the driving force actually "sees" and are numerically equal to the values defined in Reference 7 divided by the square of the normal-mode function at the driving point.

In its lower vertical modes of vibration the ship may be considered to have an effective mass at the stern equal to about one-quarter of the sum of its actual mass and the virtual mass of the surrounding water. It has also been shown by experiment that it has an effective inherent damping constant in the lower modes equal to about 0.3 percent of critical damping. This is the same order of damping as found for the bent. Hence the laboratory setup on the bent should enable one to predict the effectiveness of such a damper on the ship for conditions under which the following quantities are invariant for both systems

$$\frac{KX}{P_0}; \quad \frac{C}{\sqrt{MK}}; \quad \frac{m}{M}; \quad \mu; \quad \frac{p}{\sqrt{\frac{K}{M}}}$$

where X is the displacement amplitude,  
 $P_0$  is the amplitude of the driving force,  
 p is the resonant circular frequency,  
 m is the mass of the damper element, and  
 $\mu$  is the coefficient of resitution.

M, K, and C are the previously defined effective values of mass, spring constant, and damping constant, respectively, at the driving point. These invariants are readily derived by the methods of dimensional analysis as discussed in Reference 9.

Suppose the vessel under consideration to have a displacement of 5000 tons and suppose the frequency of its third vertical mode to be 360 cpm ( $p = 37.6$  rad/sec). With an allowance of 100 percent of the mass of the vessel for virtual mass of the surrounding water, the effective mass of the system referred to the after perpendicular would be about 14,500 lb-sec<sup>2</sup>/in. The effective mass of the bent at the driving point was found experimentally to be 4.96 lb-sec<sup>2</sup>/in. Its effective spring constant was 53,200 lb/in, and the corresponding value for the ship for the third vertical mode would be  $20.6 \times 10^6$  lb/in. The effective inherent damping constant for the bent referred to the driving point was found experimentally to be 3.54 lb-sec/in. The corresponding value for the ship is estimated to be 3280 lb-sec/in.

The available data on driving forces acting on ships are as yet very scant, but if a force of 5000 lb were assumed to be acting at the stern the resonant amplitude would be 0.040 in. To represent this condition on the bent it would be necessary that  $KX/P_0$  be the same as for the ship. This value for the ship is 165 and, since K for the bent is 53,200,  $X/P_0$  would have to be 0.0031. The experimental value of this ratio was 0.0025, showing that the conditions for similitude for the forced vibration with the damper out of action can be fairly well satisfied.

It then follows from the conditions of similitude that in order to obtain a 15-percent reduction in the resonant amplitude on the ship the damper would have to have elements whose total mass would bear the same ratio to the effective mass of the ship as the corresponding ratio for the bent. This gives a mass of 11.2 lb-sec<sup>2</sup>/in or 4330 lb. This weight of itself would not be excessive, but in order to maintain the same coefficient of restitution it would obviously be necessary to produce this mass by using a very large number of steel balls. If the complete damper were built as a unit, it would probably weigh at least ten times as much as the balls or about 40,000 lb.

A 15-percent reduction in amplitude such as was obtained on the bent in the laboratory would hardly warrant the installation of such a device on a naval vessel, but it is not even certain that the 15-percent reduction in amplitude could be realized on the ship. The amplitudes on the bent were at least twice as great as would be expected on the ship. It might appear that the impact damper would be relatively more effective at reduced initial amplitudes. This, however, was not found to be the case in the laboratory as has previously been pointed out. It was only when large amplitudes were used that the balls would bounce over a sufficient clearance to produce any noticeable damping action. This was probably due to the friction in the damper itself, which might be avoided in a vertical unit. However, in designing a vertical damper, gravity introduces a major problem. The balls, if free, would remain at the bottom of their individual cells unless the acceleration exceeded g. Such an acceleration due to vibration, however, is seldom reached even in the most severe cases. Hence, some form of flexible support would have to be provided to maintain them halfway between their stops under the action of gravity. In view of the space limitations within the cells the introduction of the flexible support would greatly complicate the design.

## CONCLUSIONS

On the basis of the experiments described in this report it is concluded that

1. The principle of the impact damper is applicable when materials giving a high coefficient of restitution are used as well as for materials with a low coefficient.
2. In order to obtain appreciable damping with the impact damper using materials giving a high coefficient of restitution, large clearances, of the order of 1 inch, would have to be used.
3. In the horizontal form of the impact damper the friction between the balls and the container walls prevents realizing its maximum capabilities.
4. To use the impact damper in the vertical form the masses would have to be flexibly mounted to eliminate the effect of gravity. This would greatly complicate the design.
5. The impact damper in the form using materials of a high coefficient of restitution is not effective enough to be considered promising for the reduction of steady-state ship vibration.
6. While the experiments here described do not show what performance might be obtained with materials giving a low coefficient of restitution, it is believed that such materials would rapidly disintegrate under shipboard service conditions and that their replacement would involve an excessive maintenance cost.

## ACKNOWLEDGMENT

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