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NO. 11

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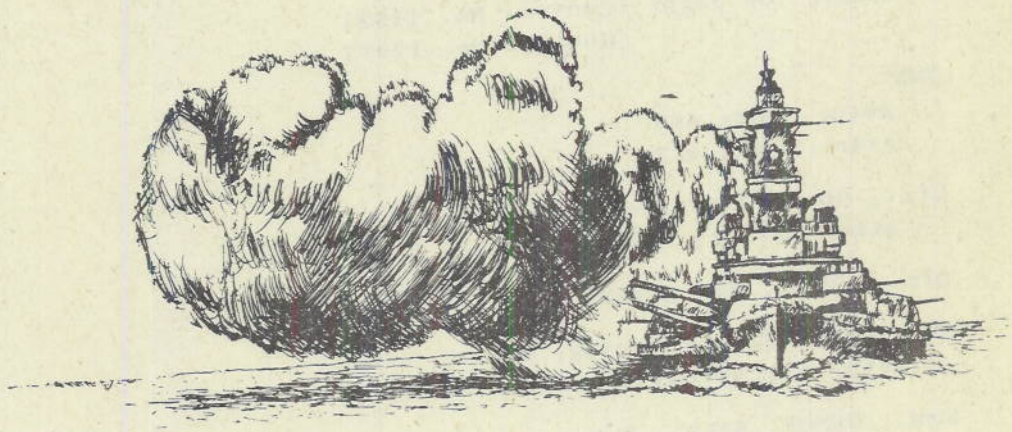

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FOREWORD

The contents of this and previous Bulletins indicate that the field of shock and vibration has few existing boundaries. Judging by the many requests for information, there are also few limits to the widespread interest in this subject. To the extent of available copies and within the security regulations this Laboratory has quite adequately supplied the diversified demand.

Future Bulletins will contain even a greater variety of technical material. The Shock and Vibration Symposia will hereafter be the official scientific information exchange in this field for the Department of the Army, the Department of the Navy and the Department of the Air Force. The meetings will be sponsored by all branches of the Armed Services and will be held at various localities throughout the country. The Bulletins will be published in cooperation with the Research and Development Board under the seal of the National Military Establishment.

To the working-level technologist interested in the field of shock and vibration, the interchange of scientific information will continue as before. Participation by personnel from various laboratories of the Army, Navy and Air Force will be most welcome.

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THE TWELFTH SYMPOSIUM

Naval Research Laboratory
16 February 1949

POSSIBILITIES AND LIMITATIONS OF MODEL TECHNIQUES FOR DESIGN AND TESTING PROCEDURES

By

L. B. Tuckerman, National Bureau of Standards

A warning is sounded to those technologists who assume that model testing data are conclusive criteria for design. The laws of physical similitude are an essential basis for all model testing, but they cannot in general be applied to all factors relating the model to the full scale structure. Success in model testing depends, therefore, on the degree with which supplemental tests and interpretation can take account of the lack of full physical similitude between model and prototype. There is an urgent need to correlate model testing data with experiments on prototypes before models can achieve their full usefulness in design.

While thinking on this subject of model techniques, I recalled a poem which I had to memorize in grammar school. Let me quote a few lines from it:

The Building of the Ship
By
Henry Wadsworth Longfellow

"Build me straight, O worthy Master!
Staunch and strong, a goodly vessel,
That shall laugh at all disaster,
And with wave and whirlwind wrestle!

} *Design against shock and turbulence*

"And first with nicest skill and art,
Perfect and finished in every part,
A little model the Master wrought,
Which should be to the larger plan
What the child is to the man,
Its counterpart in miniature;
That with a hand more swift and sure
The greater labor might be brought
To answer to his inward thought.

} *Model designing*

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"Broad in the beam that the stress of the blast,
Pressing down upon sail and mast,
Might not the sharp prow o'er whelm
Broad in the beam, but sloping aft
With graceful curve and slow degrees,
That she might be docile to the helm,
And that the current of parted seas,
Closing behind, with mighty force,
Might aid and not impede her course."

} *Streamlining*

Some time ago my colleague, Mr. McPherson at the National Bureau of Standards, and I were discussing what we have rather wishfully called a dynamic model of a certain airplane. During the discussion he remarked that he preferred to think of it as an analogue computer

It was a new and illuminating thought to me. When we think of analogue computers we are inclined to think of such things as the differential analyzer of Vannevar Bush or the complex electric circuits used to solve simultaneous linear equations. Mr. McPherson's remark taught me that such thinking is much too narrow. Models also are, in fact, special analogue computers and many other devices, which we do not ordinarily associate with the words, are analogue computers.

The designation "analogue computer" has only recently been introduced, but the fundamental ideas involved in analogue computers are old.

Whenever the mathematical relations which describe two different physical systems, even involving entirely different phenomena, are the same, the behavior of one system may be used to compute the behavior of the other.

Thus for years scientists have used the phenomena of gravitational attraction to describe some of the behavior of electric charges and magnets. Thus they have used coupled pendulums and gear trains to predict the rise and fall of the tides; have used electric circuits to solve problems in acoustics. They even use soap films to compute the torsional stiffness and strength of structural members. Many other examples could be cited.

The physical systems which we study by means of models are mechanical systems. The equations which define mechanical systems are differential equations involving the physical properties of the materials of which the system is composed and its boundary conditions. The boundary conditions involve the geometrical shape of the system and the mechanical forces applied to it.

If a mechanical model could be made to be a perfect analogue computer, all of these physical and geometrical conditions which are significant in the behavior of the structure which we study, would have to be so proportioned in the model as to give a one-to-one correspondence between all the details of the

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behavior of the model and those of its prototype. They would have to be dynamically similar.

Although we know that this ideal cannot be achieved, we can state what would be necessary to achieve it, and then see how closely we can approach it. We know, from many model test, that in many cases, a sufficiently close approach can be made to yield useful results.

Examples are (all these are stationary phenomena):

a. Towing tests of ships.

b. Wind tunnel tests.

c. Venturi tube, centrifugal tachometer, liquid filled magnetic compass, liquid inclinometer, windage resistance in gyroscopes, torsion viscosimeter, lubrication of journal bearings.

d. Pendulum in a viscous fluid.

e. Bridges, dams, pressure airships.

f. Flow in tubes.

g. Orifice meters.

In none of these examples were the conditions of dynamical similitude fully complied with. The tests needed further interpretation to take account of this failure.

The theory of dynamic similitude is well known. Beginning with Newton's classical work, it slumbered for nearly two centuries. Fourier, Stokes, William Thomson, J. J. Thomson, and Helmholtz made contributions to it. Froude and Osborne Reynolds discovered experimentally certain of its laws in the field of fluid dynamics.

It was Rayleigh who first realized the importance of dimensional reasoning in these problems and extended the concept from dynamic to more complete "physical" similitude involving other than mechanical phenomena.

Examples of other than mechanical models:

a. Submarine cable.

b. Scattering of light, blue sky.

c. Cooling problem.

The work of Rayleigh, Riabouchinsky and Buckingham, as cooked up in "Buckingham pie" has so simplified and standardized the technique of dimensional reasoning in problems of "physical similitude" that anyone with a few sheets of paper can figure out the conditions necessary for any kind of a model test, provided he has a clear picture of the physical problems involved.

The advantage of the formulation of the law of physical similitude in the form of Buckingham's pi theorem lies in the fact that the attention can be concentrated on the purely physical aspects of the problem; that is, on listing, with their dimensions, all of the quantities upon which the particular dynamic behavior under investigation materially depends.

The formation of (n-m) independent pi's is then a matter of routine. After any set of (n-m) independent pi's has been found, the arrangement of them into physically more significant groupings is a matter of simple inspection.

The practical application is not so simple. Frequently the theoretical conditions are conflicting and still more frequently construction and testing of a model meeting all these conditions proves

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impossible or impractical. If, in spite of this, model tests are to be made, compromises must be reached and a more detailed study be made in each case to estimate the effect of the deviations from similarity and correct for them.

In many cases, a knowledge of the differential equations of the system makes it possible to waive the strict requirement of a dynamically similar model and work instead with an affine model. The model may be affine in the strict geometric sense, as in Reynold's estuary models or Durand's surge chamber models, or it may be affine in a more general dynamic sense. For instance, the horizontal components of load might have a model ratio different from the vertical, or the deformations in horizontal and vertical directions might not be in the same ratio, although the linear dimensions were.

By using affine rather than similar models, conflicting requirements of similarity or difficulties of construction may sometimes be reconciled, and a model test made feasible.

However, every departure from strict physical similitude demands a more detailed knowledge of the equations of the system than is contained in the theory of physical similitude. Every such departure must be carefully scrutinized to make sure that the dynamical relations may be safely considered affine.

Examples of affine models are:

a. Rivers, harbors, estuaries - scouring and erosion.

b. Flow in chutes and storm sewers and over dams.

c. Surge tanks.

d. Propeller flutter.

e. Corrugated diaphragms for aeronautic instruments.

To emphasize the care which must be exercised in applying the principles of model testing, let me quote from masters of this subject, on the following pages 5 through 7.

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AERONAUTICAL RESEARCH COMMITTEE
Reports and Memoranda, No. 732.
(M. 2.)

Note on the Application of Dimensional Theory To Notched-Bar Impact Tests

By
R. V. Southwell

January, 1921

p. 4 "It is of course true, as has been previously remarked, that dimensional theory is not strictly applicable to this problem, since the grain-size of the material will not in practice vary in proportion to the dimensions of the specimen. The point is of importance, since grain-size may well have an important influence upon the resistance of a material to shock, and should such be the case the conclusions obtained from dimensional theory will not, of course, be valid. Experiment alone can decide the point, but the general confirmation of these conclusions, mentioned above, would seem to justify the hope that this additional factor may prove to be negligible."

pp. 5-6 "Thus, on the assumption that fracture occurs when a definite stress, or stress difference, p is reached at some point, and that p is a constant for the material considered, we are led to the conclusion that "scale-effect" should not be a characteristic of the test."

p. 7 "Hence, in the case of materials which exhibit properties of the nature of viscosity, there would appear to be no theoretical means of separating the effects of the independent variables V and L; in other words, "scale-effect" is to be expected."

pp. 7-8 "...The application of dimensional theory requires that the presumed geometrical similarity shall extend to all parts of the system into which the stress is propagated, and we know from elastic theory that the two principal velocities with which stress is propagated through steel are of the order of 17,000 and 10,500 ft. per sec. Now the time of fracture, according to the table on p. 20 of Dr. Stanton's and Mr. Batson's paper, ranges from 1 to 4 thousandths of a second, and in such time the faster wave would travel a distance of from 17 to 70 ft., so that many reflections of stress must have occurred during the process of fracture. Reflections of stress are, of course, taken into account by the dimensional theory, so long as absolute similarity of the specimens and apparatus is maintained; but is it certain that this condition has been realised hitherto in every part of the apparatus which lies within 70 ft. of the point of impact?..."

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p. 8 "To obviate the danger of inaccuracies arising from this cause, it would seem that the apparatus ought to be designed with the definite aim of confining the total energy to the specimen, striker and anvil - the two latter parts being kept strictly to scale."

JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS
Volume XXXIII, No. 2, May, 1921.

On the Use of Models in Investigating the Resistance of Ship Structures to External Explosions

By
E. Buckingham

p. 229 "The foregoing paragraph contains the principle on which all model experiments must be conducted in order to furnish reliable information about the full scale originals. The model is made geometrically similar to the original, with its size diminished in some known ratio. It usually happens that some of the physical properties, P, can not be adjusted arbitrarily. With a ship model, for example, it may not be practicable to use any other liquid than water, so that if the density of the medium is one of the P's which affect what happens, this P has to remain fixed. Some of the outside actions may also have to be left unaltered, gravity for example. But if there remains a sufficient number of the P's and A's which can be controlled and given arbitrarily chosen values, the model may be made to behave like the original. The problem of conducting valid model experiments consists, therefore, first in finding out what these adjustments must be, and second in selecting the materials of construction of the model and arranging the circumstances of the model experiment so that the requirements shall be satisfied.

"The first part of the problem is purely theoretical, and its solution tells us what ought to be done. The second part consists in the carrying out of these instructions, which may or may not be practicable. But in any event and whatever the nature of the phenomenon under consideration, the first thing to be done when considering the possibility of getting useful information from experiments on models, is to investigate the conditions which must be satisfied in order to make the model experiment "correspond" to the full scale one, so that the behavior of the model shall be "dynamically similar" to that of the original."

p. 244 "Upon looking over all these requirements, it is evident that they can not be satisfied very accurately. But it seems highly probable that the errors and uncertainties due to this lack of exactness will be no greater than the variations of strength between ships built to the same drawings and specifications or, especially, than the differences, under service conditions, between the effects of different mines, bombs, etc., of the same lot. Model experiments of the sort outlined will certainly give reliable data on relative strength; and if the conditions are carefully attended to, they will give reliable, even if only approximate,

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information regarding absolute strengths."

p. 719 "In concluding these notes, which are merely a series of remarks that are obvious as soon as made, one final remark may be added. It is, that what is needed for the successful use of dimensional reasoning is neither algebra nor metaphysics, but only practice and a very modest portion of that physical common sense which the late Lord Rayleigh exhibited in so eminent a degree."

To Buckingham, the necessary amount of "physical common sense" may well have been "very modest." To me it seems far from modest.

DISCUSSION

E. Klein, NRL:

For the benefit of the designers, I asked Dr. Ball to look up material to see if any use has been made of Buckingham's dimensional analysis.

L. W. Ball, NOL:

We were not able to find any. That doesn't mean there wasn't any. At the beginning, Dr. Tuckerman, I think you talked about analogues in the sense of measuring quantity. Will you comment on the degree of diagnosis and qualitative observations involved in the use of models? Very often in our work on shock we are seeking to diagnose what mode of failure, what particular phenomenon might occur. We do have the need to diagnose qualitatively the type of thing that might occur, as well as to measure quantitatively the effect if it does occur, or do you consider that models are useful only for quantitative studies?

L. B. Tuckerman, NBS:

Qualitative differences, such as you mention, are the result of quantitative differences and may be considered as furnishing rough estimates of the quantitative differences which cause them.

As an example, some years ago we investigated for the Navy the results of hard landings of two different airplanes. In one the bottom chord members of the fuselage failed in compression. In the other, the top chord members failed in tension. The quantitative differences, under bending moments, of the stresses in the chord members of these two airplanes fully accounted for the "qualitative" difference between these two failures.

Models can therefore be useful for both qualitative and quantitative studies. However, although we can make models which will simulate fairly well the behavior of mechanical structures within the limits of the application of Hooke's Law, when you go beyond that, you find yourself in all sorts of difficulties. It is frequently beyond the limits of the applicability of

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Hooke's Law that the "qualitative" results you are interested in occur.

It was not a shock testing job I was doing, but when we were working on a model of a semi-rigid airship, the RS-1, we wished to know what forces the ship could stand.[1] We built a model and studied that model very thoroughly. We measured what happened to it under changing pressures and loads so low that our model was not damaged. Then we extrapolated to higher pressures and loads sufficient to cause failure. For the extrapolation we tested full size patches to see what stresses they could stand, full size girders, to see what they could stand, etc. Then we compared these stresses with those computed from our model under low loads. The limit of practical application of Hooke's Law required this further interpretation to judge what might happen to the ship under more extreme conditions.

The ship was flown successfully. It was finally scrapped after some years only because its fabric envelope had deteriorated till it was no longer safe.

You can study the behavior of your model within limits before failure, then extrapolate by means of full size tests to failure on various parts. That is what has been most successful in my experience.

Of course, there are the further difficulties which Southwell points out, of the speed with which the stresses are applied. In any extended structure this may make a great difference in the way it will behave. At the Taylor Model Basin, I saw a movie[2] of the explosion effects on a model. How anybody could have told how that apparatus which was hung around on the inside of that model was going to behave when the explosion came on the outside, I don't know. Shortly after the explosion, they jumped around every which way much before the main shoving in of the wall. The slower motion of the wall may perhaps have followed the predictions of dimensional reasoning. The wall didn't break, but I should think possibly that if you knew the characteristics of the material, how it would respond to three-dimensional stress suddenly applied, you could reason from what did happen under the explosion to what would happen under larger forces.

An example of the effect of stress propagation in the case of suddenly applied loads was the case of sledge hammer blows on the heads of traffic sign supporters, being driven into the ground. The stress travelled as a compression wave to the ground and was reflected as a tensile stress. In a number of cases, several inches of the top of the support broke off under tension and flew up in the face of the workman.

W. H. Hoppmann, Johns Hopkins University:

I would like to make this remark, because I think it is a particularly serious and significant one. I had the privilege in the middle

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[1] L. B. Tuckerman, Water Model Tests for Semi-rigid Airships, N.A.C.A. Technical Report No. 211, 1925.

[2] Shock and Vibration Bulletin No. 3, p. 23.

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thirties, under Captain Forrest, to analyze the archives of the Bureau of Ships at that time with regard to the explosion problem. There has been made available a series of tests by a certain Italian named Kidaul, and the original work that Admiral Stocker had done in 1914 that Admiral Schuyler referred to last time. In addition to that, there were later pieces of work on half scale, very valuable work by Commodore Schade, and finally the work that Captain Forrest and Admiral Cochrane performed in 1946 or 1947 on the half-scale models of the battleships USS NORTH CAROLINA and WASHINGTON. No one certainly would want to pretend that he can give a very complete and accurate description of the action which is obviously very complicated from a physical standpoint. It would be foolhardy to say that the effects of the explosion bore any definite ratio, between the model and prototype. One of the important parameters to be measured was the penetration factor or the distance inboard that the explosion would travel. For example, we would say it was 18 or 20 feet in the ship. As a result of the analyses, it appeared that the scale factor was considerably less than the "3" Buckingham called for on the basis of his analysis. Otherwise, if I recall correctly, Buckingham said that it should vary with the cube of the scale ratio, but that did not turn out to be the case as far as we were concerned. The depth of penetration was important because the engine room and boiler room on these ships are concerned and I think that point should be emphasized. Considering such a complicated physical phenomenon, I do not see how one can use such a naive approach to the study. Impulses and wave transmissions are very complicated and do not follow those simple laws which Buckingham says he has to offer. A lot of the factors that Buckingham did not consider are crucial and the reason we don't get the correlation is that we have not summed up properly all the physical situations.

Tuckerman:

That is the problem, to sum up the physical situations. If you don't do that, all the physical dimensional reasoning doesn't mean a thing. I think I am right in saying that if you don't consider all of those things and if you don't try to consider them and build your model reasonably dynamically similar or dynamically affine then you are not going to get results. You may not get results after all that effort, but you won't get it without those considerations.

Captain Hinners, TMB:

With particular reference to Dr. Hoppmann's remark, I would like to suggest one respect in which the attempt to use small models of steel structure, representing a ship or a portion of a ship, to predict full scale results could be misleading and possibly could explain the differences which he said he observed in the study of these things. A great deal has been learned in the last few years about the size effect in steel, even under static conditions as well as what you get with underwater explosions, and it is now well known that you can get ductile behavior of the thin sections which you might use in model testing, whereas you can get brittle behavior of that same material when it is used in the relatively thick full scale structure. It is further known that when you have this brittle behavior of the full size

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material very little energy is required for fracture, even though the stress at fracture is about the same as the thin section, whereas when you have the ductile behavior of the thin section, a great deal of energy is absorbed by the material. That being the case, couldn't we expect that our model, when loaded explosively, might behave in a ductile manner so that a considerable amount of energy will be absorbed before fracture, whereas the full scale structure when tested would not absorb much energy in the initial fracturing process. That energy is still there and has to be absorbed somewhere, so you could expect that the full scale behavior would be considerably influenced by these properties of the material.

Tuckerman:

If we can determine the behavior of the material before fracture, and then test our materials on full scale and find these differences, it may be possible to interpret our tests on even smaller models. If a certain deformation occurred in the model material, and more brittle material would have been fractured by it, the model would then be sufficient, but that is going beyond your dimensional reasoning and extrapolating. Let us take an example which has nothing to do with the shock question; the test of the Stevenson Creek dam in California.[3] It was felt that model tests should be made to compare with the tests on the actual dam. A model that was constructed at Princeton under Professor Beggs was made of celluloid and loaded from behind with mercury. The dimensional reasoning told what the size of the model dam must be and we could calculate the conditions for dimensional similarity, but the one thing we could not do was to adjust Poisson's ratio (0.42) for the celluloid to match that ratio (0.15) for the concrete of which the dam was made. It was, however, found possible because of the conditions of the test, to make computational adjustments later and to correct for that difference.

The model with the higher value of Poisson's ratio deformed vertically, relatively far more than the corresponding deformations in the dam, but by making a correction for that we were able to match fairly well the stress distribution in the model and in the dam.

Try to make your model of the same material, get as near to dynamical similarity as you can, later do some reasoning with the full scale material to find its properties and then interpret. That must be done if you expect to solve these problems.

W. Mostow, NOL:

Dr. Tuckerman mentioned the use of dimensional analysis in connection with model scaling for underwater explosion tests on ships' structures. Dimensional analysis is essentially a "dodge" for arriving at a relationship between the variables involved in a physical problem when it is either impossible or impracticable to determine the relationship analytically. In other words, if the physical situation is accurately represented by a partial dif-

[3] Report on Arch Dam Investigation, Proc. A.S.C.E., Vol. 54, pp. 217, 219-230, 271, May, 1928.

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ferential equation which one cannot solve, recourse is had to dimensional analysis, and the numerical coefficients are determined from a few carefully chosen experiments. If a parameter such as a Reynolds' number is involved, then care must be taken to make allowance for its variation with scale factor.

In the case of loading a structure by an underwater explosion, however, the physical mechanism may vary with the scale factor. That is, the differential equations representing the physical problem on two fairly widely separated scales may be significantly different not so much in the parameters as in the terms involving time and space dependent variables. The manner in which the test structure modifies the loading function may actually vary with scale. One cannot, therefore, assume a priori that a scaling law is implicit in the mere geometrical scaling of the test structure per se when such a law may not, in fact, exist.

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APPROACHES TO DESIGN CRITERIA FOR MILITARY AND CIVILIAN USES

By

E. H. Jones, Bell Telephone Laboratories

Differences in concepts of design are outlined in terms of applications and overall requirements.

This is obviously an extremely broad subject, far too broad to treat here in detail in the time available. However, an attempt will be made to outline some of the essential differences between military and civilian requirements as we see them. It is hoped that in these remarks and in the ensuing discussion questions will be raised, the answers to which, whether available now or only after further investigation, will aid the designers of military equipment.

The value of any design, for whatever use is of course, measured by the extent to which it meets the overall requirements imposed by its intended use. Therefore the first requisite to a successful design is a careful and thorough evaluation of these requirements and a complete understanding of them by the designer or design organization.

Considered in their broadcast sense many design requirements are common to both military and civilian applications, for instance, any satisfactory design must meet suitable requirements as to performance, reliability, maintenance, safety, weight, physical size, etc. These requirements will, of course, be weighted somewhat differently with respect to each other for practically every job that comes to hand.

We have found that this matter of achieving an adequate knowledge of these true underlying requirements is usually one of the most difficult stages in the orderly development of a satisfactory design for any job. For many reasons it is usually even more difficult to establish them for military design than for civilian designs.

The responsibility for evaluating many of the wartime field conditions must of necessity lie with the military organizations. Much work along these lines has of course been done and many papers dealing with various phases of this problem have been presented at prior meetings of this group. Investigations of other phases of this problem are continually being carried on by the appropriate military organizations. Wherever possible these field conditions are being reduced to reproducible laboratory simulations.

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Since many of the conditions of use and unavoidable abuse to which military equipment is subjected in time of war cannot be directly reproduced at will, the importance of this work cannot be overstressed. This is true first, because these simulations are the best and frequently the only "yardstick" available to the designer for the evaluation of his work as it progresses, and second, because of their value as a basis for final acceptance of the design by the purchaser.

Many of these tests or simulations and many of the detailed requirements as to finishes, materials, etc. to be used on military equipment as now found in service specifications were devised and specified during the period of World War II and were based on the best information available at that time. This work was done under the pressure of wartime conditions by many divisions of the various branches of the armed services, each acting, of necessity, more or less independently. This has resulted, in some cases, in an undesirable lack of uniformity between various specifications which has imposed an additional burden on both the designer and the manufacturer. In the post war period considerable progress has been made in correcting this condition but there is much yet to be done and it is hoped that effort along these lines will be continued to the greatest degree possible.

The requirements for any design may be established by consideration of points such as the following:

a). Is it to be permanently installed or mobile?

b). Where is it to be used?

c). Will environmental conditions at place of use be favorable or unfavorable?

d). To what degree will shipment to place of use be hazardous?

e). What maintenance facilities will be available?

f). What is a reasonable cost for this equipment (in terms of dollars and man hours)?

g). Are suitable materials available?

h). Weight and size limitations.

This list is, of course, not by any means comprehensive but it at least does cover some of the major factors likely to be encountered. Let us then consider them briefly, from the standpoint, first, of a civilian project and second, a military project.

While public and private means of transportation are responsible for a large portion of the total demand for civilian material there are vast quantities of goods produced for civilian use which are permanently installed at their use location. Probably a majority of the remaining civilian articles, even though not actually permanently installed, are only transported for appreciable distances at infrequent intervals.

In the case of military equipment on the other hand, mobility is of the utmost importance. In many instances, it is a working part of the vehicle in which it is installed, as in the case of radio or radar equipment on Naval vessels, aircraft or tanks. Even where this is not so it must usually at least be capable of being frequently transported over considerable distances. The times required to dismantle for transit and to set up for operation at the new location must be held to a minimum.

Most civilian goods will very obviously be consumed or used in

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relatively densely populated regions. This means that the environmental conditions at the place of use will in general be relatively favorable. Facilities for maintenance or replacement are likely to be close at hand. Transportation facilities can normally be considered adequate and comparatively little hazard of damage in transit need be considered.

But when we consider these questions from a military standpoint we find quite a different picture. Wars may be and frequently are fought in the far places of the earth without regard to the nature of the climate or terrain or whether the transportation or maintenance facilities available are adequate.

Military equipment must therefore be designed to function under the most adverse environmental conditions. It must be able to withstand extremes of temperature, humidity, dust and sand, or in short anything that may be encountered from the hot desert countries, through the jungles to the extreme cold of the arctic regions. It must survive high impact shock and vibration that would wreck much civilian equipment. What would be considered severe abuse of civilian equipment will frequently necessarily constitute only normal use for military equipment. Since maintenance facilities too must be transported great distances, the design should be such as to require a minimum of them. Maintenance procedures should be such that comparatively unskilled personnel working under adverse conditions will be able to make the necessary repairs.

Most articles for civilian use are to some extent at least designed to a price and most types of merchandise are available in a wide range of quality with a correspondingly wide price range. Experienced concerns in every field have a wide background of information on which to base their price decisions.

For most military equipment, however, there is not the same freedom of action. Quality cannot be compromised for the sake of price. Even the best is all too often none too good. This has at times resulted in a feeling that the cost of military equipment is relatively unimportant. The designer should beware of this attitude! It is a complete fallacy! Quite aside from its effect on that unpopular item, the tax rate, is the fact that costs in dollars also mean costs in man hours and materials and both are always in critically short supply in wartime.

It follows then that the designer should exert every possible effort to insure that the manufacture of the equipment will consume the least number of man hours and the smallest quantity of material consistent with the required results.

Some further discussion of the choice of materials seems to be in order. It goes without saying that careful consideration of the most suitable materials should be a part of any design program. There is one phase of the question of suitability, however, that is particularly pertinent to military projects. This is the matter of availability. As we found during the last war, many materials that were completely available in peacetime were only available in very limited quantities or not at all in wartime. The designer should wherever possible make use of those materials that are expected to be most available in time of war.

Size and weight have long been important factors in the design of military equipment, but with the growing complexity of mechanized

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warfare and the constantly increasing need for more and more kinds of equipment in our Naval vessels, airplanes, tanks, etc., the need for still further reduction in size and weight becomes more urgent every day. Much effort is being executed on this problem both by the military services and industry, and the designer should keep abreast of this rapidly developing field.

By way of summary of the above I think we can say that the really fundamental differences between military and civilian design concepts are few. Sound engineering principles and methods will apply to any job. The differences in specific requirements, however, are many but they all seem to point in one direction, that of vastly increased severity.

The designer whose experience has been confined to civilian products invariably has difficulty in achieving a real appreciation of the extreme severity of military requirements.

We have found some educational work along these lines very helpful. For this purpose we have made use of demonstration tests on shock and vibration machines, high speed motion pictures of high impact shock tests, display boards illustrating corrosion resulting from various kinds of exposure, etc.

It is possible that an educational program along these lines sponsored by the military organizations would be helpful to many design organizations engaged in the development of military equipment.

DISCUSSION

Prof. Macduff, Brooklyn Polytechnic Institute:

Do you think the present control procedures in vibration and shock are good for laboratory tests, and are they sufficient for educating the personnel?

E. H. Jones, BTL:

We don't have sufficient resources to determine the degrees of parallelism between the tests now specified and actual field conditions.

I. Vigness, NRL:

We believe that the shocks that are provided by our high impact shock machines are sufficiently severe to simulate most field conditions. Of course, all the field work that is going on at the present time gives us added knowledge along this particular line, so that we can change the types of shock we give to more closely simulated field conditions. It is probable that if we take into account the location on board ship that we may have different shock tests corresponding to equipment

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location on board ship. Up to this time, it has not been possible to get answers from individuals as to where any partidular kind of equipment might be located on board ship, away from the hull structures, or at more or less protected locations. At the present time, most of our shocks are somewhat more severe than they might be if we could get definite commitments as to where equipment might be. I would like to ask you a question also. You have made some very good equipment for shipboard use, could you tell a little bit about the procedures that you follow in obtaining an equipment which can pass shock and vibration tests? How much calculation do you begin with? How much do you depend on tests?

Jones:

In general, we try to do as much testing as possible along the line. We are fortunate in having a light weight high impact machine at our disposal in our own test laboratory, and we make tests on complete units where their weight limit is within the weight limitations of that test machine, and in other cases, we make use of a simulated procedure wherever possible. Mr. Muller should be able to supplement my answer.

J. T. Muller, BTL:

First of all we get some idea what kind of shock we have to encounter then the designers on the drawing board get their instructions as to what the size of the equipment is going to be. From then on, we follow the arbitrary rules. For instance, we expect the shock in the bottom of the equipment to be more severe and for the components inside the equipment, the shock is somewhat attenuated. The equipment that is most rugged will be put in the bottom and vaccum tubes are put more towards the center of the equipment, cabinet, etc. If you follow this design procedure, you are building the type of equipment to withstand shock. Now in doing this, you have to consider the design values of your structures, and usually it is important to make your framework rather sturdy and independent of the screen which goes around it.

Klein:

It is unlikely that we can bring out the "design bible" before us and go through the various steps of designing a particular equipment. I would like to interpose at this point a question about one remark which Mr. Jones made. They are making things smaller so as to get more equipment into a tank or a plane, but what can be done about the operator with regard to miniaturizing?

Jones:

That is one contract we don't want to tackle.

G. R. Gohn, BTL:

I would like to emphasize that for the past six or seven years I have been engaged in trying to reconcile differences between service specifications and civilian specifications with respect to critical materials. In military application, the designers should not specify scarce materials without considering its availability. As an illustration, we attempted to sell the Services the use of low tin solders for military applications. Practically all your specifications call for 50-50 or 60-40 solder. We made very little headway even though tin was very critical and was needed for

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other applications. Moreover, there were no technical justifications for it. Investigations have shown that where shock or impact is a factor in your equipment, the high lead solder (more than 85 percent of lead) are much better from an impact standpoint, particularly at low temperatures. Now trying to sell that to the Services was rather difficult. Please bear in mind that you must justify the need for using critical materials. Don't assume that because it is for military applications, you can get all you want.

N. E. Lee, Coles Signal Lab:

Mr. Jones, do you very often find yourself in the position where the Service requirements lead to an impossible situation? For example, they give you rather rigid requirements as to size, weight, etc., also shock and vibration requirements. The request is to make an equipment light enough, and small enough to pass the shock and vibration test. If you are in this position, how do the Services generally reconcile the dilemma? If you ask them which of these particular considerations they most desire, what approximate sequence of importance do they come up with?

Jones:

In answer to the first question, yes, we do find more than occasionally that some of the requirements seem to be a little more than our organization can cope with. We have found, in general, that the Services have been more than willing to cooperate. It is our feeling that it is not a bad idea to try to "hitch your wagon to a star" and we will agree to make that a design objective rather than a requirement. I think in all fairness to the customer in this case, without these continually tightened requirements, progress would be very slow.

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VIBRATION IN TURBO-JET AIRCRAFT

by

H. A. Magrath and G. L. Getline

Vibratory conditions prevailing in different parts of turbo-jet aircraft are discussed. Data presented suggested to the designers of aeronautical structures and equipment those environmental conditions pertaining to vibration which should be taken into consideration.

The Dynamics Branch of the Aircraft Laboratory of the Air Materiel Command at Wright Field has responsibility for vibration control in Air Force aircraft. It has been found that information as to the vibrations that are likely to be encountered in aircraft is of considerable interest to other AMC laboratories and Government agencies; particularly to those responsible for the installation of equipment in the airplane. Recently, inquiries have been received as to the vibration characteristics of turbo-jet powered aircraft. This paper is an attempt to describe the vibrations which are likely to be encountered in such airplanes. The data upon which the discussion is based are the results of a series of tests carried out upon a Lockheed F-80A airplane.

In these tests, it was decided to attempt to measure the vibrations in such a manner that it would be possible to classify them according to magnitude, origin and distribution over the airplane. Since the vibration responses of an airplane are a function of the natural frequencies of the structure, data from ground vibration tests on the airplane were first gathered. Data were also gathered concerning the power plant and its operation in order to determine in advance the vibratory forces likely to be imposed on the aircraft structure by the power plant.

These considerations indicated that vibrations in the frequency range of 6 to at least 200 cycles per second (and possibly higher) were likely to be encountered. Seismic velocity type pickups were chosen as primary detecting instruments. Recording oscillograph elements of high natural frequency (2000 cycles per second) were used. Calibrations of the instrumentation at amplitudes of .0005", .001" and .020" were made.

The arrangement of pickups on the airplane is shown in Figure 1. Pickups were distributed spanwise along the wing and chordwise over

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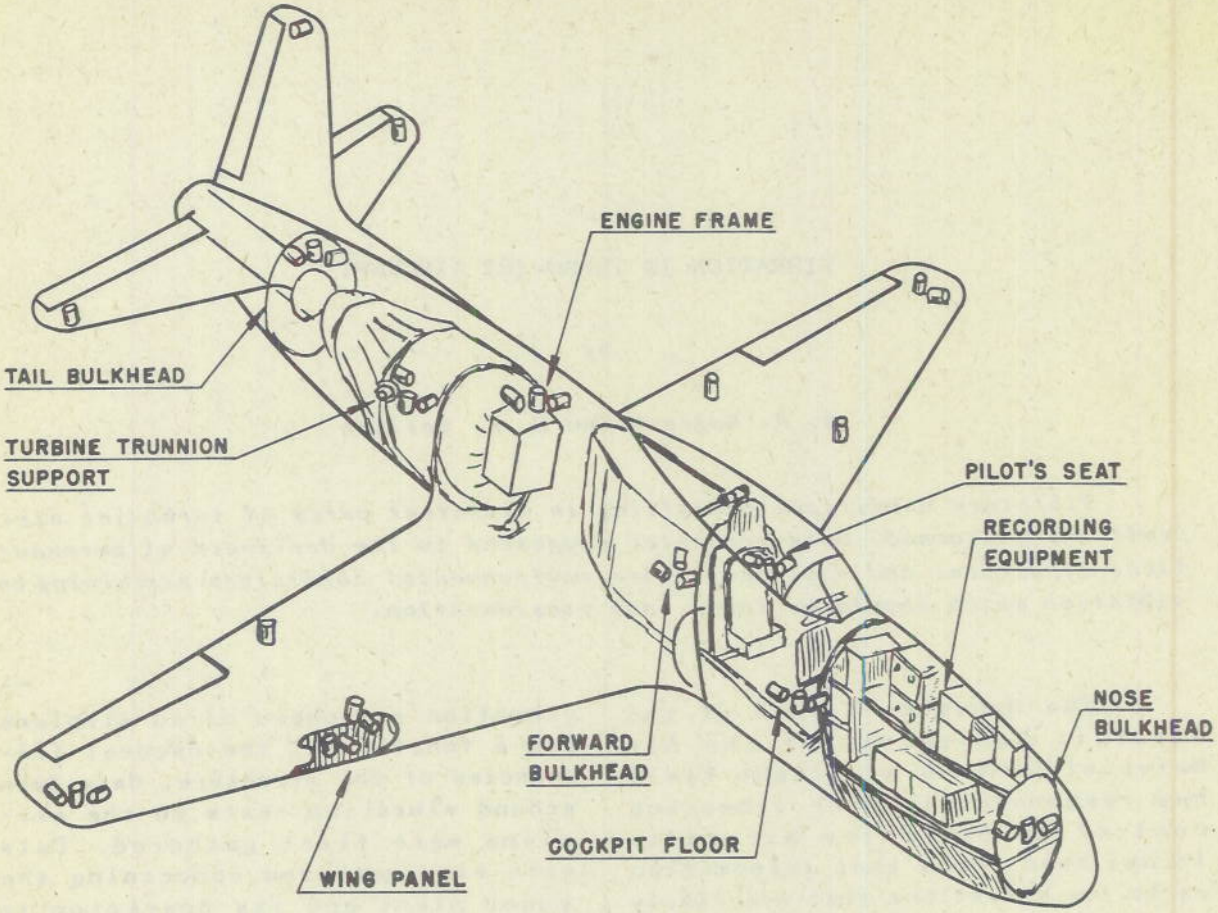

Figure 1 - Vibration pickup installation in F-80A No. 44-85348

it to measure its vibratory responses and obtain an indication of its vibratory modes, i.e., bending and torsion. Pickups were also distributed along the fuselage to measure its vibratory responses and modes. The fuselage pickups also measured the vibration of primary structure of the fuselage at (1) the nose bulkhead in the armament compartment; (2) the cockpit floor, pilot's seat and cockpit bulkhead; (3) the engine accessory case; (4) the engine trunnion support; and (5) the tail bulkhead. Pickups were also installed on the tips of the horizontal and vertical tail surfaces to measure their bending vibrations. In the armament compartment of the airplane, ten channels of integrating amplifiers and a recording oscillograph were installed. The recording equipment was remotely controlled by the pilot.

Tests were conducted at an altitude of 10,000 feet. Vibration records were taken when in level flight from 100 percent rated turbine speed (11,600 rpm) down to 80 percent rated turbine speed at

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increments of 5 percent of rated speed. Power was such that indicated airspeeds from 450 to 275 mph were obtained. In order to isolate the vibratory effects produced by the engine, vibration records were obtained at the same airspeeds but with the engine idling (35 percent rated speed). This was accomplished by diving the airplane to a speed slightly higher than desired, levelling out and taking the vibration record as the airspeed dropped across the desired value.

The large number of vibration oscillograms accumulated during the test were analyzed systematically in groups corresponding to the major airplane components; that is, the wing, fuselage and empennage. The character of the vibrations was noted; that is, whether they were intermittent, or transient vibrations, or periodic, steady-state vibrations. In all cases the amplitude and frequency were determined. When the recorded vibrations were complex; that is, contained more than one predominant frequency, methods of harmonic analysis were used to determine the amplitude and frequency of each component.

In order to present briefly what are considered to be significant results, the vibrations of the aircraft structure have been classified as to their frequencies, maximum amplitudes and maximum velocities. The distribution of the maximum velocities over the airplane was then compiled.

Figure 2 presents a summary of the frequencies of the predominant vibrations measured in the tests. These range from 500 to 12,000 cycles per minute. The frequencies between 500 and 1000 cycles per minute were invariably found to be transient vibrations and involved motions of the wing tips and forward part of the fuselage.

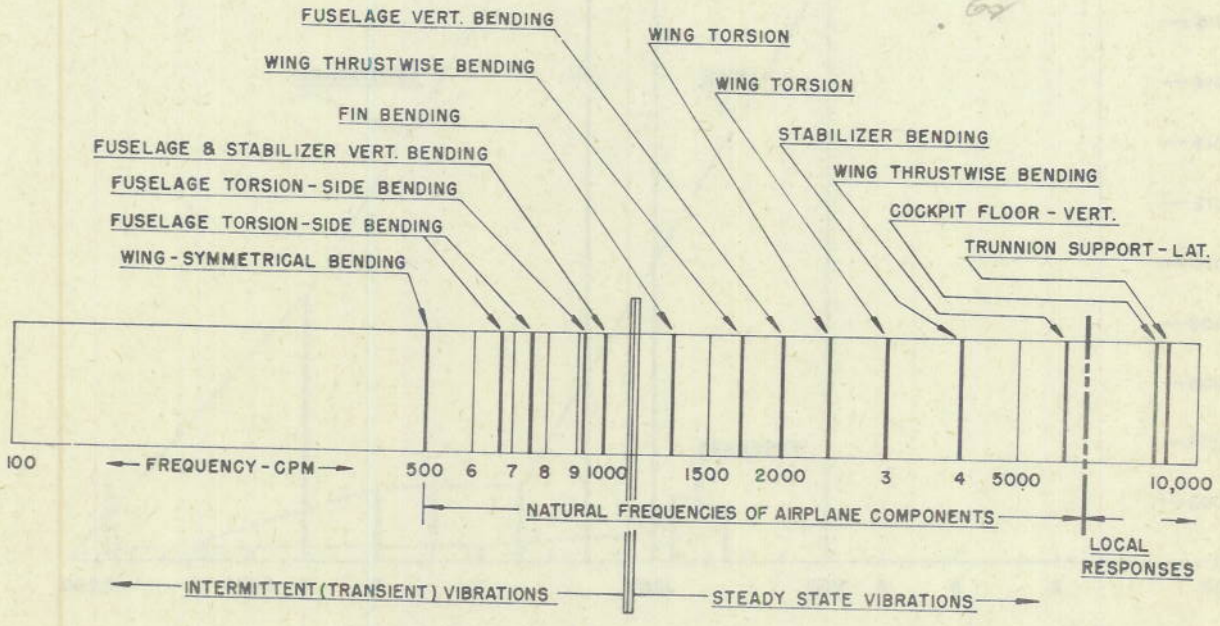

Figure 2 - *Important vibration response frequencies of F-80A airplane measured in flight*

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The vibrations between 1300 cycles per minute and 6000 cycles per minute were, in general, of a steady-state type and were due primarily to vibration of the tail of the airplane, but also involved thrustwise vibrations of the wing. The higher frequencies around 10,000 cycles per minute were also steady-state vibrations and were measured in the engine compartment and, to a lesser extent, in the cockpit. Their frequencies were proportional to the rpm of the engine.

Figure 3 summarizes the maximum amplitudes measured at the foregoing frequencies and indicates which major components of the airplane structure were involved. The double amplitude of the wing (0.020 inch) involved motion of the wing tips and corresponded to a transient excitation of the fundamental wing bending mode (500 cpm). The amplitudes of the wing became smaller at the higher frequencies (.002 inch) and became more of the nature of steady-state vibrations. They are comparable in magnitude to the fuselage vibrations; in fact, at these higher frequencies, the fuselage vibration coupled to that of the wing. The fuselage amplitudes were small (one to two thousandths of an inch) at the lower frequencies,

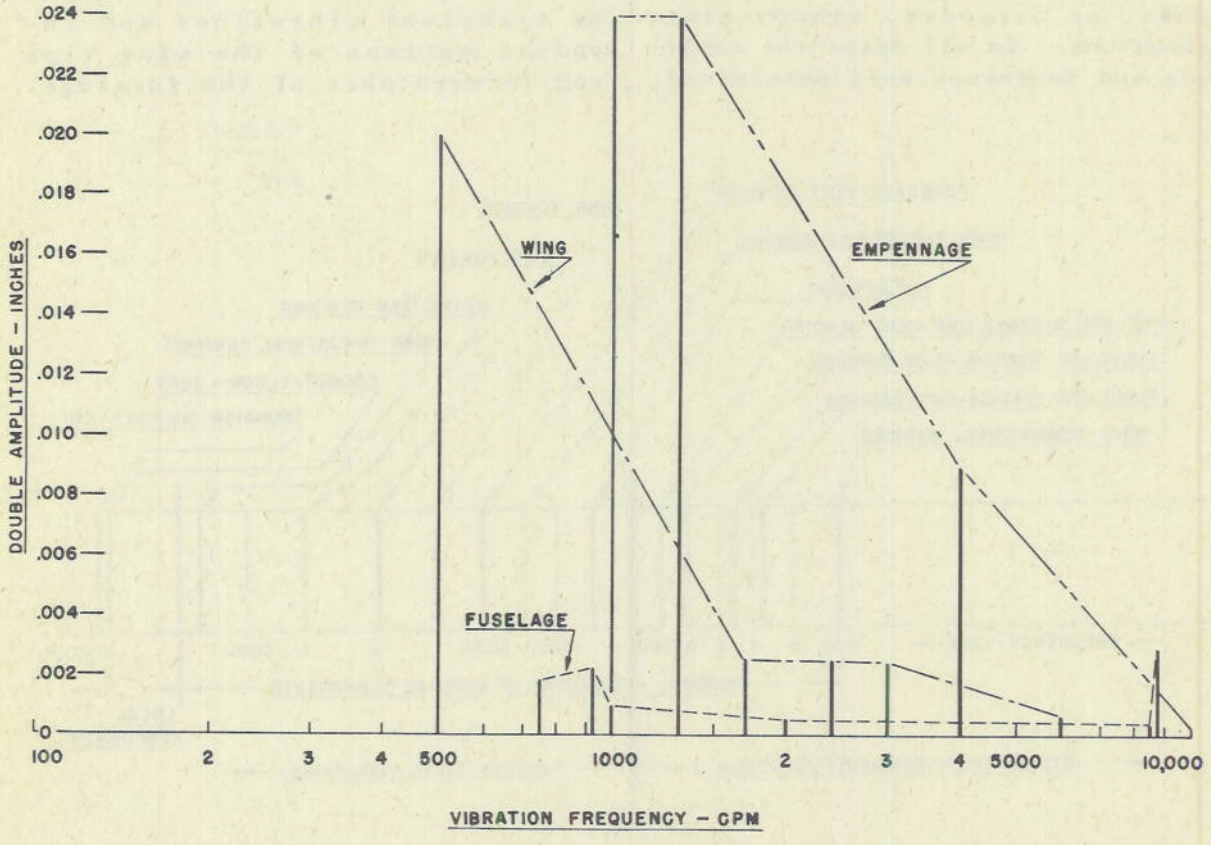

Figure 3 - Maximum amplitudes of major airframe components as a function of frequency F-80A airplane

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corresponding to transient vibration of the fuselage in side bending modes. The steady-state amplitudes of the fuselage were significant at high frequencies corresponding to engine rpm. The largest vibration amplitudes were measured on the tail. The vibrations occur at structural natural frequencies, stabilizer and fin bending and torsion modes coupled with fuselage vibration, and at frequencies corresponding to engine rpm. The large amplitude vibrations are considered to be excited by the wing or other wake effects.

In describing the vibrations, specification of both amplitude and frequency has been necessary. To show the distribution of vibration over the airplane it was decided to describe the vibrations in terms of vibratory velocity since this parameter gives equal weight to amplitude and frequency. Figure 4 shows this distribution. The maximum amplitudes of vibratory velocity at several points on the aircraft structure are shown by the heavy arrows. This Figure may be considered to express, in terms of vibratory velocity, maximum

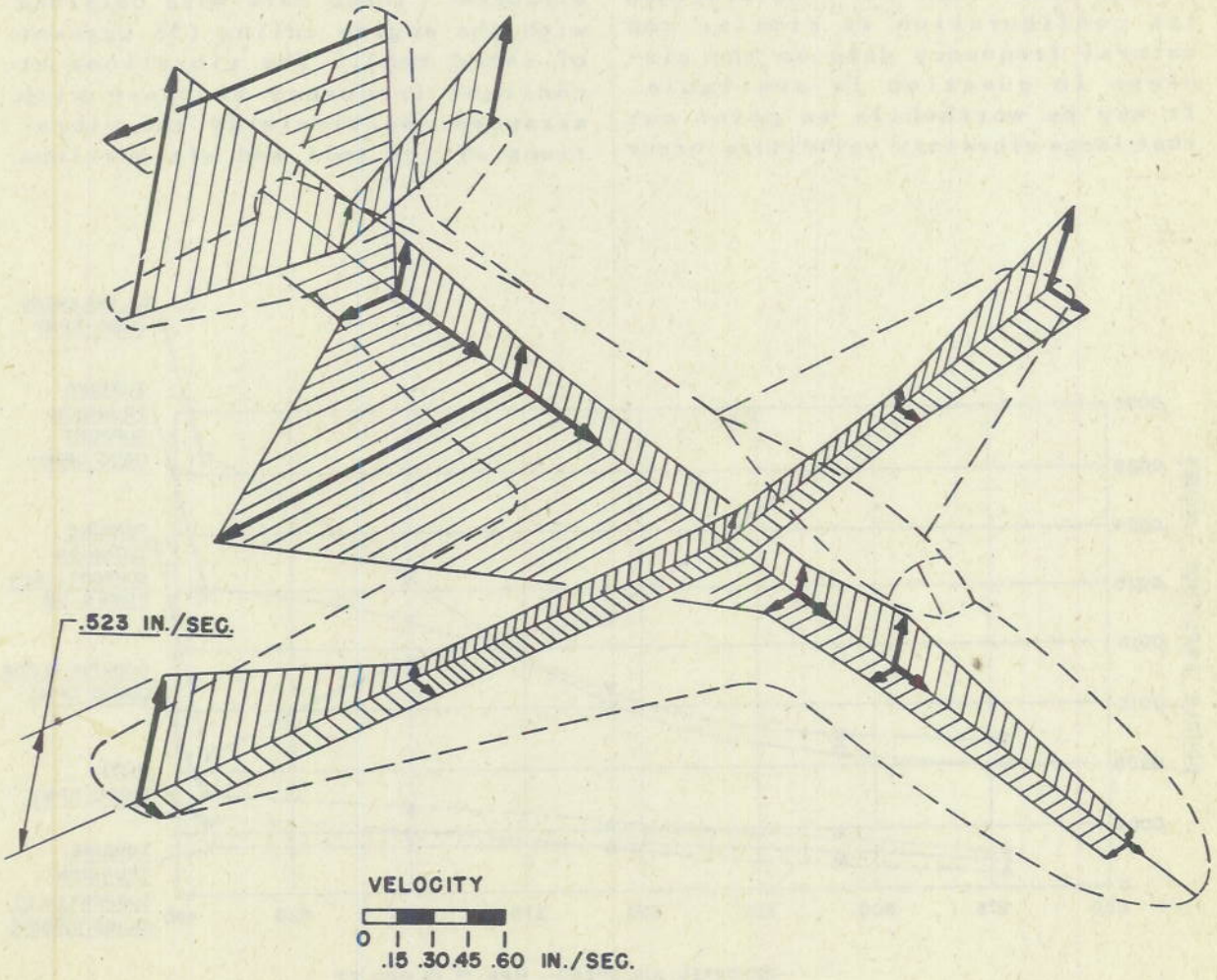

Figure 4 - Maximum vibration velocity diagram of F-80A airplane

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vibratory amplitudes since the velocities shown, if divided by a proper value of frequency, will give a maximum amplitude of vibration. However, this will be strictly true only for the specific locations at which vibrations were measured. This Figure is considered to represent, nevertheless, a diagram of limiting vibrations which are likely to be encountered in the airplane in general; and it may be useful as a guide to designers interested in environmental vibrations in this airplane. Further, it is considered that this diagram might possibly be applied to airplanes other than the F-80 provided its configuration is similar and natural frequency data on the airplane in question is available. It may be worthwhile to point out that large vibratory velocities occur in the fuselage in the engine compartment. Since the amplitudes measured were small (0.003 inch) the large vibratory velocity is due to the high frequencies (12,000 cpm). In this case, vibratory accelerations are also large. On the other hand, vibration in the pilot's cockpit is small.

So far, maximum values of vibratory amplitude and velocity have been discussed. Another interesting point is the variation of vibration with airspeed. Figure 5 shows vibratory responses of the fuselage and tail plotted against airspeed. These data were obtained with the engine idling (35 percent of rated rpm). The vibrations at constant frequency increase with airspeed, particularly the vibrations of the tail and aft portions

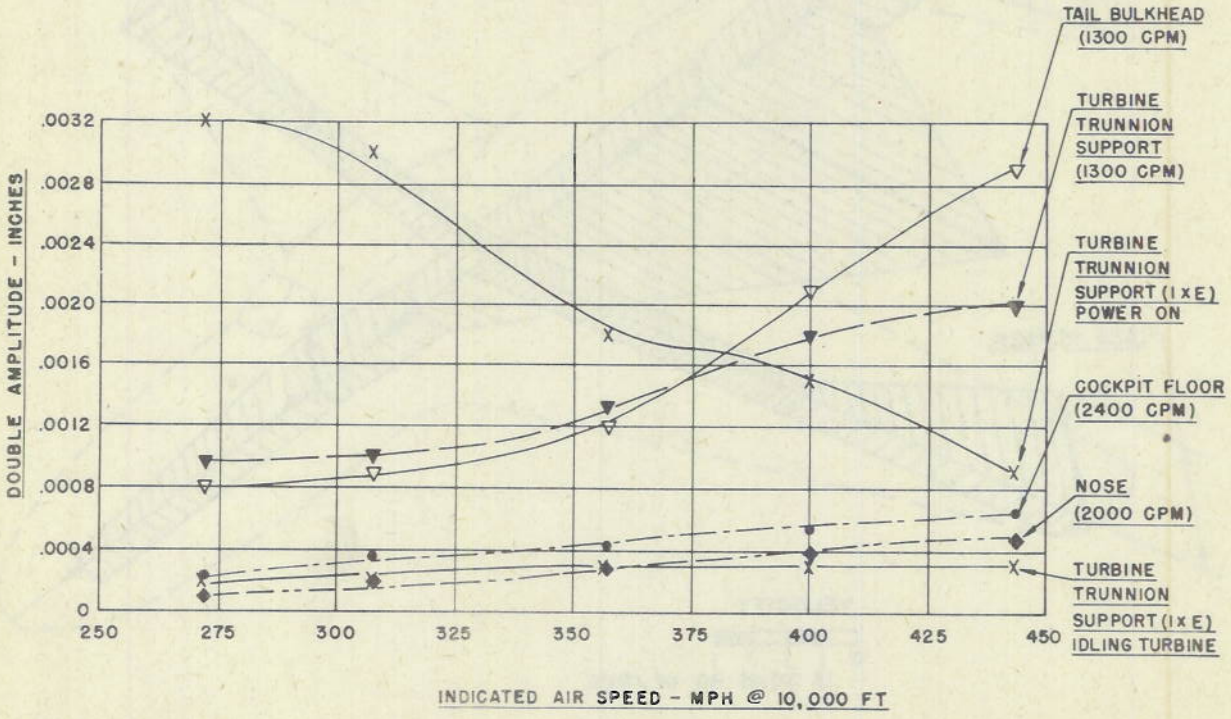

Figure 5 - Vibration response vs indicated airspeed F-80A airplane fuselage

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of the fuselage. Thus with increasing airspeed, vibration becomes increasingly important. Also shown on this Figure is the vibration component, the frequency of which corresponds to engine rpm. With the engine idling, the vibration is very small at all airspeeds. However, when varying the airspeed by varying engine thrust (that is, by operating the turbine) the vibrations are much larger (as could be expected) but show a decrease with airspeed. Of course in this case the controlling factor is engine rpm; the conclusion is reached that the 1 x E forcing function is exciting some natural mode involving the engine and its mounts in the neighborhood of 8800 cycles per minute. This illustrates the necessity of isolating the real variable in interpreting vibration data. In general, significant vibration involves natural frequencies; when the natural frequencies are all well below the fundamental vibration frequency generated by the power plant, the vibrations are proportional to airspeed; when a natural frequency coincides with a fundamental vibration generated by the power plant resonance occurs and the vibration response is a function of engine operating rpm. In aircraft equipped with engine-propeller combinations many structural natural frequencies coincide with engine generated frequencies, resulting in large vibrations at relatively low frequencies and involving resonant modes of major aircraft components. In turbo-jet powered aircraft, the high rotating speed of the turbine separates the engine generated frequencies from the natural frequencies of major structural components of the airplane which has the result of localizing the high frequency vibrations in the engine compartment.

In reviewing the test data it appears that the following conclusions are justified:

1. Transient vibrations of large amplitude involving the wing tips and smaller vibrations of the forward portion of the fuselage are excited by air gusts, i.e., rough air.

2. Forced vibration of the tail involving large amplitudes of fin and stabilizer tips and the aft portion of the fuselage occur at constant natural frequencies; the amplitudes increase markedly with increase in airspeed. The vibrations may be excited by wing and other wakes. The mechanics of this excitation have not been definitely determined.

3. Local resonances involving the power plant and its mounts are excited principally by the 1 x E power plant forcing function producing comparatively large vibratory velocities and accelerations in the engine compartment.

4. Vibration responses in the pilot's cockpit are below the threshold of severe vibration on accepted indices of human reaction to vibration but are above the comfort level for prolonged exposure and hence may contribute to pilot fatigue.

5. Vibration characteristics of turbo-jet powered aircraft are expected to be influenced by the position in the airplane of the power plant and jet exhaust.

An attempt has been made to analyze the vibrations in a turbo-jet powered aircraft to indicate the character of the vibrations, whether transient or steady-state; the frequencies of the vibrations and their maximum amplitudes; and to show how these vibrations vary in different parts of the airplane. It is hoped that these data will

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be useful to those needing information concerning environmental vibration in aircraft for design or test purposes. It is realized that the data presented reflect vibratory conditions in one airplane only when flying in level flight at subsonic speeds. It is also realized that other vibratory conditions of importance, such as the response of the structure to the firing of guns or to landing impact was not investigated. It is planned, however, to continue tests to obtain needed data; and to extend vibration investigations to jet-powered aircraft other than the fighter type.

DISCUSSION

W. Ramberg, National Bureau of Standards:

Structural designers are particularly interested in stress and strain. Have you tried to convert velocity amplitudes into stress amplitudes?

H. A. Magrath, Wright-Patterson Air Force Base:

Unfortunately we have not done that. The data that was shown was rather new and we have not converted all data into elastic strain curves, positions of maximum stress, or things of that nature.

G. L. Wannall, Glenn Martin Co:

Various aircraft companies are apparently observing specifications for mounting equipment which have been primarily designed around reciprocating engine aircraft, where isolation from the reciprocating engine frequencies are considered of primary importance. We are still going to be confronted with the responses to transient loads, and therefore, will require isolation from those frequencies. We are now faced with the problem of how to tackle the engine frequencies, and also putting more emphasis upon the shock loads which you mentioned. What do you foresee in the future as possible changes in specification requirements for both vibration and shock mounted equipment, specifically in different aircraft due to firing and landing loads?

Magrath:

Well, I think what is going to happen, instead of writing specifications around the reciprocating engines, we are going to have to take into consideration the dynamics of guns and also do a little dynamic load work. To answer the question, I agree to a great extent, the specifications were developed a long time ago around what appeared to be very important factors. They are likely to be considered not so important anymore.

G. W. Dorr, NRL:

Along this same line, you have been speaking primarily of vibration and structure of aircraft in reciprocating engines. We must also take into consideration sound level. Is the noise level

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in the turbo-jet engine intense enough to be objectionable from a physiological standpoint?

H. C. Parrock, Wright-Patterson Air Force Base:

Noise level of those frequencies which are important in understanding human speech are much worse in jet aircraft than they are in conventional propeller-driven aircraft.

J. H. Van Trump, NBS:

Are plans being made to investigate the frequencies and amplitudes in the higher range? Are they considered important?

Magrath:

So far we have found that frequencies above 6000 cycles per second are important from the point of view of the engine, turbo, and vibration matters of that kind. They are also important from the standpoint of acoustics. With the equipment that we are using we feel that we can't say very much about this subject. In other words, the higher frequencies are a matter of more investigation.

C. B. Cunningham, NRL:

Similar tests have been conducted at NRL and the results will be described by Mr. Feher at this time.

L. H. Feher, NRL:

Generally, available instruments to measure such vibration in the vehicle are cumbersome and unsuitable for installation in small aircraft. Furthermore, the records when obtained were difficult and tedious to analyze and often involved mathematical efforts for considerable time. A telemetering system which is less involved has been developed at NRL. This technique together with vibration records obtained from two types of aircraft are outlined.

Figure 1 shows a velocity pickup and an FM transmitter together with a dynamotor which were installed in a Navy type SNB aircraft. The total weight of the plane installation including dynamotor was less than 15 pounds.

Power for this equipment was obtained from the aircraft itself. Radio voice communication with the pilot was maintained by means of VHF radio as vibration data from ranges up to 85 miles was telemetered.

Although only one or two channels were used, the variable area recorder can permit the simultaneous recording of 13 channels and operates by photographing the shadows of vibrating galvanometer strings on 35 mm film.

Figure 2 shows the vibration spectrum obtained from a type SNB airplane on its second experimental flight. The dominant frequencies in order of their magnitude are 68, 136, 204, and 34 cps. These frequencies were attributable to vibrations caused by two twin bladed propellers rotating at 2000 rpm. An 8 cps component of small amplitude, believed to be the result of wing vibration was also found. Vibration was measured halfway back on the fuselage at an electronic equipment location.

The graph, Figure 3, represents part of the work done on a problem assigned to NRL by the Bureau of Aeronautics. The purpose of this frequency analysis was to

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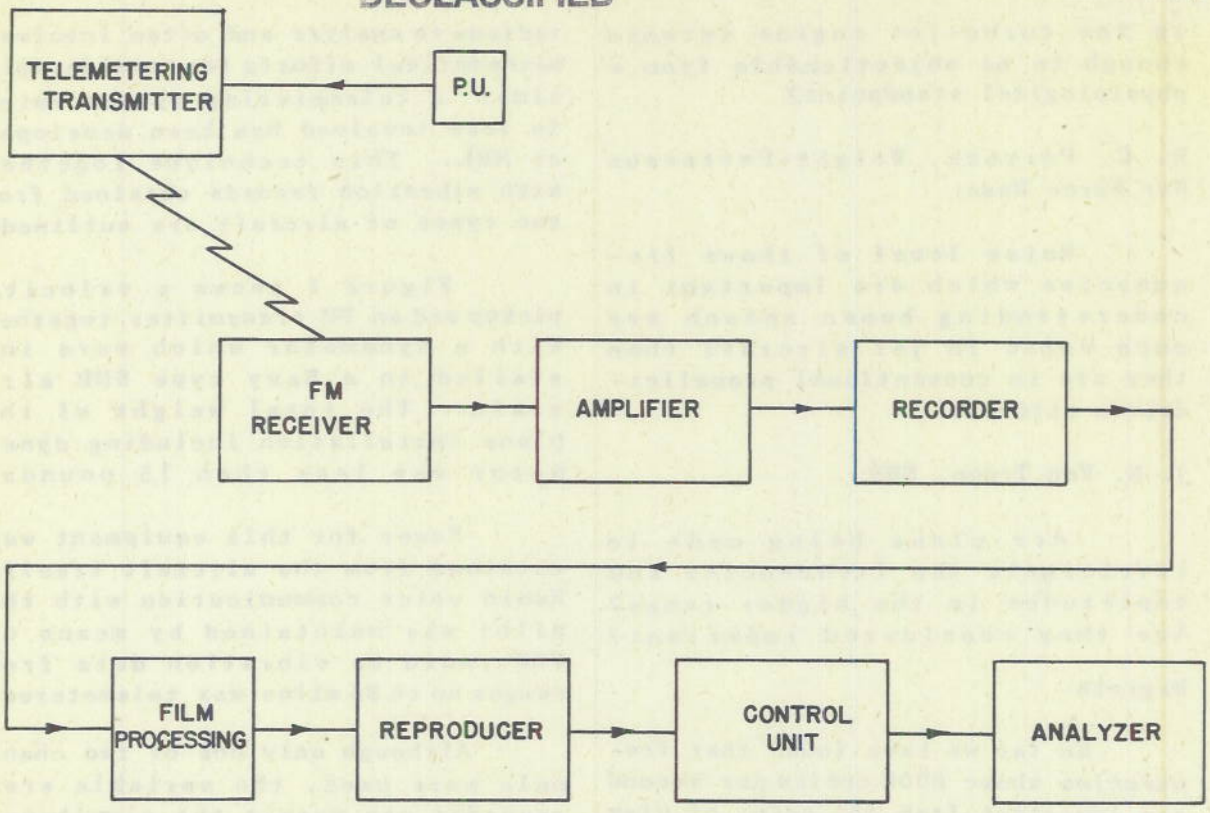

Figure 1 - NRL vibration recording and analyzing system

determine the frequencies at which there was no excitation so that adequate shock mounts could be designed to protect electronic equipment. Vibration in this case was measured at a wing tip where the wing supported a radar nacelle housing the APS-19. Vibration frequencies here encountered are attributable either to vibration caused by the four-bladed propeller driven at an engine speed of 2400 rpm or to wing vibration excited by turbulence.

The complete system as described is capable of recording and analyzing vibration frequencies from 2 to 6000 cps and vibration amplitudes from one mil to one inch.

Klein:

The purpose of injecting this particular series of measurements here is to show a new technique which ordinarily is quite difficult to achieve, which has been achieved, and can be used on any type of aircraft or vehicle.

J. P. Walsh, NRL:

These results which Mr. Feher has shown bring up again this question of specifications that was raised by Mr. Wannall. The equipment was breaking up under landing impact with values of acceleration in the order of 20 g in the aircraft. Vibration isolators which

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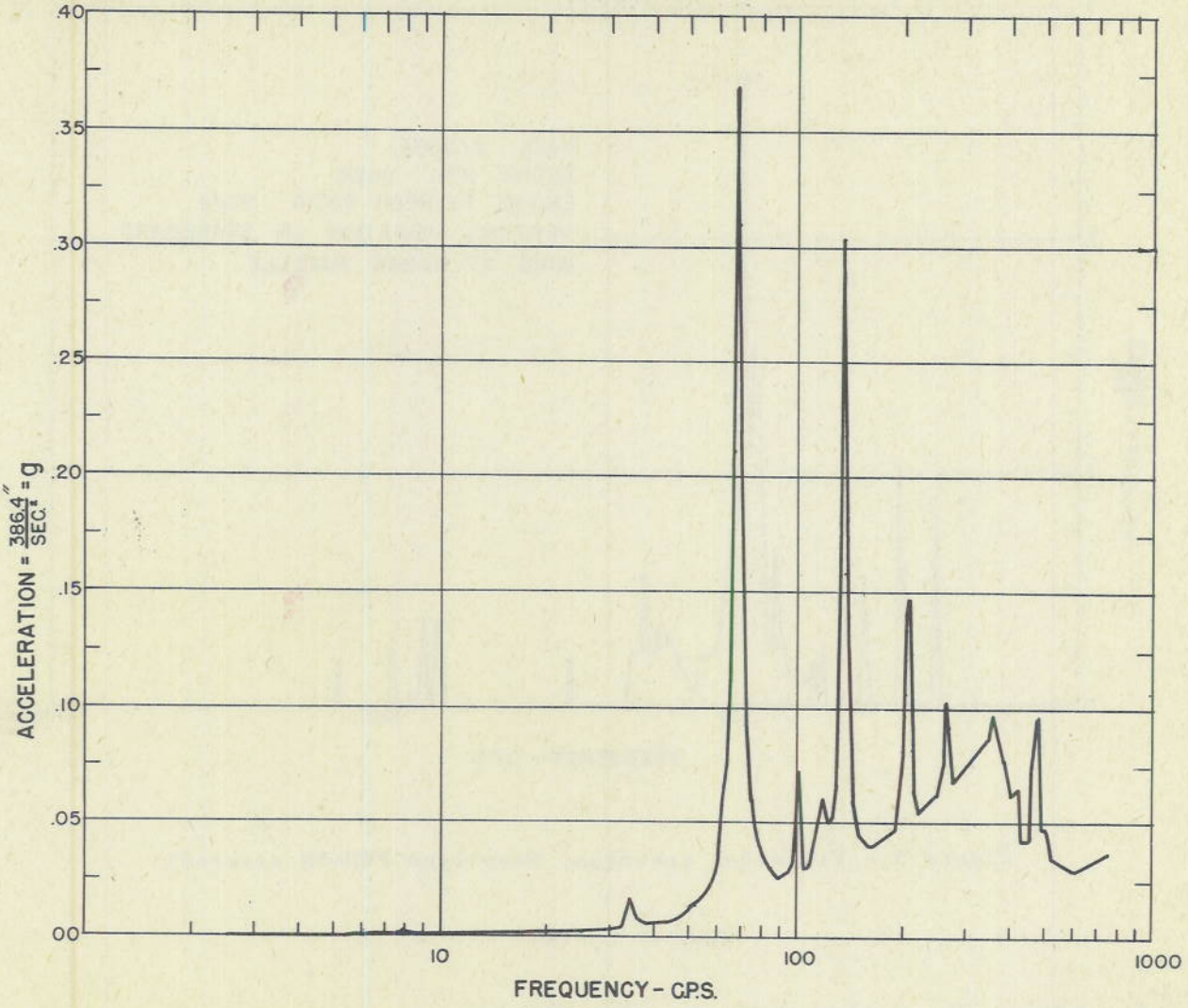

Figure 2 - Vibration spectrum, Navy type SNB aircraft

had been tested were found inadequate to protect this equipment. This was not a vibration problem but actually a shock problem. That again emphasizes this point that specifications are lacking in many cases.

D. E. Weiss, NAMC:

I am very much interested in a velocity pickup which gives halfway decent data below 10 cycles per second. Can you give me any information with respect to such a velocity pickup?

Feher:

The type velocity pickup we used was calibrated on a vibration table at frequencies all the way down to about three cycles. The spectrum as shown in Figure 3 was corrected for frequencies below

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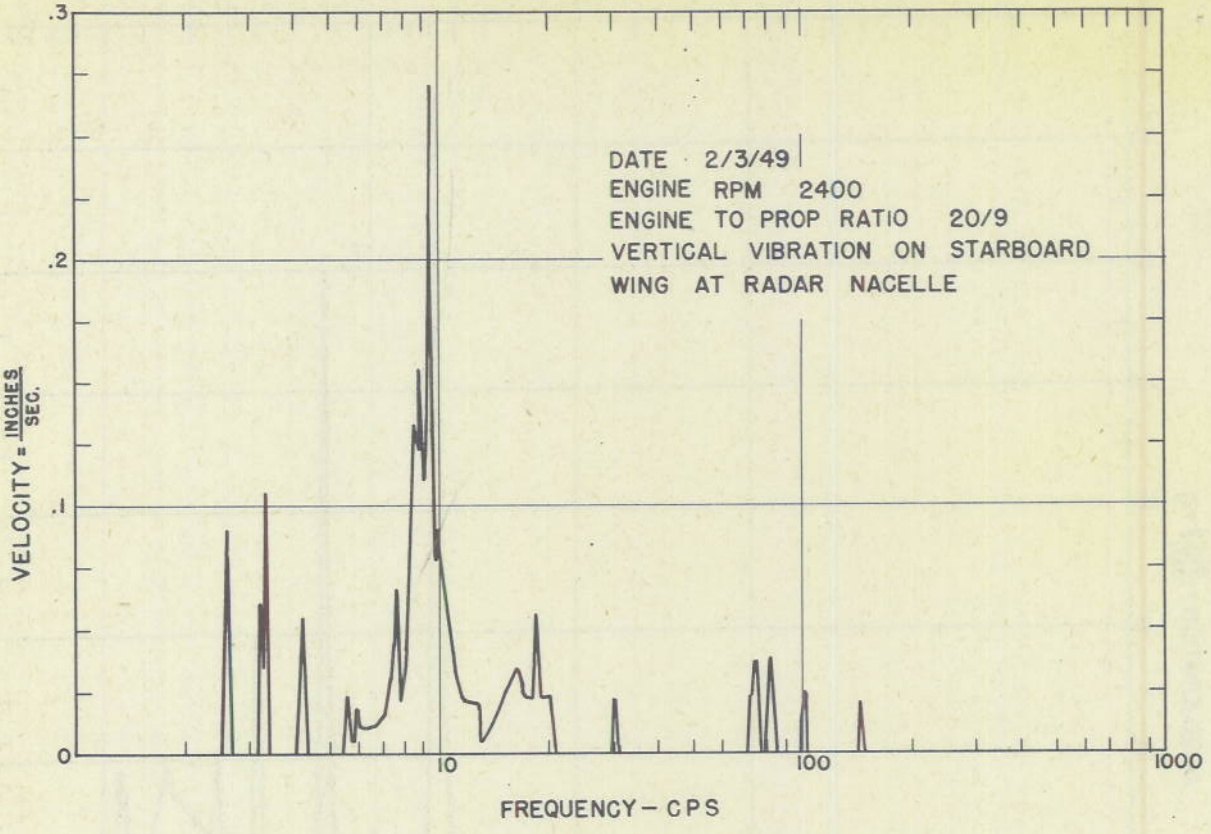

Figure 3 - Vibration spectrum, Navy type F4U-5N aircraft

5 cycles. The pickup used was the consolidated type 4-102.

F. K. Priebe, Squier Signal Lab.:

I would like some information concerning telemetering systems used in individual channels.

Feher:

We used a 75 kc channel. On that channel we made provision for a sequence timer, but we did not use it for more than two pickups. One telemetering channel can be used for any number of pickups providing suitable sequence switching arrangements can be made. Our channel was assigned to one pickup.

Priebe:

Do you transmit and receive information from several pickups simultaneously?

Feher:

Not simultaneously, one channel of 75 kc is used to transmit information from various pickups. A sequence time switched from one pickup to another and the information from the pickups was transmitted on that one channel (frequency) allotted to us.

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Gohn:

I think the type of data here presented are absolutely essential. In order to help the materials man and the designer to select the proper materials, the speed of testing is important also. Fatigue strength at various frequencies and stress amplitudes must be known.

Tuckerman:

I am not interested in amplitude vibration as is the designer who refuses to pay attention to what we have said about fatigue failures and stress. I have seen a recent number of cases of serious accidents where every single precept that we have set up to avoid stress concentrations were ignored.

G. L. Getline, Wright-Patterson Air Force Base:

I want to point out that at AMC our harmonic analysis doesn't take a corps of mathematicians. We have five. It is just a matter of breaking down your fundamental from any number of components you want from the first to the hundredth harmonic regardless of frequencies.

Priebe:

Work similar to this has been done during the war on landing vehicles by the Chrysler Laboratories of the Ordnance Dept. of the Army and the spectrum obtained for a large number of failures in vehicles, were subsequently analyzed.

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NEW PROPOSALS FOR THE STRENGTH DESIGN OF GUNS, MOUNTS, AND TURRETS

By

Edward Wenk, Jr., Taylor Model Basin

The development in recent years of improved experimental techniques has permitted the widespread checking of strength criteria for guns and turrets and their auxiliary components. The measured stresses have been found to be usually lower than those computed; and the discrepancy has been attributed to the conservative nature of assumptions in design regarding the exciting gunfire forces, and the boundary conditions of the structure. In the absence of definitive requirements for stiffness, the measured displacements have furnished the basis for stiffness criteria.

To obtain structures in the future which may be lighter in weight, and more efficient in proportions, but which will nevertheless possess the required strength and stiffness, there are proposed several new techniques of analysis which blend experimental and computational processes. The application of this technique to the design of components of ship structures other than turrets is also discussed.

INTRODUCTION

By the end of this year, over half a billion dollars will have been invested by the Navy in the construction of five new ships which are to function tactically as floating platforms for guns. Three vessels are of the SALEM Class which mount 8-inch guns, and two vessels are of the WORCESTER Class which mount 6-inch dual-purpose guns. Certainly the most important military characteristic of these ships is their greatly increased rate of gunfire over that of earlier classes, for each 8-inch gun may fire 10 rounds per minute, and each 6-inch gun may fire 12 rounds per minute. There is thus placed at the disposal of ship commanders a concentration of firepower which permits full exploitation of tactical elements of surprise in surface engagements, and improves protection against high flying speedy aircraft or missiles which may penetrate air borne defense. It appears that this mode of antiaircraft protection will be sustained for some time to come.

To translate the concept of rapid gunfire into practice, several unique ordnance systems have been developed, such as the use of case ammunition with guns as large as 8-inches in bore diameter, and special mechanical and structural arrangements for swift ammunition handling. Even though these new

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developments were solved on the drafting table, the importance of proper operation of the guns and turrets was of such paramount significance, that pilot models of some turrets were assembled and tested in advance of construction of the ships themselves to confirm the design characteristics.

Those designers in both the Bureau of Ships and Bureau of Ordnance who conceived and developed these turrets and engineers at the Naval Gun Factory who designed the components should be credited with products which performed almost faultlessly upon initial operation. These designers should also be credited with having recognized that perfect functioning was not the sole objective of rational design; for the use of conservative approaches which insure satisfactory operation may coincidentally introduce excessive weight or size factors in the ships. After all, numerous approximations are required in the course of a design, and experimental proof of the validity of assumptions has not generally been obtainable until recent years when adequate scientific instrumentation was developed. Thus, with the opportunity presented during service tests of pilot turrets, the design bureaus requested the Taylor Model Basin to collect information so that the existing design criteria could either be confirmed, or refined. Not overlooked were the potential generalizations that could be developed for the future design of ship structures other than turrets which experience dynamic loads in service.

In this discussion, I should like to present samples of these test results, compare them with the behavior computed by existing criteria, and where discrepancies occur, to offer an explanation.

As a matter of personal opinion, I should like to propose several new approaches to design which differ from contemporary procedures in that experimental analyses are made an integral part of the design process.

To furnish background information for those of you who possibly may not be familiar with modern turrets, I should like to preface this discussion with a brief account of turret operation.

DESCRIPTION OF THE 8-INCH 55-CALIBER GUN TURRETS

The most comprehensive recent study of guns and turrets has been made of the 8-inch 55-caliber size guns and turrets, but the contemporary principles involved are applicable to all sizes.

A ship of the SALEM Class is shown in Figure 1; the armament includes 20 millimeter and 3-inch 5-caliber antiaircraft batteries, 5-inch dual purpose secondary batteries, and a main battery consisting of three 8-inch 3-gun turrets.

Typical details of operation and construction of one of these turrets are shown in Figure 2. The primary components are the cylindrical turret foundation, and the rotating structure, with functional separation of the two by a horizontal roller bearing or track. Suspended from the rotating structure are the ammunition storage and handling platforms.

When discharging guns automatically, the only personnel required in the turret beside the gun captains are at the lower levels to insert projectiles and powder cases in their respective hoists, on opposite sides of each gun. These hoists lift projectiles and cases

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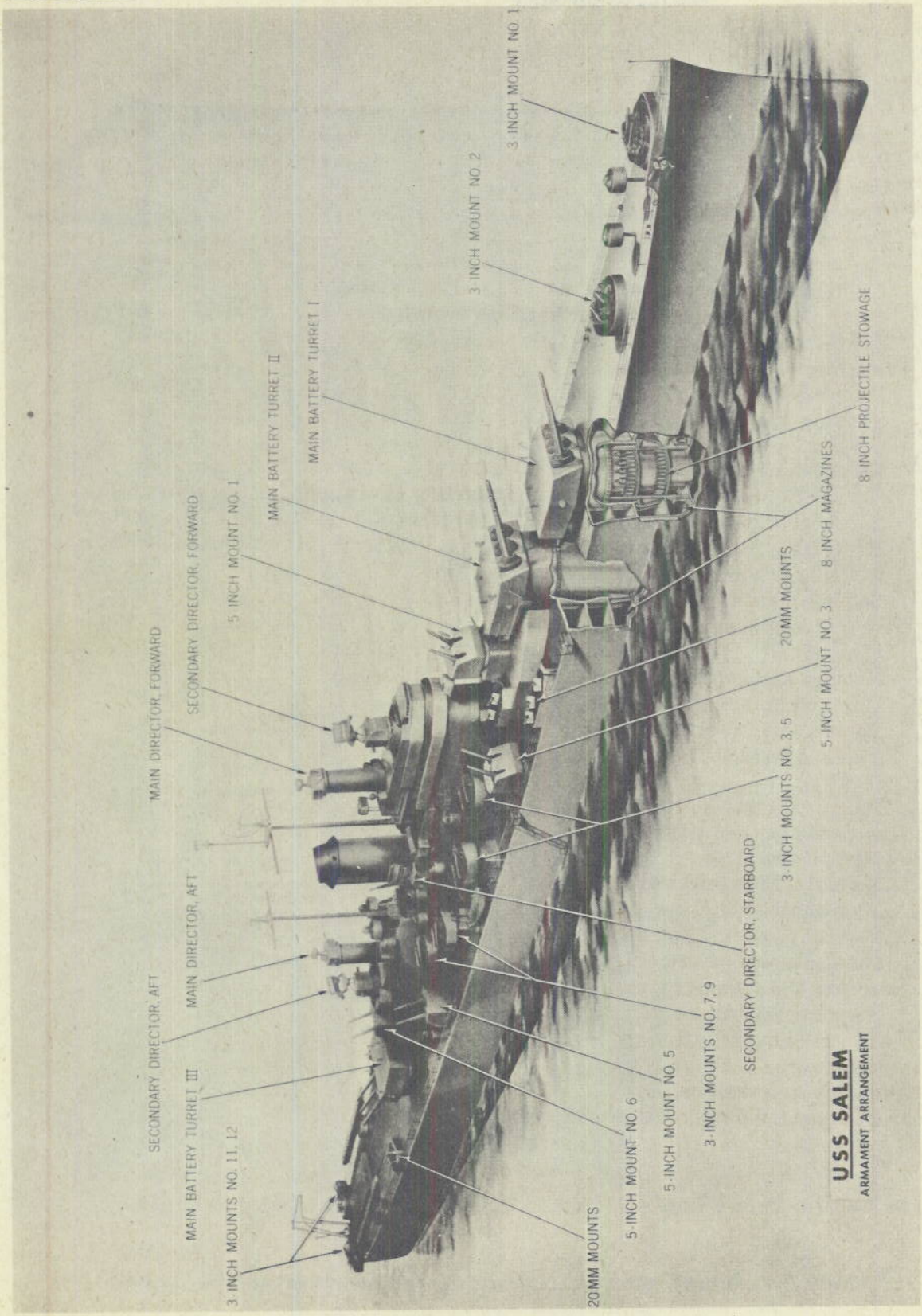

Figure 1 - USS Salem—armanent arrangement

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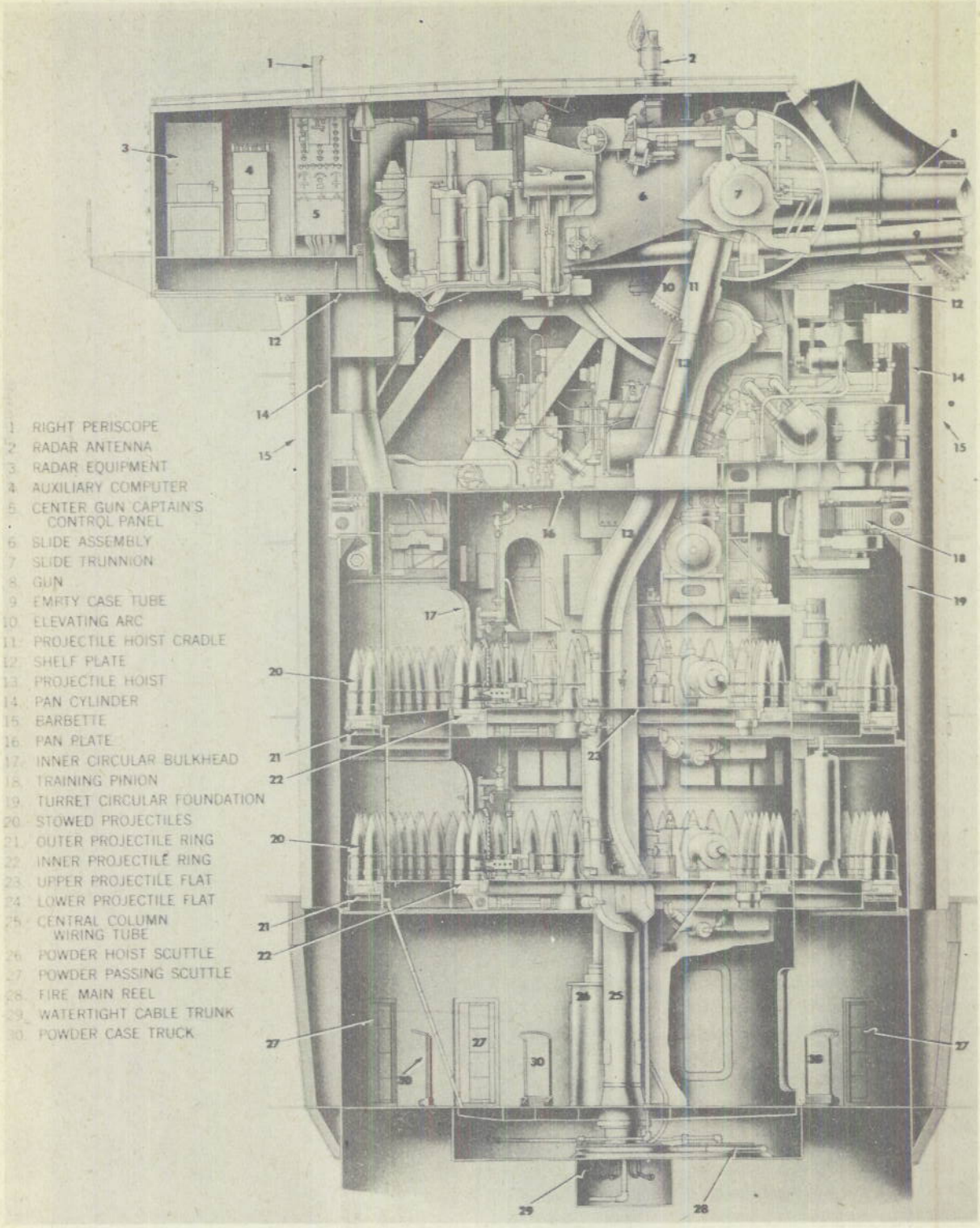

Figure 2 - Turret general arrangement, longitudinal section

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to cradles attached to the gun slides. When loaded, the cradles are pivoted about the gun trunnion and latched to the slide, with the gun either stationary at any elevation, or in motion; the projectiles and cases are then transferred from their cradles to trays which are then aligned with the barrel. Then, with the breech ready to be charged, the projectile and case are rammed home simultaneously, the breech block closed, and the gun fired. The empty case is ejected and the gun prepared for another cycle of continuous gunfire.

Novelty in design was not restricted here to the ammunition handling arrangements, for the structure which supports the guns differed radically from that in earlier turrets. Most important, as shown in Figure 2, the inboard gun girders were designed as welded trusses assembled with rolled I-sections in contrast with solid plate girders heretofore employed.

In accordance with general ordnance practice, the 8-inch guns are provided with a recoil system so that the reaction from gunfire loading is absorbed over a controlled space-time interval with a consequent minimum intensity of force imposed on supporting structure, and on the training drive. In addition, a counter-recoil system is provided to return the gun to battery after it has recoiled. As shown in Figure 3, for the 8-inch guns, the recoil mechanism functions as a hydraulic buffer which absorbs energy by accelerating buffer fluid through annular orifices formed between the throttling rods and holes in the piston. The counter-recoil mechanism, in which air is initially stored at high pressure, also absorbs a random quantity of the recoil force; but it then functions as a compressed spring to return the gun to battery. So as to obviate shock on return to battery position, the return motion is arrested by the reaction of the counter-recoil hydraulic buffer at the muzzle end of the recoil cylinder.

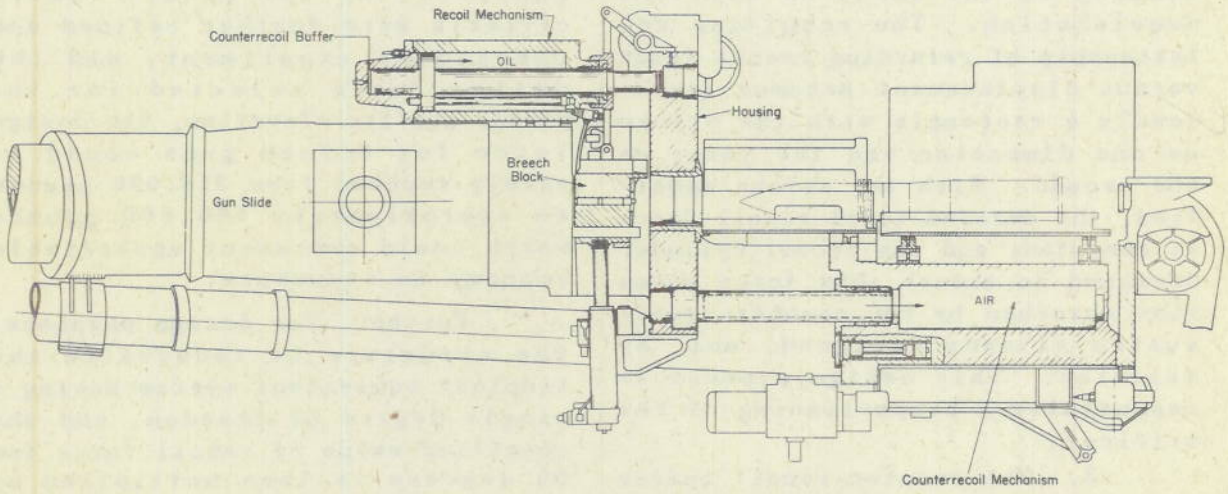

Figure 3 - The recoil and counterrecoil mechanisms for the 8-inch 55 caliber gun

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DESIGN OF THE RECOIL SYSTEM

Time for this discussion is too limited for any but a brief commentary on the procedures employed in the design of this complex of machinery and structure just described, so that most of my remarks will be confined to the recoil system and structure.

The recoil-counter-recoil system for naval guns is designed in the following sequence.

1. First of all, the counter-recoil system is chosen to hold the gun in battery position when at maximum true elevation with a spring force equal to the gravity component of the recoiling mass, plus friction. The maximum true elevation for these guns is 60 degrees, comprising the maximum gun elevation of 40 degrees plus the maximum angle of ship roll at which guns are fired, 20 degrees.

2. The motion characteristics of the freely recoiling system produced by gunfire are then computed at maximum true elevation, and the retarded recoil motion defined usually on the basis of constant deceleration. The resulting relationship of retarding recoil force versus displacement becomes graphically a rectangle with the stroke as one dimension and the force as the second. With the stroke specified, the desired total recoil force is computed; and the recoil cylinder designed to absorb this force minus that absorbed by the counter-recoil system already designed, and by friction. This design process is essentially a proportioning of the orifices.

3. The counter-recoil buffer is then designed to arrest the gun with a uniform force when it is being returned to battery.

The total recoil force and its constituent elements were measured accurately for the first time during tests of the 8-inch guns, and are compared with design values in Figure 4. It is apparent that the desired intensity and time-history of the recoil force, is not completely realized, but agreement is very good in view of the critical effect on this relationship of only the size of the orifice. Based on these tests, refinements are now being developed for existing design criteria and it is presumed that these be accepted for future design so that the new formulas behave as all valid formulas should; namely, to predict accurately the result of experiment.

For the design of the supporting structure and training drive, the recoil force is specified as a rectangular pulse of intensity 1.25 times the previously computed value of recoil load at maximum elevation as shown in Figure 5; the arbitrary factor of 1¼ actually represents a factor of safety. This value of design force is applied in design at 0 degrees to the structure, even though it is based on high angle gunfire. If the recoil system criteria were further refined and checked by experiment, and the design force selected for the proper gunfire elevation, the design force for 8-inch guns could be safely reduced from 215,000 pounds to approximately 160,000 pounds which could represent appreciable economy in structure.

Further, for design purposes, the structure is reduced to the simplest equivalent system having a single degree of freedom, and the specified value of recoil force for 60 degrees is then multiplied by still another factor to account for the dynamic character of loading. As is now well cited in literature

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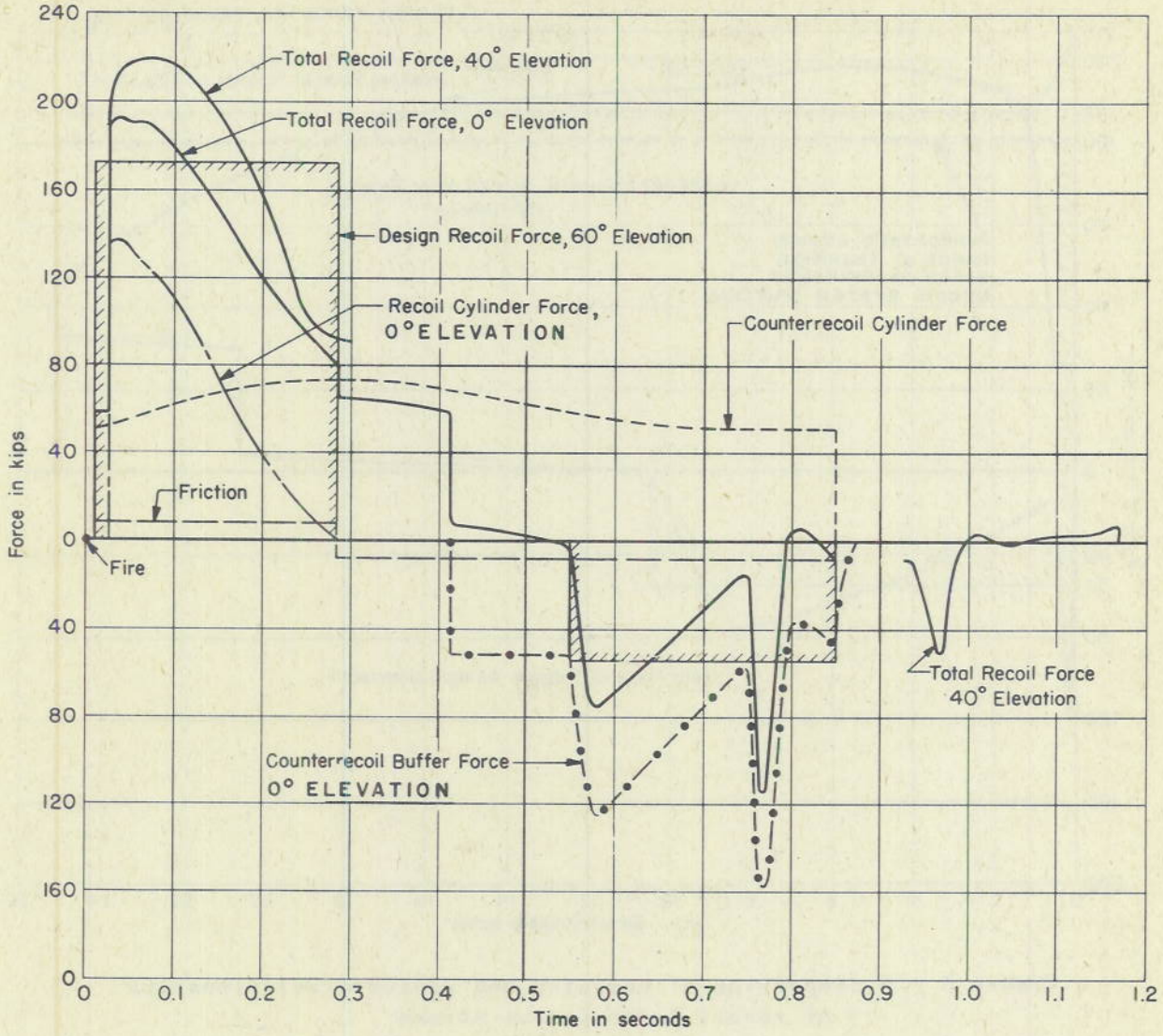

The total recoil force comprises recoil cylinder, counterrecoil cylinder, counterrecoil buffer and frictional forces.

Figure 4- A comparison of computed and measured recoil forces for the 8-inch 55 caliber gun at various angles of elevation

by Frankland, Hudson et al, the dynamic factor for a single degree of freedom system depends only on the shape of the exciting pulse and the natural frequency of the structure. Thus the time history of recoil force may be as significant as the maximum value. Theoretically, at least, using the assumed shape of recoil load, the dynamic factor for turret foundations should be almost 2, and the values of 1.3 as measured on several tests are not easily explained except on the basis of effects of flexible supports. However, more recent measurements employing the most reliable techniques now available show that the dynamic

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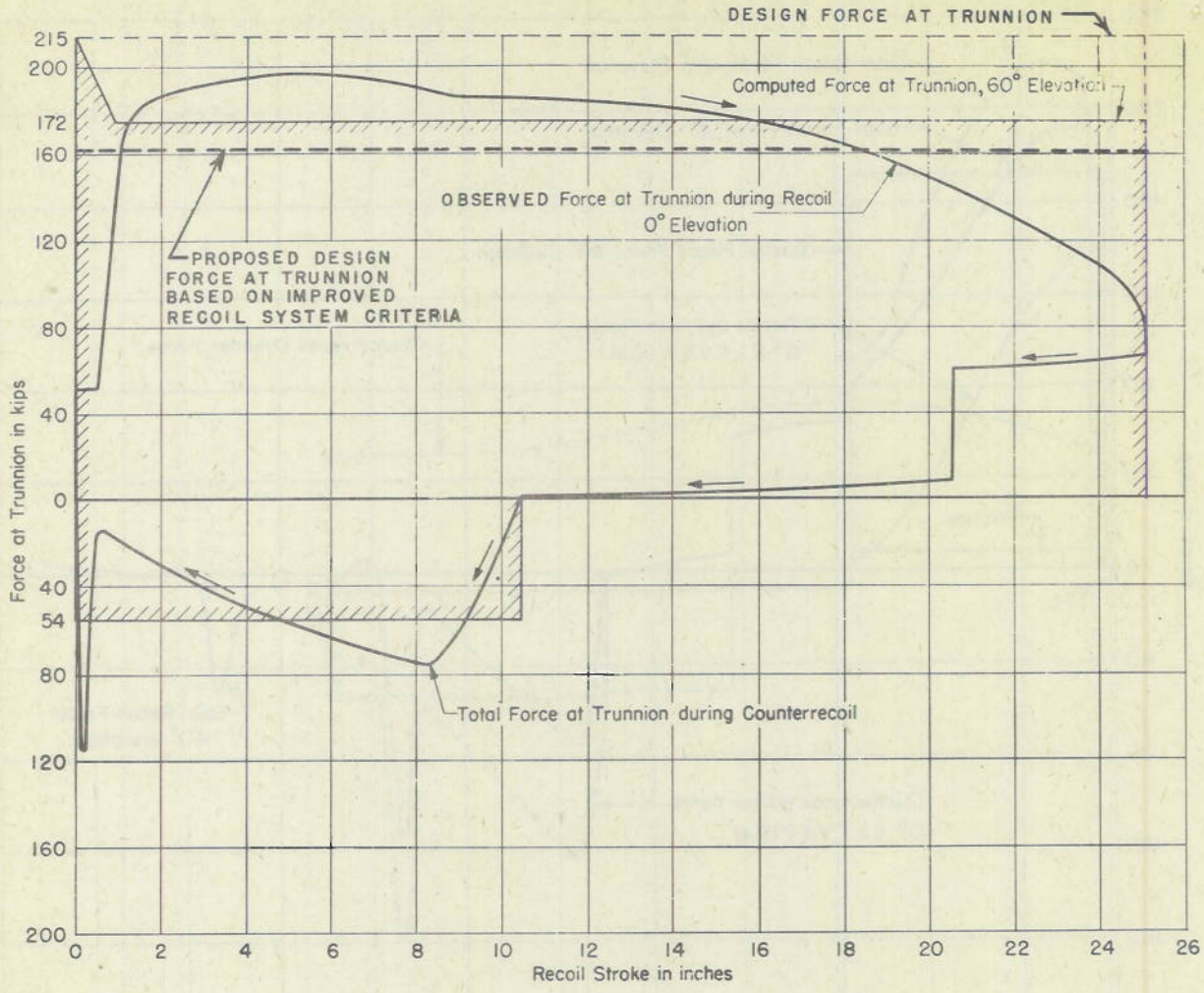

Figure 5 - A comparison of computed and measured relationships of recoil force versus stroke

factor is difficult to establish closer than 20 percent because the observed magnitude of strain and deformation that are developed during gunfire are often so small that the dynamic factor is masked by experimental errors. It is indicated, however, that the dynamic factor ranges from 1.2 to 1.6 and, contrary to previous beliefs, is found as equally affective in the rotating structure as in the circular foundation. As will be pointed out later, other factors in the structural design are more important in their influence on choice of dimensions, so that the importance of assigning an exact value to the dynamic factor could be de-emphasized. In all cases, however, during the computation of force and dynamic factor, the structural designer could best employ the intensity and time-history of recoil force as actually measured for that angle of fire used in design rather than the value postulated by the gun designer for the maximum angle of gunfire. That is, the structure should be custom-built for a specific gun recoil load.

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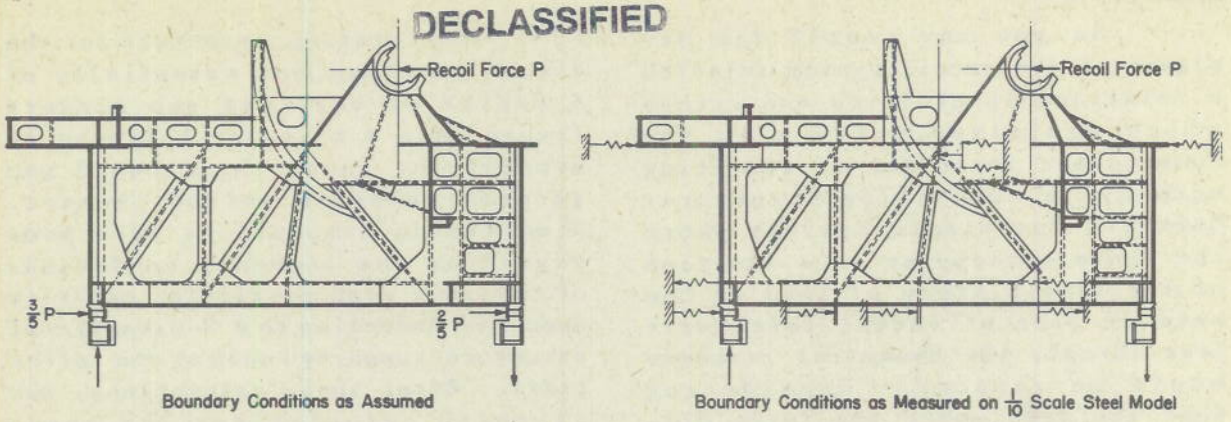

Figure 6 - A comparison of boundary conditions assumed in design of the 8-inch 55 caliber truss, with those conditions actually measured on both a 1/10-scale model and the full-scale pilot turret

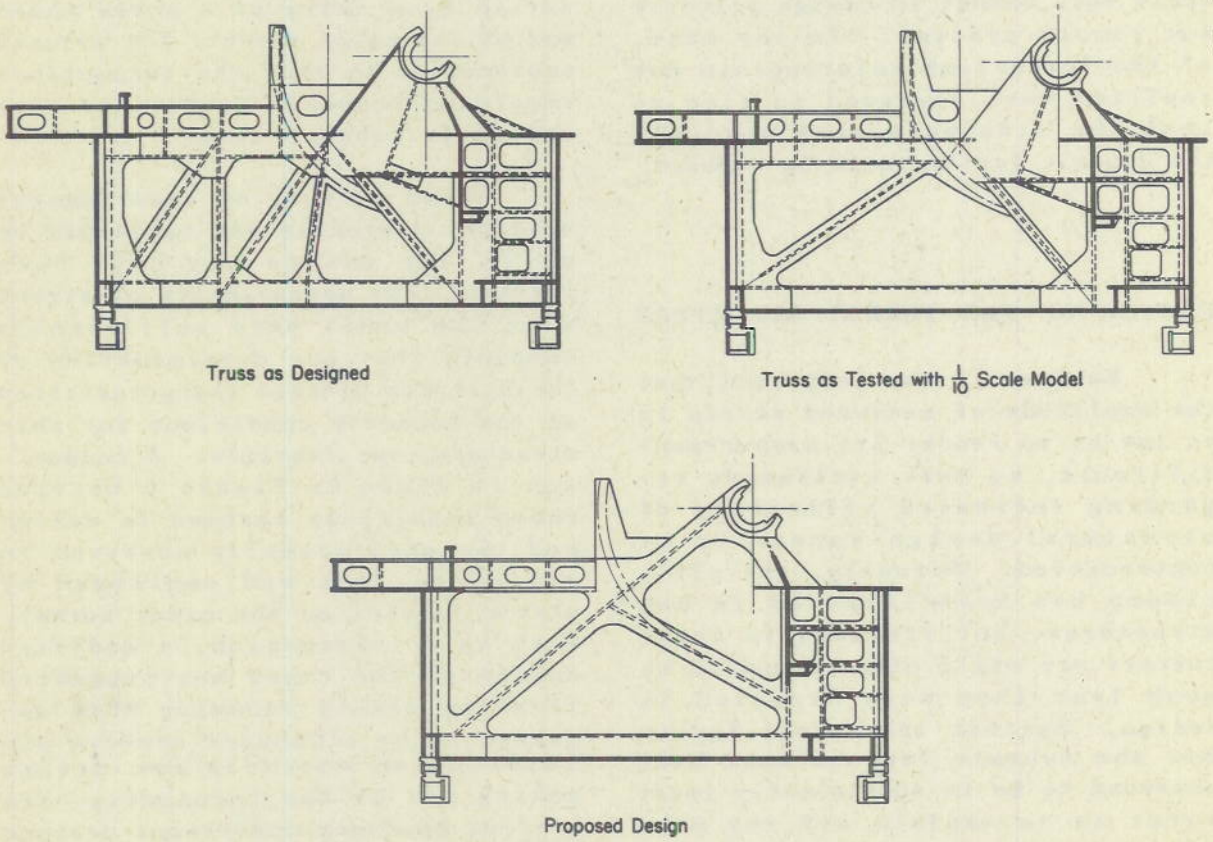

Figure 7 - Rational development of the truss for the 8-inch 55 caliber turret based on experimental determination of boundary conditions

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As you may recall the designer of the recoil system selected a relationship of force and stroke which minimized force with the commendable intention of effecting economy in supporting structure. However, for simple systems where the dynamic response is a function of the time-history of load if the rate of rise of recoil force were less abrupt, the dynamical response would be reduced. Considering that the area under the force displacement curve must be constant, this modification of a rectangular force curve for reduced rate of rise must result in a trapezoid with increased magnitude; but the reduction in the dynamic factor may more than compensate for the increased force, and may make worthwhile this change in design criteria for recoil systems. In any case, if the force-time relationship now realized were reversed in time so that the rise were more gradual, the dynamic factor would be reduced.

DESIGN OF THE TURRET STRUCTURE

Earlier it was indicated that the amplitude of measured strain is so low as to render its measurement difficult, so that statements regarding increased efficiency of structural design appear to be contradicted. Actually, the efficiency has been improved in new structures, but stresses in these turrets are still often found to be much less than were expected in design. Neither the recoil forces nor the dynamic factors have been observed to be in sufficiently great error as to explain all the discrepancies between computed and observed structural response.

To resolve this problem, let us examine the structure in the 8-inch turret.

The rotating structure for the 8-inch guns consists essentially of a series of vertical gun girders framed into a circular bulkhead to support the guns; the inboard gun girders comprise welded trusses. Simplifying assumptions were made regarding the boundary conditions of the truss with particular emphasis upon its reduction to a 2-dimensional structure supported only at the roller tract. After these assumptions, the strength calculations for these trusses were made as though a static load were applied transversely to the trunnion having the same magnitude as the design recoil force for 60 degrees (see Figure 6). The computed stress and mode of stress distribution however was found to differ considerably from that observed from tests of a scale model and of the pilot turret. The primary explanation is that the assumptions regarding boundary conditions were extremely conservative.

From a study of experimental results, a process was suggested by which the design may have been improved. The measurements of stress with the model were sufficiently complete that the determination of the discrete elastic characteristics of the boundary conditions for this structure, was feasible. A comparison is shown in Figure 6 between those conditions assumed in design and the ones actually observed in the model test and confirmed by static loading on the pilot turret. With this information, a modified design of the truss was suggested first by simply removing what appeared to be redundant members and later by an entirely new design making use of the information regarding boundary conditions derived from the model tests. A comparison of these three designs is shown in Figure 7 and it is immediately apparent that a saving in the material would have resulted from a design approach of this type.

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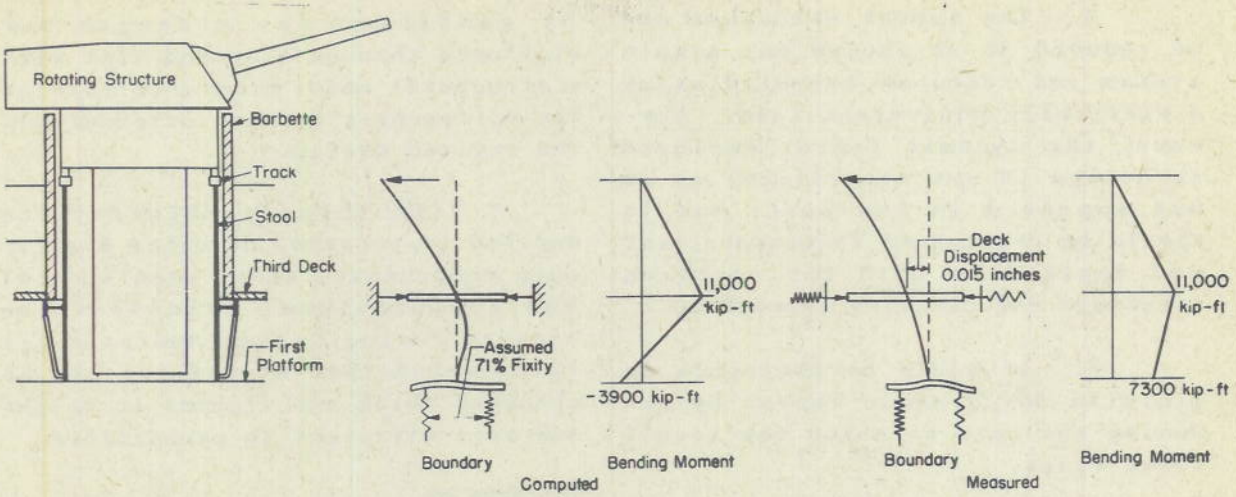

Figure 8 - A comparison of boundary conditions assumed in design of the 6-inch 47 caliber circular foundation with those conditions actually measured on a 1/8-scale model

The importance of boundary conditions applies also to the design of the circular foundation. With the WORCESTER Class of ships, the computed elastic behavior was checked by model tests and it was found that the assumption in design that the decks remain stationary relative to each other, produced serious discrepancies between computed and observed behavior. The comparison with the different boundary conditions is shown in Figure 8. In all cases, however, the stresses were relatively low because of the thickness of material required in the foundation for splinter protection; the strength calculations may thus become only a check rather than a controlling factor.

As refinements appear in strength calculations the designer must take increasing precautions to insure adequate stiffness of structure, for excessive flexibility may impair accuracy of gunfire or increases wear between mechanisms located on structure perturbed by gunfire loading. The criteria for stiffness have in the past been arbitrarily established from experience; and may be subject to further refinement. Removal of material not required for strength may not seriously decrease rigidity, and this process may be checked by static tests of models.

SUMMARY

1. Strength Calculations.- The exact magnitude of force for the design angle of gunfire should always be determined in advance of design by the measurement of the recoil force on a sample gun. By this procedure uncertainty as to the exact magnitude and time history of recoil force and need for a factor of safety for load could be removed. Considering ship roll, the design load at a trunnion should be the recoil load at maximum roll plus the component of the weight of the gun corresponding to that angle.

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2. The turret structure can be reduced to an equivalent simple system and response computed using a statically equivalent load. However, the dynamic factor employed in design is not as critical as it has appeared in the past, but it should be determined from experiment and applied to both the rotating structure and circular foundation.

3. It would be desirable to minimize the dynamic factor by reducing the rate at which the recoil force rises.

4. The last and most emphatic of all, is the proposal that the characteristics of the structural boundary conditions be determined so that the designer may effect great economy in proportioning. It is suggested that these boundary conditions may best be determined by tests with structural models; a preliminary design using any conservative set of boundary conditions should serve as the basis for construction of a model which includes not only elements of structure under study but also those elements of the ship which support that structure. Experimental measurements with this model could be analyzed to determine the boundary conditions and with this information the final structural design can be crystallized. The final design could be confirmed for strength and stiffness through a second test with a structural model which incorporates the differences between original and the revised design.

With these considerations applied to turrets, or other similar ship components, where experimental and computational processes are blended during design analysis, it is believed that structures can be obtained which are lighter in weight and more efficient in proportions.

REFERENCES

Summaries of pertinent criteria heretofor employed in design are given in these references. They, in turn, contain extensive bibliographies.

(1) McGoldrick, R.T., "Model Tests of Turrets and Foundations and their Influence on Full-Scale Design." TMB Conf Rpt 498, Nov. 1943.

(2) Ferris, L. W., "Structural Design of Turrets and Gun Foundation." BuShips 447 Battery Conf Memo 34, Jan. 1945.

(3) Wenk, E. Jr., "The Dynamical Response of the Rotating Structure of Turrets with Particular Reference to the 8-inch 55-caliber Size." TMB Restricted Rpt. C-81.

DISCUSSION

Hoppmann:

I gather from what Mr. Wenk says that present design procedure is based solely on recoil forces of the gun. Is it possible under shock conditions with enemy attack that your firing power would be annihilated?

E. Wenks; TMB:

Yes, turret structures may be rendered inoperative or even destroyed by an enemy attack, and it is

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not believed practicable to design for such severe conditions. However, to insure continued operation after near hits, or underwater explosion attack, it is believed that the structural designer should consider inertia effects of dead weights, perhaps by introducing as additional external loading multiples of the gravity forces on the order of five. Further, even though the recoil force is now the principle loading considered in turret design, a check should be made of effects on structures of ship motion in a seaway, of vibrations, and particularly of large angular accelerations generated by high speed training drive.

N. H. Jasper, TMB:

Shouldn't the boundary conditions be determined from a full scale test on actual ships with similar turrets rather than from a model test?

Wenk:

The most important application of this proposed design process occurs before you have a ship to test; that is, with newly conceived ship designs. I believe that one of the more important applications of experimental analysis is with an uncrystallized design.

However, you do suggest one important question as to the validity of the process. The agreement between model and prototype behavior of the CA139 Class turret rotating structure has recently been investigated by elaborate static tests on a 1/10 scale model, by static and gunfire tests of a full-scale pilot model, and finally by shipboard firing trials. Good correlation of results could be obtained at every phase, and boundary conditions observed with the model were confirmed. The strains measured on shipboard were approximately half the intensity computed by the designer who lacked information concerning boundary conditions, but shipboard measurements agreed better than 20 percent with predictions from model tests made four years earlier. From these results, both the structural and ordnance designer should derive comfort as to the validity of the process.

L. W. Ferris, BuShips:

Early tests of structural models were conducted with models built to 1/3 or 1/4 scale, whereas, the ones you described were smaller. What are the relative advantages of very small models?

Wenk:

The selection of scale of structural models is always a compromise.

Small size is desired because of the economy in time and effort devoted to fabrication. Small size is also desired so that it will be possible to apply scaled-down loads in the laboratory that are within capacity of available loading devices. Generally, the intensity of these scaled-down loads are raised to 5 or 10 times the design value so that the load carrying capacity and mode of failure of the model may be determined. These intensities would be difficult to achieve with large models.

There is a lower limit on size of model resulting from minimum thicknesses of steel which can be satisfactorily welded. With brazing,

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this minimum, is 0.025 inches. Unless geometric similitude is sacrificed, this represents the scaled-down thickness of the thinnest plate in the prototype, and the largest scale ratio derived accordingly.

Caution must be exercised when using the minimum thicknesses because irregularities which inadvertently occur during fabrication may alter the mode of failure by precipitating premature buckling. Model data would of course, be on the side of safety. Except for this latter effect, the scale has no important influence on similitude of elastic behavior of model and prototype, both with regard to stress distribution and elastic stability.

T.L. SooHoo, BuShips:

What is the effect of the roller track on the dynamic response of the structure?

Wenk:

The roller track is an obvious discontinuity of structure that would appear as a nonlinear link. However, all recent experimental observations indicate that the two elements of structure above and below the track are displaced under dynamic load as though the two components were locked together, and oscillations occur as with a monolithic structure.

Further, the time history of all displacements and strain in both rotating structure and stool, show that maximums are achieved simultaneously, and that with these structures, with gunfire loading, the system approximates one with a single degree of freedom. The stress distribution in the circular foundation, however, is compatible with the discontinuous boundary at the track. Structural dynamic stresses suggestive of multiple impact phenomena are observed in the vicinity of the track, but not elsewhere in the foundation and rotating structure.

SooHoo, BuShips:

If model tests are made as suggested, how much more time would be required to complete a design, assuming that a reasonable priority could be given to the tests?

Wenk:

I can understand that your question is of fundamental importance to the designers who are always furnished with deadlines for completion of computations. Speaking on a basis of past performance, in 1944 when everybody competed for high priorities, we were able to have the model of the CA139 Class rotating structure constructed and tested, and results analyzed to the point of recommendations to designers in a period of four months. Six months were required for the test of the CL144 turret stool model, but there elaborate new test facilities also had to be provided. Both of these models were the most complex to be built up to that time so that many fabrication difficulties had to be overcome. With facilities available, and with sufficient priorities, models on the order of complexity of the turrets should be constructed, tested and results analyzed in four months.

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ATTITUDES IN THE DESIGN OF ELECTRONIC EQUIPMENTS

By

Norman E. Lee, Coles Signal Laboratory

A brief summary of replies to a questionnaire on electronic components is presented. Shock and vibration environmental conditions related to design specifications are discussed.

This is a discussion of the answers to a questionnaire sent out by the Subpanel on Shock and Vibration of the Panel on Components, of the Committee on Electronics, RDB.

It was the function of this Subpanel to deal with the problems of shock and vibration as related to electronic components.

The opinions expressed are essentially a personal expression since the rest of the committee has not yet had the opportunity of going over the returns to the questionnaire.

I might say a few words on what motivated the committee to undertake this questionnaire. It was generally appreciated that research on shock and vibration had made tremendous strides during the war but there was not a commensurate reflection of this work in the governmental agencies requirements on components and on equipments. Since much of the general purpose component development had been directed toward better electrical performance, smaller size and less weight, there was a lack of organized design information, and little evaluation of the capabilities of standard components with respect to shock and vibration. There was an obvious need for a better correlation between component and equipment specifications. While equipment specifications had generally been made more exhaustive, component specifications still had little in common with them. It was recognized that in the press for procurement in the last war, much design and acceptance was done on a trial and error basis. It was the hope of this committee that we could amass enough information and recommend more adequate standards so that electronic equipment design could be processed in a more rational fashion. The Armed Forces consolidation with its interchange of responsibility and mutual use of equipments in itself necessitated a coordination of service requirements. Lastly, there was some concern over the fact that while there were many scattered programs for improving general purpose components towards shock and vibration resistance, they showed little coordination with each other or with equipment specifications.

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I have made an arbitrary distinction between general purpose equipments and special purpose equipments. General purpose equipments here are communication sets, radar sets, direction finders and the host of electronic devices which may be used in many installations and generally have commercial type components. Special purpose equipments are the electronic devices used in mines, torpedos, fuses, and guided missiles in which the electronic equipment is designed specially for the particular application.

Since the action of the committee was to be on a broad basis, it was decided to write a questionnaire to those people designing and installing electronic equipment, who might be concerned with shock and vibration as an important environment. The membership of the committee was representative and it has a good deal of material directly available. The questionnaire was to supplement this information and to get a survey of attitudes and practices. Aside from the specific information requested additional or correlative information was solicited. Governmental agencies were fairly well covered and commercial firms and universities were sampled.

The response to the questionnaire is informative in itself. The total number of letters mailed was 166. Because in a sense we were "fishing" there were many parallel letters sent to individuals in the same organization. On receipt and consolidation of the answers, it appeared that there had been 90 valid inquiries to governmental agencies and 25 to commercial firms and universities. There were approximately 12 letters from government agencies which contained detailed answers to all the questions. The commercial agencies submitted two informative answers which showed identity in attitude. Most all of the answers evidenced a desire to be as helpful as was possible, within the scope of the background of the organization. Considerable interest was expressed in the program of the Subpanel. The consolidated total of answers was about 39.

Quantitatively the letters showed that in general the electronic engineer does not concern himself with shock and vibration, and seems exclusively occupied with electrical characteristics and performance. Information on shock and vibration is localized in special groups in those agencies where shock and vibration is an obviously important consideration. Equipment designing seems to have become departmentalized. It does not appear, however, that there are sufficient groups in shock and vibration.

The questions were as follows:

A. What is the maximum value of shock and vibration to which electronic equipment is subjected under service conditions? (The source of the information was requested)

Data was submitted on ballistic impact and gun recoil, rough handling, water impact, projectile acceleration, ship vibration, vehicle vibration, rocket acceleration, parachute opening acceleration, airplane vibration, etc. Most of the data was properly qualified as to circumstances, applicability, and accuracy. The range of stimuli applied to general purpose equipments was quite interesting. Shocks ranged from 40 G for one second to 1000 G at 1 millisecond. Vibratory excitation extended roughly from 4 G at 2 cycles per second

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to 10 G at 200 cycles per second. These were not isolated extremes. There was much data filling the spectrum between these limits and some even beyond. Accelerations and vibrations for special purpose devices had an enormous spread. Shocks for example ranged from 5 G to 50,000 G and the necessity for specially designed components was evident. The most explicit information came from those agencies which dealt with mines, depth charges and the like, as might be expected, because of the nature of these devices. The information received from people whose electronic devices were used in the new carriers such as guided missiles and rockets, was also quite explicit but limited probably due to the newness of the field and difficulties in instrumentation. The data reported on general purpose conveyances for electronic equipment such as ships, aircraft, and tanks did give specific values for specific types of carriers. Whether this information is representative and can be extended to other types will be investigated. One thing which was evident was the need for an extension of the information on prosaic services such as rough handling and commercial transportation. For a large class of equipments it is likely that this is the maximum condition they will have to withstand, and practically all equipments endure this. It is general knowledge that much equipment, while satisfactory in field use, showed a large number of failures on being transported to the field in the last war.

B. What do you consider average service conditions of shock and vibration to which electronic equipments are subjected under field operations?

The general consensus was that for the military services average conditions were without much significance, as might be expected. Information was received on some average measurements on aircraft and vehicles, but it appeared that it was submitted solely to answer the inquiry; little design use being made of average values. This uniform attitude is important specification-wise.

C. Suppose you had to design a component for some service condition which you have already described. What would you use for design conditions, maximum value or average values?

The military services, as indicated in the previous question, quite generally desired to design for maximum values. Mention was made of exceptions where the components available were not capable of resisting the maximum shock, and designs had to be accepted and used with knowledge that a certain percentage of failures will occur. There was also the statement that where safety is involved, design values considerably in excess of the maximum were used.

D. What type of instruments do you employ in recording shock and vibration values? What are the instrument's overall characteristics?

A wide variety of instrumentation was reported here. There was little uniformity since the people who reported seem quite sophisticated in their respective field, and had evolved instruments specially suited for their problems. This information will be matched against Question A (maximum conditions) to see how the data must be qualified because of instrument characteristics. It

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was of interest to note how many laboratories had obtained equipment with a frequency response up to 1000 cycles per second.

E. How do you simulate or synthesize field conditions in your Laboratory or in your test requirements?

Considered as a whole there is an impressive number of types of testing machines in this country. The range of stimuli which can be applied is enormous. Because of this variety the possibility of selecting testing equipment for standardization appears good. For those equipments that had unusual environments, the government agencies had devised many tests for duplicating these conditions. The general purpose equipment people primarily used vibration tests, and in a few instances drop and shock tests. Referring to the data reported in A (maximum conditions), the testing devices showed serious limitations. For example: Very little of the vibration test equipment extended high enough in frequency. There were few requirements specifically designed to account for transportation conditions. As a general proposition it appears that the testing art has not yet caught up with the data on field conditions. Only one or two agencies appear to be adequate in this respect.

F. Is it felt that these tests constitute an adequate examination of this equipment so far as it is satisfactory and reliable in the field? If not, to what extent should field conditions be duplicated?

Of all the questions asked, this particular one received the most qualified answers. Apparently data on performance of equipment in the field is limited or unavailable and there is a good deal of uncertainty as to whether the tests assure satisfactory equipment for field use. It was of interest to note that the two comments by commercial agencies both stated that they felt that field conditions had not been well reproduced in tests because of their excessive severity. They both pointed out that this compromises designs. The people dealing with special purpose equipments listed several phenomena for which satisfactory tests had not yet been developed. The answers to the question from general purpose equipment people showed a little unhappiness with their tests.

G. In the design of equipment do you evaluate and specify shock and vibration environmental factors or is the burden placed on the contractor for a set design?

In general the military forces stated that they set up shock and vibration tests which the contractor is required to pass. The purpose of the question was to determine how many agencies were buying equipment according to definite tests since some years ago much equipment was bought with rather vague proviso's in the specifications.

H. Do you anticipate the need for modifications in existing component requirements in view of the research and development program?

It was generally agreed on this question from almost all parties that it would be both desirable and in many cases necessary to evolve new components with better properties towards shock and vibration. Factors such as reliability, microphonics, and new equipment uses, were all given as justification.

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There was almost complete uniformity on this question.

I. Is it necessary in your activity to develop special components with regard to better shock and vibration characteristics, and if so, what work is in progress, or what is considered desirable in this respect?

There was some mention of specific work underway on relatively few general purpose components. There was not the widespread effort that the answers to the previous question indicated as necessary.

J. So far as the use of present available components which of those are considered most satisfactory and what seems to be the failure mechanism?

Although the previous question which inquired into present developing of special components brought forth a few explicit answers, this particular question gave many articulate detailed answers, showing a fairly consistent opinion. There were large numbers of unsatisfactory components relative to shock and vibration characteristics, particularly tubes, relays, resistors, capacitors, indicating instruments, fasteners and component brackets, shock mounts and shock mounting systems.

Some of the generalizations made from the answers to the questionnaire are naturally based on the omissions in the information submitted rather than on a literal interpretation of the information submitted.

It is quite evident that the knowledge in shock and vibration has not spread from those engaged in research in this field. There is not enough reflection in specifications and designing of the latest thinking and data on shock and vibration. This in particular applies to specification requirements and the scarcity of design data on electronic components.

In the field of general purpose electronic equipment there is not enough correlation between component and equipment requirements, but the answers to the questionnaire showed that there is enough information available to improve this situation now. This sometimes gives rise to difficulties in determination of responsibility in procurement. For example: A contractor whose specification calls for passing certain equipment tests, and yet is bound to use JAN components may find these two requirements impossible to achieve.

Fundamentally it appears that people occupied in the design of general purpose electronic equipment are mainly concerned with performance and consider shock and vibration resistance, which is important in terms of reliability, as a secondary characteristic. This is an unfortunate circumstance since the adequacy of a piece of equipment in the field just as much depends on mechanical characteristics as on electrical performance. In the actual design of electronic equipment, electrical and mechanical problems are most intimately linked with one another. It is extremely difficult to take a design based on purely electrical considerations and without seriously compromising it, make it satisfactory mechanically. It is almost an absolute necessity for efficient overall design that the electronic engineer have more familiarity with shock and vibration work.

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It appears that improvement in the design of electronic equipment lies in two avenues:

(1) Make better use of the information available. What is particularly required here is to circulate the data and the appropriate design techniques to the equipment designers, and to make better specifications for both equipments and components.

(2) Design of components with better shock and vibration possibilities for general purpose equipment. The tenor of the answers in this questionnaire was quite strong on the point that if adequate components, in this sense, could be obtained, the savings in weight and size, the elimination of shock mounts and mounting complexities would result in designs that are much more satisfactory. While it may be argued that in research work on new components incorporating these attributes would be an unnecessary constraint on improving electrical performance, the experience in the last war has indicated how difficult a problem it is to subsequently try to tack new attributes on an essentially designed piece of equipment.

In general those people occupied with special purpose electronic equipments are in a straightforward position so far as rational shock and vibration work is concerned. The environments these devisers must work in, forces an appreciation of the mechanical factors involved, and the engineers in the field seem to have a coherent program for the collection of data, testing, and the evolution of these electronic devices. In the field of general purpose equipments the situation is not as clear cut since for normal use shock and vibration is reflected usually not in immediate complete inoperableness, but often subtle erratic degradation in performance. The need for better practices and new components is therefore not as easy to justify, particularly since field reports give so little organized information.

While the development of components for special purpose equipments is a problem already underway, there is information enough available at present for the start on development of more satisfactory components for general purpose equipments and the evolution of better specifications for the equipments themselves.

In philosophical terms, the importance of this kind of shock and vibration work is as follows: It is almost unanimously held in light of the developments in the latter part of the last war and in these intervening years that if there is another conflict, it would be quite different from the last one in this sense: The conflict will be both destructive and short; instead of taking years it may take days or weeks. It is for this reason that our military equipment must be reliable in the highest degree. There will be no time to manufacture or repair equipment; that which we have on hand must function satisfactorily. Since shock and vibration are environmental factors which are most important in determining the reliability of electronic equipment, it therefore is of the utmost importance that mechanical design of our equipment be the best that can be obtained.

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VIBRATION TESTING AS A GUIDE TO EQUIPMENT DESIGN FOR AIRCRAFT

By

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One of the greatest single factors which made possible the modern aircraft reciprocating engine, with all its refinement and dependability, was the development of vibration isolating systems in the engine supports. Certainly, aircraft engines in current use would not be practical without their vibration isolating suspensions. Effective as these isolating systems are, nevertheless, considerable vibration is transmitted into the aircraft structure, much to the annoyance of aircraft equipment designers. Also, there are many other sources of vibration which cannot be controlled by the power plant mountings, such as, air-borne vibrations induced by the propeller, air-flow disturbances over various surfaces of the airplane, machine gun firing, flight through rough air, taxiing over rough ground, etc. The particular aircraft structure responding in its various modes to the existing sources of vibration, determines the nature, severity, and extent of vibration at any particular point in the airplane under any given set of operational conditions.

The damaging effect of vibration in aircraft structure forced designers into early efforts to insure proper operation and adequate service life for equipment which must be attached to such structure. It was too costly, time consuming, and dangerous to depend upon accumulation of service hours for proving equipment designs. Therefore, it was imperative to have methods for approving and accepting aircraft equipment from a vibration standpoint by means of laboratory testing. Consequently, an improvised laboratory vibration testing procedure was adopted which essentially consisted of uniformly varying the applied frequency of vibration from 10 to 55 and back to 10 cycles per second in one-minute intervals with a constant applied double amplitude of 0.030 or 0.060 inch depending upon the discretion of the specification writer.

It is now common knowledge that the actual frequencies of vibration in flight are not encompassed by the vibration testing method just described. In 1945, the Equipment Laboratory, Air Materiel Command, recognizing after considerable study the need for more rational and modernized vibration testing procedures, set up vibration testing requirements based upon actual service conditions. These were included in AF Specification 41065 along with testing requirements for other environmental conditions, such as heat, cold, fungus, humidity,

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salt fog, sand and dust, etc. Since that time the vibration testing requirements have been revised and improved somewhat but the basic testing methods in essence have remained unchanged.

The purpose of setting up vibration testing procedures to be included in AF Specification 41065 was twofold: First, to provide standard vibration testing procedures based upon service conditions which would apply universally to the development of the many and diverse items of equipment for which the Equipment Laboratory is responsible. There are over one thousand of these items most of which are involved in the following general categories: Electrical power systems, circuits, lighting, and electrical controls for aircraft; automatic control and guidance systems for aircraft, targets, and missiles; aircraft instruments and navigational equipment; fire detecting and extinguishing systems; anti-icing systems; aerial pickup and delivery systems; automatic parachute and escape capsule equipment, to mention only a few.

Secondly, the specification was intended to facilitate standardization within the Equipment Laboratory for items developed for interchangeability in many types of aircraft and various locations. There is rarely freedom of choice for locating equipment and appliances in aircraft. Location usually is dictated by functional requirements, space limitations, and system proximity requirements. It was necessary, therefore, for the testing specification to be very general and in accordance with accepted aircraft design standards, yet not overconservative to the extent that undue weight or space penalties would be imposed upon the equipment design.

Considerable data were compiled on the vibration frequencies and amplitudes measured on many types of airplanes under conditions of operation to establish a basis for laboratory testing procedures. Most of the data were obtained from flight vibration tests conducted by the Aircraft Laboratory, Air Materiel Command. The data were grouped into three elementary categories: aircraft structural vibration, reciprocating engine vibration, and vibration of elastically mounted assemblies. Points of maximum amplitude were plotted on amplitude versus frequency graphs and envelopes encompassing the points were established. A few points based upon more recent data lie outside the original envelopes. However, these points do not appear to be sufficiently representative of typical service conditions to warrant changing the original envelopes at this time. These envelopes were considered to be typical of the most severe conditions of vibration likely to be encountered in each of the three elementary categories for any aircraft powered by reciprocating engines.

Figure 1 shows the envelope and corroborating data for aircraft structural vibration. The points plotted in Figure 1 are identified in Table 1. In the range of 5 to 10 cycles per second, the double amplitude of the envelope was established at 0.050 inch to take account of vibration induced by propeller unbalances and other possible low frequency disturbances. The double amplitude was reduced to 0.036 inch from 10 to 75 cycles per second and from 75 to 300 cycles per second, a constant vibratory acceleration of ± 10g was maintained. From 300 to 500 cycles per second, the acceleration was increased and held constant at ± 15g.

The envelope for reciprocating engine vibration was not so easily

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established. Most of the engine vibration measurements were made with paired pickups fed into mixing circuits which separated the translational from the rotational components of vibration. The assumption is made that the engine vibrates as a rigid body. Figure 2 shows the envelope and pertinent data for engine translational components, and Figure 3, the envelope and data for rotational components. The points shown in Figures 2 and 3 are identified in Table 2. The envelope for rotational components was transformed into translational vibration at a 2-foot radius from the power plant center-of-gravity. This transformed translational envelope was then added to the envelope for center-of-gravity translation and the resultant translational envelope shown in Figure 4 for a locus at 2-foot radius, was obtained. The points plotted in Figure 4 and identified in Table 2 represent vibration measurements at particular locations on a power plant in the C-74 airplane. The 2-foot radius arbitrarily was chosen as the average distance of power plant accessories from the center-of-gravity. The resultant envelope entails a double amplitude of 0.100 inch from 5 to 10 cycles per second to accommodate low frequency resonances of the power plant. The double amplitude decreases to 0.060 inch from 10 to 60 cycles per second. Above 60 cycles per second the following values of constant vibratory accelerations are maintained: ± 11g from 60 to 75 cycles per second; ± 15g from 75 to 100 cycles per second; and ± 20g from 100 to 500 cycles per second.

Figure 5 shows the envelope for components of elastically mounted assemblies and the corroborating data are identified in Table 3. A large double amplitude of 0.300 inch is maintained from 5 to 10 cycles per second to cover the high amplitude, low frequency transient resonant conditions characteristic of elastically mounted assemblies. From 10 to 55 cycles per second the relatively mild vibratory acceleration of ± 1.5g is maintained in anticipation of the vibration isolating effect of the elastic mounts.

The next problem consisted of evolving a test procedure based upon the data envelopes. It was considered that a logical approach to the problem might be patterned after the accepted approach to structural design problems. That is, the greatest possible loads under any possible service condition are determined and then the design is developed to accommodate these loads (with appropriate safety factors) no matter how remote the possibility of these loads ever being realized. In reference to aircraft equipment, the most severe possible vibration applied at critical or resonant conditions of the test specimen would appear to induce the greatest possible stresses and deterioration in the specimen. However, these conditions, if met in service, would be unlikely to occur for any great length of time under any operational sequence. Hence, an endurance test at these critical conditions with the most severe vibration likely to be encountered in service appears to represent an acceleration of service time.

In the light of these considerations, the procedures outlined in AF Specification No. 41065 were established. Three vibration testing methods were included which correspond to the three data envelopes. Method 61, applies to equipment which attaches to aircraft structure (such as motors, actuators, inverters, lighting assemblies, fire detectors); Method 62, equipment which attaches

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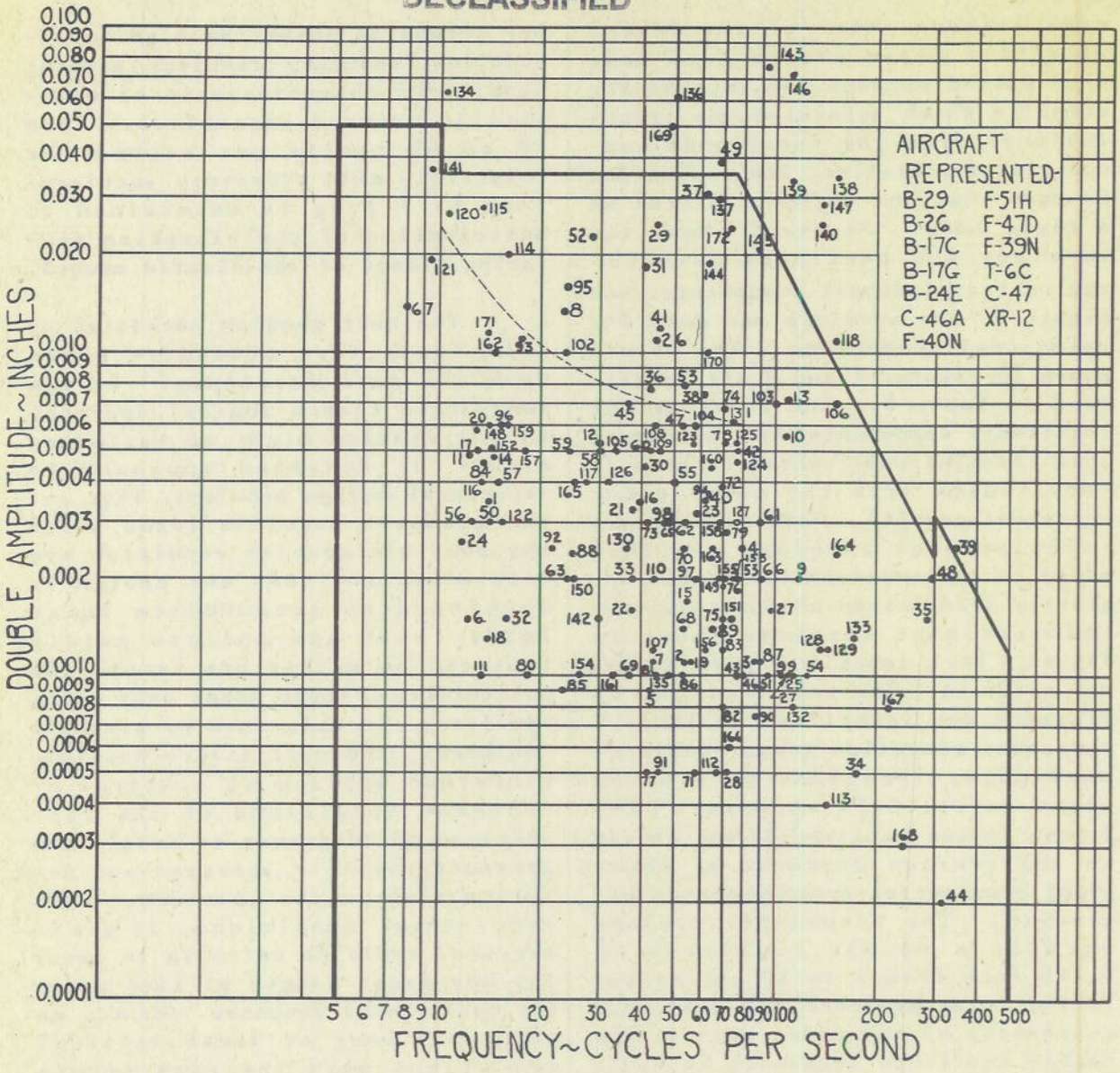

Figure 1 - Vibration characteristics of aircraft powered by reciprocating power plants

to reciprocating engines (generators, pressure and temperature transmitters, starters); and Method 63, equipment which as minor assemblies attach to vibration isolated major assemblies (instruments, items mounted on equipment racks, components of vibration isolated amplifiers). The data envelopes for all these methods are plotted in Figure 6 for purposes of comparison. The procedure for each method follows the same general pattern which essentially consists of surveying for resonance in the frequency range of 5 to 500 cycles per second for structural or engine mounted equipment, and in the range of 5 to

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Table 1

Identification of Points Plotted in Figure 1
for Structural Vibration of Conventional Aircraft in Flight

No.	Bibliography Reference No.	Aircraft	Location	Direction	Source	Engine Speed RPM	Frequency CPS	Double Amplitude Inches
1	2	B-29	Navigator's Comp't.	Vertical	1XP	2400	14	0.0035
2	2	"	" "	"	4XP	2300	53.7	0.0011
3	2	"	" "	"	2XE	2600	86.7	0.0011
4	2	"	Wing Spar near Airplane c.g.	Lateral	2XE	2400	80	0.0025
5	2	"	"	Vertical	4XP	1800	42	0.0009
6	2	"	"	"	1XP	2100	12.1	0.0015
7	2	"	"	"	1XE	2600	43.3	0.0011
8	2	"	Secondary Structure	Thrustwise	Buffet	-	23.3	0.0135
9	2	"	At Radar Antenna	"	8XP	2500	116	0.002
10	2	"	" " "	Lateral	8XP	2200	105	0.0056
11	2	"	" " "	"	1XP	2100	12.2	0.0049
12	2	"	" " "	Vertical	1XE	1800	30	0.0053
13	2	"	" " "	"	2-1/2 XE	2600	108	0.0072
14	2	"	" " "	"	1XP	2500	14.5	0.0048
15	2	"	Radar Rack	Lateral	4XP	2300	54	0.0017
16	2	"	" "	Vertical	4XP	1700	39.7	0.0035
17	2	"	" "	Vertical	1XP	2200	12.8	0.005
18	2	"	Fuselage Structure at Radar Scope	Lateral	1XP	2400	14.0	0.0013
19	2	"	" " "	"	4XP	2400	56	0.0011
20	2	"	" " "	Vertical	1XP	2200	12.8	0.0058
21	2	"	" " "	"	4XP	1600	37.3	0.0033
22	2	"	Tail Cone	Lateral	4XP	1600	37.3	0.0016
23	2	"	" "	Vertical	4XP	2500	58	0.0032
24	2	"	" "	"	1XP	2000	11.6	0.0026
25	2	"	" "	"	8XP	2400	11.2	0.001
26	3	B-26B	Nacelle Structure	Thrustwise	1XE	2630	44	0.011
27	3	"	" "	"	3-1/2XE	1650	96	0.0016
28	3	"	" "	"	2XE	2120	71	0.0005
29	3	"	" "	Vertical	1XE	2640	44	0.025
30	3	"	" "	"	1-1/2XE	1620	40.5	0.0045
31	3	"	" "	Lateral	1XE	2420	40.3	0.0185
32	3	B-26B	Nacelle Structure	Lateral	1/2XE	1900	15.8	0.0015
33	3	"	" "	"	1-1/2XE	1500	37.5	0.002
34	3	"	" "	"	4-1/2XE	2320	174	0.0005
35	3	"	" "	"	6-1/2XE	2600	282	0.0015
36	3	"	Pilot's Comp't.	Thrustwise	1XE	2540	42.3	0.0075
37	3	"	" "	Lateral	1-1/2XE	2450	61.2	0.031
38	3	"	" "	"	2XE	1820	61	0.0075
39	3	"	" "	"	7-1/2XE	2730	340	0.0025
40	3	"	" "	Vertical	1-1/2XE	2450	61.2	0.0035
41	4	"	Wing Tip	"	1XE	2700	45	0.012
42	4	"	" "	"	2XE	2300	77	0.005
43	4	"	" "	Lateral	2XE	2300	77	0.001
44	4	"	" "	"	7XE	2700	315	0.0002
45	4	"	" "	Thrustwise	1XE	2190	36.5	0.007
46	4	"	" "	"	2-1/2XE	1900	79	0.001
47	4	"	Fin Station	Vertical	1-1/2XE	2100	52.5	0.006
48	4	"	" "	"	6-1/2XE	2700	292	0.002
49	4	"	" "	Lateral	1-1/2 XE	2700	67.5	0.039
50	4	"	" "	Thrustwise	1/2XE	1900	15.8	0.003
51	4	"	" "	"	3XE	1900	95	0.001
52	5	B-17C	Nose at Bombsight Base	"	1XE	1700	28.3	0.023
53	5	"	" " "	"	3XP	1900	53.2	0.008
54	5	"	" " "	"	6XP	2200	124	0.001
55	5	"	" " "	"	2XE	1500	50	0.004
56	5	"	" " "	"	1/2XE	1500	12.5	0.003
57	5	"	" " "	"	1XP	1600	15	0.004
58	5	"	" " "	Vertical	1XE	1800	30	0.005
59	5	"	" " "	"	1XP	2450	23	0.005
60	5	"	" " "	"	3XP	1500	42.2	0.005
61	5	"	" " "	"	6XP	1600	90	0.003
62	5	"	" " "	"	2XE	1600	53	0.003
63	5	"	" " "	Lateral	1XP	2400	22.5	0.002
64	5	"	" " "	"	2XE	1600	53	0.003
65	5	"	" " "	"	3XP	1700	47.8	0.003
66	5	"	" " "	"	6XP	1600	90	0.002

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Table 1 (Cont.)

No.	Bibliography Reference No.	Aircraft	Location	Direction	Source	Engine Speed RPM	Frequency CPS	Double Amplitude Inches
67	6	B-17G	Nose at Bombsight Base	Lateral Transient Freq	-	-	4 - 8	0.014
68	6	"	" " " "	Thrustwise	3XP	1900	53	0.0014
69	6	"	" " " "	"	1XE	2200	36.7	0.001
70	6	"	" " " "	Lateral	3XP	1900	53.3	0.0025
71	6	"	" " " "	Vertical	3XP	2100	59	0.0005
72	7	B-24E	Nose at Interval Control	Vertical	3XP	2400	69	0.0039
73	7	"	" " "	Thrustwise	1XE	2500	41.6	0.0030
74	7	"	" " "	Lateral	3XP	2500	70	0.0068
75	7	"	Fwd. Bomb Rack	"	3XP	2400	69	0.0015
76	7	"	" " "	Vertical	3XP	2400	69	0.0019
77	7	"	Aft Bomb Rack	Thrustwise	1XE	2500	41.6	0.0005
78	7	"	" " "	Lateral	3XP	2500	70	0.0053
79	7	"	" " "	Vertical	3XP	2500	70	0.0028
80	8	C-46A	Pilot's Seat	Lateral	1/2XE	2200	18.3	0.001
81	8	"	" "	"	1XE	2630	44	0.001
82	8	"	" "	"	1-1/2XE	2680	67	0.0008
83	8	"	" "	"	2XE	2100	70	0.0012
84	8	"	" "	Vertical	1/2XE	1620	13.4	0.0046
85	8	"	" "	"	1XE	1400	23.3	0.0009
86	8	"	" "	"	1-1/2XE	2100	52.5	0.001
87	8	"	" "	"	2XE	2680	89	0.0011
88	9	F-40N	Cockpit Floor	Thrustwise	1/2XE	2960	24.7	0.0024
89	9	"	" "	"	1-1/2XE	2600	65	0.0014
90	9	"	" "	"	2-1/2XE	2100	87.5	0.00075
91	9	"	" "	"	1XE	2700	45	0.0005
92	9	"	" "	Lateral	1XP	2600	21.7	0.0025
93	9	"	" "	Vertical	1XP	2100	17.5	0.011
94	9	"	" "	"	3XP	2600	65	0.0035
95	10	P-51H	Head Rest	Thrustwise	1XP	3000	24	0.016
96	10	"	Pilot's Seat	Vertical	1XP	1900	15	0.006
97	10	"	" "	"	1-1/2XE	2300	57.5	0.002
98	10	"	" "	"	1XE	2800	46.6	0.003
99	10	"	" "	"	2-1/2XE	2500	104	0.001
100	10	"	" "	"	4XP	3000	95.5	0.001
101	10	"	" "	"	3XE	1900	95	0.001
102	10	P-51H	Windshield	Lateral	1XP	3000	23.9	0.010
103	10	"	"	"	2-1/2XE	2400	100	0.007
104	10	"	"	"	4XP	1800	30	0.006
105	10	"	"	"	1XE	2000	33.3	0.005
106	10	"	"	"	3XE	3000	150	0.007
107	11	F-47D	Pilot's Seat	Vertical	1XE	2500	41.7	0.0012
108	11	"	" "	Thrustwise	1XE	2600	43.3	0.006
109	11	"	Rudder Bar	"	1XE	2700	45	0.005
110	12	"	Pilot's Seat	"	1XE	2600	43.3	0.002
111	12	"	" "	"	1/2XE	1600	13.3	0.001
112	12	"	" "	"	2-1/2XE	1600	67	0.0005
113	12	"	" "	"	3-1/2XE	2450	143	0.0004
114	13	F-39N	Gear Box	Lateral	1XP	2100	15.8	0.020
115	13	"	" "	"	1XP 1XE	1600	12- 13.3	0.028
116	13	"	Tail Section	"	1/2XE	1640	13.6	0.004
117	13	"	" "	"	1XE	1640	27.3	0.004
118	13	"	" "	"	3XE	3000	150	0.011
119	13	"	" "	Vertical	1/2XE	1800	15	0.004
120	14	B-29	Engine Nacelle	Lateral	1XP	1800	10.4	0.027
121	14	"	" "	Vertical	1XP	1600	9.35	0.0195
122	14	"	" "	Thrustwise	1XP	2600	15.1	0.003
123	15	B-24E	Bombardier's Comp't.	"	3XP	2000	57	0.0053
124	15	"	" "	Lateral	3XP	2740	77	0.0046
125	15	"	" "	Vertical	3XP	2740	77	0.0053
126	16&17	T-6C	Front Cockpit	Lateral	1XE	1900	31.7	0.004
127	16&17	"	" "	"	2XE	2390	80	0.0033
128	16&17	"	" "	"	3-1/2XE	2290	134	0.0012
129	16&17	"	" "	"	4-1/2XE	1920	144	0.0012
130	16&17	"	Rear Cockpit	"	1XE	2160	36	0.0025
131	16&17	"	" "	"	2XE	2220	74	0.006
132	16&17	"	" "	"	3-1/2XE	1930	112	0.0008
133	16&17	"	" "	"	4-1/2XE	2280	171	0.0014
134	18	C-74	Engine Mt. Member	Vertical	1XP	1600	10.2	0.063
135	18	"	" " "	"	1XE	2300	38.3	0.001
136	18	"	" " "	"	2XE	1500	50.0	0.061

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Table 1 (Cont.)

No.	Bibliography Reference No.	Aircraft	Location	Direction	Source	Engine Speed RPM	Frequency CPS	Double Amplitude Inches
137	18	C-74	Engine Mt. Member	Vertical	2-1/2XE	1600	67	0.030
138	18	"	" " "	"	3-1/2XE	2800	163	0.035
139	18	"	" " "	"	4-1/2XE	1500	112	0.034
140	18	"	" " "	"	5-1/2XE	1500	137	0.025
141	18	"	" " "	Lateral	1XP	1500	9.5	0.037
142	18	"	" " "	"	1XE	1800	30	0.0015
143	18	"	" " "	"	2XE	2800	93.5	0.077
144	18	"	" " "	"	2-1/2XE	1500	62.5	0.019
145	18	"	" " "	"	3-1/2XE	1500	87.7	0.021
146	18	"	" " "	"	4-1/2XE	1500	112	0.072
147	18	"	" " "	"	5-1/2XE	1500	137	0.029
148	19	XR-12	Front Wing Spar	Vertical	1XP	2000	14.1	0.006
149	19	"	" " "	"	4XP	2400	68	0.002
150	19	"	" " "	"	1XE	1500	25	0.002
151	19	"	" " "	"	2XE	2200	73	0.0015
152	19	"	Rear Wing Spar	"	1XP	2000	14.1	0.005
153	19	"	" " "	"	4XP	2700	76.5	0.002
154	19	"	" " "	"	1XE	1600	26.6	0.001
155	19	"	" " "	"	2XE	2100	70	0.002
156	19	"	" " "	"	2-1/2XE	1500	62.5	0.0012
157	19	"	Camera Section	"	1XP	2400	17	0.005
158	19	"	" "	"	4XP	2400	68	0.003
159	19	"	" "	Lateral	1XP	2300	16.2	0.006
160	19	"	" "	"	4XP	2300	65	0.0045
161	19	"	" "	"	1XE	2000	33.3	0.001
162	19	"	Pilot's Comp't.	Vertical	1XP	2000	14.2	0.010
163	19	"	" "	"	4XP	2300	65	0.0025
164	19	"	" "	"	8XP	2700	152	0.0024
165	19	"	" "	"	1XE	1500	25	0.004
166	19	"	" "	"	2XE	2200	73	0.0006
167	19	"	" "	Lateral	12XP	2500	213	0.0008
168	19	"	" "	Thrustwise	12XP	2700	230	0.0003
169	20	B-24E	Nose at Interval Cont.	"	Mach. Gun	-	47.3	0.0496
170	20	"	" " " "	Lateral	" "	-	62.5	0.0098
171	20	"	" " " "	"	" "	-	131	0.0115
172	20	"	" " " "	Vertical	" "	-	73	0.0245

55 cycles per second for components of elastically supported assemblies. An endurance test is then conducted at the conditions of resonance with the applied amplitude indicated by the pertinent data envelopes. For the frequency survey and subsequent endurance test linear vibration is directed consecutively along three mutually perpendicular axes of the test specimen.

For a working assumption, the tentative correlation factor between laboratory testing time and service time arbitrarily has been established as approximately one to ten, that is, one hour laboratory testing time is equivalent to approximately ten hours service time. Hence, a typical 12-hour laboratory vibration test represents at least 120 hours of service. In general, this is considered to be conservative.

The following assumptions have been made in formulating AF Specification No. 41065:

1. Equipment will be subjected to the most severe conditions of vibration which have been measured on any airplane in flight.

2. The severity of conducting an endurance test at resonant or critical frequencies constitutes an acceleration of the deteriorating effect of vibration.

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3. In service, the equipment will operate at resonant or critical conditions less than one-tenth of the total flight time, or if there are no critical conditions, operation will entail average magnitudes of vibration only one-third as severe as those required in the laboratory test.

4. Such conditions as icing of propellers, flying through rough air, unsteady airflow, taxiing, and landing will cause vibration having large amplitudes and low frequencies.

5. Frequencies of vibration in aircraft above 500 cycles per second

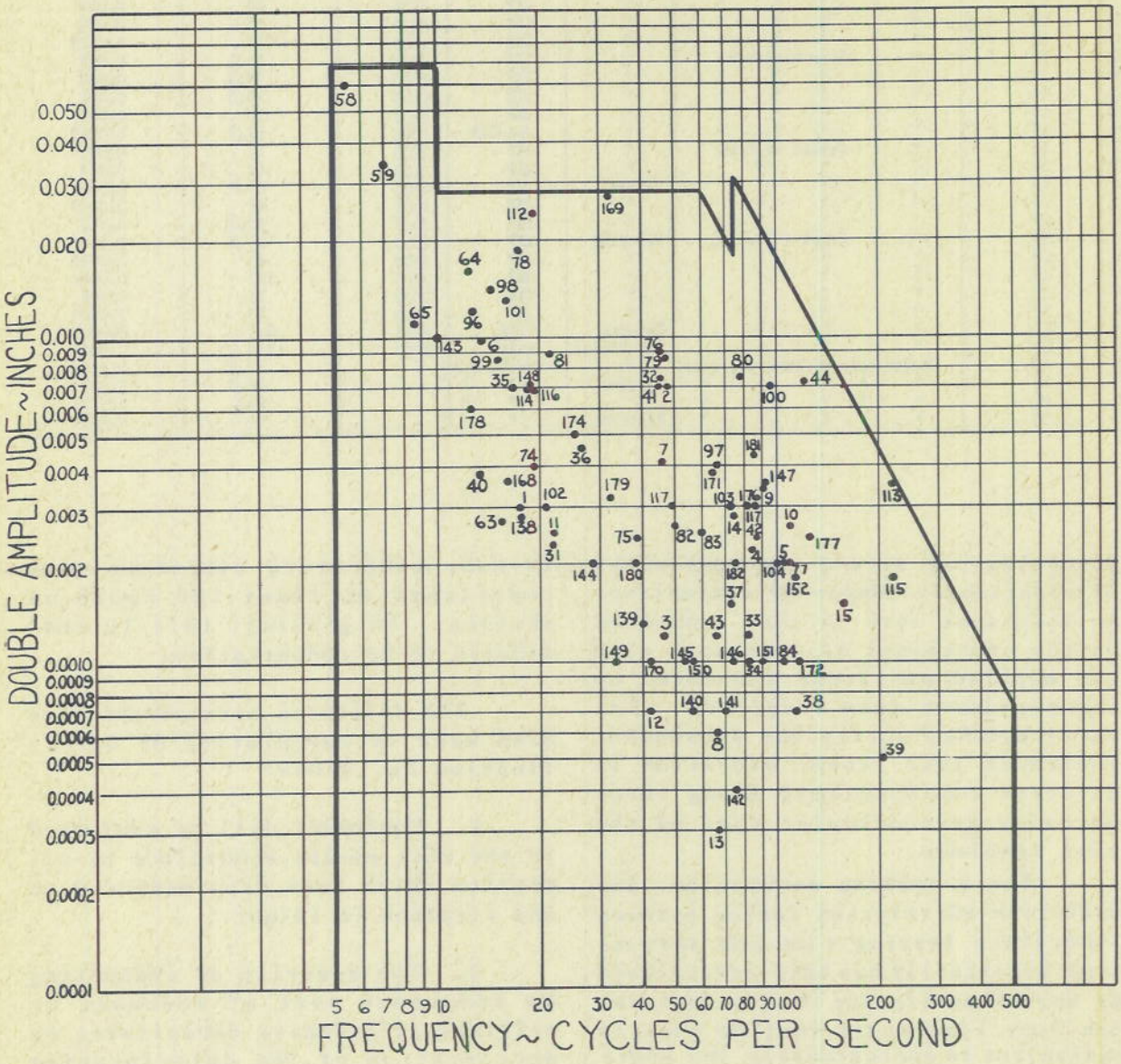

Figure 2 - Vibration characteristics of reciprocating engines-translational components

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have no significant effect upon the endurance and performance of equipment.

AF Specification 41065 now has been in effect for four years and the vibration testing procedures have been applied during this period with encouraging results. The procedures have been very helpful in detecting design weaknesses and in improving immunity of equipment to vibration failure.

In some instances it has been necessary to establish special vibration testing procedures directed toward specific vibration problems.

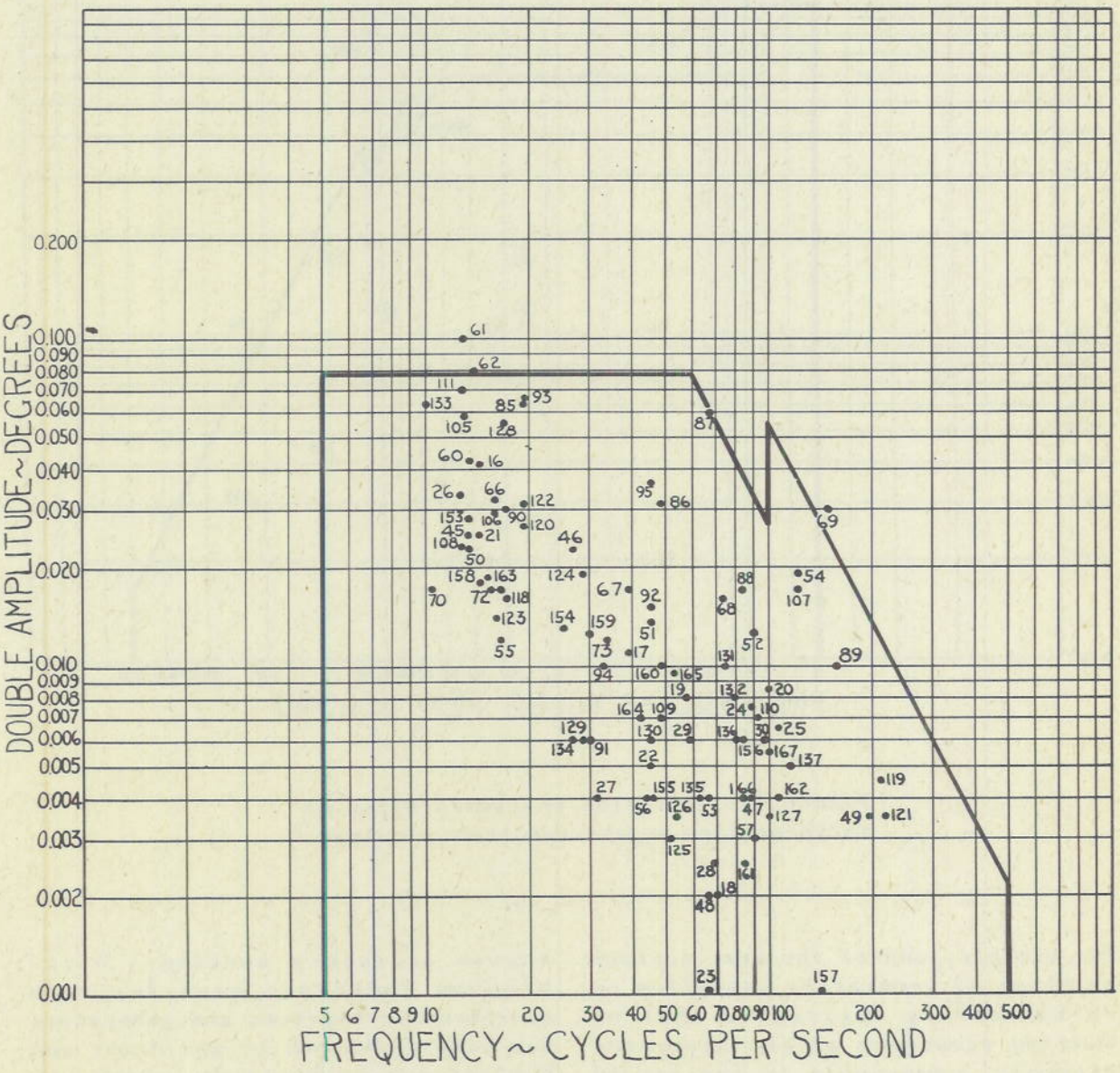

Figure 3 - Vibration characteristics of reciprocating engines-rotational components

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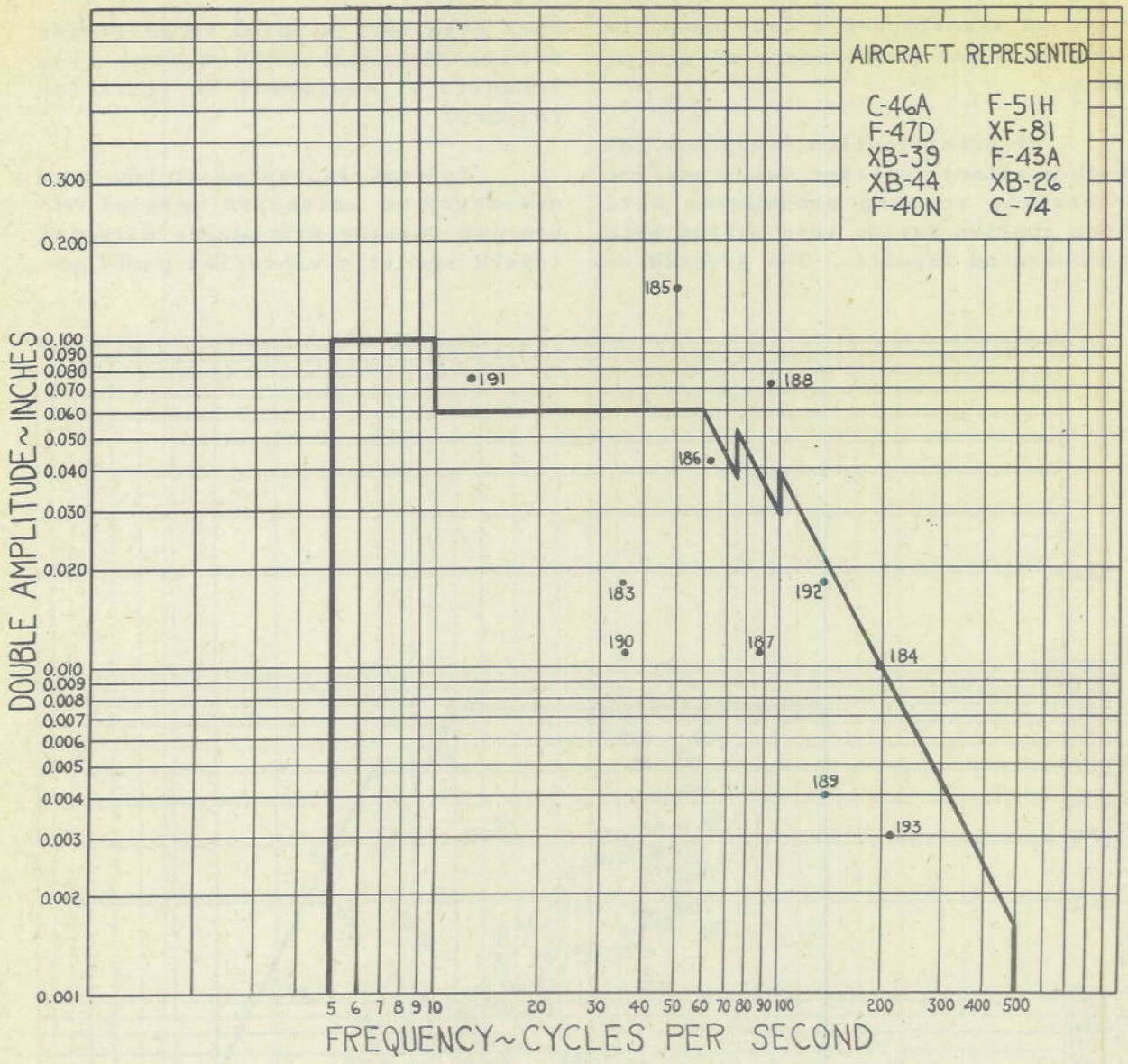

Figure 4 - Vibration characteristics of reciprocating engines-resultant envelope

For example, one of the most serious problems of generators installed on reciprocating engines is that of bending resonance of the generator assembly, comparable to the action of a cantilever beam. Such resonances may cause failures of mounting studs, mounting flanges, or failures of the engine accessory section. A vibration endurance test for such equipment as starters and generators has been developed by the Pratt and Whitney Engine Division, which requires the accessory to be subjected to fundamental cantilever bending resonance involving a flexural

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Table 2
Identification of Points Plotted in Figures 2, 3, and 4
for Aircraft Reciprocating Engine Vibration in Flight

No.	Bibliography Reference No.	Aircraft	Engine	Motion or Location	Source	Engine Speed RPM	Frequency CPS	Double Amplitude Inches
1	8	C-46A	R-2800-51	Thrustwise Translation	1/2XE	2100	17.5	0.003
2	8	"	"	" "	1XE	2730	45.5	0.007
3	8	"	"	" "	1-1/2XE	1840	46	0.0012
4	8	"	"	" "	2XE	2500	83.3	0.0022
5	8	"	"	" "	3-1/2XE	1840	107	0.002
6	8	"	"	Lateral "	1/2XE	1630	13.5	0.0097
7	8	"	"	" "	1XE	2730	45.5	0.0041
8	8	"	"	" "	1-1/2XE	2630	65.5	0.0006
9	8	"	"	" "	2XE	2630	91	0.0034
10	8	"	"	" "	3-1/2XE	1860	108	0.0026
11	8	"	"	Vertical "	1/2XE	2640	22	0.0025
12	8	"	"	" "	1XE	2500	41.7	0.0007
13	8	"	"	" "	1-1/2XE	2680	67	0.0003
14	8	"	"	" "	2XE	2200	73.4	0.0028
15	8	"	"	" "	3-1/2 XE	2680	156	0.0015
16	8	"	"	Roll	1/2XE	1700	14.1	0.041
17	8	"	"	"	1XE	2300	38.3	0.011
18	8	"	"	"	1-1/2XE	2800	70	0.002
19	8	"	"	"	2XE	1700	56.7	0.008
20	8	"	"	"	3-1/2XE	1700	99	0.0085
21	8	"	"	Pitch	1/2XE	1700	14.1	0.025
22	8	"	"	"	1XE	2680	44.7	0.005
23	8	"	"	"	1-1/2XE	2650	66.3	0.001
24	8	"	"	"	2XE	2680	89.7	0.0075
25	8	"	"	"	3-1/2XE	1800	105	0.0065
26	8	"	"	Yaw	1/2XE	1520	12.6	0.033
27	8	"	"	"	1XE	1840	30.6	0.004
28	8	"	"	"	1-1/2XE	2730	68.3	0.0025
29	8	"	"	"	2XE	1750	58	0.006
30	8	"	"	"	3-1/2XE	1650	96.5	0.006
31	12	F-47D	R-2800-21	Thrustwise Translation	1/2XE	2620	21.9	0.0023
32	12	"	"	" "	1XE	2700	45	0.0074
33	12	F-47D	R-2800-21	Thrustwise Translation	2XE	2440	81.3	0.0012
34	12	"	"	" "	2-1/2XE	1970	82	0.001
35	12	"	"	Lateral "	1/2XE	2000	16.7	0.007
36	12	"	"	" "	1XE	1580	26.3	0.0045
37	12	"	"	" "	2XE	2170	72.5	0.0015
38	12	"	"	" "	2-1/2XE	2700	112	0.0007
39	12	"	"	" "	4-1/2XE	2700	202	0.0005
40	12	"	"	Vertical "	1/2XE	1600	13.3	0.0038
41	12	"	"	" "	1XE	2670	44.6	0.007
42	12	"	"	" "	2XE	2600	86.7	0.0024
43	12	"	"	" "	2-1/2XE	1580	66	0.0012
44	12	"	"	" "	4-1/2XE	1600	120	0.0072
45	11	"	"	Roll	1/2XE	1600	13.3	0.025
46	11	"	"	"	1XE	1590	26.5	0.0225
47	11	"	"	"	2XE	2620	87.5	0.004
48	11	"	"	"	2-1/2XE	1600	66.6	0.002
49	11	"	"	"	4-1/2XE	2600	194	0.0035
50	11	"	"	Pitch	1/2XE	1600	13.3	0.0225
51	11	"	"	"	1XE	2660	44.3	0.0135
52	11	"	"	"	2XE	2700	90	0.0125
53	11	"	"	"	2-1/2XE	1580	66	0.004
54	11	"	"	"	4-1/2XE	1600	120	0.019
55	11	"	"	Yaw	1/2XE	1970	16.4	0.012
56	11	"	"	"	1XE	2600	43.3	0.004
57	11	"	"	"	2XE	2600	90	0.003
58	21	XB-39	V-3420-11	Lateral "	1XP	1000	5.4	0.060
59	21	"	"	Vertical "	1XP	1325	7	0.034
60	21	"	"	Roll	1/2XE	1600	13.3	0.042
61	21	"	"	Pitch	1XP	2400	12.8	0.100
62	21	"	"	Yaw	1XP	2600	13.7	0.079
63	22	XB-44	R-4360-33	Thrustwise Translation	3-1/2XE	2630	15.4	0.0027
64	22	"	"	Lateral "	1XP	1940	12.3	0.016
65	22	"	"	Vertical "	1XP	1350	8.55	0.011
66	22	"	"	Roll	1/2XE	1900	15.8	0.032
67	22	"	"	"	1XE	2300	38.4	0.017

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Table 2 (Cont.)

No.	Bibliography Reference No.	Aircraft	Engine	Motion or Location	Source	Engine Speed RPM	Frequency CPS	Double Amplitude Inches
68	22	XB-44	R-4360-33	Roll	2XE	2150	71.7	0.016
69	22	"	"	"	3-1/2XE	2550	149	0.030
70	22	"	"	Pitch	1XP	1620	10.3	0.017
71	22	"	"	"	1XE	2000	33.3	0.012
72	22	"	"	Yaw	1XP	2400-2600	15.2 - 16.5	0.017
73	22	"	"	"	1XE	2000	33.3	0.012
74	9	F-40N	V-1710-81	Thrustwise Translation	1/2XE	2300	19.1	0.004
75	9	"	"	" "	1XE	2300	38.3	0.0024
76	9	"	"	" "	1-1/2XE	1800	45	0.009
77	9	"	"	" "	2-1/2XE	2600	108	0.002
78	9	"	"	Lateral "	1/2XE	2280	17.3	0.0185
79	9	"	"	" "	1XE	2800	46.7	0.0085
80	9	"	"	" "	1-1/2XE	3130	78	0.0075
81	9	"	"	Vertical "	1/2XE	2560	21.3	0.0088
82	9	"	"	" "	1XE	3000	50	0.0026
83	9	"	"	" "	1-1/2XE	2400	60	0.0025
84	9	"	"	" "	2-1/2XE	2500	104	0.001
85	9	"	"	Roll	1/2XE	2280	19	0.063
86	9	"	"	"	1XE	2900	48.3	0.031
87	9	"	"	"	1-1/2XE	2650	66.3	0.059
88	9	"	"	"	2XE	2500	83.5	0.017
89	9	"	"	"	3XE	3120	156	0.010
90	9	"	"	Pitch	1/2XE	2040	17	0.030
91	9	"	"	"	1XE	1800	30	0.006
92	9	"	"	"	1-1/2XE	1800	45	0.015
93	9	"	"	Yaw	1/2XE	2300	19.2	0.065
94	9	"	"	"	1XE	1940	32.4	0.010
95	9	"	"	"	1-1/2XE	1800	45	0.036
96	10	F-51H	V-1650-11	Thrustwise Translation	1XP	1600	12.7	0.012
97	10	"	"	" "	2-1/2XE	1600	66.7	0.004
98	10	"	"	Lateral "	1XP	1800	14.4	0.014
99	10	"	"	" "	1/2XE	1800	15	0.0085
100	10	"	"	" "	2-1/2XE	2300	96	0.007
101	10	"	"	Vertical "	1XP	2000	16	0.013
102	10	"	"	" "	1/2XE	2500	20.8	0.003
103	10	F-51H	V-1650-11	Vertical Translation	1-1/2XE	2900	72.7	0.003
104	10	"	"	" "	2-1/2XE	2400	100	0.002
105	10	"	"	Roll	1XP	2700	12.9	0.058
106	10	"	"	"	1/2XE	1900	15.8	0.029
107	10	"	"	"	2-1/2XE	2900	120	0.017
108	10	"	"	Pitch	1XP	1600	12.7	0.023
109	10	"	"	"	1XE	2900	48.3	0.007
110	10	"	"	"	2-1/2XE	2200	92	0.007
111	10	"	"	Yaw	1XP	1600	12.7	0.070
112	23	XF-81	T-31	Lateral Translation	1XP	13,000	19.1	0.024
113	23	"	"	" "	1XE	13,000	217	0.0035
114	23	"	"	Vertical "	1XP	12,500	18.3	0.0068
115	23	"	"	" "	1XE	13,000	217	0.0018
116	23	"	"	Thrustwise "	1XP	13,000	19.1	0.0068
117	23	"	"	Lateral " Gear Reduction		13,000	49 & 85	0.003
118	23	"	"	Roll	1XP	12,200	17	0.016
119	23	"	"	"	1XE	12,400	207	0.0045
120	23	"	"	Pitch	1XP	13,000	19.1	0.0265
121	23	"	"	"	1XE	13,000	217	0.0035
122	23	"	"	Yaw	1XP	13,000	19.1	0.031
123	24	F-43A	R-1830-49	Roll	1XP	1700	15.8	0.014
124	24	"	"	"	1XE	1700	28.4	0.019
125	24	"	"	"	3XP	1800	51	0.003
126	24	"	"	"	2XE	1600	53	0.0035
127	24	"	"	"	2-1/2XE	2400	100	0.0035
128	24	"	"	Pitch	1XP	1800	16.8	0.055
129	24	"	"	"	1XE	1700	28.4	0.006
130	24	"	"	"	3XP	1600	45	0.006
131	24	"	"	"	2XE	2200	73	0.010
132	24	"	"	"	2-1/2XE	1900	79	0.008
133	24	"	"	Yaw	1XP	1800	10	0.062
134	24	"	"	"	1XE	1600	26.7	0.006
135	24	"	"	"	3XP	2200	62	0.004
136	24	"	"	"	2XE	2400	80	0.006
137	24	"	"	"	2-1/2XE	2700	112	0.005
138	24	"	"	Thrustwise Translation	1XP	1900	17.7	0.0028

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Table 2 (Cont.)

No.	Bibliography Reference No.	Aircraft	Engine	Motion or Location	Source	Engine Speed RPM	Frequency CPS	Double Amplitude Inches
139	24	F-43A	R-1830-49	Thrustwise Translation	1XE	2400	40	0.0013
140	24	"	"	" "	3XP	2000	56	0.0007
141	24	"	"	" "	2XE	2100	70	0.0007
142	24	"	"	" "	2-1/2XE	1800	75	0.0004
143	24	"	"	Lateral "	1XP	1800	10	0.010
144	24	"	"	" "	1XE	1700	28.3	0.002
145	24	"	"	" "	3XP	1900	53	0.001
146	24	"	"	" "	2XE	2200	73	0.001
147	24	"	"	" "	2-1/2XE	2200	92	0.0035
148	24	"	"	Vertical "	1XP	2000	18.7	0.007
149	24	"	"	" "	1XE	2000	33.3	0.001
150	24	"	"	" "	3XP	2000	56	0.001
151	24	"	"	" "	2XE	2700	90	0.001
152	24	"	"	" "	2-1/2XE	2700	112	0.0018
153	25	XB-26	R-2800	Roll	1/2XE	1620	13.4	0.028
154	25	"	"	"	1XE	1500	25	0.013
155	25	"	"	"	1-1/2XE	1820	45.5	0.004
156	25	"	"	"	2XE	2500	83.5	0.006
157	25	"	"	"	3-1/2XE	2400	140	0.001
158	25	"	"	Pitch	1/2XE	1720	14.3	0.018
159	25	"	"	"	1XE	1800	30	0.0125
160	25	"	"	"	1-1/2XE	1920	48	0.010
161	25	"	"	"	2XE	2520	84	0.0025
162	25	"	"	"	3-1/2XE	1800	105	0.004
163	25	"	"	Yaw	1/2XE	1800	15	0.0185
164	25	"	"	"	1XE	2520	42	0.007
165	25	"	"	"	1-1/2XE	2140	53.7	0.0095
166	25	"	"	"	2XE	2520	84	0.004
167	25	"	"	"	3-1/2XE	1720	100	0.0055
168	25	"	"	Thrustwise Translation	1/2XE	1930	16	0.0036
169	25	"	"	" "	1XE	1920	32	0.027
170	25	"	"	" "	1-1/2XE	2520	42	0.001
171	25	"	"	" "	2XE	1930	64.5	0.0038
172	25	"	"	" "	3-1/2XE	2000	116	0.001
173	25	"	"	Lateral "	1/2XE	1520	12.6	0.006
174	25	"	"	" "	1XE	1520	25.3	0.005
175	25	XB-26	R-2800	Lateral Translation	1-1/2XE	1520	38	0.002
176	25	"	"	" "	2XE	2420	81	0.003
177	25	"	"	" "	3-1/2XE	2140	124	0.0024
178	25	"	"	Vertical "	1/2XE	1520	12.6	0.006
179	25	"	"	" "	1XE	1930	32	0.0032
180	25	"	"	" "	1-1/2XE	1520	38	0.002
181	25	"	"	" "	2XE	2520	84	0.0043
182	25	"	"	" "	3-1/2XE	2130	74.7	0.002
183	26	C-74	R-4360	Fuel Pressure Trans. Vertical	1XE	2100	35	0.018
184	26	"	"	" " "	4-1/2XE	2600	195	0.010
185	26	"	"	F&A " "	2XE	1500	50	0.141
186	26	"	"	" " "	2-1/2XE	1500	62.5	0.042
187	26	"	"	" " "	3-1/2XE	1500	87	0.011
188	26	"	"	" " "	4-1/2XE	1500	94	0.072
189	26	"	"	" " "	5-1/2XE	1500	137	0.004
190	26	"	"	Torque Press. Trans. Vertical	4XP	1400	35.6	0.011
191	26	"	"	Tach. Gen. Drive Vert.	1XP	2170	12.7	0.075
192	26	"	"	" " " "	8XP	2600	132	0.018
193	26	"	"	" " " F&A	4-1/2XE	2800	210	0.003

double amplitude of approximately 0.015 inch measured at a point ten inches from the mounting base. Throughout the endurance period the accessory is to be attached to an actual engine accessory section so that resonance involves a combined flexure of the accessory section and the accessory itself.

Special vibration testing procedures also have been evolved for aircraft instruments. The standard vibration error and vibration failure

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tests require the application of circular vibration in a plane tilted 45 degrees to the vertical plane of the instrument mounting panel. The vibration error test requiring a constant double amplitude of 0.003 to 0.005 inch through the frequency range of 5 to 50 cycles per second, is intended not so much to simulate service conditions of vibration as to calibrate the instruments under standard conditions of vibration. Vibration failure tests conducted with the frequency cycling between 5 and 50 cycles per second in one-hour periods at a double amplitude of 0.018 to 0.020 inch for a duration of 3 hours, are

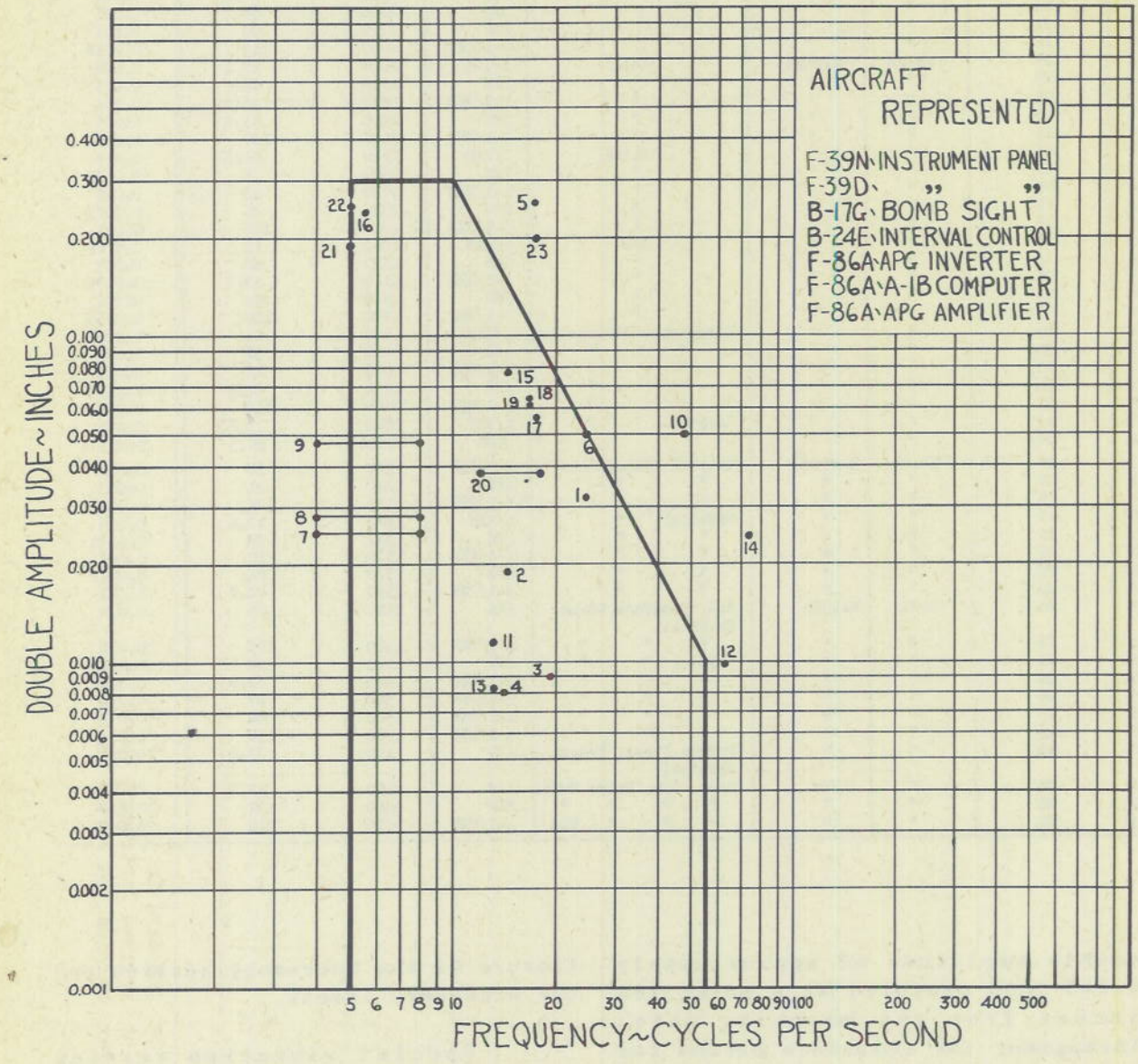

Figure 5 - Vibration characteristics of elastically mounted equipment

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Table 3
Identification of Points Plotted in Figure 5
for Flight Vibration of Elastically Mounted Assemblies

No.	Reference	Aircraft	Engine	Component	Motion	Source	Engine Speed RPM	Freq. CPS	Double Amplitude Inches
1	27	F-39N	V-1710-83	Instrument Panel	Lateral	1/2XE	2940	24.5	0.032 *
2	27	"	"	" "	"	1XP	1790	13.4	0.019 **
3	27	"	"	" "	Vertical	1/2XE	2300	19.2	0.009 *
4	27	"	"	" "	"	1XP	1800	13.4	0.008 **
5	28	F-39D	V-1710	" "	Lateral	1XP	1860	17.2	0.255
6	28	"	"	" "	Thrustwise	1XP	2650	24.5	0.050
7	6	B-17G	R-1820-97	Bombsight	"	Transient	-	4-8	0.025
8	6	"	"	"	Lateral	"	-	4-8	0.028
9	6	"	"	"	Vertical	"	-	4-8	0.0473
10	7	B-24E	R-1830-43	Bomb Internal Control	Thrustwise	Machine Gun	2700	47.3	0.0496
11	7	"	"	" "	Lateral	"	2700	13.1	0.0115
12	7	"	"	" "	"	"	2700	62.5	0.0098
13	7	"	"	" "	Vertical	"	2700	13.1	0.0082
14	7	"	"	" "	"	"	2700	73	0.0245
15	29	F-86A	J-35	APG Inverter	Lateral	"	-	14.5	0.077 ***
16	29	"	"	" "	Thrustwise	"	-	5.5	0.240 ***
17	29	"	"	" "	Vertical	"	-	17.4	0.056 ***
18	29	"	"	A-1B Sight Computer	Lateral	"	-	16.7	0.063 ***
19	29	"	"	" "	Thrustwise	"	-	16.7	0.061 ***
20	29	"	"	" "	Vertical	"	-	12-18	0.038 ***
21	29	"	"	APG-5 Servo Amplif.	Lateral	"	-	5	0.190 ***
22	29	"	"	" "	Thrustwise	"	-	5	0.25 ***
23	29	"	"	" "	Vertical	"	-	17.3	0.20 ***

* Standard Instrument Panel
** Revised Instrument Panel
*** Machine Gun Firing on Ground

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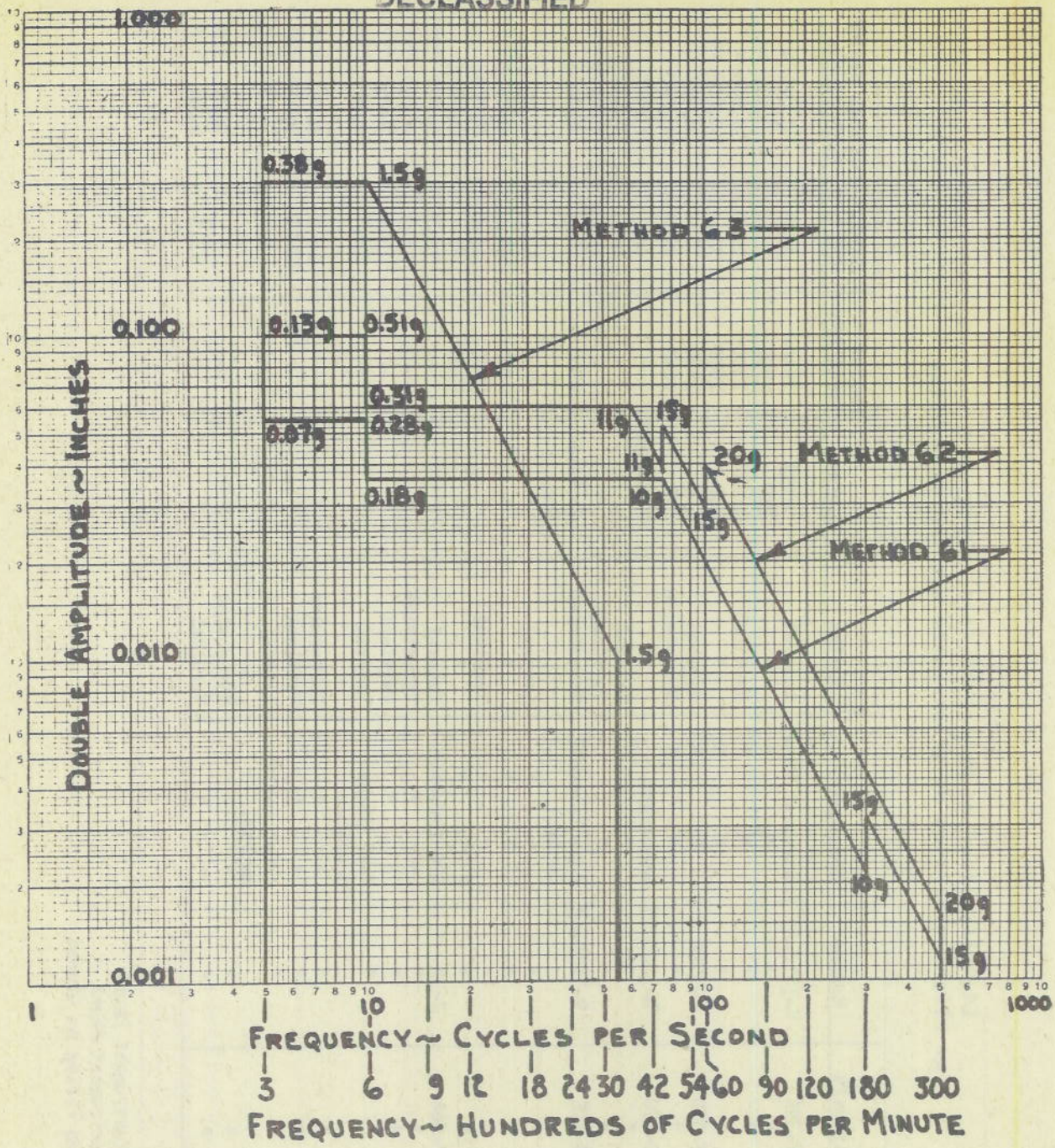

Figure 6 - Range curves for vibration tests outlined in Army Air Forces specification 41065

intended to detect faulty materials or workmanship. Testing of instrument designs with respect to vibration endurance follows Method 63 of AF Specification 41065 which more nearly simulates conditions of service vibration.

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In regard to jet aircraft, the tendency has been to over-simplify the vibration problem and to assume that the presence of vibration in such aircraft is of minor importance. It was considered that possibly the vibration testing procedures for equipment attached to the structures of conventional aircraft (Method 61, AF Specification 41065) might involve conditions of unwarranted severity when applied to jet aircraft. However, a plot of the meager amount of test data available for jet aircraft reveals a scattering of points of maximum amplitude, showing a surprising contiguity to the data envelope established for structural vibration in conventional aircraft. These data shown in Figure 7 and identified in Table 4 do not appear to justify deviation in the case of jet aircraft, from the vibration testing requirements for structural mounted equipment in conventional aircraft.

As might be expected, the dominant source of vibration on turbo-jet engines is turbine wheel unbalance. The frequency corresponds to the rotating speed of the turbine wheel with appreciable harmonics, such as three times rotating speed. Tolerances for the maximum amount of vibration on turbo-jet power plants due to wheel unbalance have been set up by the manufacturers. These tolerances vary from a double amplitude of 0.003 inch for the J-33 power plant to 0.005 inch for the J-35 power plant. In addition, there are present certain conditions of combustion and gas flow within the turbines which cause vibration and which are characteristic of each individual aircraft. Other important sources of vibration present in conventional aircraft are also present in jet aircraft such as vibration caused by air flow disturbances, flying through rough air, machine gun firing, taxiing, etc.

At first thought, many engineers considered the ram jet as the ideal vibration-free power plant because it involves no moving parts. However, a so-called resonant burning phenomenon causes vibration which measurements have shown to be as high as 0.20 inch double amplitude at a frequency of 60 cycles per second with a corresponding vibratory acceleration of ± 36g. This is approximately three times the most severe vibratory acceleration found on reciprocating engines in the same frequency range.

The test apparatus used to accomplish the vibration testing according to the requirements of AF Specification 41065 is the electro-magnetic type vibration exciter. The Westinghouse Type HI and GS vibration exciting equipment is shown in Figure 8 and the MB Type S-3 vibration exciting equipment in Figure 9. When necessary, a mechanical resonator system such as that shown in Figure 10 for the Pratt and Whitney type accessory vibration table, is utilized to augment the vibratory output of the electro-magnetic exciters. Mechanical type vibration tables such as the All-American Tool machine shown in Figure 11, also are used when the resonant frequencies are under 60 cycles per second as in the case of elastically supported equipment. Reaction-type mechanical vibration stands are employed for instrument vibration testing. The Krouse instrument vibration test stand shown in Figure 12 is an example in this category.

The practical effect of the vibration testing procedures outlined in AF Specification 41065 with respect to aircraft equipment design is to require the use of effective vibration isolating mounts for all inherently fragile, delicate, and sensitive equipment. Assemblies

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containing vacuum tubes, relays, switches, solenoids, meters, gyros, electro-mechanical valves, pressure-sensitive bellows, and voltage regulators should be vibration isolated to avoid impairment of functioning and to insure maximum service life. From the space and weight-saving standpoint, the vibration isolating system should be designed as an integral part with the functional components of the assembly. It is inefficient to add the vibration isolating system as an afterthought or as an extra which is unrelated to the functional components. However, in many instances it has been possible to add or revise the vibration isolating system to "save" the equipment from an expensive redesign. If

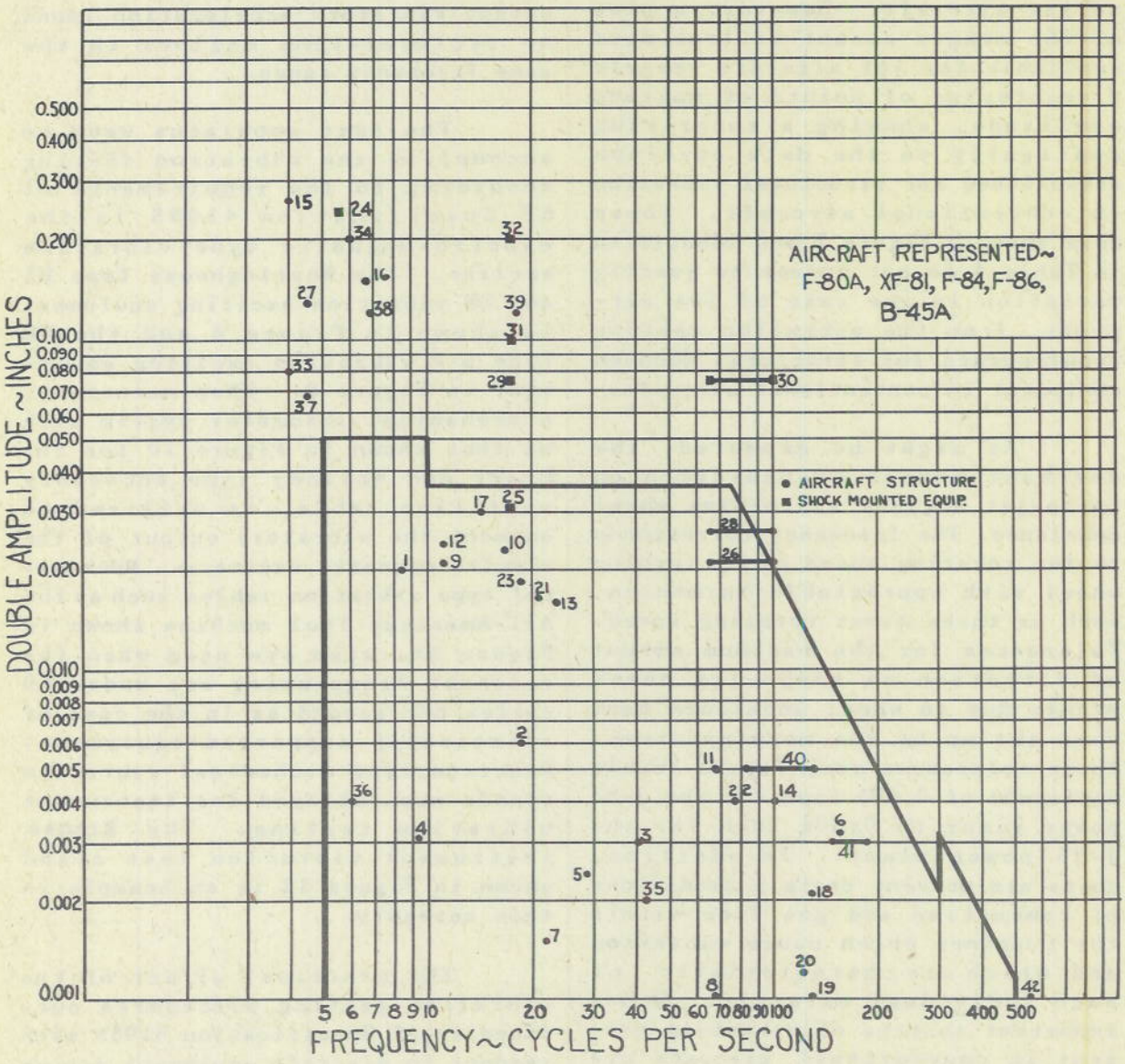

Figure 7 - Vibration characteristics of jet-powered aircraft

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the case assembly and mechanical layout of such items of equipment had been considered from the standpoint of vibration isolation, smaller and lighter end products would have resulted.

Table 4
Identification of Points Plotted in Figure 7
for Structural Vibration of Jet-Powered Aircraft in Flight

No.	Ref. No.	Aircraft	Location	Direction	Source	Engine Speed RPM	Freq. CPS	Double Amplitude Inches	Test Condition
1	18	F-80A	Wing Tip	Vertical	Wing Sym. Bending	Idle 4030	8.3	0.020	443IAS
2	18	"	" "	"	Wing. Asym. Bending	10,400	18.3	0.006	357IAS
3	18	"	" "	"	Wing Torsion	Idle 4030	40	0.003	443IAS
4	18	"	" "	Thrustwise	Fuselage Side Bending	11,500	9.2	0.003	443IAS
5	18	"	" "	"	Sym. Wing Bend.	11,500	28.3	0.0024	443IAS
6	18	"	Engine Trunnion Support	Lateral	1XE	9,020	152	0.0032	272IAS
7	18	"	" "	Thrustwise	-	11,500	21.6	0.0015	443IAS
8	18	"	" "	"	-	11,500	67	0.001	443IAS
9	18	"	Stabilizer Tip	Vertical	Fus. Torsion	Idle 4030	11.2	0.021	443IAS
10	18	"	" "	"	Stab. Sym. Bending	11,500	16.6	0.023	443IAS
11	18	"	" "	"	Stab. Tors.	11,500	67	0.005	443IAS
12	18	"	Fin Tip	Lateral	Fus. Torsion	11,500	11.2	0.024	443IAS
13	18	"	" "	"	Fin Bending	10,900	23.4	0.016	400IAS
14	30	"	Wing Tip	"			100	0.004	
15	30	"	" "	Vertical	-	-	4	0.26	
16	30	"	" "	"	-	-	6.7	0.15	
17	30	"	" "	"	-	-	14.1	0.030	
18	31	F-84	Engine Trunnion Support	Thrustwise	1XE	7,700	128	0.0021	200IAS
19	31	"	" "	Lateral	1XE	7,700	128	0.0010	150IAS
20	31	"	" "	Vertical	1XE	7,300	122	0.0012	100IAS
21	23	XF-81	Cockpit	Vertical	1XP	13,000	19	0.0165	Rated Turbine Speed
22	23	"	"	"	4XP	12,500	76	0.0045	95% Rated Turbine Speed
23	23	"	"	"	1XP	12,500	18.4	0.0185	280IAS
24	29	F-86A	APG Inverter	Thrustwise	Machine Gun	-	5.5	0.240	Ground Test*
25	29	F-86A	Fuselage Nose	Lateral	Mach. Gun	-	17.3	0.031	Ground Test
26	29	"	" "	Thrustwise	" "	-	65-100	0.021	" "
27	29	"	A-1B Sight Head	"	" "	-	4.5	0.128	" "
28	29	"	Fuselage	"	" "	-	65-100	0.026	" "
29	29	"	A-1B Sight Dynamoter	"	" "	-	17.4	0.075	" "
30	29	"	" "	"	" "	-	65-100	0.032	" "
31	29	"	" "	Lateral	" "	-	17.4	0.100	" "
32	29	"	APG-5 Servo Amplif.	Vertical	" "	-	17.3	0.200	" "
33	32	B-45A	APA-59 Stabilization Unit Position***	Vertical	-	-	4	0.080	Take Off**
34	32	"	" "	Lateral	-	-	6	0.200	" " **
35	32	"	" "	"	-	-	42	0.002	" " **
36	32	"	" "	Vertical	-	-	6	0.004	Bomb Bay Open
37	32	"	" "	"	-	-	4.5	0.067	Dive 500 MPH**
38	32	"	" "	Lateral	-	-	6.8	0.120	" " " **
39	32	"	" "	Thrustwise	-	-	18	0.120	Landing**
40	Allowable Vibration Due to Turbine Wheel Unbalance in J-35					Power Plants			
41	"	"	" " "	" "	" J-33	" "			
42	33	YF-80A	Fwd Station of Turbine Support	Lateral	3XE	10,900	550	0.001	Wind Tunnel

* Shock Mounted Inverter.

** Series of Transients

*** Under Floor Navigator's Compartment

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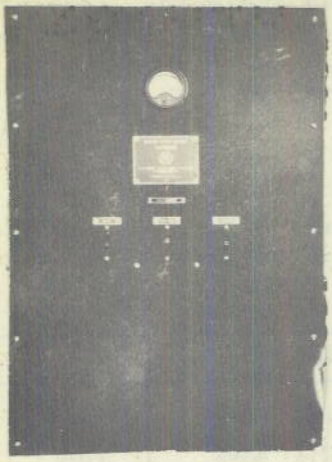

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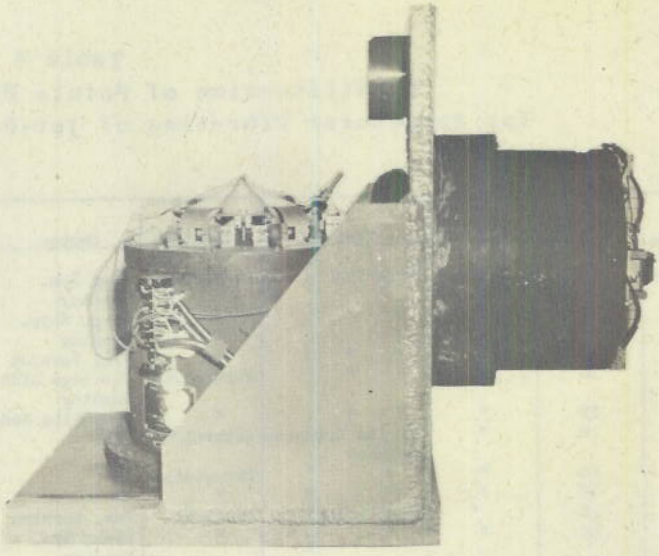
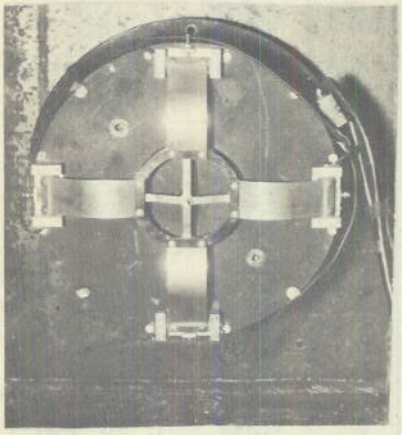
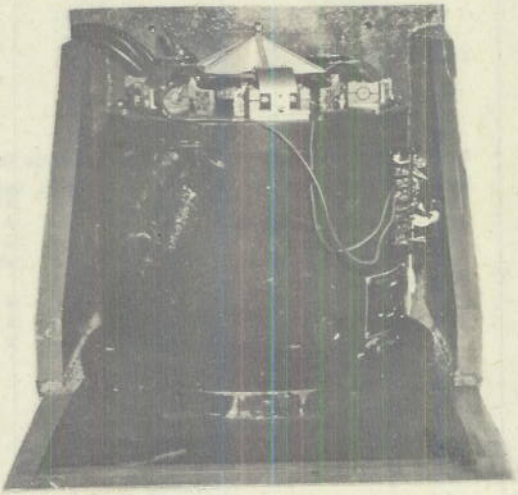


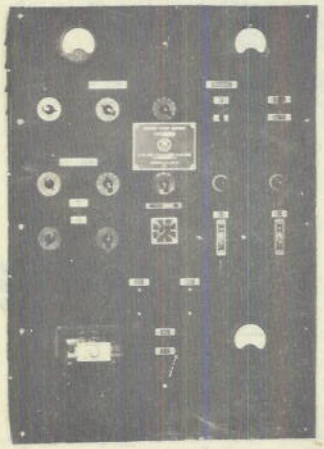
TYPE GS AND HI EXCITERS MOUNTED



TYPE GS EXCITER (TOP VIEW)



TYPE HI EXCITER (FRONT VIEW)



40 WATT AMPLIFIER

Figure 8 - The Westinghouse type HI and GS vibration exciting equipment

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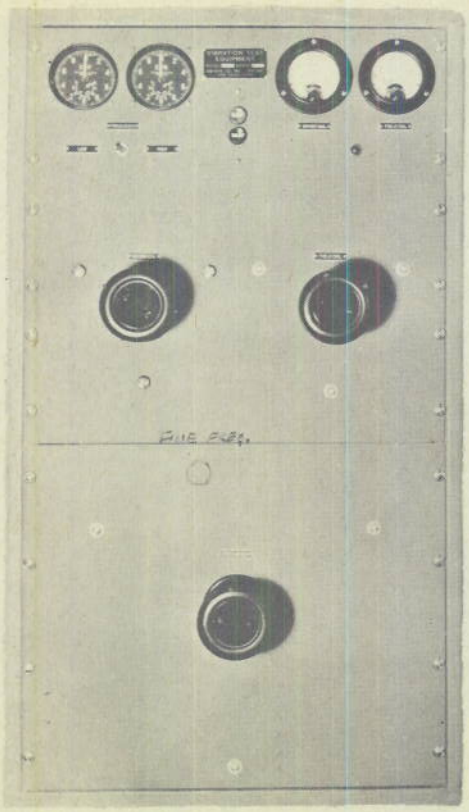
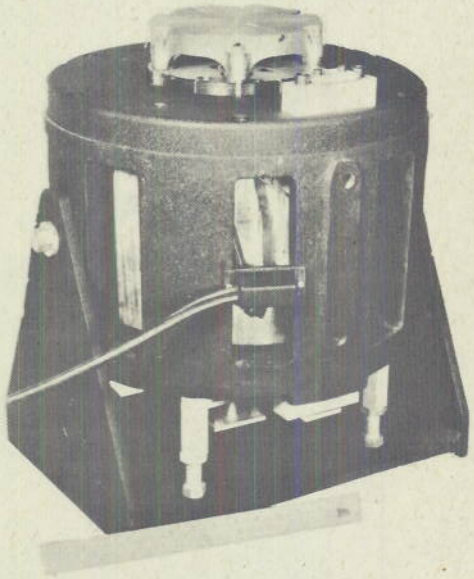
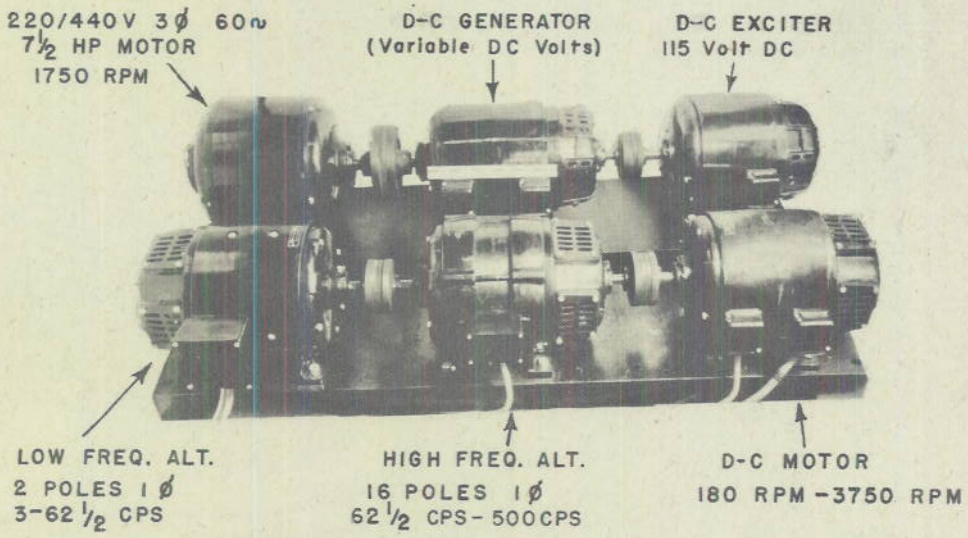


T-15 CONTROL PANEL



S-3 EXCITER



P-15 ROTATING POWER SUPPLY

Figure 9 - The MB type S-3 vibration exciting equipment

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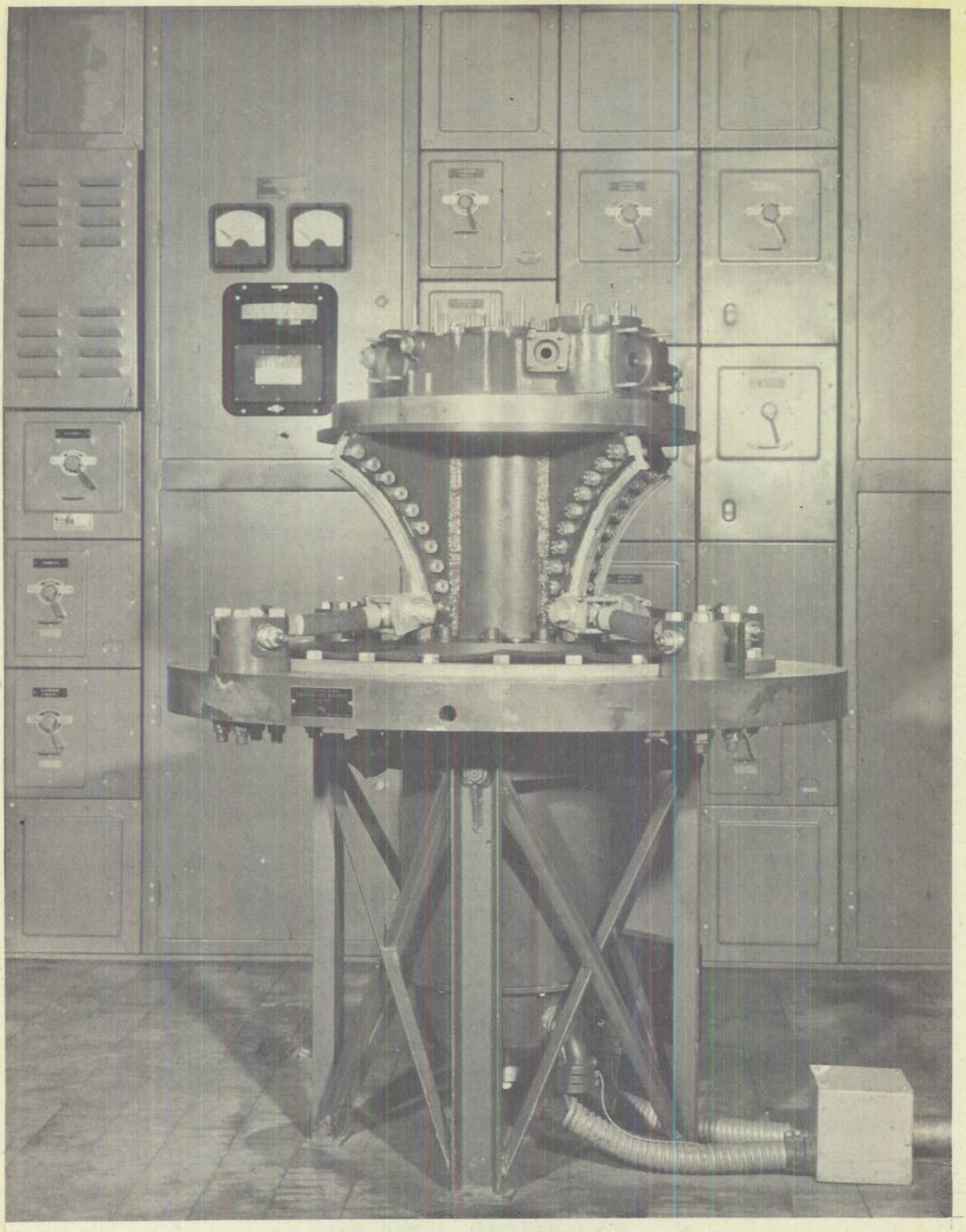

Figure 10 - The Pratt and Whitney type engine accessory vibration table

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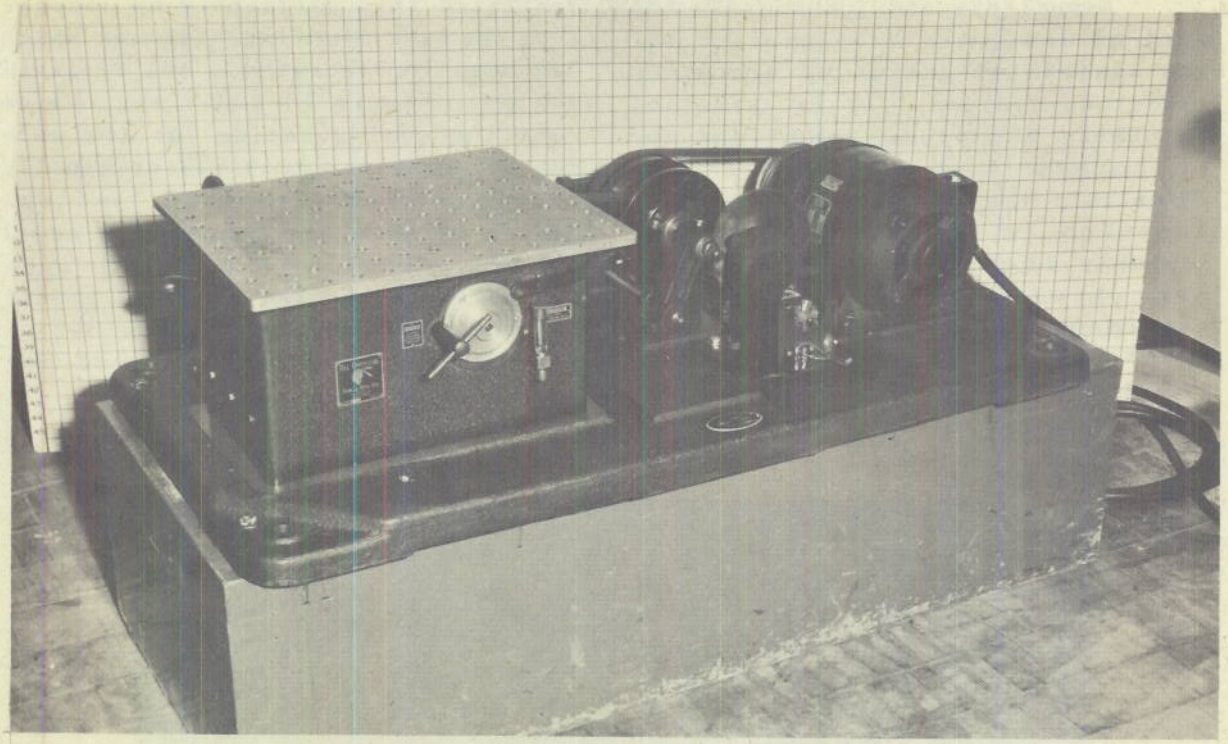

Figure 11 - The All American Tool Model 100A mechanical vibration table

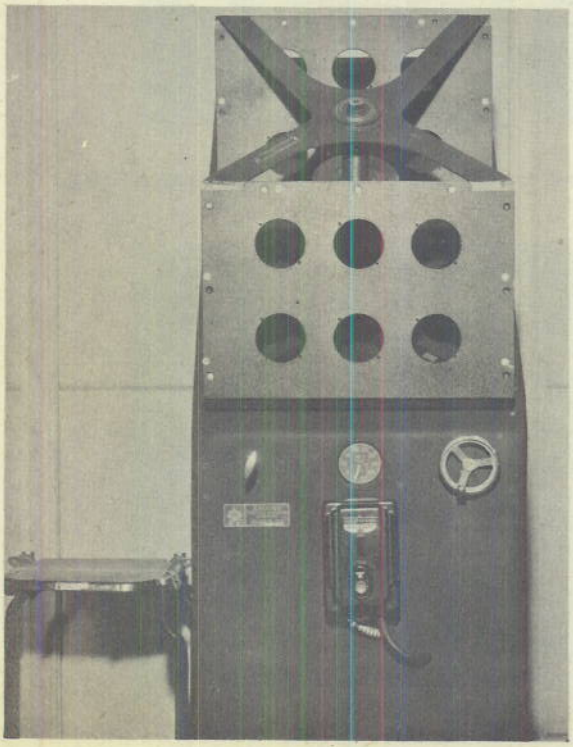

Figure 12 - The Krouse instrument vibration test stand

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Experience and judgment on the part of designers are required for the successful design of vibration isolating systems. It is essential to choose the proper mountings and to locate them correctly. To illustrate the importance of correct location, a compass transmitter assembly submitted to the Equipment Laboratory failed to pass the pertinent vibration test required by AF Specification 41065 because of vibration-induced malfunctioning. The Equipment Laboratory easily corrected this difficulty by relocating the same mounting elements which had been incorporated in the original design. Figure 13 shows the bottom mounting configuration for the compass transmitter as it was originally submitted for vibration testing. Figure 14 shows the engineering model of the center-of-gravity type suspension using the original mounting units which permitted the transmitter to pass the required vibration tests.

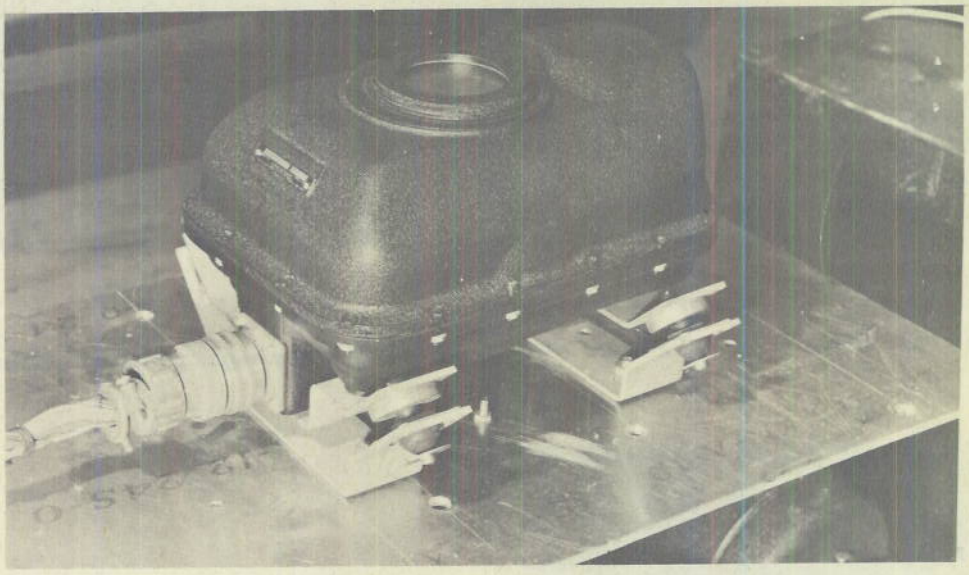

Figure 13 - Remote compass transmitter with original bottom mounting

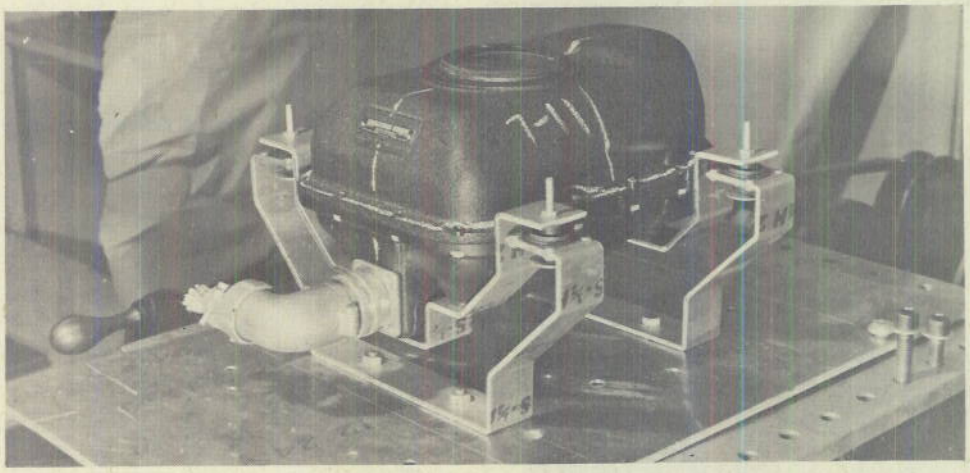

Figure 14 - Remote compass transmitter with improved center-of-gravity mounting

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In conclusion, the vibration testing procedures outlined in AF Specification 41065 are not perfect nor final, but their use has proved invaluable in improving aircraft equipment from a vibration standpoint. The procedures have a rational basis and should be considered as an attempt to establish the best possible engineering approach to the vibration testing problem pending the accumulation of more service data. To substantiate and improve the vibration data envelopes included in this specification, more flight vibration surveys on all types of aircraft are definitely needed. Additional flight test data will undoubtedly point the way to establishing refinements in the classification of aircraft equipment from a vibration standpoint. Refined classification may in some instances require augmenting, and in others reducing the severity of presently required testing conditions. This will permit an increase of design efficiency. Nevertheless, even on the basis of present procedures, more careful attention on the part of designers in incorporating vibration isolating systems as an integral part of each design, will improve the functional and design efficiency of aircraft equipment in general.

Four years of vibration testing experience with AF Specification 41065 in force, has demonstrated that its twofold purpose of standardizing laboratory testing procedures and standardizing equipment design on a rational basis vibration-wise, is being accomplished. The specification has contributed immeasureably to the mechanical development of aircraft equipment and is pointing the way to future aircraft equipment of superior mechanical design.

DISCUSSION

Macduff:

What is the frequency of "3 times" which you mentioned?

D. C. Kennard, Wright-Patterson Air Force Base:

There has been measured in the F-80 airplane the vibration induced by unbalance in the turbine wheel. There has also been measured 3 times that frequency and what the source is I am not prepared to say, but it has been measured.

W. J. Howard, Los Alamos Scientific Laboratory:

We have had occasion to use the spec's that you people outlined. Is it practical to measure amplitude to thousandths? Are the limits in these spec's plus and minus five thousandths?

Kennard:

There have been no tolerances established on the application of the vibration. It is assumed that the

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vibration measuring equipment tolerances are applied; which are, I think for conventional equipment, an accuracy of 10 percent. Did I understand your question correctly?

Howard:

You admitted 10 percent. That is a pretty good percentage.

Kennard:

I think usually it can be much better than that. For instance, the vibration data on which these were established, the overall accuracy is within 10 percent. So that would seem a reasonable accuracy to maintain. However, I am sure we can do much better in our own laboratory. Equipment may not be available outside our laboratory to do better than that.

Priebe:

I would like to know how you make available equipment for reproducing these field conditions and follow the envelope on the diagram which you have set forth for the single tests. Does that equipment which is available, stay close enough to the envelope established within 50 to 100 percent variations?

Kennard:

I think possibly you misunderstand the testing procedures of the specification. The procedure does not imply cycling over this envelope that you saw. It implies making a natural frequency survey or a survey for resonances within that frequency range and then where resonances appear to increase the applied vibration to the corresponding point on the envelope in order to run the endurance test.

Priebe:

How can you determine all resonances if some of the vibrating elements are concealed in the instrument which are resonating inside the case?

Kennard:

That is sometimes a problem. However, we have found that approximate resonances of any significance will affect the functional properties of the specimen. The specifications also imply functioning the equipment during vibration. So that, if we make the proper measurements on the functional characteristics of the equipment, usually resonance will show up as a change in the functional characteristics.

C.E. Crede, Barry Corporation:

What time duration of test is used in your procedure?

Kennard:

The time we have specified is 4 hours of vibration along each mutually perpendicular axis.

Crede:

Can you assure yourself that the equipment is standing up for the 12 hours when installed in a plane? Do you have some means of taking into account the fatigue properties?

Kennard:

That 4 hours of testing or 12 hours of testing are under extreme severe conditions. That is, it is not only applying it to maximum amplitude that you would expect to find in service, but it is applying it at the worst possible conditions,

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mainly resonance, that is considered to be a relative factor in service time. We are going more or less arbitrarily on the basis that one hour in the laboratory simulates at least 10 hours service time and in most cases, considerably more than that. What we are aiming at is to make equipment that will last as long as the airplane will and I believe that is true of most of the equipment which we have tested. The 41065 specification is tending to show the endurance property of approximately the maximum amplitude measured in aircraft.

Crede:

I do not understand where the 10 to 1 factor comes in. Is that more or less arbitrary?

Kennard:

We do not have enough data at the present time to make a better estimate than that. We assume that resonance conditions on which the endurance tests run in the laboratory will not be held in service more than one-tenth of the time, and probably much less than that, so we are on the conservative side in that respect.

Muller:

What consideration have you given to damping in your investigations? If you have enough damping, the amplitude might be very small and would conclude there is no resonance.

Kennard:

That is correct, if the damping is sufficient to prevent a magnification of the amplitude, we would assume that if resonance is there, it is not one of the critical resonances. Usually resonance is very general here. In external assembly or anything of that nature there is usually one outstanding resonance that has very little damping in it.

D. E. Goldman, NMRI:

What specifications are there on those parts of the aircraft with which personnel come in contact such as pilot's seats, instrument panels, etc?

Kennard:

Specification 41065 has method 63 which would apply the vibration of instrument panels. Considerable vibration has been measured in the pilots compartment and that seems to fall well within the range of these envelopes I tested. In fact, in one aircraft in particular, the B-26, the vibration level in the pilot's compartment was very high, and there was no way to get rid of it.

Ball:

Supplementing Dr. Goldman's question, if the vibration in new planes were to exceed the limit specifications, would the planes be rejected on that ground?

Kennard:

These methods are not final as I have indicated and they should be considered as a developing process in airplane design.

C. B. Cunningham, NRL:

Mr. Kennard presents a desirable approach for determining vibration specifications for aircraft equipment. In addition to vibration the problem of shock protection of aircraft equipment is also important, particularly for Naval aircraft where hard carrier

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landings or catapult launchings are encountered. To adequately determine design figures for aircraft vibration isolation systems more shock as well as vibration field data are needed, particularly on the newer types of jet aircraft and on the many types of guided missiles. It is hoped that a program of revision of shock and vibration testing specifications for Naval aircraft equipment will be forthcoming along the lines indicated by Mr. Kennard.

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THE SHOCK INDUCED STRESSES IN BEAMS

By
R. E. Blake, Naval Research Laboratory

Practical design procedures for structures to be subject to Naval shipboard shock have not been able to utilize available theories because of their complexity. Possible compromises which might be made to achieve more accurate or easier design applications are discussed. The application of a fairly exact theory has been reduced for the case of beams to a simple, although rather tedious, procedure.

The problem of how to design structures to withstand naval shipboard shock is still far from a satisfactory solution. A great many papers have been published on the subject of elastic structures under shock which should apply to this problem. What seems to be the first of these was published by M. A. Biot in 1933. While application of this theory to design would result in rather good accuracy, it is too complicated and lengthy a process to use except in rare instances.

The present procedure, of necessity, relies heavily on shock machines to check finished designs and to furnish empirical design rules. Most design calculations which are made are based upon assumption of some value of steady acceleration which the entire structure is designed to withstand. This design acceleration at least reflects the fact that stresses which occur in a structure during a shock motion are caused by the inertia of the parts. It is the simplest possible design condition, yet it can become very difficult to apply to complicated, redundant shapes.

This calculation procedure can be shown by Biot's theory to be highly inaccurate. The immediate problem then, from the designer's point of view, is not to find refinements which make the Biot theory even more exact. This seems to be unprofitable at this time because we are unable to define the shock motion to be designed for with very great accuracy. Rather it seems desirable to study the effect of various simplifying assumptions on the accuracy of design calculations. Apparently the designer should have available several methods of design of successively greater accuracy but of successively greater complexity. Then according to his need for accuracy and the time available for design calculations, he can choose the method best suited to his case.

SINGLE-DEGREE-OF-FREEDOM-SYSTEMS

A possible design method which suggests itself is one which agrees with the Biot theory when the structure is a single-degree-of-freedom system. It is more complicated than the first design method in that the natural frequency of the structure must be found in order to

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specify a design acceleration. It becomes inaccurate as the structure deviates from a simple system, which is a concentrated mass on a massless elastic member. Calculation of maximum stress in the elastic member can be made if the maximum acceleration of the mass is known. For a given shock motion, the maximum acceleration of the mass depends only upon the natural frequency of the simple system. Therefore, the only data concerning a shock motion which the designer needs is a curve showing design acceleration as a function of natural frequency. Such a curve can be found from the time record of a shock motion by using a graphical method, mathematical machines, or electrical or mechanical analogs. A curve can also be obtained by subjecting a set of representative simple systems to the shock motion itself. The multifrequency-reed-gage has been used recently for this purpose. The reed gage records the maximum deflection (caused by the shock motion) of each reed. The ratio of this deflection to the static deflection of a reed under its own weight is calculated. Since the static deflection of the reed corresponds to the deflection which it would have for a steady (static) acceleration of one "g," this ratio is called the equivalent static acceleration of the shock motion in "g" units. It is the design acceleration for all simple systems having the same natural frequency as the reed from which it is calculated.

This curve of equivalent static acceleration versus natural frequency is of importance in several aspects of shock work and has been called a shock spectrum. It is identical to the plot which Biot called an "earthquake spectrum" in a paper on the design of earthquake-proof buildings.

MULTIDEGREE-OF-FREEDOM-SYSTEMS

Since all practical structures which can withstand shipboard shock have distributed mass and elasticity, the simple system approach necessarily is inaccurate. It is desirable therefore to reduce Biot's theory to as simple a calculation procedure as possible in order that its use will require the least possible time and effort. In order to ascertain how difficult Biot's method might be to apply to distributed mass systems, Mr. J. P. Walsh at NRL has worked out the procedure necessary to apply it to beams under transverse shock. He found that the calculation is rather tedious but it required only the plotting of curves and use of a calculating machine.

The first step requires the finding of the shapes of the normal modes of vibration of the beam and their corresponding natural frequencies. There have been several recent papers on this problem. Most of the methods are based upon Stodola's iteration method, which always converges to the first mode. Beskin and Rosenberg have recently presented a method by which higher modes can be obtained by a modification of Stodola's Method.

Corresponding to each normal mode (which is a deflection curve of the beam) there is a normal load. A normal load is that distributed load on the beam which will cause the deflection curve of the beam to be a normal mode. It can be shown that any distributed load on the beam can be represented by superposition of normal loads of the proper amplitude. This is analagous to the representation of any arbitrary curve by a Fourier series of sines and cosines of the proper magnitude.

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The second step is relatively easy and consists of finding the normal loads from the normal modes. The third step consists of representing the distributed weight of the beam by a series of normal loads. The amplitude of each normal load is formed by evaluating two integrals numerically. This is analagous to the determination of the coefficients of a Fourier series.

The fourth step consists of calculating the stress in the beam due to each of these normal load components of the weight of the beam itself. Since the weight of the beam produces an effect on the beam identical to that of a steady one "g" acceleration of its supports, we have actually found what might be called the normal stress components corresponding to a one "g" acceleration.

Thus far the process has been concerned with decomposing deflection, load, and stress patterns in the beams into their "normal components." The reason for this is that each of these normal modes of the beam responds to a shock motion independently, just as if they were single-degree-of-freedom systems. Thus the response of simple systems to the shock motion which were mentioned earlier can be applied directly at this point. Corresponding to the frequency of each mode, an equivalent static acceleration can be found from the shock spectrum. The maximum stress contribution of each mode is this acceleration multiplied by the stress found for a one "g" acceleration. The maximum possible stress in the beam due to the shock motion is the sum of these maximum contributions of the various modes.

This method of using the shock spectrum infers the conservative assumption that the maximum stress contributions of all modes occur simultaneously. By using the complete time history of the response of simple systems instead of the maximum response, the complete time history of the stress in the beam can be defined. This of course requires more work.

While Biot's theory may be too complex to apply to many design problems it is the most accurate method available. It is hoped that this method might be used to permit accurate, easy design of common types of structures by producing handbook data for these common types. Many of the steps required need only be done once for each type of structure.

DISCUSSION

M. St. Denis, TMB:

I should like to take advantage of the opportunity afforded by the reading of Mr. Blake's most interesting paper on "The Shock Induced Stresses in Beams" to make a few comments on how the naval architect designs structures subject to transient loadings.

Undoubtedly the design of a vessel is the most difficult structural problem in the world. The structure itself is most complex, consisting as it does of an almost incomprehensible assembly of material, and the external forces acting thereon, and arising either from the interplay of the elements or through the action of man, are

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dynamic in nature. To estimate the external forces is not an easy task. Neither adequate theory nor sufficient experimental data are available for dealing with this phase. Yet even if available they would not necessarily be sufficient. To be of utmost value the method of estimating the external forces must be simple so that it can be readily applied in the drafting room. For if a procedure of design, no matter how accurate, or clever, or appealing takes more than a minimum amount of time, it cannot well be used.

Pressed by the constant necessity of obtaining solutions to a design that fluxes continuously because of developments and changes in requirements, the naval architect must have a tool that he can use quickly and upon which he can rely. For him time is always of the essence. And yet, he can never compromise with safety.

According to the procedure in use at present in the drafting room, all dynamically applied external loadings are at first reduced to equivalent static loadings. This reduction is accomplished by a sort of alchemy in which both theory and empiricism are mysteriously fused. Sometimes the method is more theoretical as, e.g., in the design of structure subject to gun blast or wave pounding; and sometimes more empirical as, e.g., in the design of structure subject to underwater explosion.

In any event, the reduction of a dynamic problem to a static one requires a knowledge of the time history of the loading. In general such knowledge is not available. The reason for this is two fold. Firstly, the theoretical approach even if the few simple cases were attempted, may run into serious difficulty and require a disproportionate amount of time. Secondly, experimental studies, until quite recently, have suffered from the lack of adequate testing technique. Even now, though equipment is available and technique developed, the large cost of obtaining the data and performing the analysis is always a major obstacle.

Having determined more or less accurately the external loading on a structure, it then becomes necessary to determine the response of the structure thereto. This response can be obtained only upon acceptance of powerful limitations. These are:

(1) The material is elastic. The yield point of the material consequently introduces a limitation beyond which the theory as developed at present does not apply.

(2) The material behaves, linearly (obeys Hooke's law). The principle of superposition thus applies. For shipbuilding steel this assumption is one of necessity, not of reality.

(3) The structure is continuous. At first impression it appears as if for welded assemblies this assumption might be fulfilled. But the generally poorer lining up of welded structural members makes this assumption even less valid than for riveted construction.

In addition to these limitations, theoretical solutions are available only for certain definite boundary conditions which rarely, if ever, apply exactly to the problem on hand.

Determination of the response of a structure is severely complicated by the fact that this response

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is also dependent upon the past history of the structure, i.e., locked-up stresses in the structure before application of the external force. These result from the rolling, cutting, and welding of the material; by the changes in temperature, loading, and constraints during the period of fabrication and erection; by the changes in support reaction during building, launching, and docking; and, finally, when sailing, by the natural elements of wind and sea. They can never be calculated, and are, therefore, ignored. Although they continue to exist and consequently affect the problem.

Even with the highly restrictive assumptions and powerful limitations, the problem of determining the response of a structure to a dynamic loading is most complex, for the structures in which we are interested, such as a ship or component parts thereof, do not constitute simple elastic systems which can be analyzed without excessive effort. However, it frequently occurs that in a structure freedom in a single direction is preponderant to such an extent that for practical purposes it determines the behavior of the system. When this occurs the complex problem can be reduced to a simple one without serious error and the analysis is greatly simplified. In practical design the conditions necessary for this simplification are always assumed to exist. These are:

(1) The external loading must be distributed with reasonable uniformity over the structure. This is to insure that only the fundamental mode of vibration of the structure is excited: or, if the higher modes are also excited, their contribution will be small. When the external loading exhibits discontinuities in distribution (e.g., a hammer blow on a plate) the higher modes are always present, and indeed, may even become dominant with the result that resemblance is lost between the structure and the simplified system that replaces it for the purpose of analysis.

(2) The duration of the pulse must be sufficiently long. This is to insure that the deflected profile of the structure correspond to the external loading. When the pulse is of extremely short duration, the structure does not have sufficient time to adjust itself to the external loading. During the period of adjustment, transient shock waves and, therefore, deformations are set up so that again similarity is lost to the simple system.

(3) There should be no coupling between the fundamental mode of vibration of the structure and the higher modes, i.e., each mode of vibration should be independent of the others. This is quite evident. The previous condition insured that the higher modes would not be excited by the externally applied pulse. This condition is to insure that they not be excited by the response of the structure itself.

If the conditions for which a complex structure can be reduced to a simple system obtain, the procedure for converting the dynamic problem to one of statics is as follows:

(1) Obtain the fundamental period of free vibration of the structure.

(2) Determine the character and duration of the pulse.

(3) Obtain, from theory, the dynamic load factor corresponding to the character of the pulse and to the ratio of pulse duration to fundamental period.

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(4) The product of peak dynamic load by the dynamic load factor gives the equivalent static load.

(5) Solve the statical problem.

The response of a structure subjected to a dynamic load having a certain peak value may differ materially from that of the same structure acted upon by the corresponding static load of constant magnitude equal to the peak value of the dynamic load.

For the behavior of a structure under dynamic loading is no longer defined solely by considerations of external loading and of elasticity, but is greatly affected by the element of inertia.

When we introduce the inertia of the system, we find that, depending on the ratio of the duration of the pulse of the load to the fundamental period of free vibration of the structure to which it is applied, the strains induced in the structure may be either over-estimated or under-estimated considerably if the dynamic nature of the load is not recognized.

A suitable system of design for dynamic loadings is to determine the dynamic load factor for the specific case and to replace the dynamic loading by an equivalent static loading. If this is not done, material waste may result or serious risk may incur.

Determination of the proper dynamic load factor to use in design requires only a knowledge of the character of the external loading and of the natural period of vibration of the structures. In practice, these are either known or can be easily obtained by calculation or from experimental records.

I have expounded such a system of design in a paper titled "Dynamic Strength" reported in the Journal of the Society of Naval Engineers for November 1948. Because in this paper I have plagiarized the work of other authors, it is probably worth reading. I recommend it to you.

M. C. Junger, Barry Corporation:

I should like to ask Mr. Blake a practical question on the method of calculating stresses. The consideration of the higher modes depend not only on the frequency ratio of the shock pulse to the fundamental mode of the structure, but also the amount of damping in the structure itself; otherwise, for a very short shock pulse you get an infinite stress because even the very higher harmonics would be excited. The amount of damping present is now known. How do you decide on what harmonics are significant? That is, what is the highest mode that you must consider in order to get significant results?

R. E. Blake, NRL:

As Mr. St. Dennis pointed out, we have to draw the line some place as to how many of these effects we are going to try to take into account. To take into account damping, you need to use a still more complicated theory. From the theory as it applies to damped structures, I think that one could more or less arbitrarily cut-off several of the higher modes. Again you are taking a chance on the design, but apparently we have to take a great many chances to reduce the design method to something that can be done in a reasonable time in the drafting room. The higher modes will produce an infinite stress if you do not also take into account shear deflections when the pulse duration is very short. For a velocity change shock, one might stop taking modes into account above the third or

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possibly the fourth. This is rather arbitrary. In one case, I know, it was safe to ignore modes above the third.

Walsh:

In the paper by Mr. St. Denis, he stated that time is a necessary dimension of stress with which I quite agree. He also makes the assumption that only the first mode need be considered. I am not willing to go along with him on that. I think that even though time is of the essence that one cannot safely say that first modes alone shall be considered and in the methods discussed by Blake, it is perfectly possible to get the contribution due to the higher modes. Also, one with a little skill in examining these shock spectra can quickly decide whether there are higher frequency components present in the excitation and then make up his mind whether he should calculate the stresses due to these higher modes. Higher modes, in general, are not important in ships.

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PROPERTIES OF MATERIALS UNDER HIGH RATES OF STRAIN

By
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The methods and factors involved in testing materials under high rates of strain are discussed with particular emphasis being given to the tension impact test. The concept of "critical velocity" is reviewed. Dynamic properties of the more common engineering materials and their significance in shock applications are presented.

In this atomic age of supersonic speed where it is possible to propel oneself across the entire United States in a matter of a few hundred minutes, or around the world in less than 100 hours, it becomes mandatory that we evaluate our engineering materials in terms of a velocity parameter. However we, who are surviving this age of mach numbers of five and better, are not the first by any means to ponder the problem of what happens to a material when it is suddenly stopped or accelerated. The problem of impact is an age old question but it has remained for the 20th century engineers, metallurgists, and physicists to so define and analyze their scope of observations that general understanding of impact phenomenon may result. However there still remains the very significant problem of translating our experimental laboratory observations into simplified design criteria for full scale machines and structures.

It is the purpose of this paper to present a general picture of what happens to the properties of materials under conditions of impact or high rates of strain. Because of the generalized treatment of the subject, some of the more complicated topics are necessarily overly simplified. It is hoped that this paper will serve as an introduction to this topic for the designer, in other words give him some "round numbers" or principles with which to proceed. Those persons interested in pursuing the subject to the "third decimal place" are referred to the many comprehensive and enlightening articles appearing in the Journals of the ASTM, ASM, Applied Mechanics Division of ASME, AIME, etc. A few selected references are included with this paper. Particularly recommended are the two Symposia on Impact Testing sponsored by the ASTM and reported in their Proceedings, Vol. 22 (1922) and Vol. 38 (1938).

What is the significance of the impact tests? McAdam and Clyne (1)* have aptly stated:

*Numbers in parenthesis refer to references at end of paper.

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"An impact test should not be viewed as a simulation of shock in service, and should not be limited to metals that are subjected to impact in service. The object of an impact test should be to determine whether a structure or machine would be in danger of failure from brittleness under service conditions."

With this in mind let us now consider briefly the essentials of an impact test, viz. the specimen, the energy producing system, and the energy measuring system.

In general, the impact test consists of rupturing a specimen by the kinetic energy of a moving mass - the energy of rupture being determinable from either the energy producing system or its manifestations in the specimen. These are the Charpy-, Izod-, and Tension-impact. The specimens used in these tests are illustrated in Figure 1. The Charpy specimen is broken as a simply supported beam, the Izod specimen as a cantilevered beam, and the Tension specimen by a simple tension loading. Any of these specimens may be "notched," about which more will be said later. Suffice it to state here that the form of the notch is a significant variable in the testing technique.

The energy producing system may be of the swinging pendulum type as shown in Figure 2, which is typical of most commercial machines; a rotating flywheel type with protruding horns as illustrated in Figure 3, and as used, with certain modifications, by Nadai, Mann, von Karman and others; a vertically falling mass, such as a drop tester, as exemplified by the Guillotine Impact Machine, Materials Laboratory, New York, and a long horizontal spring

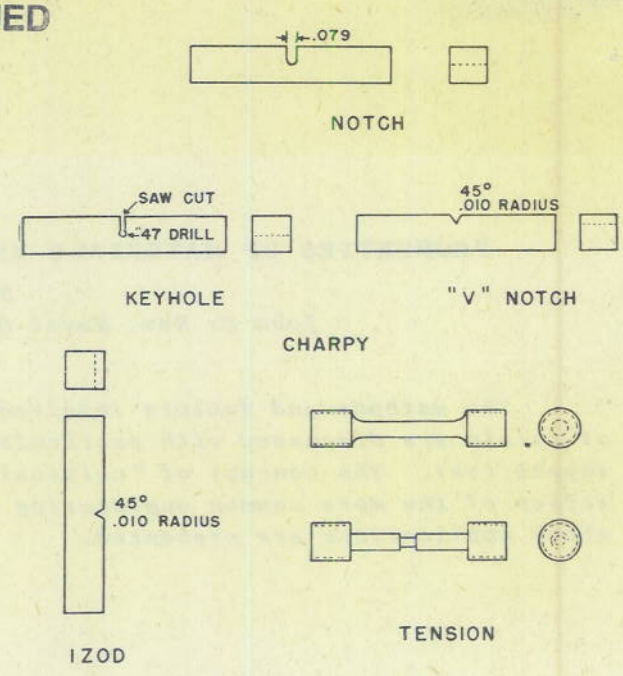

Figure 1- Impact test specimens

loaded rod as used by the University of Cambridge, England. Not to be excluded are those impact tests in which the specimen itself is the moving mass. Examples of this type are the incremental drop tests, such as is used for plastic telephone equipment.

The energy measuring system is a question of proper and adequate instrumentation - the details of which are not within the scope of the present paper. The reader may find these details among the references given.

The velocity of impact in the commercial pendulum machines is generally 11 or 17 ft/sec., whereas the rotating flywheel machines may have a variable velocity up to 1,000 ft/sec. The Materials Laboratory Guillotine is capable of velocities up to approximately 200 ft/sec.

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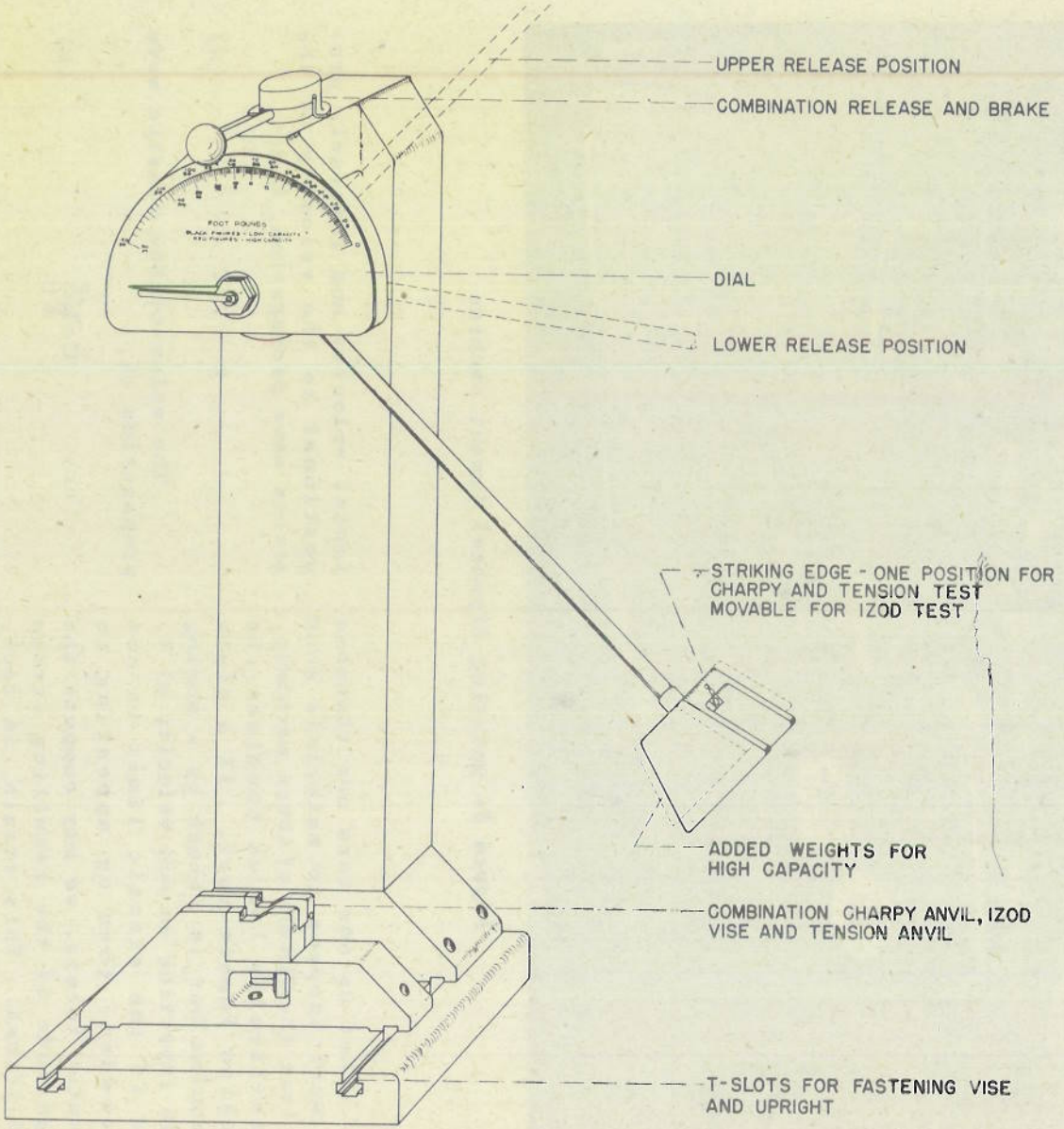

Figure 1
STRIKING EDGE

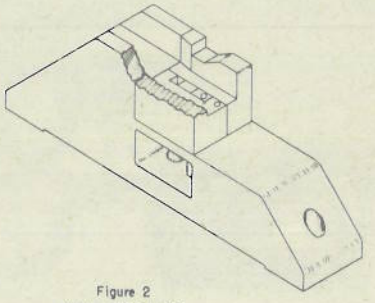

Figure 2
ANVIL - VISE UNIT

OVERALL DIMENSIONS
Width - 20 inches
Depth - 19 inches
Height - 50 inches

FLOOR SPACE REQUIRED
20 x 72 inches

Figure 2- Charpy-, izod-, tension-impact testing machine for metals

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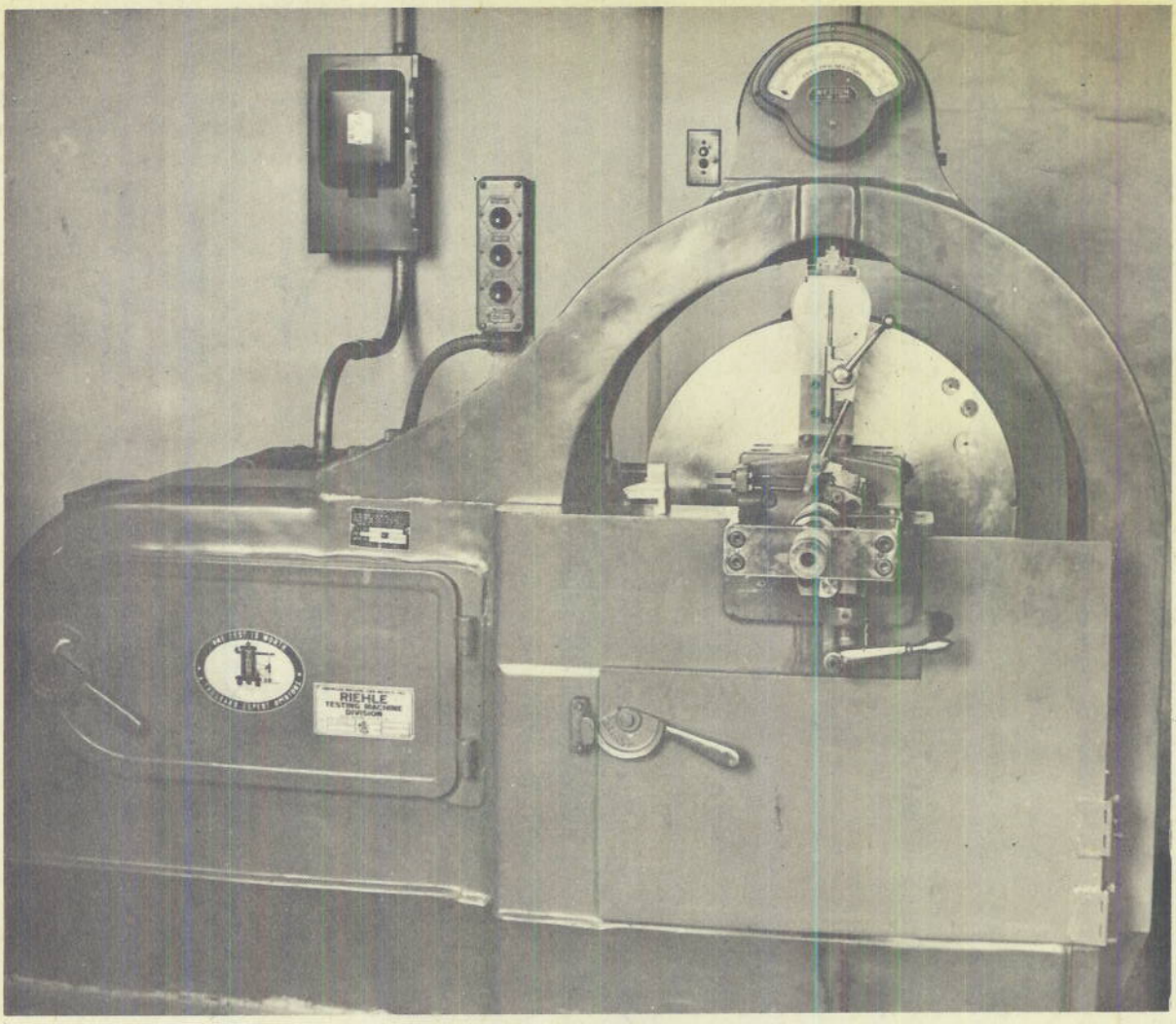

Figure 3- Rotating flywheel impact machine

Let us now turn our thoughts to what happens to materials when we test them in one of these machines. Specifically let us consider the tension impact test. If a plain circular bar is struck by a moving mass imparting an end velocity of V, and if the elastic limit is not exceeded, then by appealing to elastic theory, we may compute the magnitude of the resulting strain and stress. This strain, in fact, is directly proportional to the impact velocity and inversely proportional to the velocity of the stress wave propagation, C or

$$\varepsilon = V/C \qquad (a)$$

The value of the elastic wave propagation is,

$$C = \sqrt{\frac{E}{\rho}} \qquad (b)$$

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where, E, is Young's modulus and, ρ, is the mass density. However, if the bar deforms plastically the previous discussion is invalidated. However extensions of this theory to the plastic state have been advanced by von Karman (2) whereby high velocity impacts may be studied. Although the analytical expressions become cumbersome, a rational basis for the concept of "critical velocity" may be stated.

Simplified, the von Karman theory relates the velocity of the plastic wave propagation, V_o, to the slope of the static stress-strain curve above the elastic limit and to the mass density, or

$$V_o = \sqrt{\frac{\partial \sigma}{\partial \epsilon}} \Big/ \rho \qquad (c)$$

Furthermore, it is shown that the impact velocity, V_c, which causes abrupt failure is,

$$V_c = \int_o^{\epsilon} V_o(\epsilon)\, d\epsilon \qquad (d)$$

Equation (d) is an analytical expression for the "critical velocity" of a material. Theoretically it can be evaluated if the engineering stress-strain curve of the material is given. The upper limit of integration, ϵ, is the strain at which necking just starts in the tension specimen. V_o, is defined by equation (b) below the elastic limit and by equation (c) above the elastic limit and is a function of the strain.

The existence of this critical velocity has been corroborated by several investigators (3). It is manifested by greatly reduced values of energy, absorption, and elongation. This characteristic of a material has been noted by other investigators, noteworthy of which is the work of H. C. Mann at Watertown Arsenal which predated the von Karman theory (4).

The velocity effect on the properties of materials is well illustrated by the data of Clark and Datwyler (5) given in Table 1. Note the consistent increase in the properties of these materials (up to 294%) when tested dynamically. Noteworthy is the drop in energy values of 18-8 and 17-ST alloys of 40% and 12% respectively. It is surmised that this is due to the critical velocity for these materials being less than 11 ft/sec. Although critical velocities have been reported for carbon steels as 30-50 ft/sec, Si-bronze as 100 ft/sec, and annealed copper and aluminum 175-200 ft/sec, these values still need experimental confirmation before being used as design values.

This same general increase in the properties of materials has been noted by Nadai and Manjoine (6) and other investigators. This general effect of strain rate is summarized graphically in Figure 4.

The significant manifestation of an impact failure is the brittle fracture which results. The question that then arises is "what causes materials to act in a brittle fashion?". The answer is not simple - many factors are involved. One of these is temperature. While a material might show a high impact energy at room temperature, this may be drastically reduced, resulting in a brittle fracture, at low temperatures. The general impact-temperature effect is shown in Figure 5. This same general trend with temperature is true for other properties of materials. It is so

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TABLE 1

Ratio of Dynamic to Static Properties at Impact Velocity of 11 ft/sec.

Material	Yield Point	Max. Load	Elongation	Red. of Area	Energy
S.A.E. 6140[1]		1.020	2.205	2.940	2.865
S.A.E. 1015[2]	1.328	1.285	1.418	1.055	1.376
S.A.E. 1018	1.995	1.397	0.992	0.946	0.964
18-8 Alloy	1.224	1.212	0.682	0.782	0.600
17S-T	1.294	1.094	1.000	1.035	0.885
Brass[2]	1.387	1.142	1.163	1.136	1.203
Aluminum	1.522	1.323	1.628	1.093	1.627
Copper	1.433	1.390	1.527	0.986	1.597

[1] Oil quenched from 1620 F.

[2] Cold-rolled.

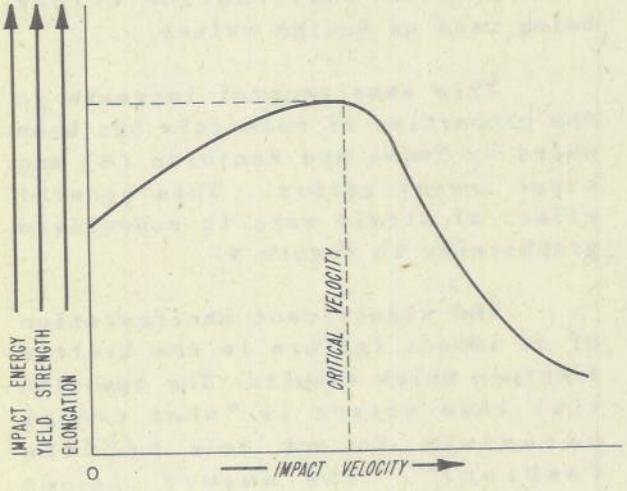

Figure 4- Properties of materials at various strain rates

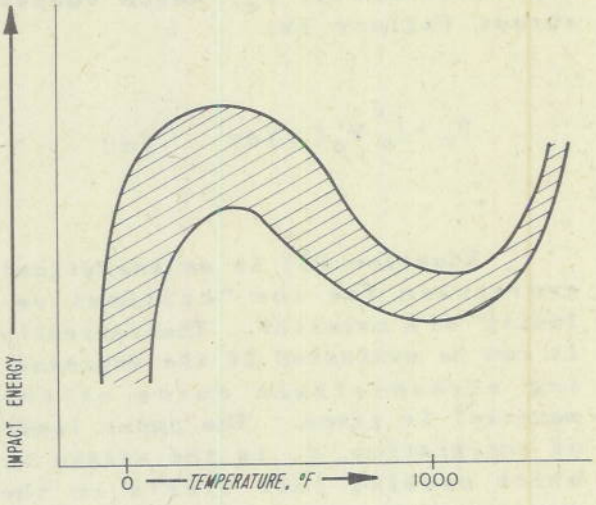

Figure 5- Impact temperature relation for ferrous metals

pronounced that it cannot be overlooked in design.

Let us now examine this question of ductility and brittleness more closely. Ludwik has given a hypothesis by which the mechanism of plastic deformation can be explained. This has been more recently discussed by McAdams (1), Hoyt (7), and others.

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In this hypothesis we appeal to the concept of the cohesive strength of a material. Cohesive strength relates to the true tensile strength of the material. It is the binding force that holds the material together. Studies of plastic deformation have shown that the process is one of slip along certain crystallographic planes. This slipping is caused by shear stresses. Every material possesses a certain resistance to this deformation. Thus we have an inherent cohesive strength to resistance of deformation ratio in any given material.

Consider now the specimens shown in Figure 6 which are of the same material and the same metallurgical condition. Assume that the CS/RD ratio is 2.5. Under a tensile loading, the tensile-shear stress ratio in the cylindrical bar is 2:1. Since this is less than the CS/RD ratio, the shearing stresses will overcome the resistance to deformation before the cohesive strength is reached and the material deforms plastically. However, in the grooved specimen assume that the tensile-shear ratio in this case is 3:1. It may be greater depending upon the sharpness of the notch. Then the cohesive strength of the material is exceeded before there can be any sliding action or deformation and the material fails with a brittle fracture. It is to be noted here that it is the specimen by virtue of the notch, not the material, that is brittle. In practice, this notch effect may be aggravated by residual stresses, inclusions, hair-line cracks, and other metallurgical considerations. Furthermore, the ratio of cohesive strength to resistance of deformation may change with temperature, plastic working, and rate of straining.

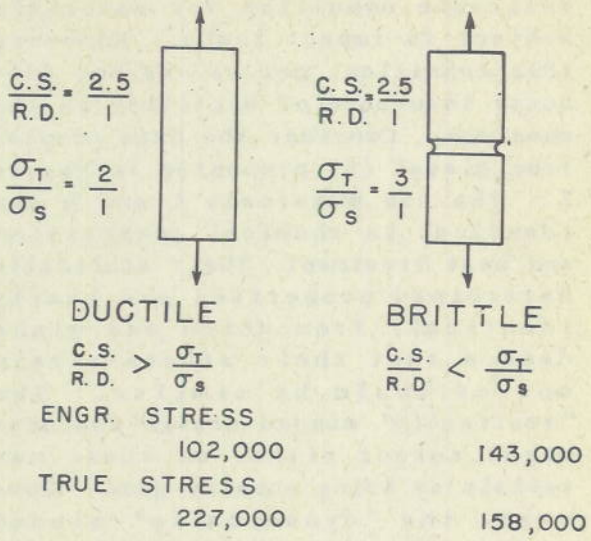

Figure 6- Effect of form on type of fracture

Unfortunately, one is not able to arrive at a simple quantitative value for the cohesive strength and resistance to deformation ratio, therefore, we seek some other parameter to gauge the strength of our materials and their notch sensitivity. Several hypotheses have been postulated. These relate to the total area under the stress-strain curve, to reduction in area, to elongation, etc. In commenting on the significance of elongation and reduction in area, Gillett (8) of Battelle Memorial Institute has this to say:

"It seems necessary to conclude that brittleness under conditions of stress concentration either under impact or repeated stress is not necessarily avoided by the presence of even very generous amounts of tensile ductility."

To be sure, a large total area under the stress-strain curve is a

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desirable condition for materials subject to impact loads. However, this condition, per se, is not adequate insurance of ductility in the specimen. Consider the data adapted from Riegel (9) presented in Figure 7. The two materials A and B are identical in chemical composition and heat treatment. Their statically determined properties are nearly identical, from which one might deduce that their stress-strain curves would be similar. The "statically" minded design engineer might select either of these materials as being equally good. However, the "dynamically" minded designer notes from the impact test that material A is good and material B is poor. Thus one high rate of loading test (notched-bar impact) gives a merit index for the material and a guide to its notch sensitivity.

As a concluding remark on notch effects and as a dictum to soothe the design engineers, the following statement of F. B. Seeley (1) is quoted:

"To what extent sharp notches create stress peaks and polyaxial tension is not definitely known; in other words, the mechanics of stresses at notches is to a large extent an unexplored field of knowledge, but

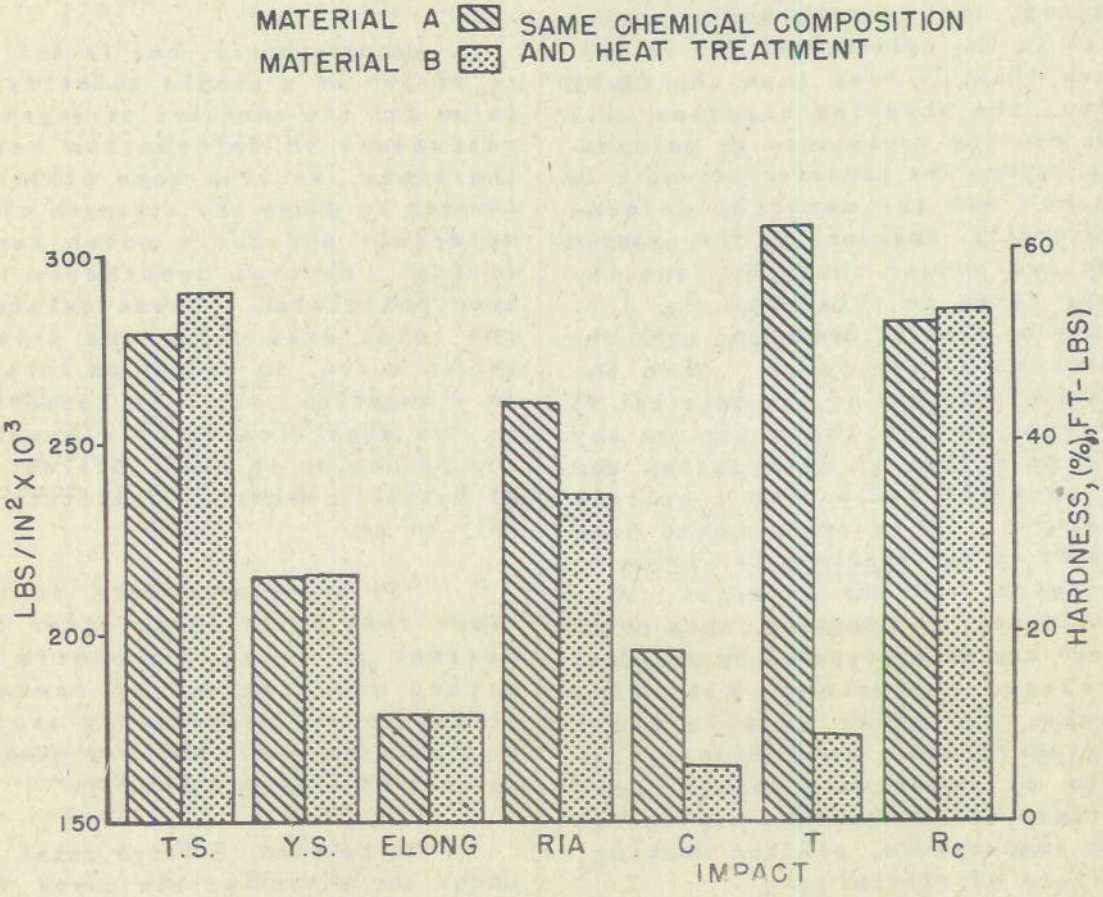

Figure 7- Relative quality of metals by impact test

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since so-called ductile materials often fail at notches, there is evidence to support the idea that the state of stress may change the mode of failure from (ductile) yielding to (brittle) fracture."

In summarizing, "what are the properties of a material which are significant in a design to resist energy loads?", for a given energy load, we can reduce the force produced by increasing the space through which it acts. Thus decreasing the cross-sectional area or increasing the gauge-length are steps in the right direction. Classical examples of this procedure are reducing the shank diameter of bolts, or drilling a hole through the center. As someone has said, "make the piece weaker where it does not break."

Considering elastic action for the moment, recall from mechanics that the resilience of a material is equal to the square of the stress divided by Young's modulus. Therefore, a material with a high elastic strength is desirable; if we double the elastic limit the resilience is quadrupled. Materials with low elastic modulii are not to be overlooked. Plastics, although notoriously poor in our standard impact tests, have yet to be evaluated under higher rates of strain. More information is needed on our light alloys such as Mg and Al and our newer materials such as Titanium.

Stresses flow through a piece like water, and consequently exert high pressures at abrupt changes in section, sharp corners, notches, etc. Thus streamlined designs have not only an esthetic value but also a functional one as well. In design, where stress raisers cannot be avoided, select materials with a low notch sensitivity as may be determined by impact tests.

Remember that most properties of materials decrease with temperature but increase with rate of loading up to a critical velocity, thereafter diminishing rapidly. The critical velocity indicates the maximum safe rate of loading. The effect of temperature on the properties of materials cannot be overemphasized. Davidenkoff (11) has even suggested that we incorporate a "temperature brittleness factor" in our design.

To date there is no single or simple relationship which relates the dynamic properties of materials to shock designs. There is a wealth of information that has been collected; it has yet to be correlated. Therefore, the shock absorber in our present designs must be a relatively large "factor of ignorance" sometimes called a factor of safety.

DISCUSSION

G.R. Irwin, NRL:

Is it not true that most brittle fractures of rods, bars, etc., under dynamically applied tensile loads occur in ways that are not described by the von Karman theory of OSRD Report 365 (1942)? White and Griffis, OSRD Report 742 (1942) furnishes a brief but fairly clear statement of the limited applicability of this theory to brittle fracturing. This point was also discussed by the writer in his paper "Fracture Dynamics" at the

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October 1947 ASM Seminar on Fracturing of metals. It seems that in addition to the influences upon development and propagation of a crack of such things as temperature, strain rate, nucleation of yielding, strain aging, relaxation, etc., one must consider the influence of specimen inertia in the case of very fast tensile testing. In 1942 von Karman and others discussed the theory of this feature separately in terms of propagation of plastic waves. Under the assumption that the effects of testing velocity upon the load deformation curve are comparatively small, one sees that it is indeed possible to apply a high enough test velocity so that necking of the test piece must occur in a vanishingly small length of specimen. However, it is possible for the energy required for tensile impact fracturing to suffer a quick drop with increasing test velocity for any given material at velocities below or above the critical velocity predicted by the plastic wave theory. This is because of influences we know to exist which are not within the scope treated by that theory.

J. C. New, NOL:

Dr. Irwin's comments have most aptly indicated some of the limitations of the von Karman theory. As originally stated by the author, this theory still lacks sufficient experimental verification to permit its use as a design criterion. Its presentation in this paper has been overly simplified to achieve brevity. The von Karman theory should not be thought of as a panacea for all high-velocity tests. However, it is one of several tools with which one may approach the problem of dynamically applied tensile loads. Within our present scope of knowledge, it is certainly true "that failure and the relative embrittlement at failure will depend not only on the impact velocity, but on the rapidity of loading" as stated by White and Griffis. This in turn depends upon the specimen inertia as pointed out by Dr. Irwin.

Klein:

Bearing on this subject, it is of interest to quote some introductory remarks concerning fatigue. The statement is from a lecture on "Fatigue of Metals" delivered at NRL on 2 March 1949 by Dr. Alfred M. Freudenthal, visiting lecturer from Columbia University.

"Fatigue research suffers from the engineering idea that metals are homogeneous, isotropic materials which respond elastically. This doctrine, emphasized in conventional elementary, intermediate and advanced engineering courses, constitutes a sterile and infertile background for approaches toward an understanding of fatigue, damping, and creep. Fatigue studies have long suffered too, for lack of distinction between the phenomena represented by the initial, falling portion and the final, horizontal portion of the curve showing stress against number of cycles, of between (a) the finite life portion, and (b) the indefinitely long endurance."

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PROTECTION AGAINST SHOCK AND VIBRATION THROUGH THE USE OF CASTING RESINS AND CEMENTS

By

P. J. Franklin, National Bureau of Standards

A discussion is presented of past experience in the use of casting resins and cements for potting and otherwise binding electronic circuit elements to afford protection against the effects of shock and vibration. Mention is made of methods which have been evolved in the National Bureau of Standards laboratories.

In the production of electronic equipment for use in devices which must withstand great stress or shocks, such as ordnance devices, and particularly in electronic devices to be included as parts of missiles, it is necessary that mechanical protection be afforded the delicate components.

Perhaps it would be well to have a brief description of the ways in which some of these devices are assembled. For the purpose of this paper, there are two major classes of assembly: (1) An assembly which uses a chassis made of insulating material, and (2) an assembly in which no chassis is used.

In the first case, this laboratory was concerned with the production of an assembly to be mounted on a chassis made of mica-filled phenolic. This chassis was a small plate with indents to accept the component parts, which were anchored by soldering to metal inserts molded in the chassis. After the soldering operation, it was, of course, necessary further to fix the components relative to the chassis block and to each other. In order to accomplish this, a styrene cement was used.

This type of cement was chosen because of the necessity for minimizing the distributed capacity, and styrene had the lowest dielectric constant of any of the suitable materials then available. Also, such cements were commercially available and had been used in industry; therefore no new production techniques had to be learned.

Temperature cycling tests with this styrene cement over a period of time showed that it possessed several disadvantages. Because of the difference in coefficients of expansion, the cement had a tendency to lift. This showed up immediately in our measurements of electrical noise, for the components vibrated relative to each other and to the chassis, thereby causing changes in the distributed capacity. When the cemented components were examined by a probe, the cement appeared to be strongly bonded to the components and to the chassis. It was obvious, however, that the apparent bonding was

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incomplete, for when the chassis block material was changed, the noise disappeared. Further investigation revealed that the adhesion was even less satisfactory than first tests indicated. When the cement was analyzed more carefully, it was found that the solvent had not been completely released. Further tests were conducted in which complete release of the solvent was obtained. As a result, the cement peeled freely from the phenolic chassis, much as dry skin peels off an onion. Sand-blasting or otherwise roughening the surface of the chassis block improved adhesion somewhat, but not sufficiently to make it satisfactory.

In order to establish a better bond between the cement and chassis, the chassis block was changed from a mica-filled phenolic material to a modified styrene. With a chassis of a material which could be dissolved and penetrated by the cement, a bond which was cohesive rather than adhesive was obtained. Thus the cement and the block became a single homogeneous unit containing the components. Because of the difficulty experienced in getting styrene cements to release their solvents in any reasonable length of time, tests were conducted with polyvinyl carbazole (a carcinogenic material, by the way). It was found that the solvents were completely removed from equivalent film thicknesses in about one-fourth to one-fifth of the time required for the styrene to release the solvent.

On the basis of homogeneity of cement and block, it was found that a phenolic cement for bonding components to the mica-filled phenolic block would also perform adequately. This was not used because of the lower dielectric constant of the styrene material as compared with that of the phenolic. The proportion of mica in the phenolic cement had to be matched to that in the block in order that the coefficient of expansion of the cement and of the block might be equal. When this coefficient of expansion was unequal, tearing took place.

This technique gave rise to the idea of further simplification by omitting the supporting chassis and potting the assembled components in a solid block of casting resin. Several factors required consideration in the choice of a suitable resin. The resin had, of course, to be mechanically strong in order to support the imbedded components under the strains of various types produced by high acceleration. The shock caused by these accelerations made high impact strength necessary.

The devices were to operate under conditions of high humidity and temperatures, so superior moisture resistance was required. Water, having a dielectric constant of approximately 80, would seriously affect the performance of a high frequency, high impedance circuit.

The devices which the casting resin was to protect utilized a high impedance, high frequency circuit; therefore low dielectric constant and low power factors were required in the frequency ranges to be used. This limited the resins to a certain few hydrocarbons. Another factor was that the shrinkage on polymerization had to be low to protect the components; similarly the coefficient of thermal expansion had to be low relative to most of the components.

Ideally, one should use a resin which matches the coefficient

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of expansion of the various components imbedded in it. These components include glass and metallic objects as well as resinous materials, such as the coatings on resistors.

Because of the desirability of minimizing the shrinkage on polymerization, a polymer was dissolved in the monomeric materials. While this did cut down on the shrinkage due to the polymerization, it increased the viscosity of the resulting solution. The amount of polymer that could be added to the monomer was limited by the tolerable viscosity of the solution, which had to be poured through a tiny orifice.

The importance of impact on the performance of the device necessitated the use of materials of high impact strength. Experience showed that impact strength as measured by Scharpy and Izod tests did not correlate well with the performance of the device. Further trials revealed that a drop test gave better correlation with performance than did the more formal tests. Scharpy and Izod tests were originally developed for testing the impact strength of steel and were adopted from the ASTM tests for those materials and applied to plastics. They are not suitable for the latter tests, for in such materials most of the strength exists in the surface fibres. The drop test used by the author consisted of throwing the test specimen out of a three-story window onto a concrete apron. Pieces which did not crack upon such test performed adequately in actual use. The NBS Casting Resin was one of the several which seldom cracked. Under such tests, the high velocity shock wave developed stresses through the material and cracked the glass of the tubes imbedded unless certain precautions were taken. In our first attempts to cushion the tube, we tried jacketing the tubes with several elastomeric materials without much success. Such materials transmitted the high-velocity shock wave as well as did the casting resin or other rigid materials. Eventually, study of the nature of the shock wave led us to the belief that only a gas would give sufficient shock absorbent cushioning to protect the tubes.

Most casting resins have their impact strength improved by incorporating various amounts of plasticizers. While these plasticizers decrease the tensile and compression strength of the material, improved impact performance outweighs these losses. In passing, we might add that resin authorities are in conflict as to the possibility of plasticizing styrene. Nevertheless, we have found that the incorporation of a plasticizer in our casting resin does improve its impact strength. Besides increasing the impact strength of the casting resin, the addition of the plasticizer decreased the shrinkage due to polymerization and reduced the viscosity of the syrup for pouring. The reduced shrinkage resulted in less likelihood of damage to, or displacement of, the components while the resin was setting.

It may be interesting to know the way in which we made a typical resin suitable for the potting of high-frequency, high-impedance electronic circuits which are to be operated under adverse conditions of humidity, temperature, and shock. This resin was first compounded to give best obtainable operating characteristics, but it was found impracticable from the standpoint of cost. Therefore, at

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a negligible sacrifice of heat distortion point, the formula was modified, resulting in a saving of between 30% and 40% of total cost. The formula of the modified casting resin is as follows:

Compounds	Amount by Weight (%)
2,5-dichlorostyrene	33.0
Poly 2,5-dichlorostyrene	21.5
Styrene Monomer (N99)	21.0
Polystyrene (Dow 411C)	11.0
Hydrogenated terphenyl (e.g., Monsanto HB-40)	13.0
Solution containing 40% divinylbenzene (e.g., Dow Q 302.4)	0.5

When completely processed, the resin presents the following characteristics:

Compounds	Amount by Weight (%)
Water absorption (24 hours immersion) %	less than 0.01
Volumetric shrinkage on polymerization %	8.0
Power factor (at 100 megacycles and 50% RH)	0.0004--0.0008
Dielectric constant (at 100 megacycles and 50% RH)	2.5
Dielectric strength (1/16 in. sample; volts/mil)	610-660
Volumetric resistivity (megohm-cm)	10^{17}

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POTTING ELECTRONIC COMPONENTS AS AN AID IN WITHSTANDING SHOCK AND VIBRATION

By
C. B. Brown, Naval Ordnance Laboratory

This paper describes techniques involved to protect vacuum tubes and other electronic components against the shock and vibration encountered in handling and firing from guns. Effect of temperature is discussed on both wax and plastic potting.

1. The techniques of potting to improve the ability of electronic components to withstand high accelerations and shock have been the subject of intensive investigation in connection with electronic devices fired from guns. As a result of these studies it is now possible to incorporate in gun-fired devices complex electronic circuits formerly considered too delicate for use in ordnance. While this paper deals largely with the application of potting to gun-fired devices, the techniques are almost universally applicable to other uses where high accelerations and high intensity shocks are encountered.

2. An examination of the nature of the forces involved in firing projectiles from guns shows that they can be classified in two general categories.

(a) The first due to longitudinal acceleration during propulsion through the bore lasts for .01 to .02 seconds taking place over a space of 10 to 30 feet. Peak accelerations of from 5000 to 20,000 G's as measured by crusher gauges are common.

(b) The second, vibration, both longitudinal and transverse, at frequencies ranging from a few thousand cycles/sec. to several hundred thousand cps arises from impact excitation of the projectile. Longitudinal vibration at the natural frequencies of the projectile are thought to be the result of compression of the projectile during passage through the bore and the sudden release of this force when the projectile emerges from the muzzle. Transverse vibrations of extremely high intensity and short duration arise as a result of "side slap" as the projectile bounces from side to side as it traverses the bore. Unfortunately very little is known as to the exact nature of these accelerations, but it is well known that their effects are much more pronounced in worn guns where opportunity for this type of excitation is increased. Shocks of the same nature are undoubtedly encountered in handling problems where the projectile is dropped on a hard surface such as the deck of a ship.

3. The clue to possible methods of protection of electronic components subjected to the acceleration described above lies in the nature of damage observed. In vacuum tubes of the most rugged construction, damage of the three types is found.

(a) Breakage of the glass envelope
(b) Breakage of the filament
(c) Distortion of the mount and supporting elements.

Damage of the latter type has been shown by high acceleration air gun experiments to be associated

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with the relatively long time longitudinal accelerations discussed in 2 (a) due to propulsion through the bore. It is immediately evident that no type of mounting can decrease these forces since they take place over many feet of travel. Hence, this type of damage must be accepted as unavoidable. The only remedy lies outside the scope of this paper in improved mechanical strength of the tube's internal structure.

Breakage of the glass envelope has been traced through air gun tests, largely to localized stress imposed by the tube mount. The inherent strength of glass envelopes properly mounted is shown by their ability to withstand breakage when subjected to centrifuge forces far greater than any encountered in ordnance applications. Breakage of filaments is thought to be almost entirely due to high intensity short duration shocks. This is borne out by the fact that drops of only a few feet onto a hard surface are sufficient to cause damage of this type while centrifuge tests indicate extreme strength.

4. Other electronic components such as condensers and resistors have very high natural resistance to the conditions encountered in firing from guns. The only problem here has been to protect against lead breakage. It is evident that the practice of supporting the component by its lead must be abandoned.

5. The principle damage observed to batteries has been in the nature of mechanical distortion, again attributed to long period longitudinal acceleration. This has resulted in open circuits and high resistance contacts both between cells and within the cells.

6. It has long been accepted that filling the voids in a packaged electronic device with some form of potting material will increase its resistance to shock and vibration. Potting should accomplish two aims: First, to furnish support for the components and, second, to cushion against high frequency shocks. The potting material must of necessity be introduced in a fluid or semi-fluid state and must not set up residual strains on setting. Waxes of various consistency have been found quite satisfactory over limited temperature ranges, i.e. (0°F-100°F). More recently plastic potting has been the subject of intense investigation in an effort to increase this range. Ideally, the potting material should have a high absorption coefficient for high frequency vibration.

7. Protecting vacuum tubes against envelope breakage by potting depends mainly on uniform distribution of the forces of acceleration on the glass envelope. One of the most successful methods is to freely float the tube by its leads and immerse it in a micro-crystalline wax whose melting point is above the highest temperatures to be encountered. Adequate flexibility of the tube leads must be made available to prevent cracking at the seals. An alternate method is to provide a flexible housing such as a rubber sock, but great care must be taken to insure uniform support within the housing. Either of these methods are capable of protecting the glass envelope adequately.

Protection against filament breakage does not appear to have been fully exploited in present techniques. It has been previously pointed out that this type of damage arises from high frequency vibration at or above the natural period

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of the filament. It should be possible to pot in a material having high absorption in this region. Experiments with foam rubber mixtures as well as spun fiber sheaths have been encouraging. Effort should be made to develop a high absorption, low acoustic velocity potting material. It might be noted that the micro-crystalline waxes presently used transmit vibration in the frequency region and hence do not offer protection against high frequency shock.

8. Although it is common practice to support resistors and condensers by their leads, this practice is detrimental to high shock resistance since the most common form of damage noticed is the breaking of lead connections. Flexible leads combined with support of the body of the resistor or condenser by potting solves this problem. The components should be oriented to place the major forces along the axis of the component to reduce bending.

9. Batteries, especially of the "Minimax" stack variety used for high potential plate supplies are not rigid structures. They can be improved by mechanical preloading axially with the stack and potting in a rigid material. Casting plastics have been used with considerable success but polyethelene in pressure presses appears to have production possibilities.

10. The most annoying difficulty encountered in the application of potting waxes is the limited temperature range of these materials. They become brittle below -20°F and crack up when subjected to high acceleration causing local stresses which result in broken components. Above 120°F they become increasingly fluid and do not offer support. The resulting component movement is a frequent cause of broken leads. As a result of these deficiencies, intensive work has been done on plastic substitutes. While the field has not yet been exhaustively studied, several polyethelenes have been used with success. These plastics have superior temperature ranges and it is hoped that potting materials having the desired acoustic properties mentioned in paragraph 7 will be obtained.

11. The technique used to study the results of various pottings and placement of components especially when using plastics consisted in the making of micrometric X-ray photographs. A long focal length Westinghouse 220KV rig was used together with an accurate indexing system of jigs in which photos were made before and after subjecting the packaged electronic device to shock. These films could then be superimposed and inspected under an illuminator. Movements as small as a few thousandths of an inch could be detected as well as voids in the potting material, and internal damage to the components. While some time has to be spent in obtaining the right exposure density, the scheme is recommended as a valuable tool in the study of potting technique.

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HIGH "G" DESIGN

By

T. E. Dinsmoor, Naval Ordnance Laboratory

A brief discussion of certain mechanical design considerations applicable to mechanisms of ordnance equipment which must be subjected to accelerations in the ranges of 0 to 100 g's, 100 to 1000 g's, and 1000 to 15,000 g's. This field covers devices weighing several hundred pounds which take accelerations as high as 400 g's as well as small units capable of withstanding 15,000 g's.

INTRODUCTION

Although almost all forms of useful devices are at times subjected to high "g" shock loading, this is a field in which little was done prior to World War II and one that is currently given only inadequate attention by many designers.

To illustrate the significance of the effects of a shock load as compared to those of a normal acceleration, consider a man sitting in a wheel chair in the center of a railroad car aisle. If, when the train starts, the man holds the wheels of his chair from turning, he will accelerate with the railroad car and in four or five seconds will have been subjected to some finite velocity change equal to that of the train. If, however, the man had chosen not to hold the wheels of his chair as the train started, he would soon find himself hard against the partition at the aft end of the car. In an equivalent period of time he would have only been subjected to the same total velocity change as in the preceding case, but in the interim he would probably have broken a number of bones.

A shock loading results whenever a force is suddenly applied to a body for a short time. It causes high accelerations of the body to which it is applied and seldom results in a simple harmonic displacement. In the example above, the impact of the wheel chair and the man against the partition very definitely represents a shock load.

An automobile traveling along a bumpy road, an aircraft-dropped mine, or a modern artillery shell illustrates the everyday employment of mechanisms subjected to shock loading causing accelerations which may be many times that of gravity. Among these examples, only the magnitude and the duration of the accelerations differ.

CLASSIFICATIONS OF SHOCK FAILURES

Designers in this field are confronted with failures occurring generally from any one or a combination of the following:

1) Physical strengths may be exceeded (either elastic or ultimate).

2) Interior impact phenomena may be occurring.

3) High accelerations occur and the duration of the shock pulse is long.

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Failures of the first type are more easily recognized. However, failures occurring from interior impacts are often neither anticipated nor recognized.

This type of phenomenon usually occurs by a momentum transferred to various elements within the shocked structure which, in turn, resonate at various natural frequencies. This may result in the collision of the various parts.

An example of a design failure involving the second type listed above occurred in an aerial-type mine in which mechanisms were suspended in the mine case by commercial-type shock mounts. Decelerations encountered on a water-entry velocity of 900 ft/sec of the mine were of such a character that the shock mounts bottomed, and the accelerations resulting from striking the stops were more damaging than the water-entry accelerations.

The design of a system which eliminated these failures is shown by Figure 1. In this design two mechanisms, "A" and "B," in cylindrical containers are positioned and supported in the mine case by the rubber rings. The rubber rings are compressed by bolting the clamping plates until a rigid suspension of the mechanisms is attained.

This design eliminated any possibility of interior impacts and simultaneously subjected the mechanisms to impact shocks of time

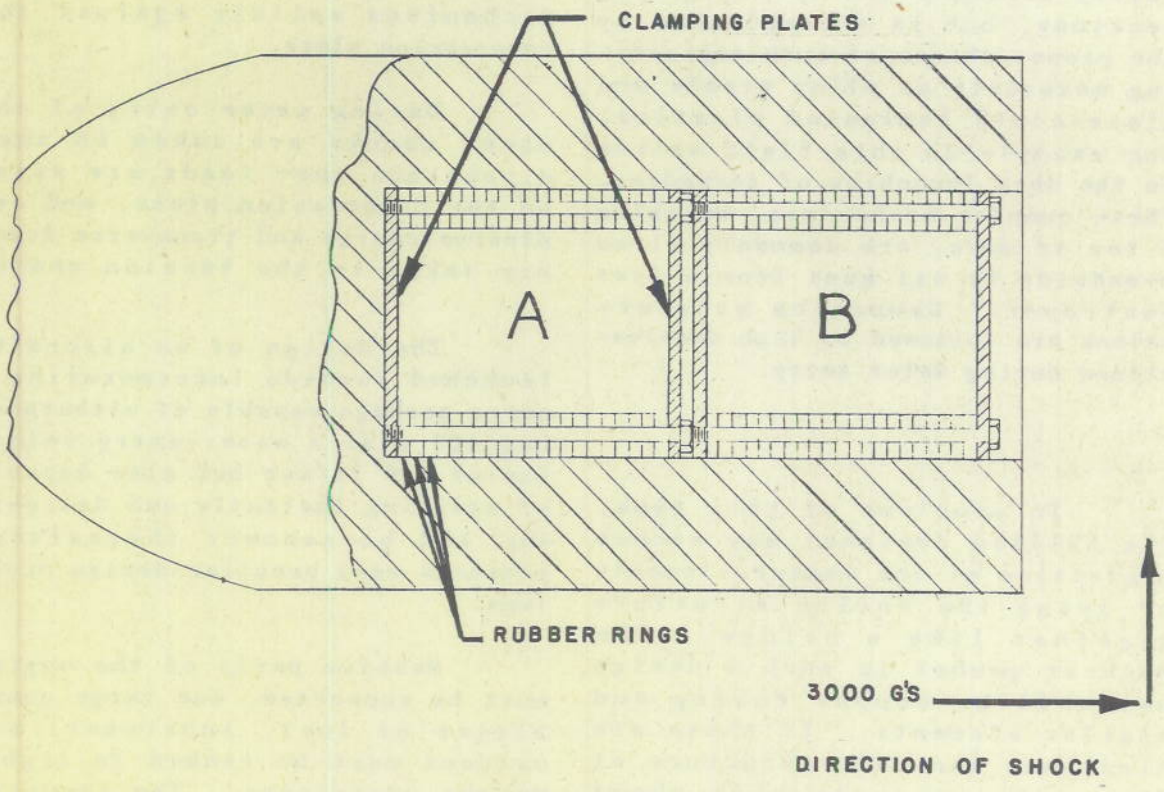

Figure 1

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durations comparable to those received by the mine case during the water-entry. As a result, physical strengths are not exceeded.

DESIGNS FOR MEDIUM SHOCK

In the more common fields, accelerations are encountered which seldom exceed 100 g. In dealing with accelerations of this order, considerable experience has been built up, and small systems may be designed with confidence that they will resist accelerations of this magnitude. A few guiding principles employed by the skilled designer in approaching problems of this type will be illustrated.

Since a high strength-to-weight ratio is desirable, the solution to such problems is not solely accomplished by increasing sections, but is accomplished by the proper choice of such engineering materials as alloy steels and glass-cloth laminated plastics. One example in this field arises in the deck launching of torpedoes. These complex mechanisms, weighing a ton or more, are commonly fired broadside by air guns from a fast destroyer. Launching accelerations are followed by high decelerations during water entry.

In problems of this type, the skilled designer may secure protection of the weaker elements by tying the entire structure together like a bridge. The weakest member in such a design may well be copper tubing and similar elements. If these are tied into the main structure at frequent intervals with short straps, very high accelerations can be resisted.

Two other factors which must be recognized are the direction of the maximum acceleration which, in the case of the torpedo, usually lies close to the axial direction and the extent of travel which must be provided in the several directions to prevent interior impacts.

Where accelerations may be confined to on-axis loads, the designer can absorb forces in simple tension or compression members, and the difficulty of beam loadings of complex systems may be avoided, as illustrated by another aerial mine design in Figure 2. In this design the mechanisms "A," "B," and "C," etc., are secured to a "T" frame which, in turn, is positioned in the mine case by the tie rods. This assembly, when installed, places the forward mechanisms solidly against the compression plate.

During water entry of the mine, shocks are taken in such directions that loads are axial on the compression plate, and explosive charge and transverse loads are taken in the tension member.

The design of an aircraft-launched torpedo incorporating a power package capable of withstanding 400 g at a water-entry velocity of 600 ft/sec but also capable of starting instantly and delivering 100 horsepower thereafter, presents many peculiar design problems.

Massive parts of the engine must be supported, and large quantities of fuel, lubricant, and oxidant must be tanked in lightweight containers. The securing of this equipment in itself is a problem.

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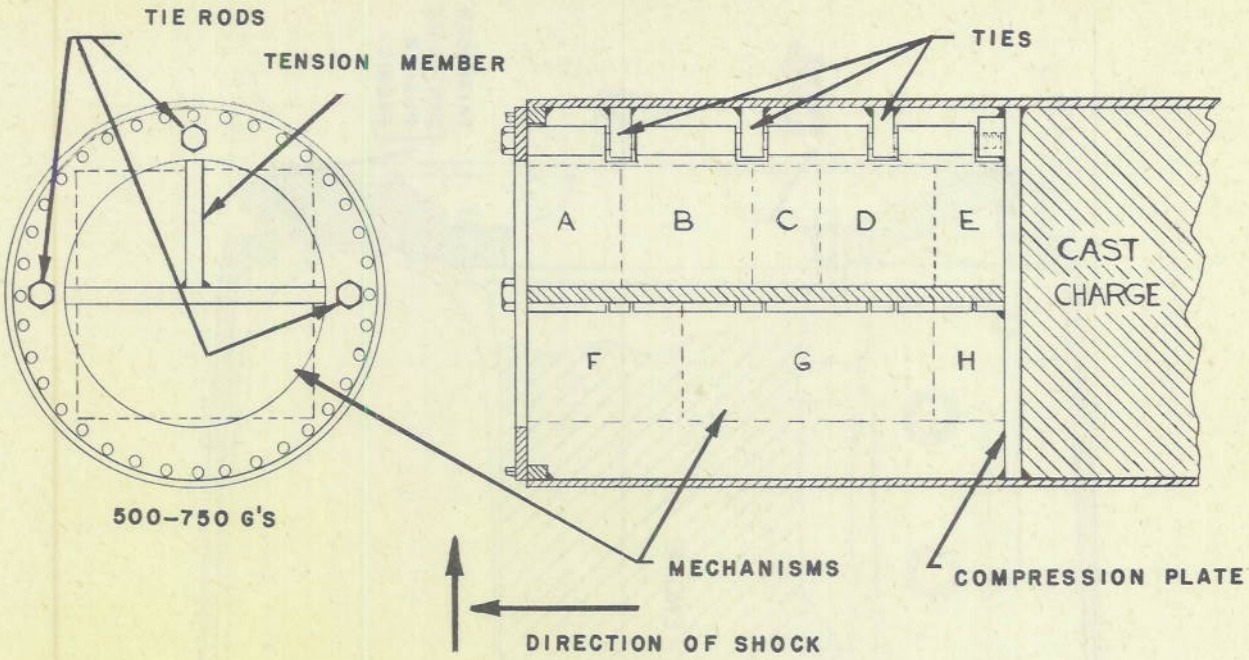

Figure 2

Figure 3 is an outline drawing of the power-plant section of the torpedo in the final form. The designers were able to employ the thick aluminum hull section as the main structural element in the design.

The cylinder block chosen for this engine also possessed inherent strength and great stiffness. Hence, the use of the cylinder block as a secondary supporting structure was indicated.

In this design, the designers chose to tie these two structural members together through the use of a textile-filled elastomer diaphragm. The one finally selected possessed moderate stiffness in an axial direction, but was very stiff to radial loads.

In this unit, the direction of maximum deceleration forces is limited to a cone of 15 degrees on either side of the axis. The missile itself will fail at water entry at steeper angles.

Since the major moving element is the crankshaft and its associated connecting rods and pistons, a study was made of the forces which would be placed on these parts during water entry.

To this end, the mock-up illustrated in Figure 4 was subjected to repeated simulated water-entry shocks in a large air gun and was subsequently operated under simulated running loads at rated speeds of rotation.

Ball-type antifriction bearings were selected for the main bearings by earlier designers. In view of the nature of the grain flow resulting from forging, the fatigue life of the ball-type bearings is less predictable than that of the roller-type bearings.

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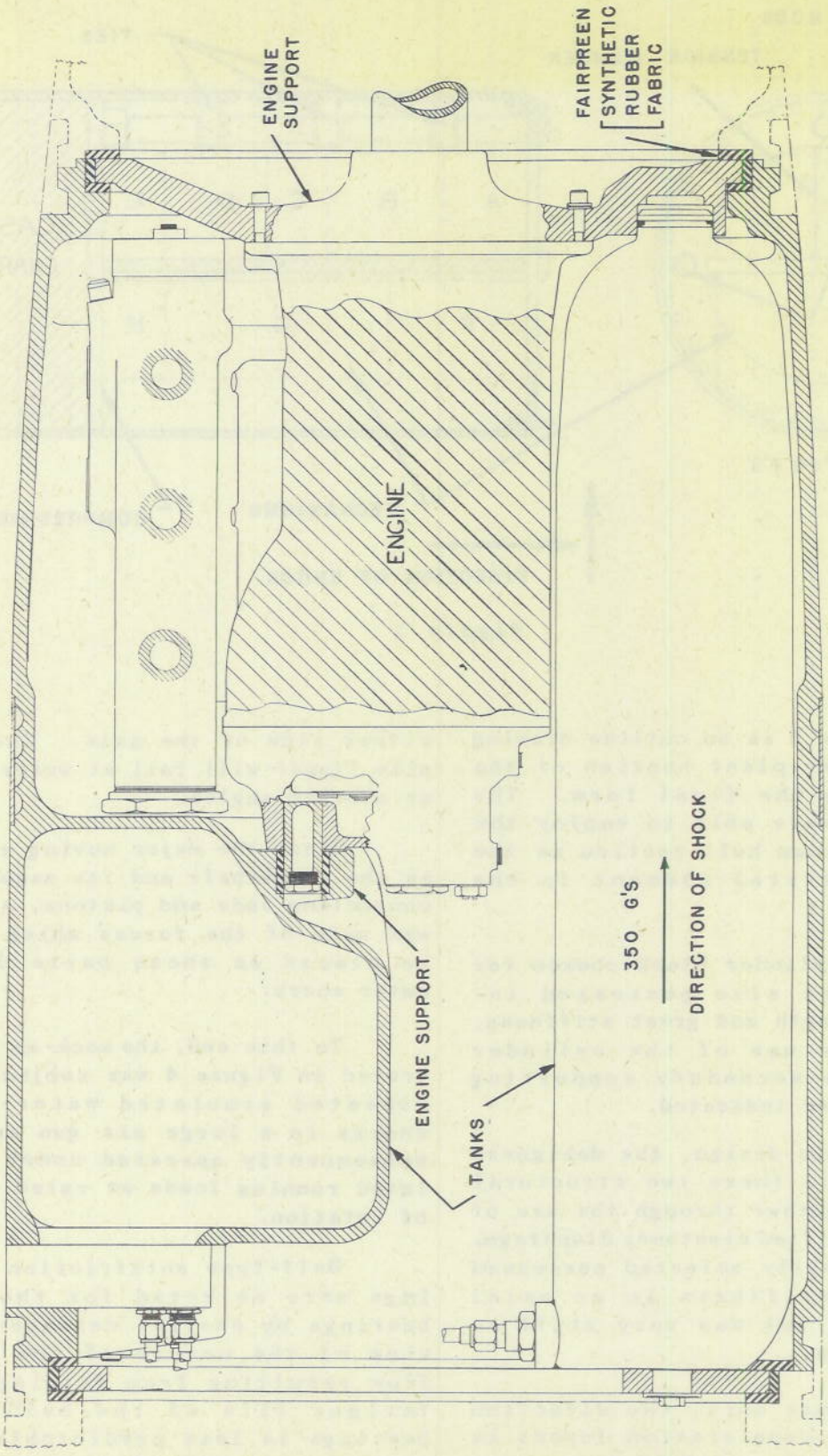

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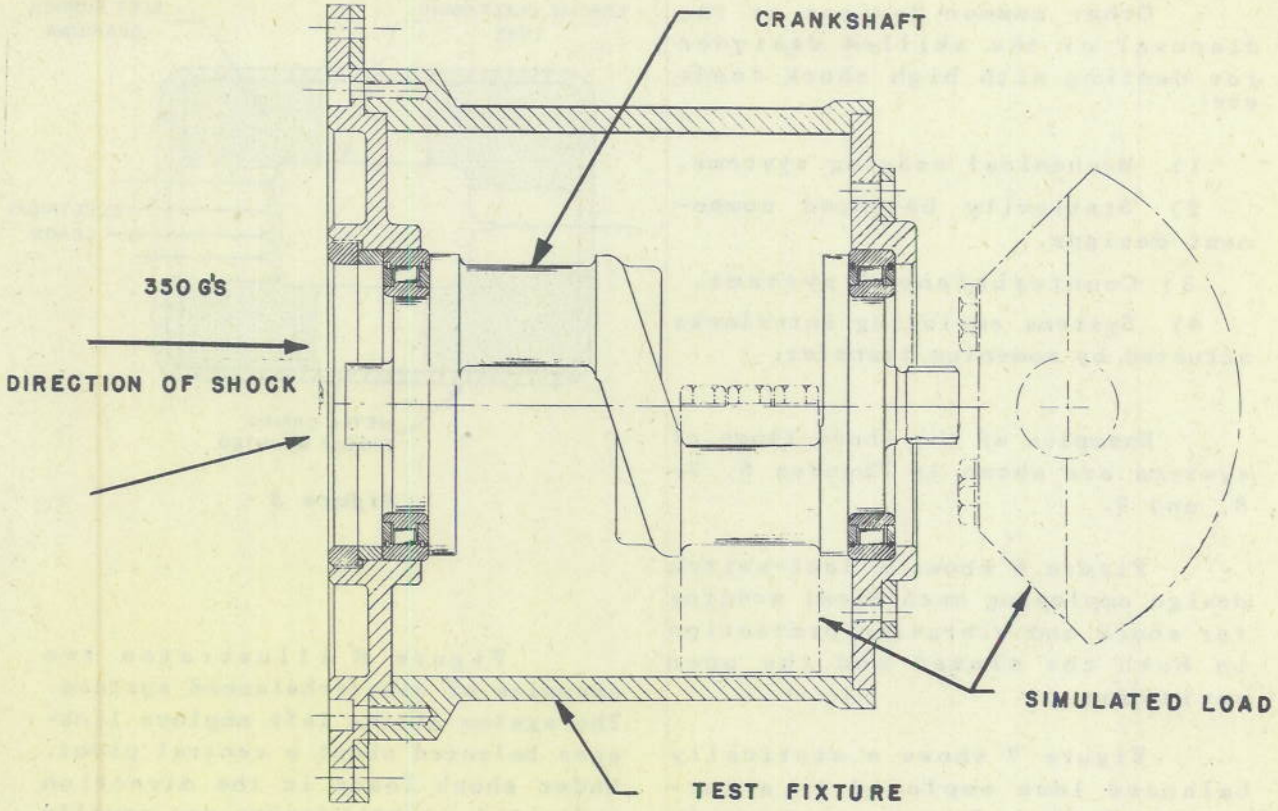

Figure 4

These roller-type bearings also appear more desirable for applications requiring large loads which result from angle entry. The major thrust loads in this design are taken by conventional thrust bearings.

Despite the fact that sea water was allowed to be mixed with the lubricating oil, the design employing stainless-steel rollers has been subjected to simulated water-entry shots at 7-degree angles followed by corrosion fatigue tests with satisfactory results.

DESIGNS FOR HIGH SHOCK

Although the use of shock mounts presents a number of special problems, shock mounts are employed to advantage in some cases. One of the best examples is that of complex electronic components which could probably be designed to resist shocks of this magnitude without damage to the components, but which will introduce spurious signals or noise into the circuits during acceleration. Fairly soft shock mounts have been successfully employed under such conditions.

Figure 5 illustrates a rather soft shock mount designed to eliminate the effects of shock and vibration such as might be set up in the service use of an aerial-dropped mine. This is a common means of dealing with accelerations of considerable magnitude.

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Other common devices at the disposal of the skilled designer for dealing with high shock loads are:

1) Mechanical wedging systems.

2) Statically balanced component designs.

3) Counterbalanced systems.

4) Systems employing interlocks actuated by momentum transfer.

Examples of the above types of systems are shown in Figures 6, 7, 8, and 9.

Figure 6 shows a leaf-switch design employing mechanical wedging for shock and vibration protection in both the closed and the open positions.

Figure 7 shows a statically balanced lock employed in a torpedo exploder. It was necessary to resort to the odd shape in order to provide the operational features desired combined with the static balance.

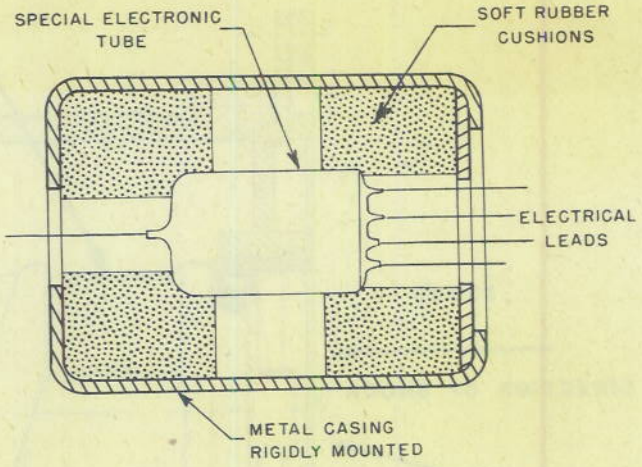

Figure 5

Figure 8 illustrates two examples of counterbalanced systems. The system at the left employs linkages balanced about a central pivot. Under shock loads in the direction indicated, inertia forces are equally opposed. The system on the right is a simple cable-and-pulley arrangement wherein the masses M_1 and M_2 are equal. Obviously, under shock

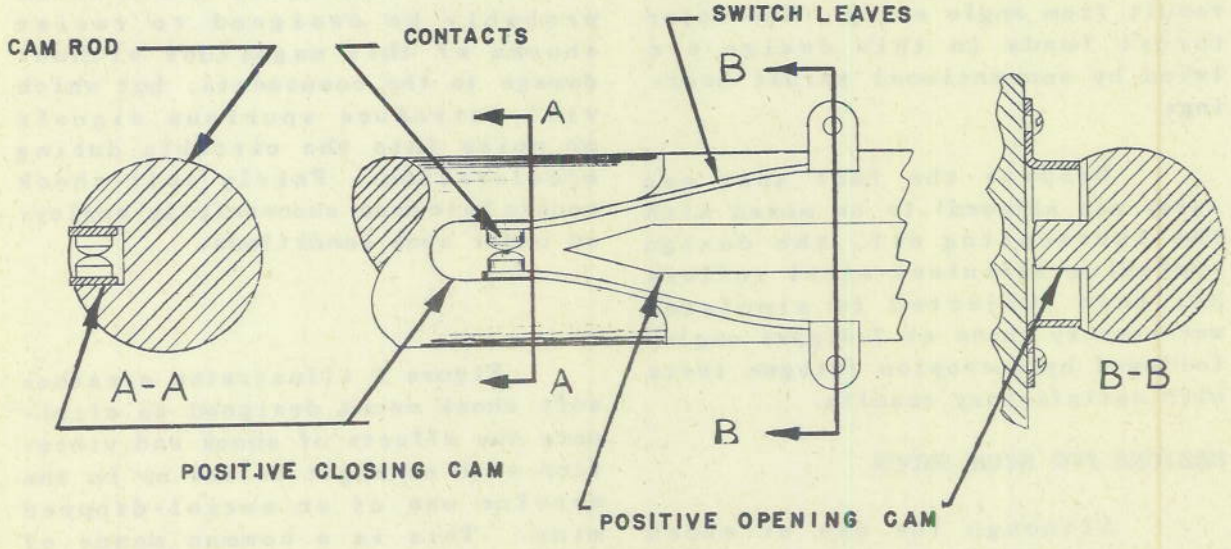

Figure 6

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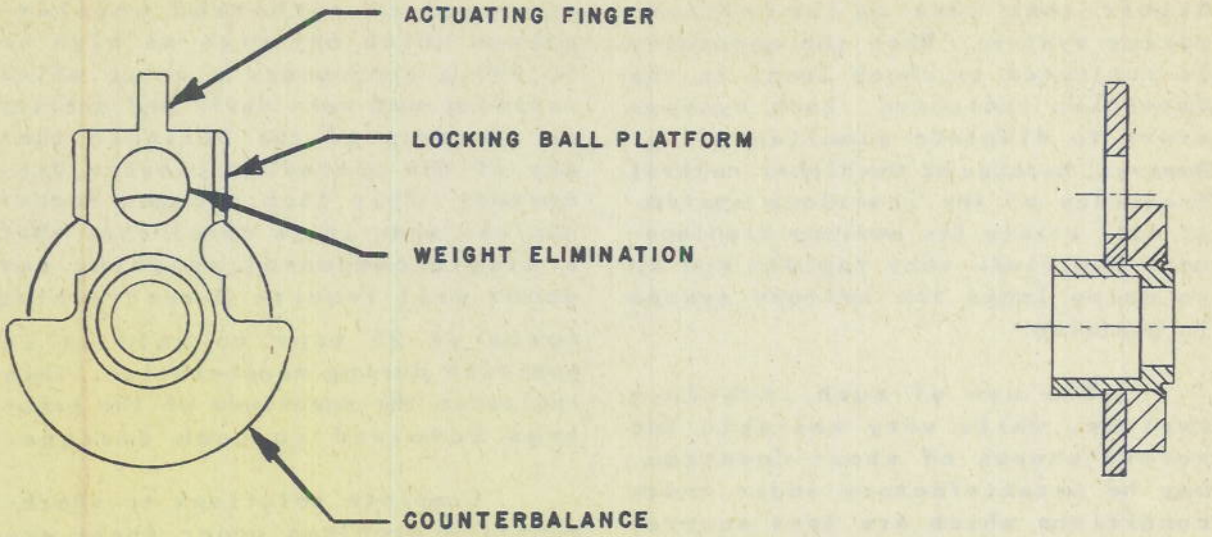

Figure 7

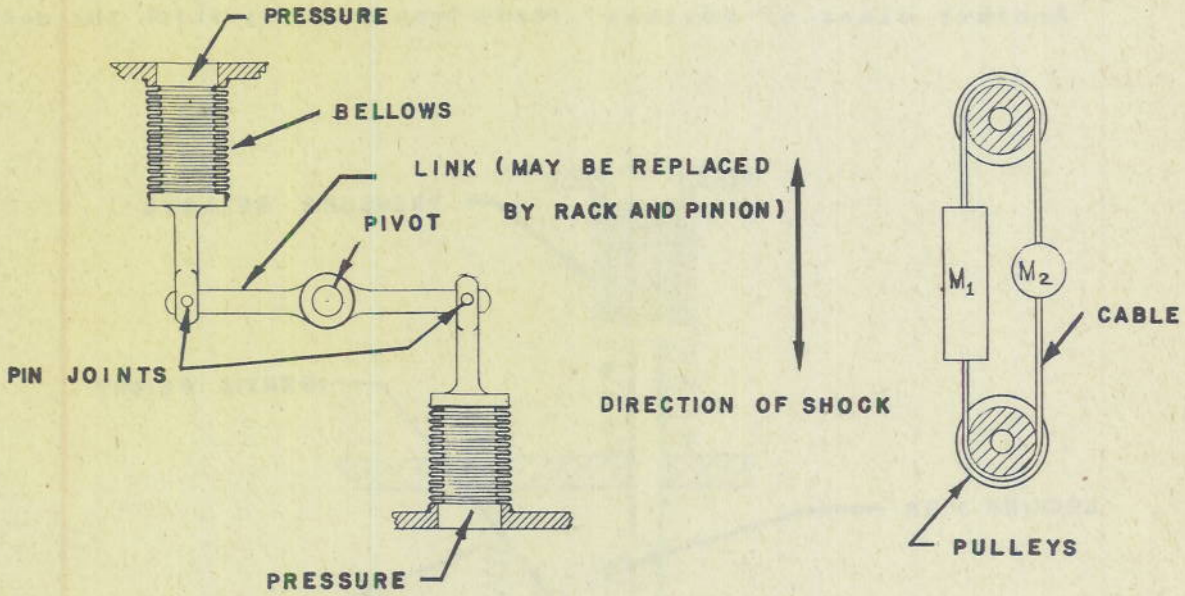

Figure 8

loads in the direction indicated, the system will not be subject to any resultant accelerations.

Figure 9 illustrates a typical system employing an interlock actuated by momentum transfer. In this particular application the designer has utilized the difference in the natural frequencies of two systems in order to obtain the desired shock proofing. The natural frequency of the pivoted inertia-weight-spring system is considerably

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higher than that of the bellows spring system. When the mechanism is subjected to shock loads in the direction indicated, both systems start to displace simultaneously. However, because of the higher natural frequency of the interlock system, it will attain its maximum displacement amplitude very rapidly and in so doing locks the bellows system in position.

The use of such interlock systems, while very reliable for severe shocks of short duration, may be unsatisfactory under shock conditions which are less severe. This results as the entire mechanism may be accelerated without actuation of the interlocking system.

DESIGNS FOR VERY HIGH SHOCK

Another class of devices designed to withstand accelerations which may rise as high as 50,000 g represents a group which calls for much more skill and ability on the part of the designer than any of the preceding classes discussed. This fact becomes rather obvious when it is considered that a single component weighing one pound will require a restraining force of 25 tons to hold it in position during acceleration. This indicates the magnitude of the problems involved in such designs.

Complete solutions to shock-mounting problems under these accelerations are practically impossible theoretically, and this approach would also be very uneconomical. In spite of difficulties, however, means have already been found by which the most

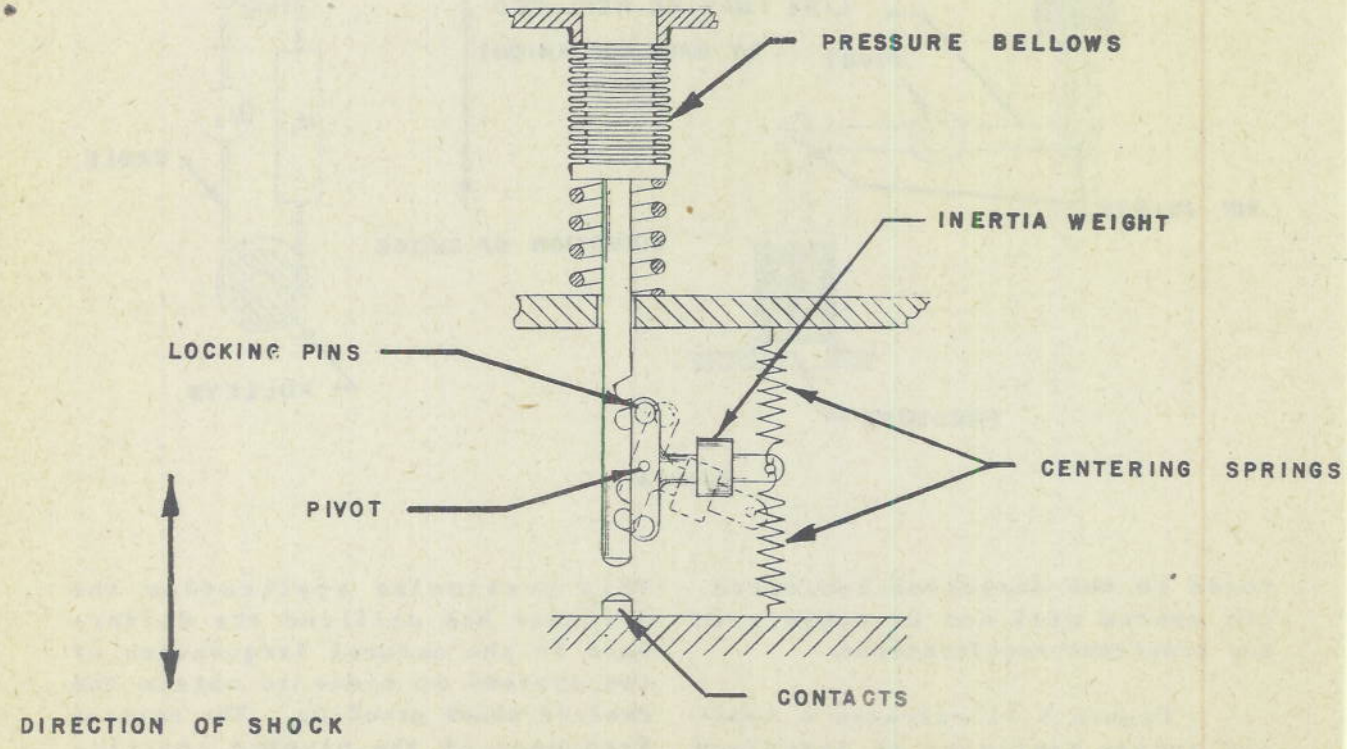

Figure 9

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unlikely components such as vacuum tubes, clocks, and other complex instruments have successfully withstood accelerations of this order, if they are skillfully designed. The proper approach is generally a combination of theory, experiment, and an invaluable experience on the part of the designer.

Consider a case where an extremely sensitive pendulum is mounted in a shell. Low-friction pivots supporting such a mass would ordinarily be damaged by the accelerating forces. However, if an automatic device is constructed which will serve to cage or clamp the pendulous mass during the peak of the acceleration, the greatest difficulty will have been surmounted.

Figure 10 illustrates one form of automatic caging device. In this design, the rather massive rotor is allowed to move during acceleration until the supporting surface has been encountered. The

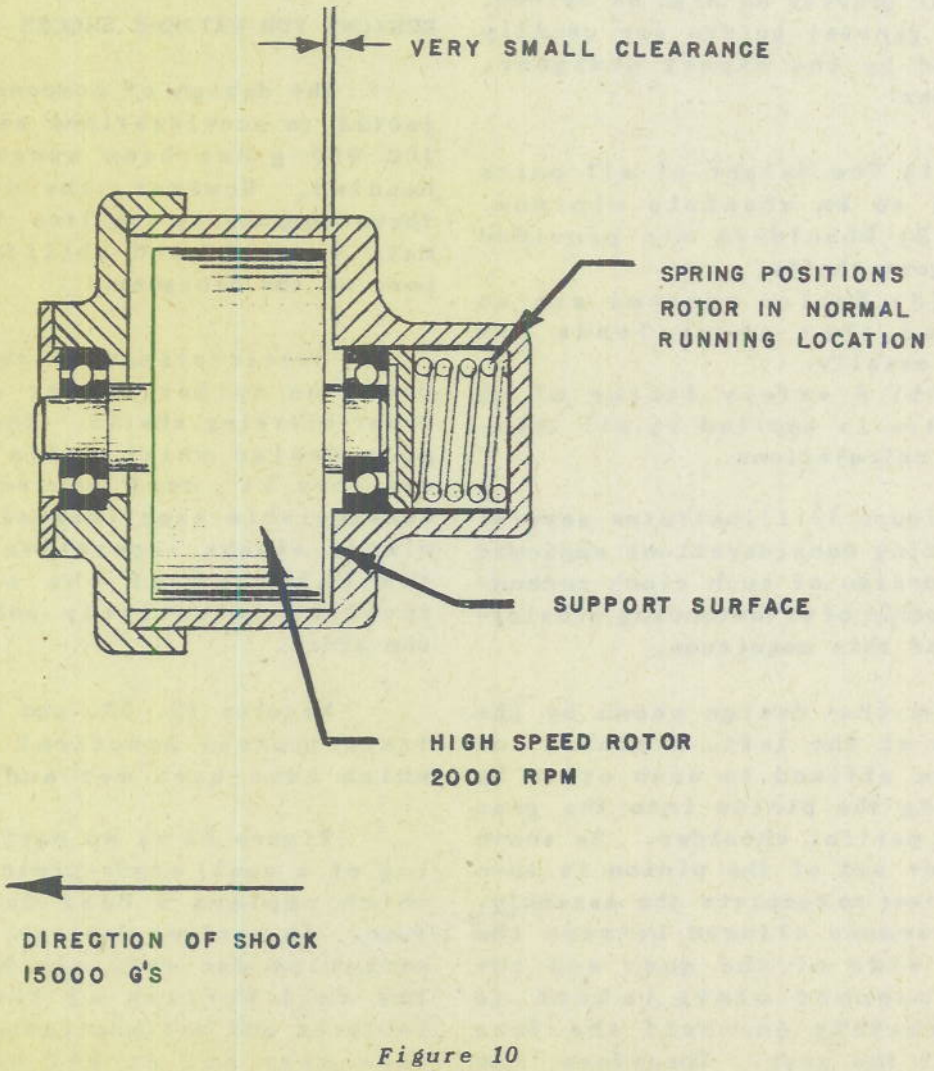

Figure 10

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delicate bearings are mounted to permit axial travel of the rotor shaft, and since acceleration is only important in a single direction, protection of the bearings is assured.

At first glance, it would appear that difficulty would be experienced as a result of bouncing of the rotor mass from the anvil. Actual firings at accelerations of gravity as high as 15,000 have demonstrated the suitability of this design.

In the design of clock mechanisms which must withstand accelerations of gravity as high as 40,000, several general guides are usually employed by the expert designer, as follows:

(1) The weight of all parts is kept to an absolute minimum.

(2) Shoulders are provided on all gear shafts.

(3) Entire systems are so designed that shock loads are applied axially.

(4) A safety factor of at least five is applied to all theoretical calculations.

Figure 11 illustrates several interesting considerations employed in the design of such clock mechanisms capable of withstanding accelerations of this magnitude.

In the design shown by the section at the left, a pinion and gear are affixed to each other by broaching the pinion into the gear up to a partial shoulder. As shown the lower end of the pinion is then peened over to complete the assembly. The clearance allowed between the bottom side of the gear and the lower support plate is kept to approximately one-half the face width of the gear. Therefore, the driving reliability of the arrangement is guaranteed even if the gear slips axially during accelerations.

Another interesting design feature is shown by the section at the right. Here the shaft is subjected to a transverse load in combination with the axial load during acceleration (the transverse load is applied to the cut-back section of the shaft). It is not sufficient merely to design adequate shaft sections to support the shock load, but it is essential that bearing lengths be made as long as possible.

DESIGNS FOR EXTREME SHOCKS

The design of components subjected to accelerations as high as 100,000 g has been successfully handled. However, the design of these components requires the ultimate in skill and ability on the part of the designer.

Accelerations of this order occur on target impact of small armor-piercing shells. The success of a design which falls in this category is most unpredictable. Considerable experimental work is almost always imperative because the behavior of the elements involved is entirely unknown at the start.

Figures 12, 13, and 14 illustrate typical practical problems which have been met and solved.

Figure 12 is an outline drawing of a small armor-piercing shell which employs a base detonating fuze. In earlier designs, the fuze mechanism was entirely housed in the tail portion of the shell. Failures and malfunctioning of the fuze were not at all uncommon.

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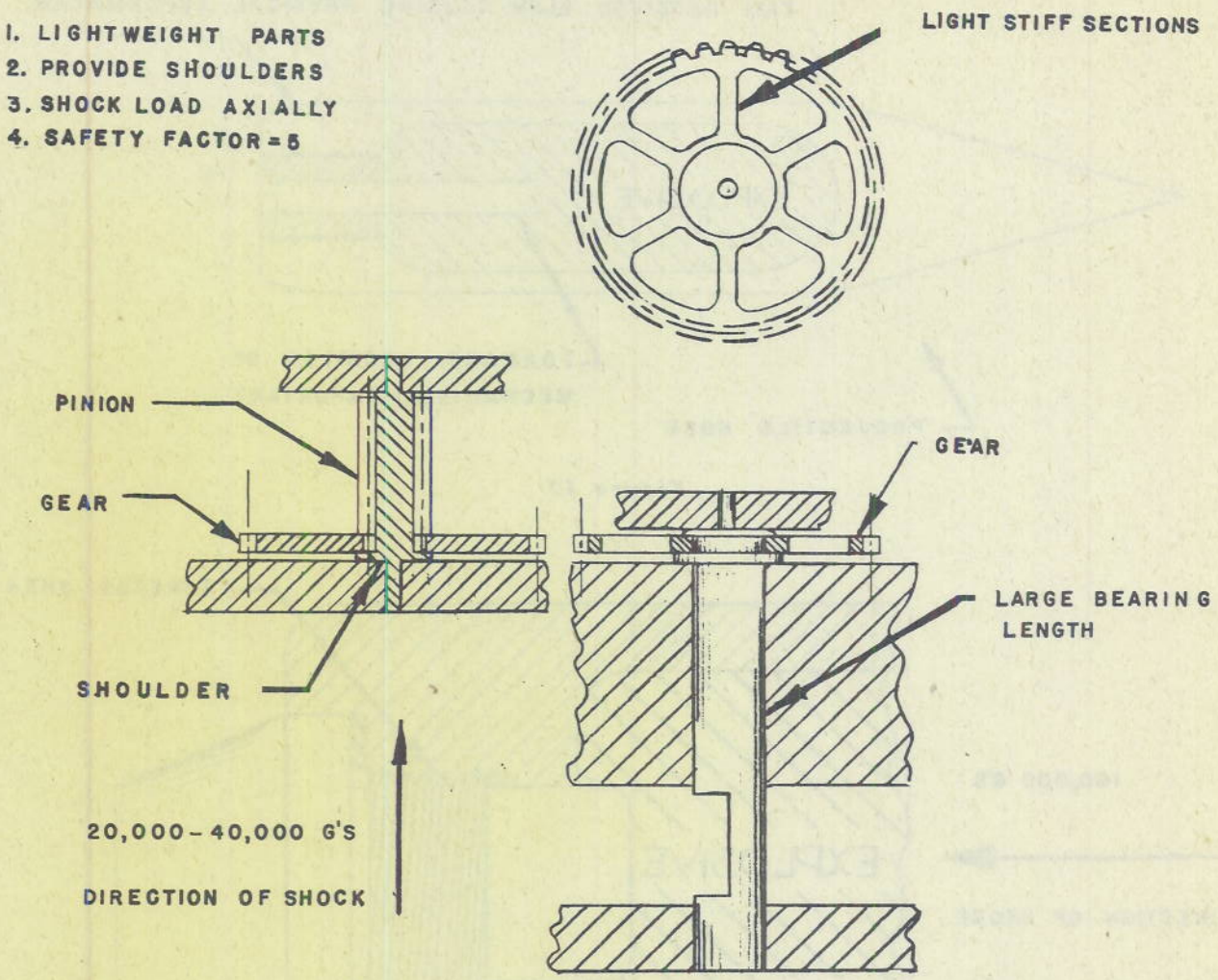

Figure 11

A careful study of experimental firings revealed that on angle-target impacts the shell would pivot slightly about its nose in such a way as to crush the tail portion of the shell as it proceeded through the armor plate. Consequently, this physical deformation of the shell tail would crush the fuze mechanism. The difficulty was readily solved by locating the functioning element of the fuze far forward, as shown by the figure.

Figure 13 illustrates an example of poor design practice for devices which must withstand shock loads of this order. The copper gasket, used to seal the assembly, acted as a soft spring under shock load. As a result, the threads used to secure the base detonating fuze in the shell were subjected to gigantic loads, and shear failure occurred. This type of failure led to low-order detonations, since the entire fuze would be expelled from the shell

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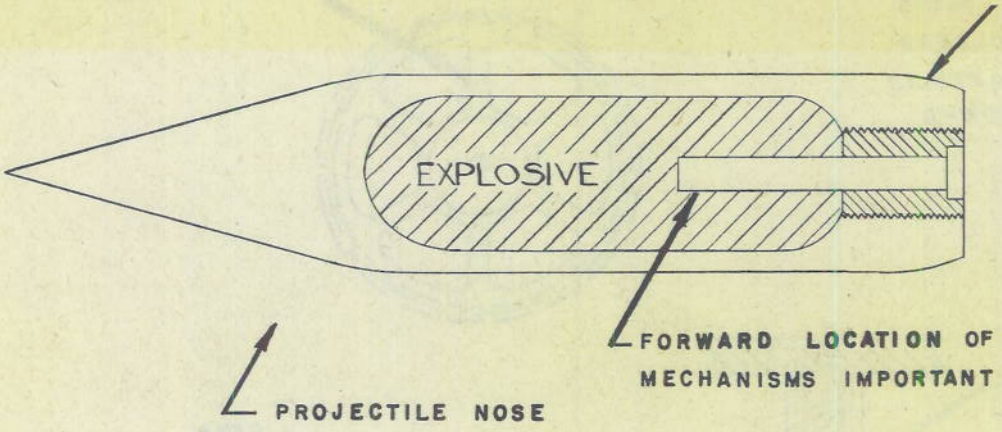

Figure 12

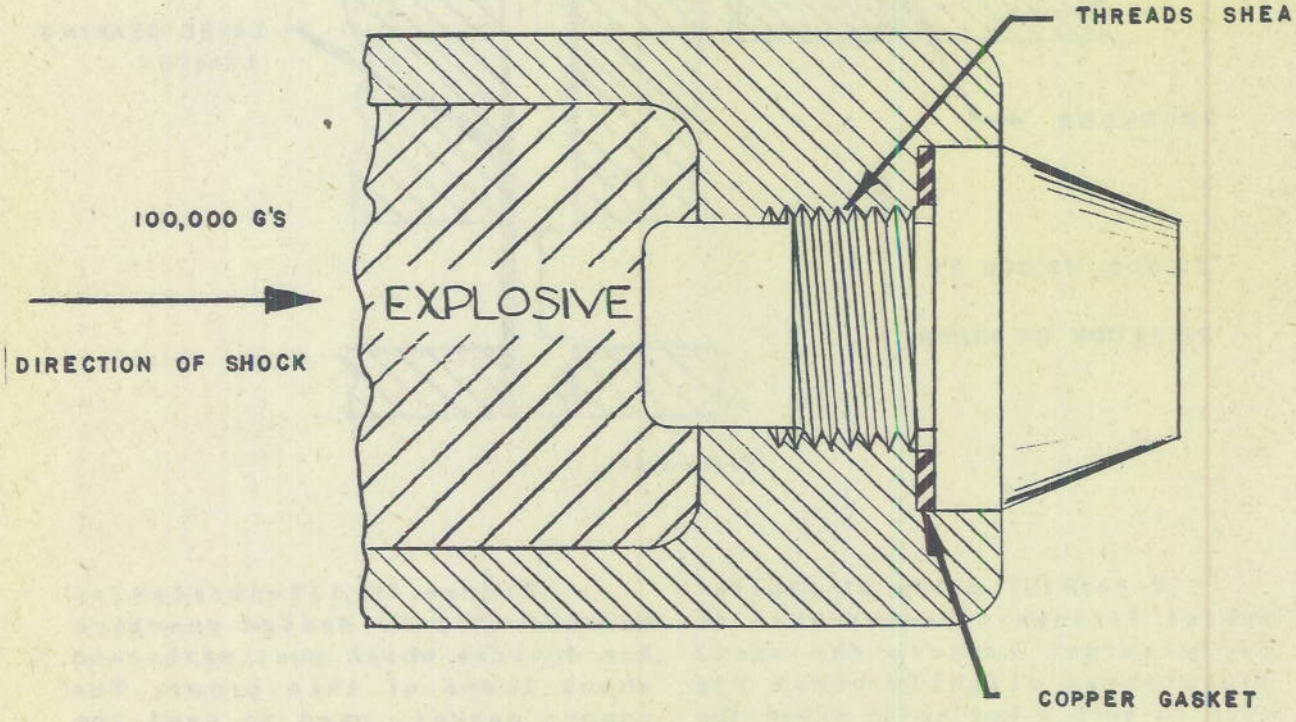

Figure 13

prematurely during the explosion. A simple solution to the problem was effected by elimination of the copper gasket. With the copper gasket eliminated, the shock load could be taken by the rather massive shoulder of the fuze itself.

Figure 14 illustrates still another technique at the disposal

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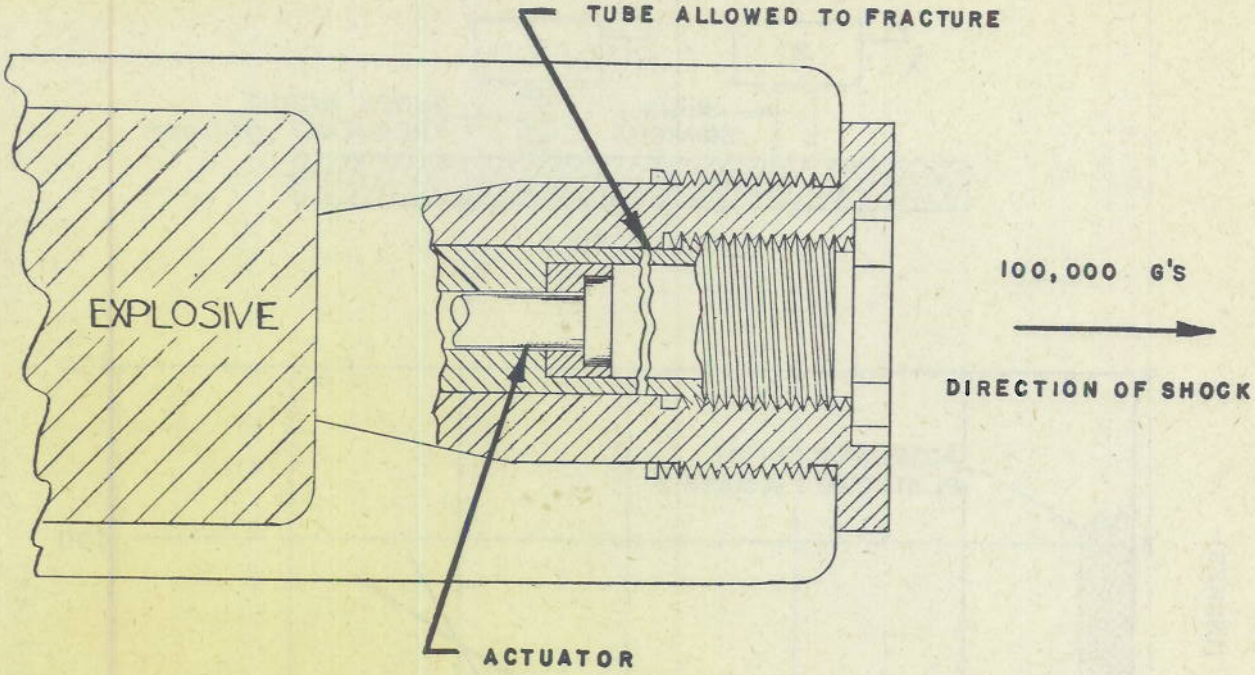

Figure 14

of the expert designer in this field. Again the problem concerned the design of a base-detonating fuze of an armor-piercing projectile. The first designs of this particular mechanism failed, as crushing of the actuator shoulder frequently occurred.

Complete satisfaction was experienced in a design which depended on the failure of a weaker element in the actuator support system. The figure shows how this result was attained by permitting a support tube to fracture under the shock load thereby providing the time delay required for the operation of this particular device

GENERAL CONSIDERATIONS

The designer of high "G" systems must always consider one other factor. Complete systems generally exhibit certain overall damping characteristics which none of its elements may possess individually. It is not always wise, therefore, to consider a simulated shock test on any individual component as entirely indicative of the results which will be experienced in field trials.

If the platform 1 and the mass 2, shown in Figure 15, are displaced (shown by the straight line) they receive a shock loading (indicated under the cross-hatched section). If, however, because of the damping capacity of a mounting, the mass 3 is displaced, (curved line on the graph) this mass is subjected to a much lesser loading (dotted section).

In conclusion, one of the most important tools available to a designer in this field is the high-speed

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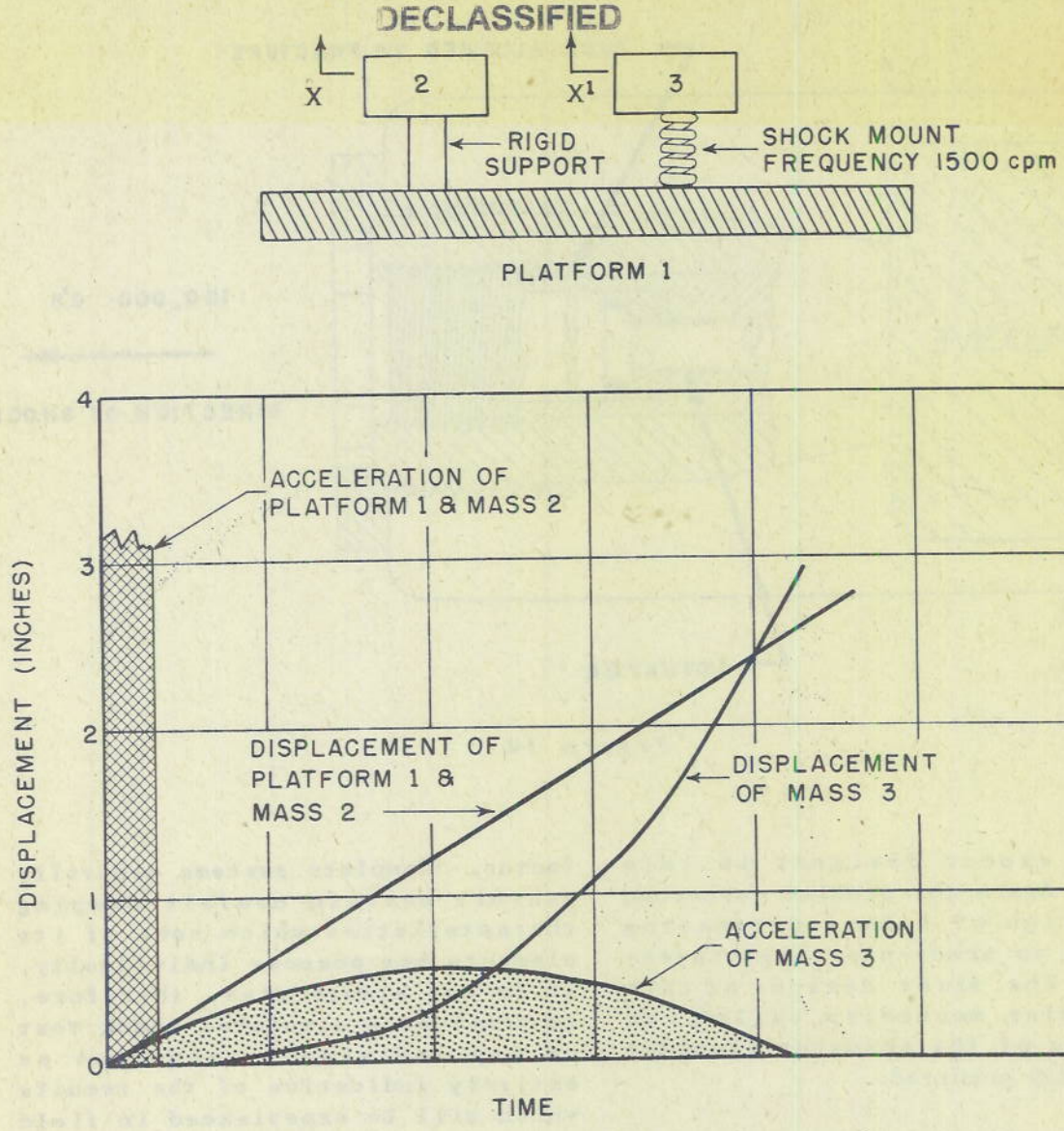

Figure 15

motion-picture camera. All too frequently it is impossible to ascertain from field tests the exact cause of a failure of malfunction, and all theory, experience, and guesses fail to solve the problem.

Figure 16 illustrates a typical problem of this sort. The device shown is an electrical component consisting of several ceramic cylinders on a steel bar, and the entire assembly is mounted to an electrical panel board.

Under shock loads, as indicated, the ceramic cylinders failed. High-speed movies taken during shock tests revealed that the steel bar was deflecting in the manner shown. As a consequence, the ceramic cylinders were subjected to severe stress

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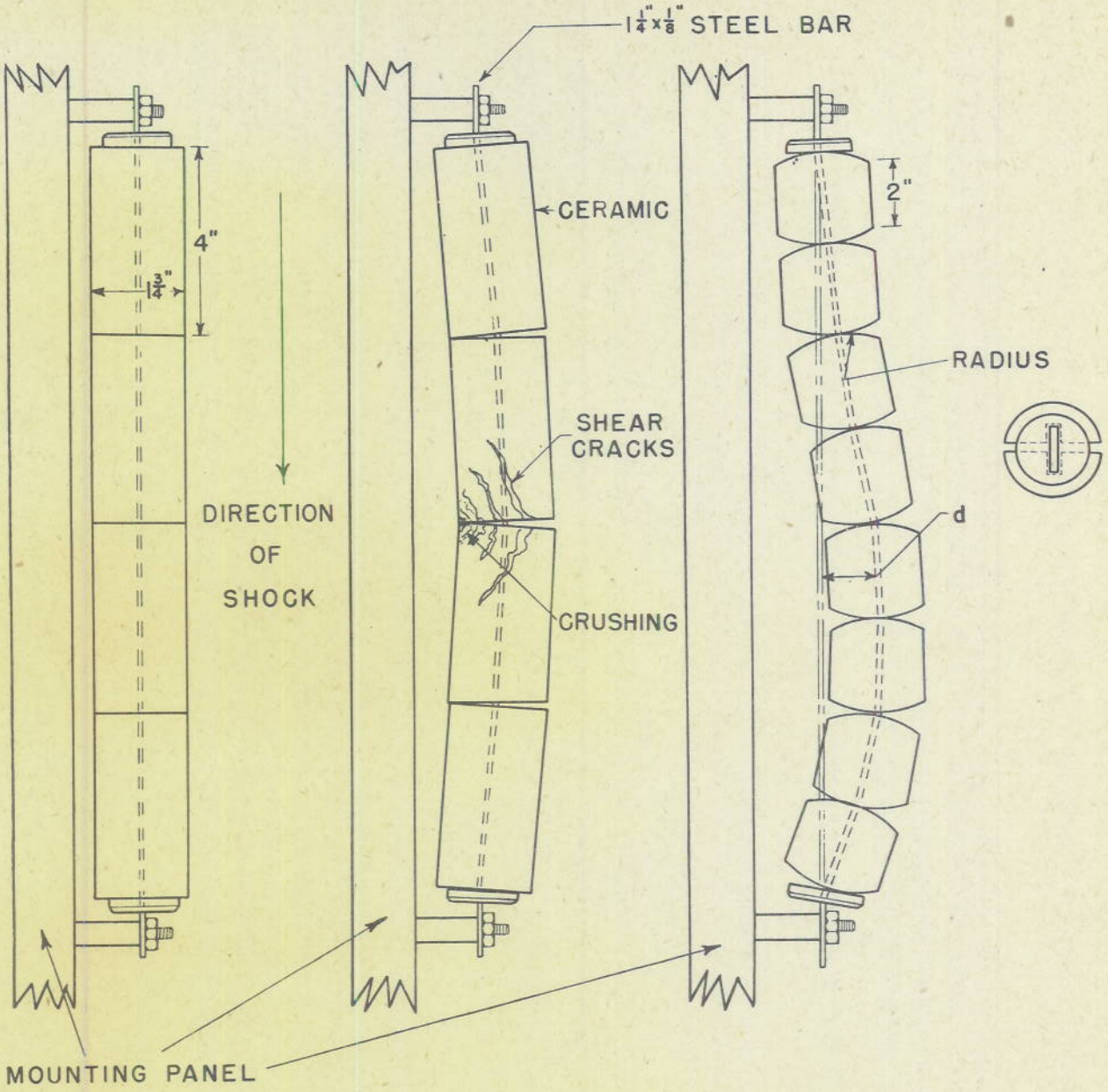

Figure 16

concentrations at the points indicated and fractures resulted.

By means of a simple redesign of the ceramic cylinders, as shown on the right of this figure, excessive stress concentrations were entirely eliminated by employing shorter sections with rounded ends instead of the former squared ends.

High-speed photography is the most direct and economical approach in problems of this type.

It has been the primary purpose of this paper to point out some of the more common problems encountered and the design techniques employed in the successful designing of mechanisms for high "g" shock loads. Practically no consideration has been given to the theoretical approach because of the limited scope of this paper.

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