REPRINT

Title: **A Low-Class Fixture Can Spoil A High Class Vibration Test**
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A vibration test is carefully tailored to simulate service loads. But sometimes a product passes a test, only to fall apart after a few days in the customer's hands. Why? Probably, because the test fixture was bad.

Vibration tests have come a long way from the old cam-driven shaker rig. Today's servodriven, computer-controlled tests can duplicate virtually any type of shock and vibration that a product will see in service. And accelerometers hooked to recorders show exactly what these service loads are. So in principle, there should be no difficulty designing a test that faithfully reproduces actual use in the field.

In practice, things don't happen that way. The instrumentation measures the service loads accurately, and the shaker table reproduces them accurately. But the loads have to pass through a fixture that holds the specimen to the table. And this fixture can distort the test input so badly that the results may be next to meaningless.

As a result, the specimen can be overtested at some frequencies and undertested at others. If undertested, the product might fail in service. If overtested, it may end up grossly overdesigned to cope with some spurious vibration encountered only on the test table but never in real life. All of this because the fixture is a simple piece of hardware that commands very little thought or attention.

There are two common mistakes that can cause poor results in a vibration test: 1. Assuming that a test fixture is doing its job so long as it does not break. 2. Assuming that test motions can always be monitored at one point on the fixture.

These assumptions are wrong because a fixture may resonate and feed excess motion to the specimen, or it may isolate motion and shield the specimen from input forces. Also, one point on the fixture may be vibrating exactly as prescribed, but the point at which the specimen is attached may be vibrating in an entirely different manner.

WHAT A TEST SHOULD DO

A vibration test is an accelerated test. It simulates, in a short time, the effects of long-duration field service. Also, the test should be standardized so that results found by one tester can be reproduced in another laboratory.

Test loads are derived from field measurements. In the past, loads generally were set at the highest levels measured in the field. Sometimes, they were set even higher to include a safety factor. Recently, though, the trend is to test at a statistical average load, resulting in more reasonable test levels.

Regardless of how the test level is derived, a laboratory test is not "realistic" simply because it is performed in a controlled environment. To fulfill its purpose, a dynamic test must generate the same *damage potential* as the real world. How to generate this potential is not of concern here; rather this article describes how to design fixtures and instrumentation to ensure a good test.

A dynamic test is an absolute must if a product is to operate in a vibrating environment. But such testing also should be considered if a product is to operate in a static environment. For example, consider a color television set. It leads a sheltered life in the average home, but it is likely

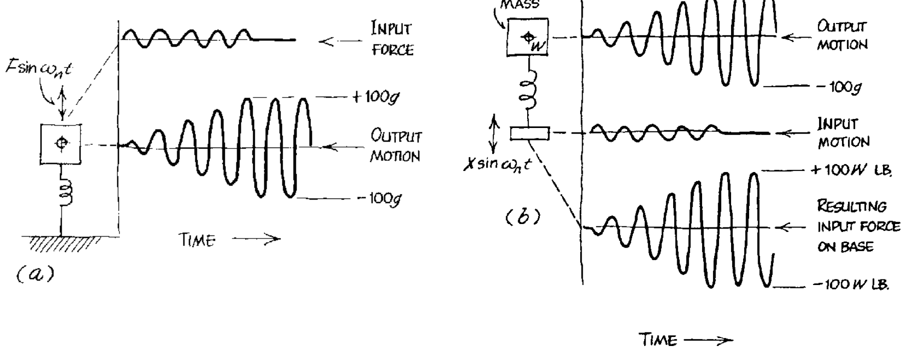

Figure 1. Illustration of mechanical impedance. In a, the mechanical impedance drops to zero at the natural frequency once the desired motion is reached, because no further input is required to maintain that motion. In b, some definite oscillating force must be applied to the base to maintain the desired amplitude. This force, divided by the input motion, is the mechanical impedance.

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to undergo considerable shock and vibration during shipping. Here, dynamic testing plays a part in product development to ensure that the product can survive shipping.

MECHANICAL IMPEDANCE

The concept of impedance helps explain the interaction between test machine and specimen. A simple definition of mechanical impedance is that it is the force required to produce a desired motion. When an alternating force is applied to a mass, Figure 1*a*, the mass moves up and down. If the force alternates at the system natural frequency, the amplitude of the mass increases continuously. Theoretically, there is no limit to its ultimate amplitude, if we assume no friction or damping.

Suppose the desired acceleration is ±100g. Once this acceleration is reached, the force input must be halted or the amplitude keeps rising. There is no damping or frictional loss, so the system continues to vibrate at ±100g without additional force input. Because mechanical impedance is the force required to produce a specified motion, the mechanical impedance of the system drops to zero at the natural frequency once the system reaches the desired amplitude.

Now suppose that the input is not a force applied to the mass, but instead is a motion applied to the base, Figure 1*b*. The amplitude again increases without limit. But if the excitation stops at the ±100g level, the alternating load in the spring is ±100 ´ W, where W = weight of the mass. The spring transmits this force to the base, and the force in the base must be balanced by the input. The result is that instead of the zero force required to maintain motion, an oscillating force of 100W peak is needed to maintain motion of the mass.

The higher the mass acceleration, the higher the force transmitted to the base. Thus for input excitation at the base (as in a vibration test), this undamped single-degree-of-freedom system exhibits a high impedance. In the limit, if the mass approaches infinite amplitude, the impedance also approaches infinity.

STATIC BALANCE

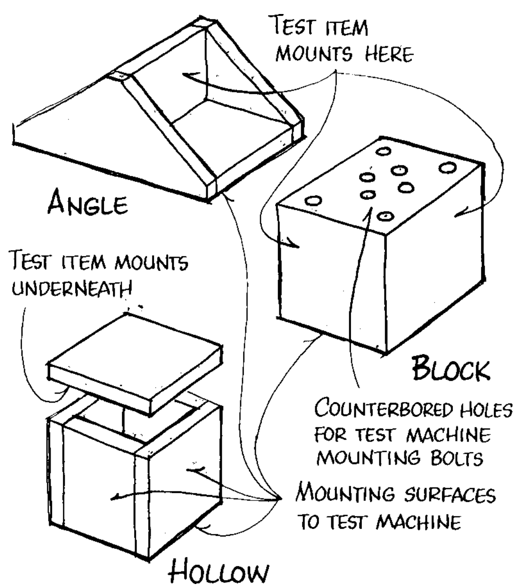
There are two cases where a test specimen may not "feel" all of the motion that the test machine attempts to impart to it. One involves test frequencies above the natural frequency of the test setup, where the stiffness of the fixture is insufficient to force the specimen to follow the motion of the test machine. So the specimen tends to float or decouple from the test machine. (See What Is Transmissibility? - page 4)

The second case concerns so-called dynamic imbalance where dynamic forces from the fixture do not pass through the center of mass of the specimen and where elasticities in the mounting allow this misalignment to produce rocking in the specimen. This rocking absorbs motion which otherwise would be imparted to the supecimen as acceleration.

Fixture design has much to do with the potential for dynamic imbalance. With angle and block fixtures, dynamic imbalance can generate large moments that can damage the specimen and the machine. One way to prevent these moments is to install two test items, if available, one on each of two opposing vertical surfaces. If two specimens are not available, static balancing with a dummy mass is preferable to no balancing at all.

With a hollow fixture, the specimen center of gravity is positioned close to the vibration machine center line. Thus, separate static balancing of the test assembly usually is not necessary. As the specimen goes through resonance and decouples, the center of gravity shifts to a position that is still close enough to the test assembly center line so as not to generate serious moments.

However, hollow fixtures usually are much more expensive than angle and block fixtures. Also, a block fixture provides greater stiffness and allows mounting several items simultaneously. So considerations other than the ease of static balancing enter into the choice of the best fixture for a particular test.



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THE CONCEPT OF INFINITE IMPEDANCE

Inherent in any standardized test program is the concept of *infinite impedance*. This term means that the dynamic input to the specimen is not altered by the specimen itself; in other words, the test machine is not loaded down by the test specimen. Thus, the ideal motion generator is one with a mechanical impedance so high that the added load of the test item does not alter the generator motion.

However, in actual testing, a product always loads its input to some extent. For instance, if guidance equipment were being built for an airplane, vibration measurements on the mounting structure would be taken first, and then the guidance package would be tested using these measurements. But if the equipment were later mounted in the plane and the vibration measurements repeated, these measurements of vibrational amplitude would be different. At some frequencies, the amplitude would be lower, and at other frequencies, the amplitude would be higher.

The important regions are where the amplitudes are lower. These points are where the guidance equipment loads down the structure, and where the equipment itself resonates (thus damping motion in the mounting structure). To avoid this resonance a "realistic" test specification calls for an input reduction at every frequency where the test item has a resonance. But the problem lies in determining how much to reduce the test level at each resonance.

This example is one where the specimen itself significantly alters the service environment. On the other hand, a relay mounted on the hull of a battle tank would not change the motion of the hull when it (the relay) resonates.

Because of the variables, uncertainties, and unknowns involved in the vast array of products and their mountings, no generalized system can be developed to compensate for mechanical impedance loading. So, rather than leave this compensation to the whim of the individual tester, most test procedures ignore impedance completely and require that the dynamic input be unaffected by the test item. As a result, a product subjected to the typical dynamic test tends to be overtested at its resonant frequencies. (The resonating parts are tested at amplitudes greater than those which they see in service.) Experts are aware of this discrepancy, and although the resonant amplitudes may be many times what they are in the field, the penalty in terms of product overdesign usually is not great.

In certain cases—especially with complex, expensive, and critical products—specimen loading should be taken into account. Doing so, however, requires special test equipment.

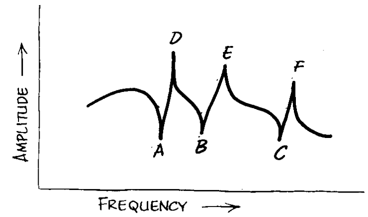

Figure 2. Vibration spectrum of a device with three nautral frequencies. Points A, B, and C are resonant frequencies of the test item. So much force is required to balance specimen movement that little is left to move the table . Points D, E, nad F are frequncies where the table and test specimen resonate against each other.

HOW THE TEST MACHINE & SPECIMEN MOVE

Shock and vibration tests excite systems by applying motion to the base. Therefore, when a system resonates, it transfers a high force to the structure to which it is connected. Complex systems, with several resonant frequencies, load down the vibration machine each time the input frequency coincides with a resonant frequency.

Consider, for example, a device with three natural frequencies, bolted to a vibration machine. Suppose that once a vibration level is set, no attempt is made to further control the amplitude of the table, but table motion is monitored with an accelerometer.

Figure 2 shows the results of a slow frequency sweep under these conditions. Where the input frequency is close to or equals any of the three natural frequencies (Points *A*, *B*, and *C*), the amplitude of the table vibration falls sharply and forms a notch in the vibration spectrum. This occurs because at a system resonant frequency, so much force is required to balance the load of the system that little force is left to move the table.

The peaks in the response spectrum (Points *D*, *E*, and *F*) are the frequencies where the mass of the table resonates

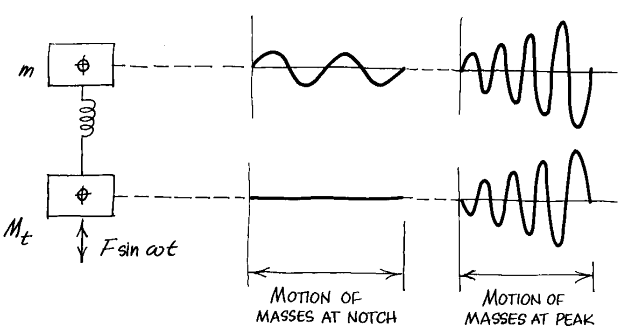

Figure 3. Motion of test specimen and vibration table at various frequencies. At the specimen resonant frequency, the amplitude of the table drops to zero. But at the overall system resonance, the table and specimen bounce against each other, and amplitude rises toward infinity.

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against the dynamic system of the test specimen. This condition is depicted in Figure 3. The vibration table M_t is part of the overall dynamic system and participates in an overall system resonance. At the resonant frequency of the specimen *m*, the input force only maintains motion of *m*, and the amplitude of M_t drops toward zero. But as the input frequency increases, a frequency is reached where the whole system resonates; in other words, where *m* and M_t bounce against each other. In this vibration mode, the input force acts directly on one of the resonating masses (M_t), so motion of the overall system easily builds up to high amplitudes. The result appears as a high point or peak in the vibration spectrum.

Test specifications usually require that the dynamic input motion to a test item be kept at a constant level regardless of fixture loading. This means that no notches or peaks are allowed. Electromagnetic and electrohydraulic vibration exciters handle this problem with servo controllers. They sense any deviation from the specified vibration level and vary the gain of the amplifiers to minimize the deviation. If the test machine keeps deviations within allowable limits, the machine is said to be equalized.

Impact or impulse-shock test machines and reaction-vibration machines cannot be adjusted to equalize the test machine against specimen loading. In these cases, the table and fixtures are made as massive as possible to minimize the effects of specimen loading.

Brute-force mechanical vibrators do not require equalization because the motion is generated by a cam, and specimen loading does not influence table motion.

WHAT IS TRANSMISSIBILITY?

Except for some transportation simulation tests where a specimen is fastened to the test machine with steel bands or nylon webbing, some sort of adapter usually is necessary to attach a specimen to the machine. This adapter can be as simple as a metal plate with properly matched hole patterns or as complex and as large as the vibration machine itself.

The problem of fixture design is tied directly to the present philosophy of dynamic testing; that is, the need for an infinite impedance test machine. Thus, the fixture must be as stiff as possible so that it is not deflected by the load and transfers motion with high fidelity. This quality is called transmissibility, which is a comparison of the output to the input. At a transmissibility of 1.0, the output faithfully follows input.

Ideally, a dynamic test fixture couples the motion from the test machine to the specimen with zero distortion; in other words, the motion at the fixture output (the connection to the specimen) is identical to the motion at the fixture input (the connection to the machine) at all amplitudes and frequencies. This ideal is approached, for practical purposes, if the test frequency range is narrow or if the test specimen is small. Usually, however, the ideal cannot be met and the limitation of the fixture must be known.

The basic fixture shortcoming is insufficient stiffness. In theory, with an infinitely stiff fixture, the natural frequency of the fixture-specimen system can be made as high as necessary to prevent resonance and to provide a transmissibility of 1.0. Suppose for example, that we have a known fixture transmissibility, Figure 4, with the natural frequency of the test setup four times higher than the highest test frequency. Therefore. the ratio of the forcing frequency to the natural frequency, F_1/F_n, is 0.25. At any ratio below 0.25, the transmissibility is close to 1.0. This is the case, for instance, if F_n = 2,000 Hz and the test specification called for frequencies from 50 to 500 Hz. The fixture output faithfully follows the fixture input in this test range, because the transmissibility stays near 1.0 as F_1/F_n varies from 0.01 to 0.25.

Generally, however, a fixture is never stiff enough so that the test system natural frequency is above the range of test frequencies. Usually, the natural frequency lies within the test specification range. In this case, the motion

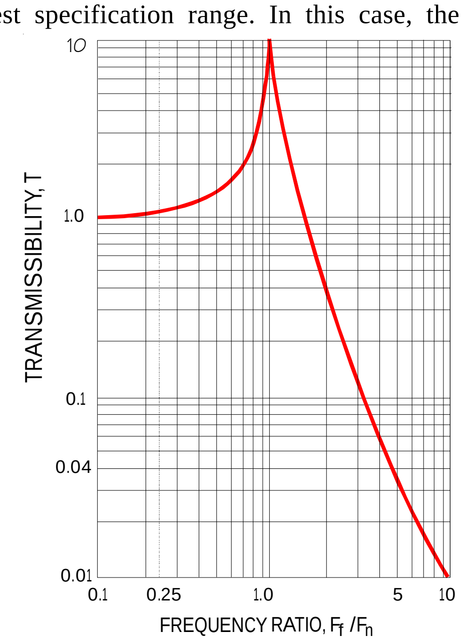

Figure 4. Transmissibility curve for a single-degree-of freedom system. At frequency ratios below 0.25, the input to the test item is identical to the input to the fixture. Above 0.25, transmissibility varies greatly, and the input to the specimen cannot be controlled accurately.

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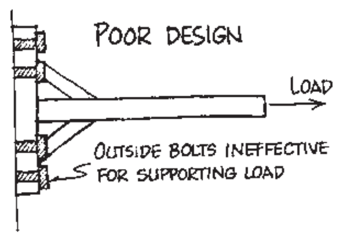
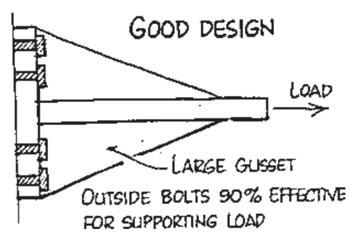
POINTERS ON FIXTURE DESIGN

Specimen Mounting: Mounting bolts should be torqued to 80% of yield, and they must be clean and rust-free for torque measurements to be meaningful. The larger the number of bolts attaching the specimen to the machine, the better - if there is sufficient stiffness in the fixture to load the bolts. To develop 90% effectiveness of a bolt some distance from a web, a good fixture uses a gusset with a 7:1 slope.

Bolting vs. Welding: If the fixture is made up of several different parts, the parts should not be bolted together.

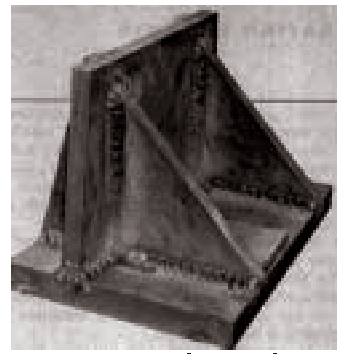




Rather, they should be welded or bonded. Bolted joints, loaded in tension, cannot transfer loads across the joint as well as a weld or bond. Also, bolts increase the expense of a fixture, because they require drilling and tapping and smoothly machined, well-fitted joints. Bolts, however, are acceptable for attaching the specimen to the fixture or for connecting the fixture to the machine.

Mounting Surfaces: Flat mounting surfaces are extremely important, and these surfaces should be protected between use. A raised burr, only a few thousandths of an inch high, can spoil a test because the fixture is not solidly supported by the surface of the test machine. The stiffness of the test setup with the burr is determined by the stiffness of the fixture alone, rather than by the stiffness of the fixture plus some reinforcement from the test machine. Any deviation from flatness decreases stiffness, thus lowering the natural frequency of the assembly. Also, the fixture may slap against the machine, producing a hashy signal.

Materials: Aluminum is light and does not weigh down the test machine, but



it is strong enough to make a good fixture. Also, it is easy to fabricate and is. relatively inexpensive. When using aluminum or magnesium, put steel washers under all bolt heads and use steel thread inserts. These inserts keep the bolts from galling or stripping the fixture material.

Accelerometers: Accelerometers should be mounted with tapped holes. These holes should not be made with hand drills because the axis of the hole must be perpendicular to the mounting surface within 1 deg to ensure accurate readout. Torquing the accelerometer down to a smooth, flat surface enhances the reliability of the measurements.

delivered to the test item is higher than the input in the region around resonance and lower than the input above resonance. Suppose the natural frequency of a test setup is 400 Hz, and the specification calls for a 5g vibration from 20 to 2,000 Hz. From Figure 4, the input to the specimen is 50g at 400 Hz (F_1/F_n = 1.0 and T = 10) and only 0.2g at 2,000 Hz (F_1/F_n = 5.0 and T = 0.04).

Most modern test facilities incorporate automatic amplitude control, so that if the vibration amplitude is measured at the input to the test specimen, a uniform 5g would be applied to the specimen. Thus we might conclude that stiffness of the fixture is of little consequence in the region of resonance. But such is not the case; fixture stiffness is still important. At 2,000 Hz, the shaker has to vibrate at 125g to provide 5g at the specimen (5/0.04 = 125). Presently, there is no equipment that can provide this high level of vibration. Therefore, the test specification cannot be met at high frequencies.

In addition, this hypothetical system consists of only a single-degree-of-freedom moving in one axis. In the real world, fixture resonances consist of multi-axis motion which causes cross-axis rocking at the specimen attaching points. These additional motions lead to poor control and ambiguity in the test output.

Thus, the most important factor in fixture design is a high natural frequency. However, weight and cost factors require some design tradeoffs. Weight is a drawback if it infringes on the capability of the vibrator to deliver enough acceleration to the specimen. Cost considerations influence whether several single-purpose fixtures or one

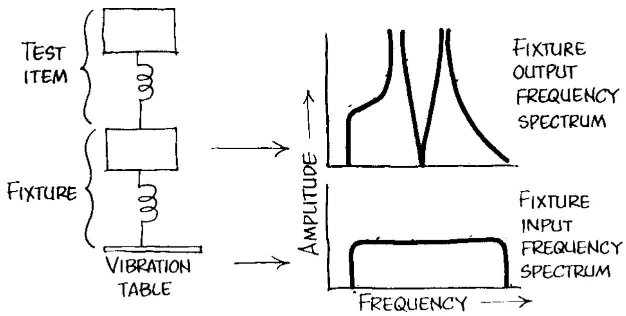


Figure 5. Fixture response when natural frequency lies within the test spectrum. The transmissibility of the fixture varies greatly throughout the test and input to the test item does not have a constant amplitude. This problem can be solved by proper placement of control instrumentation.

general purpose fixture will be built. Weight and cost considerations almost always compromise fixture performance; therefore, at least some thought must be given to intelligent design tradeoffs.

FIXTURE-SPECIMEN RESONANCE

A simple way to model the fixture-specimen dynamic system is to assume that the test specimen s an ideal, resonant-free mass that loads the fixture. An idea of the fixture design problem (in terms of how stiff the fixture must be) can be obtained from the formula $D = (3.13/F_n)^2$, where D = the fixture static deflection caused by the specimen that produces the desired F_n; and F_n = the desired resonant frequency of the specimen-fixture system. Preferably, F_n is higher than the test spectrum, but usually this ideal is impractical or in possible.

For instance, many specifications call for testing up to 2,000 Hz. If, to keep transmissibility close to 1.0, F_n is targeted for 3,000 Hz, the above equation shows that the fixture must not deflect more than Im in. under the weight of the specimen. In most cases, such a stiff fixture is not feasible. Thus, often it is more convenient to develop a fixture, compute stiffness, and then solve for F_n using the above equation. Of course, this approach is crude. But it does provide order-of-magnitude information.

The important factor left out of the equation is the mass of the fixture itself which acts to reduce F_n. The amount of reduction varies according to the test configuration and the ratio of the fixture mass to the specimen mass, but a reduction of 20% to 30% is not uncommon.

THE RESONANT FIXTURE

Virtually every test with frequencies above 1,000 Hz shows moderate to severe loss of motion control. Often this loss of control occurs at frequencies as low as 200 Hz, and little can be done to alleviate the problem.

For example, consider a fixture with one resonant frequency within the test frequency spectrum, Figure 5. Suppose the motion of the fixture is collinear with the input motion from the vibration machine; in other words, no rocking is present. Sinusoidal, constant-amplitude motion of the vibration machine gives the test fixture response shown. Insofar as the test specimen is concerned, its input certainly does not have constant amplitude. Fortunately, though, this problem can be solved with proper instrumentation.

Figure 6 shows the effect on specimen input as the control point (accelerometer mounting point) moves from the fixture input toward the fixture output. In each case, the shaker table (because it is servodriven) maintains constant amplitude vs. frequency at the control point. Therefore, the control point must be as close as possible to the specimen input, Figure 6*d*.

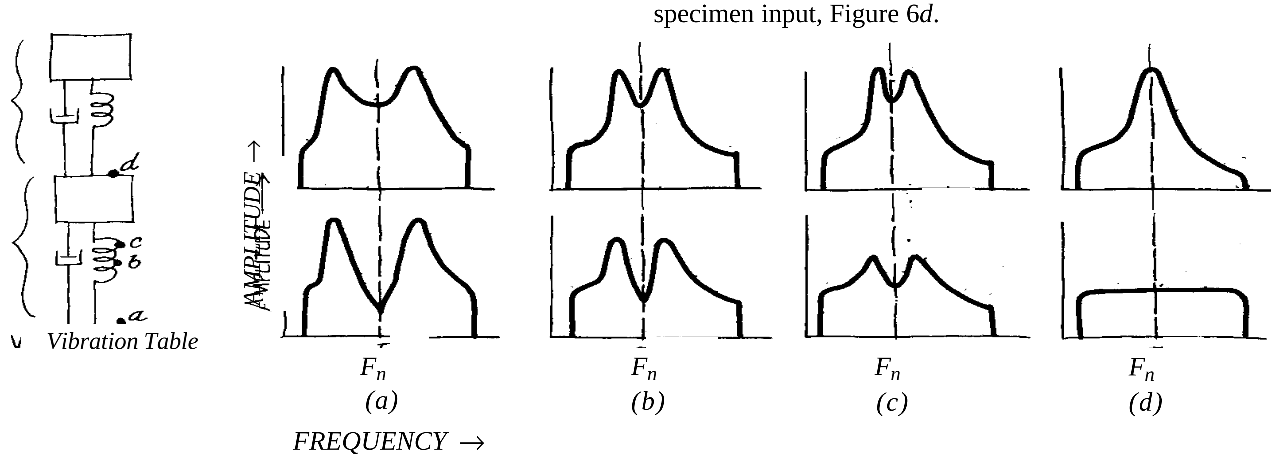


Figure 6. Specimen input for various control point locations. With the control point as close as possible to the specimen attaching point, the input to the specimen approaches the requirement for constant amplitude at all frequencies.

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In addition, several accelerometers should be used to sense the specimen input, and these accelerometers should be located immediately adjacent to the specimen mounting points (the fixture output). Relying on one accelerometer, mounted on the test machine table, is poor practice to be condoned only if the specimen is small (less than 100 cu in. or 10 lb.), if it is mounted directly to the table, and if frequency is below 500 Hz.

Taking rocking into account clouds the picture. The rocking motion at the specimen input might be caused by a resonance on the fixture or by a dynamic imbalance moment. Rocking is particularly prevalent on a fixture mounted directly to a test machine table operating at high test frequencies. The result is that the specimen experiences uncontrolled multi-axis motion rather than uniaxial motion. The use of several accelerometers will alleviate this problem somewhat because they will detect rocking and then generate an average input control signal to prevent over or undertesting.

There is another, more subtle, advantage to using several accelerometers. For instance, consider a test setup using only one control accelerometer. If the device is mounted at or near a vibration node, it will detect no vibration and will signal for ever-increasing input from the shaker table. This action could damage the specimen and produce ridiculous test results.

RESONANT SEARCH

How well a product will stand up under service conditions usually is determined by how its natural frequencies relate to loading frequencies. Sometimes natural frequencies can be computed during design. But often the stiffnesses and masses cannot be predicted accurately enough for meaningful calculations. Here, a resonant search can identify critical frequencies.

Resonant search is used in two ways. First, it can identify the frequencies at which delicate subcomponents in a product are likely to be damaged. Suppose, for example, that a product is to be tested to the specification shown in Figure 7, and that a resonant search with accelerometers attached to critical subcomponents revealed that an important part had a resonance at 150 Hz.

Because this resonance lies within a portion of the test spectrum imposing a severe loading, we might consider stiffening the component supports to increase its resonance to 300 Hz. Or we might reduce support stiffness to move the resonance below 75 Hz.

The second use for-resonant search is as a proof test where the important factor is the overall resonant frequency of the entire product. Here, a test is run of the final hardware to identify frequencies at which subsequent testing should be performed.

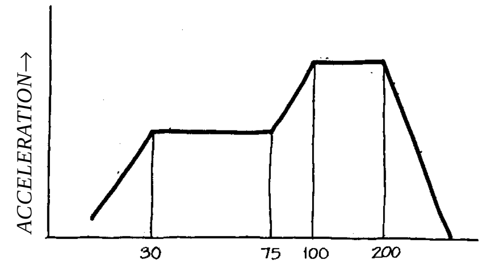


Figure 7. Typical vibration test spectrum. If a resonant search reveals that a critical component has a resonance in the severe (F> 75 Hz) portion of the spectrum, the stiffness of the component should be altered.

During a resonant search, accelerometers should be used not only to determine motion of critical components, but also to monitor input to the assembly. This measurement is a reference signal to the servo control equipment, ensuring constant amplitude input, which in turn ensures that measurements from other accelerometers truly indicate response of the items in question. If servo control is not used, then the reference signal is used in data reduction wherein the true response of the specimen can be obtained by dividing each response signal by the reference signal.

Figure 8 illustrates the use of a reference signal to find true response. Curve *A* is the true response (obtained

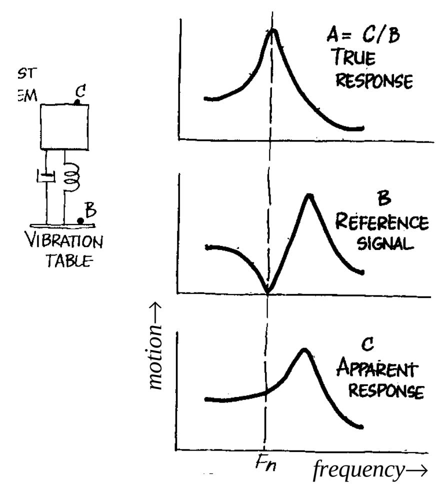
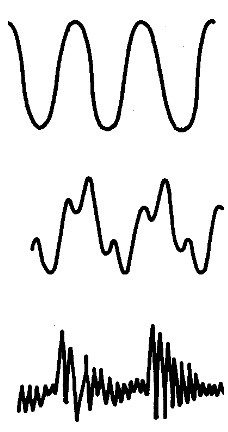


Figure 8. How a reference provides true response. Without reference signal B, the response of the test item would appear to be Curve C. But Curve C is not the true response of the item. Dividing Curve B by Curve C produces the true response and identifies the true resonant frequency, Curve A.

from an accelerometer mounted on the mass) of a single-degree-of-freedom system subjected to a constant-amplitude vibration of the base. Without servo control, the vibration machine input to the test assembly develops a notch-peak response, Curve *B*. With this input, the test mass output looks like Curve *C*. The resonant peak does not appear at the resonant frequency of the test item, but at the resonant frequency of the system that includes the mass of a vibration table. This information obviously is erroneous. Dividing Curve *B* by Curve *C* generates Curve *A*, the true response. But if the control signal (Curve *B*) were not available, there would be no way of knowing that Curve *C* is not the true response.

Additional sources of erroneous information are the nonlinearities in the vibration machine itself. Armature resonance can be excited by harmonics from the power amplifiers (usually the third harmonic). Thus, if the armature has a resonance at 1,800 Hz, it can be excited by a 600-Hz signal, and a peak appears in the accelerometer response at 600 Hz, falsely suggesting system resonance.

Two paths lead out of this trap. One is to use tracking filters in the accelerometer circuits. These filters remove all harmonics and noise, leaving the fundamental frequency of interest. If tracking filters are not available, then an oscilloscope should be used to monitor signal quality. A true resonance produces a signal that looks like a pure sine wave, Figure 9. If the signal builds up, but looks hashy or consists primarily of frequencies other than that to which the vibration machine is adjusted, then there is no resonance at that frequency.



TRUE RESONANCE SIGNAL

HIGH SIGNAL CAUSED BY RESONANCE AT OTHER FREQUENCIES; NOT AT FREQUENCY OF INTEREST

HASHY SIGNAL CAUSED BY RATTLING

Figure 9. Oscilloscope traces of accelerometer signal quality. A pure sine wave indicates a true resonance at the frequency of interest. The other signals indicating "apparent" resonance are caused by noise and interference from the vibration machine.

If these procedures are too complicated, or if you don't have access to the inside of the product, then resonance can be judged using sound or feel. Another approach is to take advantage of the notch-peak phenomenon described earlier. In this approach, the test item is secured to the vibration table, an arbitrary vibration level is applied, and a spectrum sweep is made. No attempt is made to control the table amplitude of motion during the sweep. The output records the response of the total system (the test item plus the vibration table). Therefore, notches in the spectrum response identify specimen resonances because these frequencies are the ones at which the test item is resonating and the table is standing still or is highly loaded.

IF THE PRODUCT FAILS

What happens if all the suggestions for controlling the vibration test are followed, and the product still fails? If the test setup is as stiff as possible, if the effect of dynamic imbalance has been minimized, and if multiple accelerometers were used to provide an average control, then in all probability the product will have to be redesigned. The design group may argue over interpretation of the test results, but since no one is likely to come up with an accurate explanation for the failure, it is usually wiser—when in doubt—to redesign.