THE DESIGN OF A TEST PROCEDURE FOR THE MEASUREMENT OF
ACOUSTIC DAMPING OF MATERIALS AT LOW STRESS
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by

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September 1983

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The Design of a Test Procedure for the Measurement of Acoustic Damping of Materials at Low Stress

by

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Lieutenant, United States Navy
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ABSTRACT

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The test procedure includes modal analysis that is expandable to other geometric shapes and varied material such as high damping alloys and composites both metallic and non-metallic.
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I. INTRODUCTION

A. GENERAL

Ship silencing continues to be a major design requirement in the construction and operation of submarines and surface ships. It has been long established that both equipment and personnel are adversely affected by unwanted noise and vibration. In today's environment both subsurface and surface, with recent advances in acoustic devices the very survivability of the platform is directly related to own ship's noise. Weapon platforms must be quiet enough to escape detection by sophisticated passive sonar devices as well as quiet enough to prevent own ship's noise from interfering with detection and prosecution of enemy targets.

The more traditional approach to ship silencing has been, and continues to be, that of vibrations isolation. This approach requires that all sources of vibration to be placed on resilient energy-absorbing mounts. This results in a significant reduction in the transmission of vibrational energy from the equipment and main engines to the surrounding environment but is not totally satisfactory because these same mounts must serve as shock mounts. In addition, any resilient mount will have a peak efficiency in particular frequency range with decreasing efficiency both above and below that frequency range. The optimal solution would be to use
resilient mounts in conjunction with other methods of vibrational control to reduce the amount of vibration and noise generated as well as reducing the amount transmitted.

As with any design, today's ships are a compromise of requirements and cost considerations. However, the trend recently has been toward the design of equipment that is inherently quieter, which results in less noise and vibrational energy at the source. This current trend in design is currently being approached from many directions. Among these considerations are:

1. Extreme care in balancing of rotating machinery with consideration of vibrations during start up and coast down as well as steady state operation.
2. Selection of operating frequency as far removed from resonant frequencies as possible.
3. Close attention to component tolerances.
4. Possible use of high damping materials for machine elements as well as machine casings and load bearing structures.

The last category, use of high damping materials, is the furthest from being utilized to its maximum potential for noise and vibrational reductions.

Only recently has the designer even considered the internal capacity of a material alone with the traditional material properties of strength, fatigue resistance, toughness and corrosion resistance.
There are several commercially available high damping alloys that meet the strength requirements for most shipboard applications. It is recognized that each of the candidate materials may have unique properties that may cast doubt on their actual usefulness in the United States Navy. A more important problem exists, however. That is, the lack of a consistent method of measuring damping for a material under low stress, and high frequency range.

B. DAMPING

"Structural damping" refers to a structure's or structural component's capacity for dissipating energy, or, more precisely, to its capacity for removing from a structural vibration some of the energy associated with that vibration. The energy removed may be converted directly into heat, transferred to connected structures or ambient media [Ref. 1].

Damping has two primary effects: (1) It limits the steady-state motions of structures or systems in situations where these motions are controlled by an energy balance; and (2) It increases the rates at which the free (i.e., unforced) vibrations of structures decay.

Consider, for example, the classical lumped-parameter mass-spring dashpot system driven by a steady sinusoidal force that acts on the mass. For such a system it is possible to make the following observations: (1) For excitation frequencies that are considerably lower than the system resonance
frequency the applied force essentially is balanced by the spring force. The mass and dashpot here have virtually no effect. (2) For excitation frequencies that are considerably higher than the system resonance frequency, the applied force essentially is opposed by only the inertia force of the mass. In this case the spring and dashpot have virtually no effect.

(3) For excitation frequency near the system resonance frequency, the spring force and the inertia force essentially cancel each other, leaving only the dashpot (i.e., the damping element) to oppose the externally applied force. One may note that changes in the damping do not affect the system response for the first two of the above cases, but for an oscillatory force acting at the system's resonance, the vibration amplitude decreases with increasing damping.

In addition to the case of resonant or near-resonant excitation, there occur several other important steady-state or near-steady-state situations in which the responses of systems or structures are controlled by a balance between the energy input and the energy dissipations. These situations include cases of (1) broad-band excitation, and (2) spatially periodic excitation where the spatial period matches that of freely propagating waves. Because broad-band excitation generally encompasses several structural resonance frequencies, the structural responses of the excited modes, and each resonant modal response is controlled by damping.
If a structure (or a mechanical system) is deflected from its equilibrium position and then released, the structure vibrates with ever-decreasing amplitude on the result of damping, i.e., as the result of energy being removed from the oscillatory motions. Greater damping corresponds to the dissipation per cycle of a greater fraction of the vibratory energy, thus resulting in more rapid decay of the vibrations. In a somewhat similar manner, increased damping also results in the more rapid decay of freely traveling waves. For example, if a long beam is subjected to an oscillatory transverse force of a constant amplitude and frequency, such as near the center of the beam, then flexural waves travel away from the driving point in both directions. As the result of damping, the amplitude of these waves decreases with increasing distance from the point of application—with greater damping leading to lesser amplitudes at a given distance.

The practical consequences of effects mentioned above generally are the reasons one is interested in systems or devices that increase damping. Reductions in resonant or random responses result in decreased oscillatory stresses and increase in the fatigue life of structures, in the reliability of mechanical devices, and in mechanical impedance (which tends to improve the effectiveness of vibration isolation).

Reduction of the spatially resonant responses of a wall or panel leads to decreased sound transmission through that
structure for frequencies above the coincidence frequency. Increased attenuation of propagating waves results in lesser transmissions of vibrations to neighboring structures. More rapid decay of free vibrations reduces the "ringing" sound of structures, thus leading to less noise, particularly from structures excited by repetitive impacts. This also tends to reduce structural fatigue.

C. MEASURES OF DAMPING

Damping may be quantified in terms of any of the previously discussed primary effects. The corresponding commonly used measures of damping, defined below, are interrelated as follows: [Refs. 2,3]

\[
\eta = \frac{\psi}{2\pi} = \frac{2.20}{f_n T_{60}} = \frac{\Delta t}{27.3 f_n^2} = \frac{\delta}{\pi} = \frac{\Delta_T}{13.6}
\]

The loss factor \( \eta \) and damping capacity \( \psi \) are defined directly in terms of the cyclic energy dissipation; the damping capacity represents the fraction of the system's vibrational energy that is dissipated per cycle of the vibration, and the loss factor similarly is defined as the fraction of the system's energy that is dissipated per radian of the vibratory motion.

On the other hand, \( T_{60} \), \( \Delta t \), are related to the rate of decay of free vibrations. \( T_{60} \) denotes the reverberation time (in seconds), defined (in analogy to the related
room-acoustic measure) as the time within which the vibration level of a system vibrating freely at a frequency $f_n$ (Hz) decreases by 60 dB (i.e., the amplitude decreases to 1/1000 of its original value). A related measure, decay rate $\Delta_t$ (dB/sec), represents the rate of reduction of the vibration (acceleration or displacement velocity) level. The logarithmic decrement $\delta$ is defined as the natural logarithm of the ratio of a peak excursion of a freely vibrating system to the peak excursion one cycle (period) later. The spatial decay $\Delta_T$ (dB/wavelength) represents the reduction in the steady-state vibration level with distance that occurs along a long beam vibrating in flexure.

It should be noted that none of the measurements of damping depend on how the energy is dissipated. Within a cycle these measures make no reference to any damping mechanism. On the other hand, some other commonly employed measures of damping are defined on the basis of viscous damping (i.e., damping that results from a retarding force that is proportional to the velocity. The ratio of the magnitude of that force to the velocity is called the viscous damping coefficient and is commonly designated by $c$.

If a simple mass-spring-dashpot system (where the dashpot provides a viscous retarding force characterized by $c$) is deflected from equilibrium and released, it typically oscillates with ever decreasing amplitude. However, if $c$ is made large enough, no oscillations occur. Instead, the system
creeps toward its equilibrium position, never traversing it. The smallest viscous damping coefficient for which this non-oscillatory behavior is obtained—i.e., the viscous damping that represents the dividing line between oscillatory and nonoscillatory behavior is called the critical (viscous) damping coefficient $c_c$. It obeys $c_c = 2\sqrt{km}$ where $m$ denotes the mass and $k$ the spring stiffness. The "damping ratio" $c/c_c$, also called the fraction of critical damping and often given in terms of "percent of critical damping", is widely used to indicate damping magnitudes.

Two other measures of damping are derived from the steady state behavior of an ideal linear mass-spring-dashpot system that is driven by a sinusoidal force of constant amplitude [Ref. 4]. The amplification at resonance, often called "the Q" of the system is defined as the ratio of the amplitude that results at resonance to the amplitude that is obtained if the force acts quasi-statically (i.e., a frequency considerably below resonance). The proportional bandwidth $b$ takes account of the damping related broadening of the peak in a plot of response amplitude versus frequency; this nondimensional bandwidth is defined as $f/f_n$, where $f$ denotes the difference between the two frequencies (one above, and one below the resonance frequency $f_n$) at which the square of the response amplitude is one-half of its maximum value. For values of damping below critical, the aforementioned measure of damping are related to each other and to the previously discussed loss factor as:
\[ \eta = \frac{c}{c_c} = \frac{1}{Q} = b \]  \hspace{1cm} (2)

D. DAMPING MECHANISMS

For linear, viscously damped systems, all of the measures of damping discussed previously are independent of the amplitude. Amplitude independence also occurs for other damping mechanisms, and approximate amplitude-independence occurs for almost all systems, provided the damping is relatively small. Thus, amplitude independence cannot be taken as an indication that a system is viscously damped; nevertheless, damping of systems with unknown energy dissipation mechanism is often characterized in terms of an equivalent viscous damping coefficient.

Indeed, much analysis is carried out with the (usually tacit) assumption that the damping is viscous—largely because viscous damping leads to linear differential equations that can be solved relatively easily. It is fortunate that for many practical problems, e.g., where only certain response maxima are of concern, the details of the damping mechanism are unimportant. Thus, one may obtain reasonable response predictions even with inaccurate damping force-versus-velocity representations. However, realistic damping models are required for the analysis of cases where one is interested in details of the response motions.
Unlike mass, a single physical phenomenon and stiffness, which results from a very few physical effects, damping may be caused by a great variety of phenomena. These phenomena include mechanical hysteresis (also called material damping or internal friction), electromagnetic effects (notably eddy currents), friction due to motion relative to fluids or solid surfaces, and energy transport to adjacent structural components or fluids (including by acoustic radiation). This great variety of phenomena that can produce damping generally make damping difficult to predict and to eliminate, but enables one to conceive a variety of means for increasing it.
II. NATURE OF THE PROBLEM

A. BACKGROUND

To date, the majority of damping research has been conducted on test specimens that are subjected to high stresses (i.e., torsional vibrations and vibrating cantilevered beams). For most United States Naval applications, structures and components are designed for operation at relatively low stresses.

Damping characteristics are shown to be highly stress dependent [Ref. 5]. Therefore, the damping values obtained at high stresses do not generally apply for most naval applications of ship silencing problems.

David W. Taylor Naval Ship Research and Development Center (NSRDC), Annapolis, Maryland is the focal point for testing and evaluating new high damping material for possible use for the United States Navy. Realizing the wide scope of the work necessary for the testing and evaluation of any new material, the author decided to approach NSRDC with a proposal to offer assistance in determining the damping of candidate material.

B. OBJECTIVE

To design a test procedure that allows the measurement of damping in a plate specimen at low stress with the following variables:
1. Plate specimen (40 inches × 14 inches × 1 inch) and 40 inches × 14 inches × 2 inches), or in proportionally reduced sizes.

2. Frequency: Damping testing will be conducted in the acoustic range (100-20,000 Hz).

3. Environment: Testing will be conducted in lab air and in nondistilled water. Temperature will range from 30°F to 90°F.

The test procedure includes a complete modal analysis that is expandable to other geometric shapes. Actual testing and verification of the procedure are conducted on the Hewlett-Packard 5451C Digital Fourier Analysis System. General information on the HP-5451C system is included in Appendix A.

C. SCOPE OF WORK TO BE COMPLETED

NSRDC has provided four specimens for testing and to establish baseline data for future work. The test specimens currently on hand are: A) Cast manganese bronze, code DEQ; B) Cast nickel-aluminum bronze, code FTC; C) Steel plate, HY-130, code FTW; and D) Aluminum alloy, 5086-H116, code ESX. It is anticipated that after baseline data has been compiled for both air and water tests that additional specimens will be provided by NSRDC. These additional specimens are expected to include high damping alloys such as Sonoston and Incramute, as well as constrained layer specimens and composites, both metallic and non-metallic.
For the purpose of this thesis the following work was completed: (A) design and construction of the test chamber, (B) design of a test procedure for the evaluation of damping of specimen provided by NSRDC, (C) complete tank characterization by impulse hammer techniques, (D) measurement of damping of the cast nickel-aluminum bronze, code FTC, specimen by impulse hammer technique. All of the above testing was completed in lab air at normal ambient temperature.
III. DESIGN OF TEST CHAMBER AND THEORY OF CHARACTERIZATION

To accomplish the aforementioned objectives four major steps must be accomplished: (A) design of a test chamber, (B) characterization of the test chamber, (C) design of experimental procedure of the measurement of damping, and (D) utilization of the above test procedure for damping measurement on a plate specimen supplied by NSRDC.

A. DESIGN OF TEST CHAMBER

Design of the test chamber was based on three major considerations: (1) size of the largest specimen, (2) weight of the largest specimen, and (3) environmental conditions desired for testing. In addition, the fixture to support the specimen had to hold the specimen rigid so that there would be no swinging.

The overall outside dimension of the chamber (Figure 1) was chosen to be a rectangular box 48 inches wide by 36 inches deep by 72 inches high. This allowed for ample clearance for the largest specimen (40 inches × 14 inches × 2 inches) on all surfaces and to limit the amount of reflected acoustic energy if acoustic absorption material was required on the inside surfaces. The test chamber was constructed from one quarter inch plate steel supported by a frame constructed from 2 inch steel thick wall square tubing. The 2 inch square tubing
Figure 1. Experiment test chamber
was welded at all mating edges to give the desired rigidity
and to prevent water entrapment (future corrosion problems).

The specimen fixture (Figure 2) was constructed from
3 inches × 2 inches thick wall steel tubing and is designed
to give a deflection, from a 400 pound specimen, of less than
0.04 inch.

To accommodate various sized specimen the random frequency
exciter base (Figure 3) can be adjusted within the 2 inch
rectangular tubing frame work of the upper half of the rear
panel (Figure 4). The random acoustic exciter can be adjusted
from 8 inches below the specimen fixture to 24 inches below
the specimen fixture. This feature allows for a wide variety
of centered excitation from the end of a specimen plate to
the center of a specimen plate.

To allow the removal and placement of various size speci-
men, the front panel of the test chamber is bolted to the
test chamber frame and a water gasket is formed by using a
RTV silicone adhesive sealant. Because of the bonding of
the RTV all water test must be conducted with the steel front
panel. For air testing the optional clear front panel may
be used because no sealant is required.

The preservation of the test chamber was accomplished by
sandblasting all steel parts, after protecting front panel
and specimen fixture bolt threads, one spray paint coat of
zinc chromate primer and one hand brushed top coat of enamel.

To allow testing in lab air and nondistilled water, the
test chamber is water tight with a 1 inch, valved, drain
Figure 3. Specimen exciter base with support mechanism
line near the bottom of the tank. Heating and cooling of the tank is accomplished by copper tubing (3/8 inch O.D.) coiled over entire outside surface of the back panel (Figure 5). The coils are spaced 3 inches apart and are 40 inches wide. To heat or cool the test chamber a mixture of 50% ethol glycol (automotive) antifreeze is supplied from a storage tank (Figure 6) at the desired temperature and is pumped through the heating and cooling coil.

The entire test chamber is insulated with one and one half inches of rigid foam insulation, with the exception of the rear panel, which is insulated with standard residential batt insulation.

B. THEORY OF THE CHARACTERIZATION OF TEST CHAMBER

The characterization of the test chamber was accomplished by use of the impulse technique for structural frequency response testing [Ref. 7] and the Fourier transfer function capability of the HP-5451C Fourier Analyzer.

1. Theory of Frequency Response Function

The measurement of the frequency response function is the heart of modal analysis. The frequency response function $H(f)$ is defined in terms of the single input/single output system, as the ratio of the Fourier transforms of the system output or response $v(t)$ to the system input or excitation $u(t)$, Equation (3).

$$H(f) = \frac{V(f)}{U(f)} \quad (3)$$
Figure 5. Cooling/heating coil on outside of rear panel
where:

\[ V(f) = \text{Fourier transform of system output } v(t) \]
\[ U(f) = \text{Fourier transform of system input } u(t). \]

The only requirements for a complete description of the frequency response function are that the input and output signals be Fourier transformable, a condition that is met by all physically realizable systems, and that the input signal be non-zero at all frequencies of interest. If the system is nonlinear or time-variant, the frequency response function will not be unique, but will be a function of the amplitude of the input signal in the case of a nonlinear system and a function of time in the case of a system with time-varying properties.

The frequency response function may be computed directly from the definition as the ratio of the Fourier transforms of the output and input signals. However, better results are obtained in practice by computing the frequency response function as the ratio of the cross-spectrum between the input and output to the power spectrum of the input, Equation (4). This relationship is derived by multiplying the numerator and denominator of the right-hand side of Equation (1) by the complex conjugate of the input Fourier transform.

\[ H(f) = \frac{G_{uv}(f)}{G_u(f)} \quad (4) \]
where:

\[ G_{uv}(f) = U^*(f)V(f), \text{ cross-spectrum between } u(t) \text{ and } v(t) \]

\[ G_{uv}(f) = U^*(f)U(f), \text{ power spectrum of } u(t) \]

\[ U^* = \text{ complex conjugate of } U(f). \]

The usefulness of this form of the frequency response function can be seen by considering the practical single input/single output measurement situation, where \( m(t) \) and \( n(t) \) represent noise at the input and output measurement points, respectively.

The measured frequency response function \( H'(f) \) is given by the equation:

\[
H'(f) = \frac{Y(f)}{X(f)} = \frac{V(f) + N(f)}{U(f) + N(f)}
\]

where the upper case letters denote the Fourier transform of the corresponding time domain signals.

In this form, the measured frequency response will be a good approximation of the true frequency response only if the measurement noise at both the input and output measurement points is small relative to the input and output signals. Multiplying the numerator and denominator of the right-hand side of Equation (5) by the complex conjugate of \( X(f) \) yields

\[
H'(f) = \frac{G_{uv}(f) + G_{un}(f) + G_{mv}(f) + G_{mn}(f)}{G_u(f) + G_{um}(f) + G_{mn}(f) + G_m(f)}
\]
Now, if the measurement noise signals \( m(t) \) and \( n(t) \) are noncoherent with each other and with the input signal \( u(t) \), then the expected value of the cross-spectrum terms involving \( m \) and \( n \) in Equation (6) will equal zero, yielding

\[
H'(f) = \frac{G_{uv}(f)}{G_u(f) + G_m(f)} = \frac{H(f)}{1 + \frac{G_m(f)}{G_u(f)}}
\]

where \( H(f) \) is the desired true frequency response function.

Thus, if the noise-to-signal ratio at the input measurement point \( [G_m(f)/G_u(f)] \) is much less than 1, the measured frequency response will closely approximate the desired true frequency response function.

It should be pointed out here that there is an inherent bias error associated with the computation of the cross-spectrum and the magnitude of this bias error is inversely proportional to the number of averages in the computation. Thus, the greater the measurement noise, the greater the number of averages required to approach the expected value of the cross-spectrum between the input and the output measurement signals. With measurement techniques employing many averages, the bias error can usually be reduced to an insignificant level so that it is only necessary to minimize the noise in the measurement of the input signal. However, if there is significant measurement noise and only a few averages are used, then the computed values of the cross-spectrum terms
involving the noise signals in Equation (6) can be large relative to the true cross-spectrum, with resulting large errors in the measured frequency response function. In general, only a few averages are used in the impulse technique; otherwise, one of its major advantages—its speed—is lost. Therefore, it is important to minimize measurement noise in both the input and output signals when using the impulse technique. The cross-spectrum bias error and its effects are discussed in more detail in Reference 8.

Coherence Function. There is another important reason for computing the frequency response function in terms of the cross-spectrum: it allows the computation of the coherence function between the input and output signals. The coherence function is defined by the equation

\[ \gamma^2_{xy}(f) = \frac{|G_{xy}(f)|^2}{G_x(f)G_y(f)} \]  

(8)

According to the definitions of the power spectrum and the cross-spectrum, the coherence function will be identically equal to 1 if there is no measurement noise and the system is linear. The minimum value of the coherence function, which occurs when the two signals are totally uncorrelated, is 0. Thus, the coherence function is a measure of the contamination of the two signals in terms of noise and nonlinear effects, with very low contamination indicated for values close to 1.
Since the cross-spectrum is included in the definition of the coherence function, the cross-spectrum bias error must be reduced to an acceptable level if a good statistical estimate of the coherence function is to be achieved. As stated above, the number of averages used in the impulse technique is usually not great enough to significantly reduce the bias error. However, the coherence function is still useful for indicating the importance of noise in the impulse technique. This is because noise in the signals causes variance in the value of the coherence function with frequency. This effect is illustrated in the section on measurement procedures.

2. Display of Frequency Response

The frequency response function is complex—that is, it has associated with it both magnitude and phase. Therefore, it can be displayed in a number of forms, including magnitude and phase versus frequency, real and imaginary magnitudes versus frequency, and imaginary magnitude versus real magnitude. Each of these types of displays has its own particular usefulness. The most common type of display for structural frequency response data is magnitude and frequency plotted logarithmically. This type of display, with the magnitude in terms of compliance (ratio of displacement to force) is called a Bode plot. In this form of the frequency response function, resonances occur as peaks in compliance plots (points of maximum dynamic weakness) and all resonance peaks of equal damping have the same width regardless
of resonance frequency. Lines of constant dynamic stiffness have zero slope, and mass-dominated frequency response lines have a -12 dB-per-octave slope. Resonances occur as nearly circular arcs in the complex plane (real versus imaginary plot) with frequency increasing in a clockwise direction around the arc. In the case of real normal modes (which occur in systems with relatively low damping and with resonances well-separated in frequency), each resonance arc is approximately tangent with, and lies below, the real axis and is symmetric about the imaginary axis when the frequency response is expressed as compliance. The complex plane plot is useful when certain types of analytical curve fitting operations are being performed on the frequency response data. The plots of the real and imaginary magnitudes of frequency response versus frequency are most useful when dealing with real normal modes. In this case the resonances will occur as peaks in the imaginary magnitude plot and the real magnitude will pass through zero at the resonance frequency when the frequency response is expressed as compliance.

The frequency response characteristics of a structural element are determined by measuring a set of cross-frequency response functions as discussed in Reference 9. The cross-frequency response functions may be obtained by exciting at one location on the structure and measuring response at various locations, or by measuring the response at a single location to excitation at various locations. The resulting
Frequency response functions comprise one column of the transfer matrix in the first case, and one row of the transfer matrix in the second case. Either set will, in general, completely define the modal characteristics of the structural element. In mathematical terms the set of frequency response functions yields the eigenvalues and eigenvectors, which are, in general, complex terms. The real part of an eigenvalue is the damping and the imaginary part is the frequency associated with a given resonance. Each eigenvector defines a resonance mode shape.

With real normal modes, each point on a structure is either exactly in-phase or exactly 180 degrees out-of-phase with any other point at the resonance frequency. Certain types of damping which are often encountered in practice will cause the eigenvectors to have non-zero imaginary components, resulting in complex mode shapes. When a mode is complex, the relative phase associated with a point on a structure is some value other than 0 or 180 degrees, with the result that node lines (lines of zero deflection) are not stationary. Precise description of complex modes requires that some type of analytical curve fitting technique be applied to the frequency response data.

The frequency response function of an operating system can be computed if the system input and output signals meet previously stated requirements of Fourier transformability and non-zero value, assuming the system input and response
can be measured. However, in practice there are usually multiple inputs to the system—either several inputs at different locations or inputs in more than one direction at a given location. In the case of multiple coherent inputs, the complexity of the analysis is greatly increased. For this reason, and the difficulty of accurately monitoring operating inputs, frequency response measurements are usually made by applying the system input "artificially" through some type of exciter. It is in the form of the input signal and the way it is applied to the structure that the wide variety of frequency response testing techniques arises.

The usefulness of the impulse technique lies in the fact that the energy in an impulse is distributed continuously in the frequency domain rather than occurring at discrete spectral lines as in the case of periodic signals. Thus, an impulse force will excite all resonances within its useful frequency range. The extent of the useful frequency range of an impulse is a function of the shape of the impulse and its time duration. For a square pulse the frequencies of the zero crossings are at integral multiples of the inverse of the time duration of the impulse, illustrating the very important inverse relationship between the time duration of an impulse and its frequency content.

The useful frequency range of an impulse is also a function of the shape of the impulse. By varying the weight and hardness of an impacting device and the manner in which
the impact is applied, the shape and time duration of the impulse produced can be varied to suit the measurement requirements.

a. Nonlinearities in the Structures

Excitation of a nonlinear system by a pure-random signal will yield the best estimate (in a mean-square sense) of the linear system response. Excitation by a pure sine wave is also useful for studying nonlinear systems because it allows precise control of the input spectrum level. However, the impulse technique, because of its very high ratio of peak level to total energy, is particularly ill-suited for testing nonlinear systems. Therefore, it is important to understand the various types of nonlinearities that can occur in structural systems and to be able to recognize nonlinearities in measured frequency response functions.

One of the most common types of nonlinearities encountered in structures is that due to clearance between parts. This type of nonlinearity is frequently encountered, for example, when testing gear systems and shafts mounted in bearings. The effects of this type of nonlinearity on measured frequency response functions when using impulse excitation are poor estimates of static stiffness values and poor repeatability of the frequency response estimates. Also, the apparent damping in the estimates will be greater than the actual examples.
The best method of dealing with this type of nonlinearity is to preload the system to take up clearances. Care must be taken when this is done, however, because any preload will change the boundary conditions of the structure and can itself lead to erroneous frequency response estimates. The usual approach is to apply the preload through a very soft spring so that the resonances associated with the preload lie below the frequency range of interest.

Another type of nonlinearity that is frequently encountered is nonlinear damping. Nonlinear damping effects are usually associated with joints in the structure, where the damping is a function of the relative displacement at the joint. In general, the frequency response estimates obtained by the impulse technique will agree most closely with those obtained with a low level of continuous excitation. However, if the point of excitation is close to a location where nonlinear damping occurs, there will be high relative motion at that location, and the apparent damping in the measured frequency response will be high. In systems with low damping, this will give the measured frequency response a discontinuous appearance, due to the varying level of damping as the response to the impulse attenuates with time.

The third type of nonlinearity that commonly occurs in structures is load-sensitive stiffness, where the spring rate of elastic elements either increases or decreases with load. The most direct way to identify this type of
nonlinearity is to measure frequency response as a function of static preload and observe the change in resonance frequencies.

b. Signal Processing

The particular characteristics of an impulsive force signal and the resulting structural response signal make the impulse technique especially susceptible to two problems: noise and truncation errors. While these problems occur to some extent with other frequency response testing techniques, their unique importance in the impulse technique requires special signal processing methods.

It was pointed out in the previous section that the usable frequency range for an impulse depends on the shape and time duration of the impulse. In order to insure that there is sufficient force over the frequency range of interest, it is necessary that the first zero crossing of the Fourier transform of the impulse be well above the maximum frequency of interest. For a given time duration the first zero crossing occurs at the lowest frequency for a square pulse. For that type of pulse the first zero crossing occurs at a frequency equal to the inverse of the time duration. A good rule of thumb, then, is to insure that the duration of the impulse is less than 2Δt, where t is the sampling interval in the analog-to-digital conversion process. This would put the first zero crossing of the Fourier transform of a square pulse at the Nyquist folding frequency, and the
first zero crossing of other pulse shapes above the Nyquist folding frequency.

The sample length is equal to \( N \Delta t \) where \( N \) is the number of digital values in each sample. A typical value of \( N \) is 1024. Thus, the duration of the impulse is very short relative to the sample length. This means that the total energy of noise represented in the time-sample can be on the order of the energy of the impulse, even for high signal-to-noise ratios. The noise problem is further aggravated when employing the zoom transform, which yields increased resolution in a given frequency band by effectively increasing the sample length.

With other techniques, the effects of noise are reduced by averaging the power spectrum and cross-spectrum functions prior to the computation of the frequency response function. However, only a few averages are usually used in the impulse technique. Otherwise, the time advantage of the technique is lost. Therefore, special time-sample windows have been developed for the impulse technique.

At first thought it might seem appropriate to just set all time-sample values beyond the impulse to zero, since it is known that the true signal value after the impulse is zero. However, this would be equivalent to multiplying the signal by a narrow rectangular window. In applying any type of window, it is important to keep in mind that multiplication by a window in one domain is equivalent to
convolution of the Fourier transforms of the window and the data in the other domain, resulting in distortion of the transformed signal. This distortion will be minimized by minimizing the width of the main lobe of the window transform and suppressing its side lobes. However, there is a fundamental conflict between these requirements and the reduction of noise in the time-sample because both the width of the main lobe and the amount of noise reduction are inversely proportional to the width of the window in the time domain. To further complicate the situation, suppression of the side lobes is generally achieved at the expense of broadening the main lobe.

A good compromise has been arrived at in practice in the form of a window with unity amplitude for the duration of the impulse and a cosine taper, with a duration of 1/16 of the sample time, from unity to zero.

Noise problems may also be encountered in the response signal, particularly when dealing with heavily damped systems and when using zoom transform analysis. In both cases the duration of the response signal may be short relative to the total sample time, so that noise may comprise a significant portion of the total energy in the time-sample even with relatively high signal-to-noise ratios. Another error in the response signal that is encountered when testing lightly damped structures occurs when the response signal does not significantly decay in the sample window. In this case
the resulting time-sample is equivalent to multiplying the true response signal by a rectangular window, with the result that the frequency resolution may not be sufficient to resolve individual resonances.

An exponential window has been developed to reduce the errors that occur in both situations described above. The window decays exponentially from 1 to a value of 0.05 in the sample time. It can be applied directly to the time-sample of the response signal or to the impulse response function. As with all windows, the exponential window does change the resulting measured frequency response function; but its only effect is to increase the apparent damping in the resonances. It does not change the resonance frequencies and, because the effect of the exponential window is the same on all frequency response measurements, it will not alter the measured mode shapes if applied to all measured frequency response functions. In addition to reducing noise and truncation errors, the exponential window will also reduce errors which often occur when testing lightly damped systems in which the damping varies with the measurement position on the structure.

Because the exponential window increases the apparent deampping in the resonance modes, there is a tendency of the window to couple closely spaced resonance modes. Zoom transform analysis may be required in some cases to allow sufficient resolution of closely spaced modes when using the exponential window.
Zoom transform analysis is discussed in some detail along with several examples in Reference 9. It is a very valuable tool in impulse testing, as it is in other frequency response measurement techniques. The effect of the zoom transform is to increase the resolution of the analysis by allowing independent selection of the upper and lower frequency limits of the analysis band. With the zoom transform, for example, it is possible to perform an analysis in the frequency range from 900 to 1000 Hz as opposed to the corresponding base-band range of 0 to 1000 Hz, resulting in a 10-to-1 increase in resolution, for a given sample size N, in the 900 to 1000 Hz band. Because of greatly increased resolution possible with the zoom transform, it can be effectively used in frequency response testing to separate closely spaced resonance modes.

There are two important effects of the zoom transform in the impulse technique, both associated with the resulting increase in sample time. The first effect is to make possible much better estimates of damping in lightly damped systems. This is due to the reduction of truncation errors in the sampled response signal. The second effect, mentioned previously, is aggravation of the noise problem in both the input and response signals. The second effect makes it essential that force and response windows be applied to the data in most cases when using the zoom transform with the impulse technique.
3. **HP-5451C Fourier Transfer Function**

A transfer function, as determined by the HP-5451C Fourier Analyzer, is a mathematical description of a system. It can be defined as:

\[
\text{transfer function} = \frac{\text{Fourier transform of output}}{\text{Fourier transform of input}}
\]

or equivalently,

\[
\text{transfer function} = \frac{\text{average cross power spectrum of input and output}}{\text{average power spectrum of input}}
\]

The coherence function measures the degree of causality between any two signals. It can, therefore, be used to check the validity of the transfer function. When a transfer function is computed it may not be obvious that there are extraneous inputs, or that the system is nonlinear. Both of these factors would introduce error in the computed transfer function. The "Transfer Function" program of the HP-5451C is used to compute the transfer and coherence functions. A program flow chart and a program listing are provided in Appendix B.
IV. THEORY OF THE TEST PROCEDURE TO MEASURE DAMPING

The design concept of the experimental procedure is based on the Hewlett-Packard modal analysis software specifically designed for the HP-5451C Fourier Analyzer. The theory of the complex modes for damped oscillatory mechanical systems is described below, much of which is explained in greater detail in the Modal Analysis Operating and Service Manual (Option 402).

A. MODAL THEORY OF OPERATION [REF. 6]

Assume that the motion of a physical system can be described by a set of n simultaneous second-order linear differential equations in the time domain, given by,

\[ \ddot{M}x + C\dot{x} + Kx = f \] (9)

where the dots denote differentiation with respect to time.

\[ f = f(t) \]

is the applied force vector, and

\[ x = x(t) \]

is the resulting displacement vector, while M, C, and K are the \((n \times n)\) mass, damping, and stiffness matrices respectively.
Our attention will be limited to symmetric matrices, and to real element values in M, C, and K.

Taking the Laplace transform of the system equations gives

\[ B(x)X(s) = F(s), \quad (10) \]

where:

\[ B(s) = Ms^2 + Cs + K \quad (11) \]

Here \( s \) is the Laplace variable, and \( F(s) \) is the applied force vector and \( X(s) \) is the resulting displacement vector in the Laplace domain. \( B(s) \) is called the system matrix, and the transfer matrix \( H(s) \) is defined as

\[ H(s) = B(s)^{-1} \quad (12) \]

which implies that

\[ H(s)F(s) = X(s) \quad (13) \]

Each element of the transfer matrix is a transfer function. The elements of \( B \) are quadratic functions of \( s \), and since \( H = B^{-1} \), it follows that the elements of \( H \) are rational fractions in \( s \), with \( \text{det}(B) \) as the denominator. Thus, \( H(s) \) can always be represented in partial fraction form.

If it is assumed that the poles of \( H \), i.e., the roots of \( \text{det}(B) = 0 \), are of unit multiplicity, then \( H \) can be expressed as
\[ H(s) = \frac{2n}{\sum_{k=1}^{n} \frac{a_k}{s-P_k}} \]  \quad (n \times n) (14)

The poles occur at \( s = P_k \) (zeros of \( \text{det}(B) \)), and each pole has \( n \times n \) residue matrix \( a_k \) associated with it. For an \( n \)th order oscillatory system, there will always be \( 2n \) poles, but they will appear in complex conjugate pairs. Each complex pair of poles causes a mode of vibration in the structure. The poles are complex numbers expressed as

\[ P_k = -\sigma_k + i\omega_k \]  \quad (15)

where \( \sigma_k \) is the damping coefficient (a negative number for stable systems), and \( \omega_k \) is the natural frequency of oscillation. The resonant frequency is given by

\[ \Omega_k = \sigma_k^2 + \omega_k^2 \quad \text{(rad/sec)} \]  \quad (16)

and the damping factor is

\[ \zeta = \frac{\sigma_k}{\Omega_k} \]  \quad (17)

These coordinates are shown in Figure 7. When \( \zeta = 1 \), mode \( (k) \) is said to be critically damped. It is also possible that \( \zeta > 1 \). For this case the poles of mode \( (k) \) lie along the real (or damping) axis in the S-plane and it is said to be super-critically damped.
Modal vectors \((u_k)\) are now defined as solutions to the homogeneous equation

\[ B_k u_k = 0 \quad (10) \]

where:

\[ B_k = B(p_k) \]

for all \(k = 1, \ldots, 2n\).

In addition, pre-multiplying \(B\) times the equation (14) for \(H(s)\), multiplying by the scalar \((s-p_k)\) and letting \(s = p_k\) gives

53
\[ B_k a_k = 0 \] (19)

It follows by comparison of (10) with (19) that every column of \( a_k \) must contain the modal vector \( u_k \), for each \( k = 1, \ldots, 2n \).

Similarly, post multiplying \( B \) times the equation (14) for \( H(s) \), multiplying by the scalar \((s-P_k)\), and letting \( s = p_k \) gives

\[ a_k B_k = 0 \] (20)

for each \( k = i, \ldots, 2n \). This can be rewritten as

\[ B_k^t a_k^t = 0 \] (21)

where \( t \) denotes the transpose.

But \( B_k \) and hence \( a_k \) is assumed to be symmetric so equation (21) is the same as (19) and it follows that each row of \( a_k \) must also contain the vector \( u_k \) for each \( k = 1, \ldots, 2n \).

In order to satisfy the above conditions \( a_k \) must have the form

\[ a_k = A_k u_k u_k^t \quad (n \times n) \] (22)

where \( A_k \) is a scalar.

This pervasiveness of the modal vectors throughout the transfer function matrix is evidence of the so-called global property of a mode of vibration.
In these terms, $H$ can be rewritten as

$$H = \sum_{k=1}^{2n} \frac{A_k}{s-P_k} u_k^t u_k$$

(23)

and this is easily written in matrix form as

$$H = ULU^t \quad (n \times n)$$

(24)

where the columns of $U$ comprise the $u_k$ modal vectors:

$$U = [u_1 \ u_2 \ \ldots \ u_{2n}], \quad (n \times 2n)$$

(25)

and $L$ is a diagonal matrix containing all $s$ dependence

$$L = \begin{bmatrix}
\frac{A_1}{s-P_1} \\
\vdots \\
\frac{A_{2n}}{s-P_{2n}}
\end{bmatrix} \quad (2n \times 2n).$$

(26)

Pre-multiplying $H$ by $U^t$, equation (13), can be written as

$$(U^tUL)(U^tF) = (U^tX)$$

(27)

so that $U^t$ transforms the spatial vectors $F$ and $X$ to vectors $U^tF$ and $U^tX$ in modal coordinates. Similarly $U^tUL$ is the modal representation of $H$. Since $B(P_k)u_k = 0$, it follows that $B(P_k^*)u_k^* = 0$, so the modal vector associated with the conjugate pole $(P_k^*)$ is $u_k^*$ (the conjugate of $u_k$).
Thus, the above \( U \) matrix always contains conjugate pairs of modal vectors, and the \( L \) matrix always contains elements corresponding to conjugate pole pairs along its diagonal.

If \( U_1 \) is defined as that \((n \times n)\) part of \( U \) associated with positive poles, then \( U_1^* \) will correspond to the negative poles. Similarly, \( L \) can be broken into two parts, \( L_1 \) comprising the positive poles, and \( L_2 \) comprising the negative poles. It can then be represented

\[
H = U_1 L_1 U_1^* + U_1^* L_2 U_1^t
\]  
(28)

or in partitioned form as

\[
H = [U_1; U_1] \begin{bmatrix} L_1 & 0 \\ \hline \hline \hline \end{bmatrix} \begin{bmatrix} U_1^t & U_1^t \\ 0 & L_2 \end{bmatrix}
\]  
(29)

Each of these sub-matrices is \((n \times n)\) and only \( L_1 \) and \( L_2 \) are functions of \( s \).

\[
L_1 = \begin{bmatrix} A_1 \\ \hline \hline \hline \hline \end{bmatrix}, \quad L_2 = \begin{bmatrix} A_1 \\ \hline \hline \hline \hline \end{bmatrix}
\]

(30)

Then \( H \) can be written as:
\[
H = \sum_{k=1}^{n} u_k u_k^t \frac{a_k}{s-p_k} + u_k u_k^t \frac{A_k}{s-p_k^*}
\]  

(31)

Each element of the H matrix has a different zero in the S-plane, depending upon the values of \(A_k\) and \(u_k\) at each point, but the poles of each element of H are common, and occur at \(s = p_k\) and \(s = p_k^*\).

1. Identification of Modal Parameters

Because of the form of the \(a_k\) matrix, only one row or column of the transfer matrix need be measured and analyzed, since all processes of measuring the transfer matrix, the unknown parameters in equation (14) (i.e., the complex values of \(p_k\) and the complex values of the elements of one row or column or the residue matrix \(a_k\)) are identified.

Once one row or column of \(a_k\) has been identified, it is then possible to construct the rest of the rows and columns in \(a_k\). For example, if the \(q^{th}\) column of \(a_k\) is given by \(a_{kq}\), then

\[
a_{kq} = A_k u_{qk} u_k
\]

(32)

where \(u_{qk}\) is the \(q^{th}\) component of the modal vector. Since \(A_k u_{qk}\) is a scalar, it is clear that the vector of modal residues \(a_{kq}\) is proportional to the modal vector \(u_k\), for each \(k = 1, \ldots, 2n\).

Since \(A_k\) is a scaling constant it can be assumed that either \(A_k = 1\) or the square root of \(u^t u = 1\) without loss of generality in the following derivation.
Suppose $A_k = 1$, then
\[ a_k = u_k u_k^t \quad (33) \]

and for the $q^{th}$ column or row of $a_k$
\[ a_{kq} = u_{qk} u_k \quad (34) \]

Hence the $q^{th}$ element of $a_{kq}$ is
\[ a_{kqq} = (u_{qk})^2 \quad (35) \]

Since $u_{qk}$ is a scalar, equation (34) can be rewritten
\[ u_k = \frac{a_{kq}}{u_{qk}} \quad (36) \]

and substituting this back into (33) gives
\[ a_k = \frac{a_{kq} a_{kq}^t}{(u_{qk})^2} \quad (37) \]

or using equation (35)
\[ a_k = \frac{a_{kq} a_{kq}^t}{a_{kqq}} \quad (38) \]

Hence the entire matrix $a_k$ and therefore, the entire transfer function matrix $H(s)$ can be constructed once one row or column of residues $a_{kq}$ has been identified as well as the pole locations $p_k$, for each $k = 1, \ldots, 2n$. The residue $a_{kqq}$
is at the driving point of the structure, i.e., the point where the structure is excited.

2. **Impulse Response of Complex Modes**

It has been shown that a mode of vibration is represented by a complex conjugate pair of poles and a complex conjugate pair of modal vectors in the transfer matrix. Hence for a single mode of vibration \(k\) the transfer matrix is written

\[
H_k(s) = \frac{a_k}{s-p_k} + \frac{a_k}{s-p^*_k} \quad (n \times n)
\]  

(39)

It is convenient to remove a factor of \(2i\) from the residue matrix, that is to define another residue matrix \(r_k\) such that

\[
r_k = 2i a_k
\]

(40)

Then the transfer matrix is written as

\[
H_k(s) = \frac{r_k}{2i(s-p_k)} - \frac{r_k}{2i(s-p^*_k)}
\]

(41)

Each component of this matrix exhibits the rectangular (or coincident--quadrature) form shown in Figure 8 for each valued \(r_k\).
Taking the inverse Laplace transform of (41) gives

\[ x(t) = \frac{r_k}{2\pi} e^{\sigma_k t} p_k^* t - \frac{r^*_k}{2\pi} e^{\sigma_k t} p_k^* t \]  (42)

\[ x(t) = |r_k| e^{-\sigma_k t} \sin(\omega_k t + \alpha_k) \]  (43)

where \( \alpha_k \) = the angle of the complex residue \( r_k \).

Equation (43) is the impulse response of a complex mode of vibration. The mode shape is defined by the magnitude \( |r_k| \) and the phase angle \( \alpha_k \) at each point on the structure.
The mode shape matrix is printed in the Modal System in terms of these magnitude and phase angles.

3. **Modal Mass, Damping and Stiffness and Scaled Mode Shapes**

   Recall that the assumed symmetry of $H(s)$ along with the global nature of mode shapes implies that the residue matrix for each mode is proportional to $u_k u_k^t$. If $Q_k = 2iA_k$, then equation (22) becomes

   $$r_k = Q_k u_k u_k^t \quad (n \times n)$$ (44)

   Once a row or column of each residue matrix $r_k$ has been identified, the mode shape vectors $U_k$ can be calculated to within a constant of proportionality. This vector can be scaled large or small, if desired, by suitable choice of $Q_k$. Criteria for choosing $Q_k$ will emerge from a discussion of modal mass, damping, and stiffness.

   Consider a single degree of freedom system represented by

   $$mx + cx + kx = f$$ (45)

   where $m$, $c$, and $k$ are scalars. The Laplace Transform gives

   $$(ms^2 + cs + k)X(s) = F(s)$$ (46)

   so that

   $$H(s) = \frac{X(s)}{F(x)} = \frac{\frac{1}{m}}{s^2 + \frac{c}{m}s + \frac{k}{m}}$$ (47)
If the damping factor is less than one, the polynomial in the denominator can be factored.

\[
H(s) = \frac{\frac{1}{m}}{(s-p)(s-p^*)}
\]  

(48)

where

\[
p = \frac{-c}{2m} - \sqrt{\frac{c^2}{4m^2} - \frac{k}{m}}
\]

(49)

\[
p = \frac{-c}{2m} + i \sqrt{\frac{k}{m} - \left(\frac{c}{2m}\right)^2}
\]

(50)

\[
p = -\sigma + i\omega, \quad \sigma > 0, \quad \omega > 0
\]

(51)

and

\[
p^* = -\sigma - i\omega
\]

(52)

To express Equation (48) in the partial fraction form of Equation (39), solve for \(a\) and \(a^*\).

\[
H(s) = \frac{\frac{1}{m}}{(s-p)(s-p^*)} = \frac{a}{(s-p)} + \frac{a^*}{(s-p^*)}
\]

(53)

\[
\left.\frac{\frac{1}{m}}{(s-p^*)}\right|_{s=p} = \left[a + \frac{a^*(s-p)}{(s-p^*)}\right]_{s=p}
\]

(54)
\[ \frac{1}{m} \frac{1}{2i\omega} = a \]  \hspace{1cm} (55)

\[ \frac{1}{m} \frac{1}{-2i\omega} = a^* \]  \hspace{1cm} (56)

\[ H(s) = \frac{1}{2i(s-p)} + \frac{1}{2i(s-p^*)} \]  \hspace{1cm} (57)

The residue \( r \) can be found from Equations (40), (41), and (57).

\[ r = \frac{1}{m\omega} \]  \hspace{1cm} (58)

Now note that Equations (50), (51), and (58) give \( m \), \( c \), and \( k \) in terms of \( r \), \( \omega \), and \( \sigma \).

\[ m = \frac{1}{r\omega} \]  \hspace{1cm} (59)

\[ c = 2\sigma m \]  \hspace{1cm} (60)

\[ k = (\omega^2 + \sigma^2)m \]  \hspace{1cm} (61)

For multiple degree of freedom systems, the analogous definitions are made for each mode \( k \).

\[ m_k = \frac{1}{Q_k\omega_k} \]  \hspace{1cm} (62)

\[ c_k = 2\sigma_k m_k \]  \hspace{1cm} (63)
Equations (44), (62), (63), and (64) are used to calculate modal mass, damping, and stiffness and scaled mode shapes in the Print Step. $Q_k$ is arbitrary so it can be chosen to give unit mass, unit $u^T u$, etc.

The modal mass, stiffness, and damping coefficients can be interpreted as being the masses, dampers, and springs of decoupled, single degree of freedom systems. These systems are equivalent to the original system under a change of coordinates.

4. Measurement Implications of Modal Theory

A fundamental assumption of modal testing is that a mode of vibration can be excited from anywhere on an elastic structure, except of course along its node lines (zero points) where it can't be excited at all. This is another way of stating the result derived earlier (i.e., that the same modal vector--scaled by a different component of itself--is contained in every row and column of the transfer matrix). In addition, modal frequency and damping are constants which can be identified in any element of the transfer matrix, i.e., any transfer function taken from the structure.

It is important to recognize that this global mode shape concept exists within certain spatial boundaries, beyond which vibrations will not readily propagate. If two linear systems are completely isolated, then a single composite mode including both systems is not meaningful.
Conversely, it is important to include enough spatial points in the measured data set to describe all of the vibration modes of interest. If some region of a bounded system is not monitored or excited, or if points are not chosen sufficiently close together, then some modes cannot be adequately represented.

When a system is represented by the partial fraction form of its transfer matrix, a closed form solution for the displacement at any point, for any combination of modes, is readily obtainable using simple matrix-vector multiplication. This is particularly helpful when the response is only to a few modes of interest.

The primary purpose for using modal coordinates to describe the dynamics of a linear vibration structure is that it drastically reduces the amount of time and effort necessary to measure the dynamics in a laboratory. The essence of the modal concept is that once one row or column of the transfer function matrix has been determined, the entire matrix and hence the entire dynamics of the structure can be specified.

In order to obtain valid modal results with the Modal Analysis system, several important assumptions of the modal theory described here must be satisfied.

1. The structure must exhibit the behavior of a linear system. If you are not able to successfully curve fit equation (14) to measured transfer function data, then the behavior may not be linear.
2. The structure must exhibit the reciprocity of symmetry property. This can be verified by comparing the transfer function obtained from an excitation measurement at point A and response measurement at point B with the transfer function obtained from an excitation at point B and response at point A. This should be the same.

3. The structure must have distinct pole pairs. Since the Modal Analysis system uses analytical expression (6) to perform the curve fitting, it cannot fit repeated poles (repeated roots of det(B) = 0 because this expression was derived under the assumption that the poles are unique. This condition can be difficult to detect in one set of transfer function measurements. To insure that the poles are unique you should measure a different row or column of the transfer function matrix, and compare the mode shapes from the two sets of measurements. If they are not identical there is a strong possibility of repeated roots.

4. There may be more than one modal vector corresponding to a given pole, due to certain kinds of boundary conditions on the structure, i.e., symmetric conditions along several axes. To insure that this is not happening, you should compare modal vectors obtained from two or more rows or columns of the transfer matrix.
5. The transfer function $H(s)$ is defined in terms of an input force and a response displacement. Response velocity or acceleration can be converted to displacement. But non-force Modal Analysis system will calculate, print, and display modal data (i.e., poles and residues) for such measurements, however. The exceptions are the four mode shape scaling procedures that produce modal mass damping and stiffness. The system logica will reject an attempt to calculate these values, unless motion in response to force was measured.
V. PROCEDURE OF TEST CHAMBER CHARACTERIZATION

A. PROCEDURE OF DATA COLLECTION

After construction of the test chamber, preservation and painting, installation of cooling/heating coils, and installation of all insulation, the following is the procedure used for final characterization of the test chamber (with front panel removed).

1. Draw a 2.54 inch × 2.54 inch grid on entire inside surface of the test chamber. This grid is used to record location of impact or excitation, as well as location of response transducer (Figure 9).

2. Re-install exciter and support framework; re-install specimen fixture with specimen in place; and re-install test chamber top. Drill 1/4 inch hole through specimen for attachment of exciter driver rod. Install exciter driver rod with Force Transducer in place (Figure 10). NOTE: Acoustic exciter and force transducer will not be used in this work. For impulse hammer testing disconnect exciter from specimen.

3. Using accelerometer mounting wax, attach pick-up transducer to desired location. Connect signal conditioners and wiring as instructed in the operating manuals. System flow chart is presented in Figure 11. Impulse hammer, signal conditioners and typical attachment of pickup transducer are
Figure 9. Detail of 10 cm × 10 cm grid on inside of experiment test chamber
Figure 10. Exciter drive mechanism with force transducer
Figure 11. Impulse hammer technique flow chart
presented in Figures 12 and 13. All calibration data of sensors and mounting and instructions are included in Appendix C.

4. Turn on the HP-5451C Fourier Analyzer (HP-2648A Graphics Terminal will automatically be energized). Insert "Modal" disk into disk drive. Energize disk drive. When "Disk Ready" light comes on "Boot" the system as outlined in volume 1 of System Operating Manual (Figures 14 and 15). Activate user assignable keys f1 and f2 on the graphics terminal to "Graphics Mode" and "Alpha Mode" respectively. Procedures for key assignments are found in volume 5 of the Computer Operating Manual.

5. From this point on, the procedure for taking impulse data is interactive. Locally generated computer programs, which are designed to allow the data generated by impulse hammer technique to be stored and later processed by the off-line Modal software, are presented in Appendix D. What follows is a typical step-by-step impulse data taking session assuming pick-up transducer at location L051 and hammer impulse location at L066.

The following abbreviations will be used:

- C = computer keyboard entry
- D = graphics terminal display
- G = graphics terminal entry
- R = user response
Figure 12. Impulse hammer with signal conditioners
Figure 13. Typical attachment of pick-up transducer
Figure 14. HP-5451C Fourier Analyzer
Step Number

1. Normal "Boot" of modal with f1 & f2 programmed.
2. D: SET UP DATA NUMBER? (1 to 5).
3. R;C: 2 [ENTER] (want to preserve Demonstration data in set up #1, set ups #2 through #5 are available).
4. D: LIST(1), EDIT(2), FINISH(3)?
5. R;C: 2 [ENTER] (must enter particular transducer specifications and general information for a particular test).
6. D: LABEL?
9. R;G: 1,6 [RETURN] (you may modify one or any sequential series of lines of the set up data).
10. D: 1. NO. OF MEASUREMENTS?
11. R;G: 16 [RETURN] (A measurement is one record of storage on the disk. One record is 1024 words in length, and a test is n measurements stored in n contiguous records on the disk).
12. D: STARTING LOCATION OF MEAS. STORAGE (1 to 485).
13. R;G: 5 [RETURN] (storage location 1 through 255 are on the removable disk, and storage locations 266 to 485 are on the fixed disk).
14. D: DATA BLOCK SIZE? (64 to 2048).
15. R;G: 1024 [RETURN] (block size of 1024 will give maximum zoom capability).
16. D: 4. POLAR TRANS. FUNCT. DISPLAY? LINEAR(1) or LOG(2) MAGNITUDE.

17. R;G: 2 [RETURN].

18. D: 5. TRANSUDER UNITS? ENGLISH(1), METRIC(2).

19. R;G: 1 [RETURN].

20. D: TRANSUDER SCALE FACTORS (VALUE, UNITS)
   1 = MV/G, 2 = VM/IPS, 3 = MV/IN, 4 = MV/LB
   A. INPUT?

21. R;G: 10.6, 4 [RETURN] (values are obtained from calibration data of sensors).

22. D: B. RESPONSE?

23. R;G: 10.12, 4 [RETURN].

24. D: G. AMPLIFIER GAINS?
   A. INPUT?

25. R;G: 1 [RETURN] (enter transducer amplifier switch setting).

26. D: B. RESPONSE?

27. R;G: 1 [RETURN]

28. D: LIST(1), EDIT(2), FINISH(3)?

   NOTE: At this point (1) may be entered and all data may be checked. If any corrections are required, enter (2) and use procedures outlined in steps 5 through 27.

29. R;G: 3 [RETURN].

   NOTE: The "set up" portion of the test is now complete. The next step is the "measure" portion.

30. D: STEP NUMBER? (1 to 5).
31. R;C: 2 [ENTER] (the computer is now in the "measure" mode).

32. D: TRANS(1), RANDOM(2), DISK(3), USER(4),
    FOURIER/ZOOM/GRAFICS(5)?

33. R;C: 5 [ENTER] (the HP 5451-C is now ready to set up for an impulse test).

34. R;C: SELECT BLOCK SIZE (1024 MAX FOR ZOOM)
    SET ADC FREQUENCY RANGE AS DESIRED
    SET TREGGER TO INTERNAL

35. R;C: [JUMP] 0 [SPACE] 1 [ENTER] (this command calls the user programs 1,50,51,52, and 59 for use).
    TRANSFER FUNCTION MEASUREMENTS
    ARE HP FILTERS INSTALLED?

36. D: (0 = NO 1 = YES)

37. R;C: 1 [ENTER]

38. D: CUT-OFF FREQUENCY (CHANNEL NO)?)

    NOTE: To eliminate unwanted or meaningless data beyond range of impulse hammer, enter channel number above which the computer will not sample data.

    EXAMPLE: 0-500 lb. hammer useful range is from 0-6000 Hz, with block size 1024 selected channels above 307 are meaningless.

39. R;C: 307 [ENTER]

40. D: IMPACT (1) OR RANDOM (2) EXCITATION?

41. R;C: 1 [ENTER]

42. D: NO. OF AVERAGES?
43. R;C: 4 [ENTER]
44. D: BASEBAND OR ZOOM MEASUREMENT?
    (0 = BASEBAND 1 = ZOOM)
45. R;C: 0 [ENTER] (first measurement must be baseband).
46. D: BASEBAND IMPACT MEASUREMENT
    PRESS CONTINUE FOR MEASUREMENT
NOTE: At this point trigger and analog-Digital Converter
levels need to be set to do this. The following keystrokes should be done.
47. R;C: SET REPEAT/SINGLE SWITCH TO REPEAT
    PRESS ANALOG IN
    SET ADC RANGE SWITCHES AND TRIGGER SLOPE AND LEVEL
    SET ADC TO SINGLE WHEN READY AND PRESS CONTINUE
48. R;C: [CONTINUE].
NOTE: Impact 4 times. The computer will beep after each
data entry. After the computer processes the data the
ready light will come on and the data in Block 0 will
be displayed on the small screen of the computer.
Processed data are stored in the following locations and
can be displayed by the key strokes:
[DISPLAY]0 [ENTER] Log transfer function
[DISPLAY]1 [ENTER] Coherence
[DISPLAY]2 [ENTER] Input power spectrum
[DISPLAY]3 [ENTER] Output power spectrum
[DISPLAY]4 [ENTER] Cross power spectrum
[DISPLAY]5 [ENTER] Raw data
To obtain hard copy of any of the above outputs execute the following key commands [USER] 5821 [SPACE] 35 [ENTER]. Then enter [USER] [PLOT] [ENTER]. To return plotting to graphics execute [USER] 5821 [SPACE] 6 [ENTER], and when the Graphics Terminal is in the Graphics mode (depress f1) and [USER] [PLOT] [ENTER] is executed all plotting to be sent to the graphics terminal. To continue, f2 must be depressed. After all preliminary graphing is completed on baseband data press [continue]. Typical plots of baseband data are presented in Figures 16 through 20.

49. D: PRESS CONTINUE WHEN READY
50. R;C: [CONTINUE]
51. D: SAVE DATA?
   (0 = NO 1 = YES)
52. R;C: 0[ENTER]
53. D: MAKE ANOTHER MEASUREMENT?
   () = NO 1 = YES).
54. R;C: 1[ENTER]
55. D: NO. OF AVERAGES?
56. R;C: R[ENTER]
57. D: BASEBAND OR ZOOM MEASUREMENT?
   () = BASEBAND 1 = ZOOM)
58. R;C: 1[ENTER]
59. D: ZOOM IMPACT MEASUREMENT
   MOVE CURSOR TO START FREQUENCY
   PRESS "VALUE" (SWITCH REGISTER 11)
Figure 16. Log of transfer function of baseband data
NOTE: The desired start and end frequency for the zoom measurement is not entered with use of cursor.

60. D: "TYPICAL" ZOOM IMPACT MEASUREMENT

MOVE CURSOR TO START FREQUENCY
PRESS "VALUE" (SWITCH REGISTER 11)
CHANNEL = 49.0000
FREQUENCY = 957.0999 Hz
AMPLITUDE = -700.0026 E -9.0000
MOVE CURSOR TO END FREQUENCY
PRESS "VALUE"

61. D: CHANNEL = 134.0000
FREQUENCY = 2618.0000 Hz
AMPLITUDE = -381.7491 E -9.0000
ANALYZE OLD OR NEW DATA?
1 = OLD (FROM THROUGHPUT FILE)
2 = NEW (WILL WRITE TO THROUGHPUT FILE)
PRESS CONTINUE FOR MEASUREMENT

62. R;C: 2 [ENTER]

NOTE: Whenever zoom is being done new data is required.

63. R;C: [CONTINUE]

NOTE: For zoom measurement the impact should be a series of rapid impacts with a varying interval between impacts. The trigger light will remain lit throughout the data taking session.

64. D: ANALYZING THROUGHPUT DATA "TYPICAL"

CNTR FREQ 1479 Hz/DIV 125.0
DF: 000.2441406 BLOCKS LEFT 25
ZOOM POWER 4
NOTE: After data has been analyzed and desired plots made as described in step 43, ZOOM data should be stored for later use by the off-line modal software. A recommended data sheet is shown in Figure 21. It is important to log number of measurements and start location of measurement storage.

65. D: PRESS CONTINUE WHEN READY

NOTE: Generally it is desired to make multiple tests at different locations at the same frequency range so that a modal analysis can be done at a later time.

66. R;C: [CONTINUE]

67. D: SAVE DATA?

(0 = No 1 = YES)

NOTE: The data taking session can be repeated by entering "0".

68. R;C: 1[ENTER]

69. D: DATA STORAGE NO.?

70. R;C: XXX[ENTER]

NOTE: Data storage registers 0 through 500 are available for permanent storage of the data.

71. D: MAKE ANOTHER MEASUREMENT?

(0 = NO 1 = YES)

NOTE: From this point the steps 57 through 71 will be repeated for as many zoom measurements as desired. Each zoom measurement will be stored in a next higher record number with the begin record number known.
<table>
<thead>
<tr>
<th>SETUP</th>
<th>TEST</th>
<th>DATE</th>
<th>TYPE</th>
<th>RANGE</th>
<th>INPUT LOC.</th>
<th>RESPONSE</th>
<th>FILTER</th>
<th>STORE</th>
<th>DATA</th>
</tr>
</thead>
<tbody>
<tr>
<td>NO.</td>
<td>NO.</td>
<td>(HorZ)</td>
<td>(MAX.FREQ)</td>
<td>(GRID NO.)</td>
<td>LOC.</td>
<td>(Y/N)</td>
<td>DATA</td>
<td>LOC.</td>
<td>(GRID NO.)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Figure 21. Sample data sheet
After all data has been taken and recorded, and the number of measurements and start record measurement has been correctly entered in the set-up table, the system is ready to proceed.

72. R;C: [SELECT STEP] Modal Analysis Controller
73. D: STEP NUMBER?
74. R;C: [X] [ENTER] where X = set up data number i.e.,
   1 for example
75. D: LIST(1), EDIT(2), FINISH(3)?
76. R;C: 3[ENTER] assuming all data correct.

From this point on, the procedure for the mode identification and damping measurement is covered in Reference 6.

For the tank characterization the modes identified and damping measured at the particular modes are presented in Table II.

B. TEST CHAMBER MAJOR MODES AND DAMPING

To identify the major modes and measure the damping of the test chamber the polar plot of the baseband band was divided into 10 sections (Figure 22), and each section was zoomed using the same pickup location and using the same impact location on left, right, deck and back panel.
Figure 22. Polar presentation of transfer function for baseband of the test chamber
A modal analysis of each section was completed. Major modes and damping for each major mode was analyzed. The results are presented in Table II. Test data are included in Appendix E.


<table>
<thead>
<tr>
<th>SECTION NO.</th>
<th>NO. OF MODES</th>
<th>AVERAGE DAMPING FACTORS (%)</th>
<th>STANDARD DEVIATION OF DAMPING</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3</td>
<td>0.5895</td>
<td>0.3059</td>
</tr>
<tr>
<td>2</td>
<td>2</td>
<td>0.3642</td>
<td>0.0035</td>
</tr>
<tr>
<td>3</td>
<td>10</td>
<td>0.8918</td>
<td>0.2458</td>
</tr>
<tr>
<td>4</td>
<td>12</td>
<td>0.7375</td>
<td>0.5084</td>
</tr>
<tr>
<td>5</td>
<td>10</td>
<td>0.3601</td>
<td>0.1934</td>
</tr>
<tr>
<td>6</td>
<td>6</td>
<td>0.2763</td>
<td>0.1065</td>
</tr>
<tr>
<td>7</td>
<td>8</td>
<td>0.3115</td>
<td>0.2176</td>
</tr>
<tr>
<td>8</td>
<td>3</td>
<td>0.2219</td>
<td>0.0627</td>
</tr>
<tr>
<td>9</td>
<td>10</td>
<td>0.1787</td>
<td>0.0873</td>
</tr>
<tr>
<td>10</td>
<td>2</td>
<td>0.1660</td>
<td>0.0525</td>
</tr>
</tbody>
</table>

The overall average damping factor of the test chamber is 0.4667% with an overall standard deviation of 0.3689%.
VI. PROCEDURE FOR DAMPING MEASUREMENT OF SPECIMEN

A. SPECIMEN SECURED IN FIXTURE INSIDE TEST CHAMBER

The test specimen (cast nickel-aluminum bronze, code FTC) was secured in the test chamber, impact locations A, B, C, are shown in Figure 23. To identify the major modes and measure the damping factor of the specimen the 0 to 500 lb. impulse hammer was used for the frequency range up to approximately 6000 Hz, and the 0 to 50 lb. impulse hammer was used for the frequency range from approximately 5000 Hz to approximately 12,000 Hz. Baseband polar plots for each impulse hammer at each location are shown in Figures 24 through 29. It should be noted that there is very good correlation between the 0-500 lb. impulse hammer and the 0-50 lb. impulse hammer. Additionally, there is very good correlation between various impact locations.

Utilizing the baseband data (0-500 lb. impulse hammer), obtained for impact location B, five frequency ranges were investigated. Additionally, using the baseband data obtained from the 0-50 lb. impulse hammer at location B, the frequency ranges of three additional ranges were investigated. Specimen frequency sections are listed in Table III. Specimen modes and damping factors are presented in Table IV. Measurement data and rough log sheets are included in Appendix F.
Figure 21. Cast nickel-aluminum bronze, code FTC specimen secured in the test chamber with impact locations identified.
THE DESIGN OF A TEST PROCEDURE FOR THE MEASUREMENT OF ACOUSTIC DAMPING OF MATERIALS AT LOW STRESS

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Figure 24. Polar presentation of transfer function for baseband (0-500 lb. impulse hammer). Impact location A
Figure 25. Polar presentation of transfer function for baseband (0-50 lb. impulse hammer). Impact Location A.
Figure 26. Polar presentation of transfer function for baseband measurement (0-500 lb. impulse hammer). Impact location B. Zoom ranges identified
Figure 27. Polar presentation of transfer function for baseband measurement (0-50 lb. impulse hammer). Impact location B. Zoom ranges identified.
Figure 28. Polar presentation of transfer function for baseband measurement (0-500 lb. impulse hammer) location C
Figure 29. Polar presentation of transfer function for baseband measurement (0-50 lb. impulse hammer). Impact location C
TABLE III

SPECIMEN FREQUENCY SECTIONS (SPECIMEN FIXED)

<table>
<thead>
<tr>
<th>SECTION NO.</th>
<th>START FREQ. (Hz)</th>
<th>STOP FREQ. (Hz)</th>
<th>CNTR. FREQ. (Hz)</th>
<th>DELTA FREQ. (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>410</td>
<td>937</td>
<td>673</td>
<td>1.395</td>
</tr>
<tr>
<td>2</td>
<td>840</td>
<td>2,598</td>
<td>1,758</td>
<td>4.882</td>
</tr>
<tr>
<td>3</td>
<td>2,598</td>
<td>3,340</td>
<td>2,969</td>
<td>1.953</td>
</tr>
<tr>
<td>4</td>
<td>4,043</td>
<td>4,512</td>
<td>4,277</td>
<td>1.221</td>
</tr>
<tr>
<td>5</td>
<td>4,747</td>
<td>5,508</td>
<td>5,127</td>
<td>1.953</td>
</tr>
<tr>
<td>6</td>
<td>5,713</td>
<td>7,276</td>
<td>6,494</td>
<td>4.069</td>
</tr>
<tr>
<td>7</td>
<td>7,276</td>
<td>9,620</td>
<td>8,448</td>
<td>6.104</td>
</tr>
<tr>
<td>8</td>
<td>9,620</td>
<td>11,380</td>
<td>10,500</td>
<td>4.883</td>
</tr>
</tbody>
</table>

Modal analysis of the specimen sections was completed and the results are shown in the following table.
### TABLE IV

**SPECIMEN MODES AND DAMPING FACTORS (SPECIMEN FIXED)**

<table>
<thead>
<tr>
<th>SECTION NO.</th>
<th>NO. OF MODES</th>
<th>AVERAGE DAMPING FACTORS (%)</th>
<th>STANDARD DEVIATION OF DAMPING FACTOR</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>0.1755</td>
<td>---</td>
</tr>
<tr>
<td>2</td>
<td>2</td>
<td>0.2173</td>
<td>0.1088</td>
</tr>
<tr>
<td>3</td>
<td>3</td>
<td>0.0559</td>
<td>0.0063</td>
</tr>
<tr>
<td>4</td>
<td>2</td>
<td>0.0883</td>
<td>0.0025</td>
</tr>
<tr>
<td>5</td>
<td>2</td>
<td>0.0958</td>
<td>0.0576</td>
</tr>
<tr>
<td>6</td>
<td>6</td>
<td>0.1310</td>
<td>0.0542</td>
</tr>
<tr>
<td>7</td>
<td>5</td>
<td>0.0917</td>
<td>0.0541</td>
</tr>
<tr>
<td>8</td>
<td>5</td>
<td>0.1000</td>
<td>0.0396</td>
</tr>
</tbody>
</table>

The overall average damping factor for the specimen is about 0.1112% with an overall standard deviation of about 0.0601%.
B. SPECIMEN REMOVED FROM FIXTURE AND TEST CHAMBER

In order to ascertain the effects of the specimen fixture and test chamber, one additional test was conducted on the same specimen. During this test the specimen was removed from the fixture and test chamber. The specimen was laid flat on 3/4 inch foam rubber. No other support was provided.

Response and impulse hammer locations are identical to the previous test. Frequency ranges for use of the large (0-500 lb.) impulse hammer and small (0-50 lb.) impulse hammer remained unchanged.

The baseband plots for large and small impulse hammer for locations A, B, and C are presented in Figures 30 through 35. For the large impulse hammer four ranges of zoom were completed, and for the small impulse hammer five additional sections were zoomed. Specimen frequency sections are tested in Table V.
Figure 30. Polar presentation of transfer function for baseband measurement (0-500 lb. impulse hammer). Impact location A.
Figure 31. Polar presentation of transfer function for baseband measurement (0-50 lb. impulse hammer). Impact location A
Figure 32. Polar presentation of transfer function for baseband measurement (0-500 lb. impulse hammer). Impact location B. Zoom ranges identified.
Figure 33. Polar presentation of transfer function for baseband measurement (0-50 lb. impulse hammer). Impact location B. Zoom ranges identified.
Figure 34. Polar presentation of transfer function for baseband measurement (0-500 lb. impulse hammer). Impact location C.
Figure 35. Polar presentation of transfer function for baseband measurement (0-50 lb. impulse hammer). Impact location C
TABLE V
SPECIMEN FREQUENCY SECTIONS (SPECIMEN FREE)

<table>
<thead>
<tr>
<th>SECTION NO.</th>
<th>START FREQ. (Hz)</th>
<th>STOP FREQ. (Hz)</th>
<th>CNTR FREQ. (Hz)</th>
<th>DELTA FREQ. (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1094</td>
<td>1954</td>
<td>1524</td>
<td>2.441</td>
</tr>
<tr>
<td>2</td>
<td>1915</td>
<td>3555</td>
<td>2735</td>
<td>4.883</td>
</tr>
<tr>
<td>3</td>
<td>3575</td>
<td>4610</td>
<td>4092</td>
<td>3.255</td>
</tr>
<tr>
<td>4</td>
<td>4610</td>
<td>5293</td>
<td>4951</td>
<td>1.953</td>
</tr>
<tr>
<td>5</td>
<td>4688</td>
<td>6299</td>
<td>5493</td>
<td>4.069</td>
</tr>
<tr>
<td>6</td>
<td>6250</td>
<td>7227</td>
<td>6738</td>
<td>2.441</td>
</tr>
<tr>
<td>7</td>
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<tr>
<td>8</td>
<td>8545</td>
<td>9473</td>
<td>9009</td>
<td>2.441</td>
</tr>
<tr>
<td>9</td>
<td>9424</td>
<td>11,190</td>
<td>10,307</td>
<td>4.883</td>
</tr>
</tbody>
</table>

Modal analysis was completed on the free specimen, and the results are tabulated below in Table VI.
### TABLE VI

**SPECIMEN MODES AND DAMPING FACTORS (SPECIMEN FREE)**

<table>
<thead>
<tr>
<th>SECTION NO.</th>
<th>NO. OF MODES</th>
<th>AVERAGE DAMPING FACTORS (%)</th>
<th>STANDARD DEVIATION OF DAMPING FACTOR</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2</td>
<td>0.0885</td>
<td>0.0064</td>
</tr>
<tr>
<td>2</td>
<td>5</td>
<td>0.1264</td>
<td>0.0774</td>
</tr>
<tr>
<td>3</td>
<td>1</td>
<td>0.0339</td>
<td>N/A</td>
</tr>
<tr>
<td>4</td>
<td>3</td>
<td>0.0265</td>
<td>0.0017</td>
</tr>
<tr>
<td>5</td>
<td>2</td>
<td>0.0295</td>
<td>0.0020</td>
</tr>
<tr>
<td>6</td>
<td>4</td>
<td>0.0250</td>
<td>0.0080</td>
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<td>0.0331</td>
<td>0.0195</td>
</tr>
<tr>
<td>8</td>
<td>3</td>
<td>0.0209</td>
<td>0.0123</td>
</tr>
<tr>
<td>9</td>
<td>3</td>
<td>0.0301</td>
<td>0.0069</td>
</tr>
</tbody>
</table>

Again, all raw data sheets are included in Appendix F.
The overall average damping factor for the free specimen is about 0.0500% with an overall standard deviation of about 0.0504%.
VII. RESULTS AND CONCLUSIONS

Mode identification was accomplished utilizing the zoom feature of the HP-5451C Fourier Analyzer. After a baseband test was conducted and results stored (Figure 22), any section could be chosen to zoom. For example, zooming section 1 of the test chamber results in a plot as shown in Figure 36. The initial display will be in polar form and must be converted to rectangular (one button function on the HP-5451C) as shown in Figure 37. As shown in Figure 38, the coherence for this test was very good, and the data may be assumed to be accurate. For mode identification it is necessary to get a Nyquist plot of the data (in rectangular form). The Nyquist plot is vector representation of the real vs. imaginary values of the data. On a Nyquist a pure mode would sweep out a perfect circle. Modes with differing damping factors exhibit circles with varying radius. Figure 39 shows the Nyquist plot of the zoom data. Note there are many modes present with greatly varying damping factors. Individual modes can be isolated by inspection of a small portion of the rectangular data. For example, Figure 40 shows the rectangular plot from about 675 Hz to about 850 Hz. If the isolated portion is converted to Nyquist, as shown in Figure 41, the major mode at 772 Hz is readily identified. The Nyquist plots can be smoothed (made more circular) by higher resolution. That is, complete a zoom measurement on a smaller frequency band.
Figure 36. Zoom of test chamber section 1
Figure 37. Zoom of test chamber, section 1. Rectangular form
Figure 38. Zoom of test chamber, section 1. Coherence
Figure 40. Zoom of test chamber, section 1. Isolated portion of rectangular data
All modes identified for the test chamber and the specimen (both fixed and free) are listed in the tables included in Appendices E and F.

It was noted that there are many more modes for the test chamber than for the specimen, but that is expected because the test chamber is a much more complex system, and the boundary conditions are considerably more complex.

The number of modes for the specimen in the test chamber is found to be 26, whereas the number of modes of the specimen separated from the test chamber is 28. This is good agreement, and it should be noted that the modes of the test chamber did not increase the number of observed modes of the specimen.

The average damping factor observed for the test chamber is about 0.47%, which is a typical value for plain carbon steel. The average damping factor for the specimen (fixed) is about 0.11%, and for the specimen (free) is about 0.05%. As expected, the fixing of one end of the specimen in the test chamber did increase the measured damping factor, but with more testing it is possible to quantify the increase and make suitable corrections.

A plot of damping factors vs. frequency for the test chamber is presented in Figure 42. The damping factor is higher at lower frequencies with the maximum damping factors being in the range of 1500-3000 Hz. After 3000 Hz the damping factor averages out to about 0.2% at higher frequencies.
For the specimen, damping factors vs. frequency (for both fixed and free) are presented in Figures 43 and 44. Like the test chamber, the damping factor appears to be higher at lower frequency for the fixed specimen, and without question this same trend is observed for the free specimen. The fixed specimen damping factor levels out to about 0.10% at higher frequencies, and for the free specimen it levels out to about 0.025%.
Figure 43. Specimen damping factor vs. frequency (specimen fixed)
VIII. RECOMMENDATIONS FOR FUTURE WORK

1. Expand the test procedure to include random excitation of test specimen for the energy input and measurement of responses up to 15,000 Hz.

2. Conduct similar tests on various materials including high damping materials such as Incramute and Sonoston.

3. Investigate the temperature dependence of viscous damping, especially for constrained layer and non-metallic composites.

4. Investigate the changes in damping for materials immersed in a water medium at varying temperatures.
APPENDIX A

THE HP-5451C FOURIER ANALYZER

The HP-5451C Fourier Analyzer System performs analysis of time and frequency data containing frequencies from DC to 50 KHz. The system analyzes time-series data such as mechanical vibrations, sonar echoes, tidal action, biomedical phenomena such as brain waves and nerve impulses, voltages and currents in electronic systems, and acoustic phenomena. These analyses may detect signals hidden in noise, or may locate critical functions in complex systems. Both continuous and transient data may be processed.

Keyboard programming allows the following operations automatically without special software:

A. Forward and inverse Fourier transform
B. Magnitude and phase spectrum
C. Power and cross power spectrum
D. Transfer function
E. Coherence function
F. Convolution
G. Auto and cross correlation
H. Hanning and other weighting functions
I. Histogram
J. Scaling
K. Ensemble averaging (time and frequency)
Six editing keys operate an on-line resident editor so that a sequence of steps configured into an automatic measurement procedure may be changed on-line without the need to do off-line editing, compiling and testing. In fact, the series of steps or programs used to perform a particular operation can be stored on the Disc for easy re-entry into the Fourier Analyzer.

Data input and output is likewise controlled from the keyboard. Data can be entered in analog form through the four channel Analog-to-Digital Converter, or in digital form.

Results of all operations are displayed on the oscilloscope. In addition, results can be printed out in decimal numbers or plotted on the Graphics Terminal or an X-Y plotter.

The Fourier Analyzer is a completely calibrated system; all displays and data output are accompanied by a scale factor relating them to physical units. This calibration results from digital techniques being used in all computations.
HP-5451C SPECIFICATIONS AND CHARACTERISTICS

SPECIFICATIONS
(Specifications describe the standard system's warranted performance.)

ANALOG-TO-DIGITAL CONVERTER
Input Voltage Range: ±0.125V to ±0V peak in steps of 2.
Input Coupling: dc or ac.
Input Channels: 2 channels wired for 4 standard. 4 channels optional with plug-in cards.
Resolution: 12 bits including sign.
Input Frequency Range: ±125Hz to ±125Hz peak in steps of 1.

ANALOG-TO-DIGITAL CONVERBER EXECUTION TIMES
Fourier Transform: <55 ms
Stable Power Spectrum Average: <80 ms
Stable Tri-Spectrum Average: <220 ms

REAL TIME BANWIDTHS
Fourier Transform: >7.5 kHz
Stable Power Spectrum Average: 5.4 kHz
Stable Tri-Spectrum Average: 1.9 kHz

MASS STORAGE SOFTWARE
MAXIMUM REAL TIME DATA ACQUISITION RATE
(Single Channel):
BS 256: 10 kHz
BS 1024: 29 kHz (0.25 kHz)
BS 4096: 80 kHz (0.5 kHz)

OFF-LINE BIFA SOFTWARE
Center Frequency Range: dc to one-half the Real Time Data Acquisition Rate.
Center Frequency Resolution: Continuous resolution to the limit of the frequency accuracy for center frequencies >0.02% of the sampling frequency.
Frequency Accuracy: ±0.01%
Bandwidth Selection: In steps of f/1n where n = 2, 3, 4, etc.
Max. Resolution Enhancement: >400
Dynamic Range: ** 90 dB from peak out-of-band spectral component to the peak level of the passband noise.
80 dB from peak in-band spectral component to the peak level of the passband noise.
Out-of-Band Rejection: >90 dB
Passband Flattness of the Digital Filter: ±0.01 dB

BASE SOFTWARE
Transform Accuracy: The expected rms value of computational error introduced in either the forward or inverse FFT will not exceed 0.1% of the rms value of the transform result.
Dynamic Range: >75 dB for a minimum detectable spectral component in the presence of one full scale spectral component after twenty ensemble averages for a block size of 1024.

EXECUTION TIMES*
Fourier Transform: <55 ms
Stable Power Spectrum Average: <80 ms
Stable Tri-Spectrum Average: <220 ms

*For band limited random noise type signals at black level 1024, no display, no histogram.
**After eight ensemble averages of a power spectrum at black level 1024. Reduced by 10 dB at the exact center of the band.
†This rate applies to systems with modules 54068 and 54074A/B having a serial profile lower than 1024.

ENVIRONMENTAL CONDITIONS
Temperature Range: 0°C to 40°C (104°F).
SUPPLEMENTAL CHARACTERISTICS
(Supplemental Characteristics are intended to provide useful information for system applications by giving typical, but not warranted, performance parameters.)

ANALOG-TO-DIGITAL CONVERTER

Input Impedance: 1 MΩ in parallel with <75 pf.
Sample Rate Control:
- Maximum Frequency Mode: Maximum frequency selectable from 0.1 Hz to 50 kHz (100 kHz optional) in steps of 1, 2.5, 5. This mode automatically sets maximum frequency independent of block size.
- Frequency Resolution Mode: Frequency resolution selectable from 0.2 Hz to 1000 Hz in steps of 1, 2.5, 5. This mode automatically sets frequency resolution and sample record length independent of block size.
Input Mode: There is a buffered and non-buffered analog mode. In the buffered mode, other operations can be performed on previously collected data while the ADC collects current input data into a buffer.

DISPLAY UNIT

Data may be displayed in single sweeps or refreshed continuously in the following forms:

<table>
<thead>
<tr>
<th>Y Axis</th>
<th>X Axis</th>
</tr>
</thead>
<tbody>
<tr>
<td>Amplitude</td>
<td>Time (Linear or Log)</td>
</tr>
<tr>
<td>Real Part</td>
<td>Frequency (Linear or Log)</td>
</tr>
<tr>
<td>Imaginary Part</td>
<td>Frequency (Linear or Log)</td>
</tr>
<tr>
<td>Magnitude (Linear or Log)</td>
<td>Frequency (Linear or Log)</td>
</tr>
<tr>
<td>Phase</td>
<td>Frequency (Linear or Log)</td>
</tr>
<tr>
<td>Imaginary Part (NYQUIST PLOT)</td>
<td>Real Part</td>
</tr>
</tbody>
</table>

CRT Positioning: Three markers to aid in adjusting trace position as well as vertical and horizontal controls are provided for display positioning.
- Origin: Left edge of display, zero amplitude.
- ±FS: Positive full scale, center of display.
- ±PS: Negative full scale, center of display.

Analog Plotter Output: Any displayed data can be output to a plotter or remote oscilloscope. HP 106408 interface is required for operation.
- Amplitude: 0.5 V per oscilloscope display division.
- Linearity: 0.1% full scale.
- Interpolation: Linear interpolation in 0.05% steps.

Type of Display: Points, bars, or continuous interpolation.

BASE SOFTWARE

System Accuracy and Range: The Fourier Transform is implemented using conditional scaling for maximum accuracy with no data overflows allowed. All calculations use floating point arithmetic on a block basis with full 16- and 32-bit arithmetic where applicable.

Data Word Sizes: 16-bit imaginary with 32 bits preserved for double precision functions. Division, addition, or subtraction operations performed in 16 or 32 bits depending on data.

Input Block Size: 64 time domain points.

Permanent Data Space: 28K words +16K words standard in systems with serial prefix below 1842, optionally expandable to 28K with option 011.

Permanent Program Space: 32K words.

BSFA SOFTWARE

Maximum BSFA Blocksize: 1024 time domain points. 2048 with option 670. See also Table 5-1 in BSFA section of manual.

MASS STORAGE

Disc Unit:
- Capacity: 2.45 megawords
- Data Transfer: 2.5 million bits/second
- Discs: 2/1 fixed, 1 removable

% of Real Time at 100 kHz ADC Sampling Rate (Single Channel):
- BS 256: 10% BS 1024: 35% (25%) BS 4096: 80% - 30%

Number of Records Per File:
- Data Blocks: 214 records =4096 blocksize maximum/record.
- ADC Throughput: 199 records 4096 blocksize max./record.
- Program Slacks: 138 records 470 steps/record.

ASCII Text: 690 records =128 words/record.

Index: 69 records 10 pointers/record.

System Coreload: 4 records =32K words/record.

Common: 286 records =56 words/record.

Overflow: 20 overlays 8K words maximum, 7K words maximum with option 261 or 265.

OPTIONS 261 & 265

The magnetic tape options are used for ADC Throughput only.

Maximum Real Time Data Acquisition Rate (Single Channel):
- Opt. 261: 6 kHz
- Opt. 265: 9 kHz

Throughput only.

Power Source: 115/230 volts ±10%, 50/60 Hz. 1800 watts typical for base system.

Power Source: 115/230 volts ±10%, 50/60 Hz. 1800 watts typical for base system.

Size: Dimensions are for a typical system excluding cabinet and


Data Transfer Rate: Option 261: 36K cps max., Option 265: 72K cps max.

Back Height: 610 mm (24 in.)

POWER REQUIREMENTS, SIZE, WEIGHT

Power Source: 115/230 volts ±10%, 50/60 Hz. 1800 watts typical for base system.

Power Source: 115/230 volts ±10%, 50/60 Hz. 1800 watts typical for base system.

Size: Dimensions are for a typical system excluding cabinet and


Data Transfer Rate: Option 261: 36K cps max., Option 265: 72K cps max.

Back Height: 610 mm (24 in.)

SYSTEM INSTALLATION

Included in the 5451C Fourier Analyzer System’s Ordering Information Guide.

SYSTEM INSTALLATION

Included in the 5451C Fourier Analyzer System’s Ordering Information Guide.

TRAINING

A course on Fourier analysis and system operation is optionally available at HP's Santa Clara, California facility. On-site training can also be provided, if desired.

These percentages apply to systems with modules 54600 and 54601A/B having a serial prefix lower than 1842.
This appendix contains flow charts and listings of the pre-written soft key programs shipped on the 5451C operating disc pack. These programs are for the Gold Key F2 (Transfer Function) and F5 (Power Spectrum) soft keys. The purposes for including these flow charts and listings are as follows:

1. For those who wish to modify the programs to better fit specific applications.
2. As an example for those who wish to write their own soft key programs.
3. As a model for re-entering portions that may have been accidentally written over on the disc (programs reside in unprotected areas).

As an aid toward rapid understanding of these programs, a certain programming “style” has been used. As a result of this style, the programs are longer than they would otherwise need to be, but are easier to comprehend. Some elements of the style are described below.

**BLOCK STRUCTURE THROUGH USE OF LABELS**

The code in the programs is organized in “blocks” — functional segments which are delimited by LABEL instructions. A convention is followed in the choice of label numbers. The beginning of each block is designated by a label number ending in a multiple of 50, e.g. L1050. The block ends with the label 9 higher, e.g. L1059. Within the block, label numbers are between these limits, e.g. L1051, L1052 etc. One block may contain others — each will use the above delimiting convention.

Although the use of these labels lengthen the programs, it makes them much easier to understand, and to correlate with the flow diagrams. For example, when several branches of a program go to the same point, each branch will jump to its own appropriate label. Since these labels all appear together at one point in the program, one can tell that multiple branches converge at that point. An example is in the Transfer Function program, stack 54, where labels 1209, 1259 and 1609 all appear together. This is just one example of the way in which complex internal linkages in the program are made more visible.

**BRANCHING THROUGH USE OF “COMPUTED GOTO’s”**

In most complex programs, branching is common. One means of branching is to use an IF statement, provided in Keyboard language by the “GOLD KEY” “SKIP” instruction. When there are more than two possible branches, however, use of IF branching tends to get complicated, involving multiple decision points. The “Computed GoTo” or “switch” type of branching statement is more suitable in such cases for simplicity of understanding. It has been used extensively in these keyboard programs, even for simple two way branching. By standardizing on it, the code becomes recognizable and easier to read.
In Keyboard language, the "Computed GoTo" is implemented by computing the number of a label and jumping to it. The following is an example.

```
L 1050    Start of branch block
Y A+ 0 1050 11D    Set variable 0 to 1050 + (the value of variable 11)
J 0D    Jump to the label number in variable 0
L 1051    Code to be executed if variable 11 = 1
J 1059    Go to end of block
L 1052    Code to be executed if variable 11 = 2
L 1059    End of branch block
```

USE OF SUBROUTINES

In these programs there are several functions which have been set up as subroutines. These include the parameter entry routine (see below) and the measurement routines. The measurement routines are handled this way to simplify the flow of the main program and allow easy replacement of the measurement code in case you modify it in some way.

PARAMETER ENTRY ROUTINE

Since these programs ask the operator for many input parameters, a single subroutine, LABEL 100 in STACK 0, handles all the parameter entries. The routine is called with variable parameters 1 and 2 equal to the lower and upper limits on the range of allowable operator inputs. The routine reads your input, checks it against the range limits, and, if it is valid, passes it back to the calling program in (floating point) variable parameter 2000. If the input is out of range, this routine notifies you and waits for the new input. The routine will not return to the calling program until a valid input has been received.

PRECAUTIONARY NOTES

The following precautions apply to the operation of the preprogrammed measurements:

1. When using the standard software zoom (BSFA), the measurement blocksize can be no larger than 1024. When using the Option 670 Fourier Preprocessor for BSFA measurements, the maximum blocksize is 2048.

2. The maximum center frequency you may specify for a BSFA measurement is 32767 Hz.

3. The messages CF WHAT? or BW WHAT? may result if the center frequency and/or bandwidth you have chosen for your measurement are such that the BSFA analysis band is either less than 0 or greater than the ADC Fmax setting. Specifying different parameters should remove this problem.
4. The message DL WHAT? may occur when performing the on-line BSFA measurement. This is because the display is active during the on-line measurement. To remove this problem, either reduce the measurement bandwidth (thereby increasing the zoom power and lowering the data rate into the computer), or edit the appropriate keyboard stacks (stack 56 for transfer function, stack 61 for power spectrum) to remove parameter n3 from the calls to User Prog 45 for the on-line measurement (refer to commented program listings which follow).

5. When performing an off-line BSFA measurement with an optional Mag Tape unit, perform the following steps before making the measurement.

Set:

- ADC SAMPLE MODE to INTERNAL KHz/μs
- MULTIPLIER to 100/10/5
- INPUT SELECTOR to A
- TRIGGERING to FREE RUN
- OVERLOAD VOLTAGE A to CHECK

Enter:

- BLOCKSIZE 4096 ENTER
- MASS STORE 32 ENTER
- MASS STORE 22 SPACE 1 SPACE 150 ENTER
- MASS STORE 32 ENTER

This writes 150 records of data on the magtape so that the magtape will be able to position to record 135 on the tape when the ADC throughput is performed. It will position by looking for the interrecord gaps written by the WRITE ADC throughput command.

6. After completing a BSFA measurement, be sure that all data space declared by the zoom programs is released by pressing RESTART.

As you go through the flow charts and commented listings, remember that these are only examples of programming the soft keys F1 through F6 on the Keyboard. It is up to you to determine which, if any, portions of these programs should be maintained. Because these programs are stored in unprotected areas of the Disc, there is the possibility they can be written over. If this should happen, you should enter the program stacks from the listing, substitute your own program, or copy from your back-up disc.

The soft key programs and the associated ASCII text and variable parameters were originally stored on the system disc pack in Files 3, 4, and 7. The records used are as follows:

- File 3 (Keyboard Programs)
  - Record 0
  - Records 51 through 62

- File 4 (Text Buffers)
  - Text buffers 51 through 55
  - ASCII records 32 through 55

- File 7 (Common)
  - Common Record 0

*This assumes that there are 5709 records in File 4. If not, the first and last ASCII text records may be computed as follows:

- First record number = NR - (5 x last text buffer number)
- Last record number = NR - (5 x first text buffer number) + 4
  where NR = number of records in File 4.

The allocations above should be kept in mind so the above records are not inadvertently altered or destroyed when using the Fourier system. Should you wish to alter the allocations, you will also have to modify the keyboard programs to reflect such changes.
TRANSFER FUNCTION
FLOW DIAGRAM

LABEL 1000
(STACK 51)

INPUT TYPE OF
TRANSFER FUNCTION

(L1050)

HP DAC ?

YES

(L1082)

INPUT DAC VOLT.

NO

(L1100)

INPUT BLOCK SIZE & FILTERS (IF AVAILABLE)

(L1150)

ADC INPUT ?

(L1151)

(L1152)

(L1153)

TURN ON RANDOM SOURCE

TURN ON HP DAC

SET TRIGGER SOURCE

SET UP ADC SWITCHES

A

134
STACK 0

Left key, jump directory

"GOLD KEV: "F9" causes jump to LABEL 1
in this stack. User can program the stack anywhere from there.
Transfer function and Power Spectrum
are preprogrammed in. These may also
be changed.

L J 1000 51
L L 1000 57

F2 TRANSFER FUNCTION
F5 POWER SPECTRUM
Label 1000 in stack 51
Label 1030 in stack 57

F1 USER ASSIGNABLE
F3 USER ASSIGNABLE
F4 USER ASSIGNABLE
F6 USER ASSIGNABLE
Pause here if unassigned function called
Loop if "CONTINUE" pressed

PARAMETERS ENTRY ROUTINE
Read user entry into variable 2000

IF 2000 1D 5 1
Check if entry = lower l.im.
Too small. Use operator with range

IF 2000 2D 5 -1
Check if entry = upper l.im.
Too large. Use operator with range

IF 2000 2D 5 1
Return to calling program—entry in
variable 2000, within specified range

=====================================================================
L -At
STACK 51
TRANSFER FUNCTION PROGRAM
Recall previous variable parameters
Read text buffer #51
Print messages 1 & 2
Set allowable range for user entry
Call user entry routine L100, stack 0

BLOCK 1050

Parameter entry
Range: 0 - 1
Variable: 12

END OF BLOCK 1050

Block size 512—user may change after pause
Pause for operator action

Parameter entry
Range: 0 - 1
Variable: 14

BLOCK 1100

Set 2 filters—auto select mode

END OF BLOCK 1100

Pause for operator action

BLOCK 1150

Parameter entry
Range: 0 - 1
Variable: 14

Fill DAC buffer and
Turn on DAC

END OF BLOCK 1150

Analog input in "REPEAT" to monitor inputs

Power spectrum (log) for user reference
Jump to next stack and continue

137
STACK 52

LOOP POINT FOR TRANSFER FUNCTION

Parameter entry
Range: 1 - 32767
Variable: 15

BLOCK 1200 (ends in stack 54)
Branch per variable 11:
to L1201, 1202 or 1203

Pause for operator action
Turn on DAC

Subroutine call - baseband measurement
Turn off DAC

then go to end of block 1200

Parameter entry
Range: 1 - 2
Variable: 16

BLOCK 1250 (ends in stack 54)
Branch per variable 16

to L1251 or 1252

Pause for operator action
Subroutine call - baseband measurement

then go to end of block 1250

Parameter entry
Range: 1 - 3
Variable: 17

Parameter entry
Range: 1 - 5
Variable: 19

BLOCK 1300 (ends in stack 53)
Branch per variable 19:
to L1301 or 1302

Parameter entry
Range: 1 - 32767
Variable: 19

Parameter entry
Range: 1 - (value of var 19 - 1)
Variable: 20
L -53 STACK 53

Y W 19 1
Y 2000
Y 2003 20010
Y Q 19 1
Y L 2004 20010
Y I 1358
Y IF 2005 20040 2 -2
Y - 2004 20030
Y 2003 20010
Y A- 2008 20090 20030
Y 20 20000
Y Y ? 2000
Y Y+ 2001 20030 20000
Y Y - 1408
Y IF 20 190 1 -2
Y Y A- 20 190 1
Y L 1389
Y L 1389
Y L 1450
Y H 1450 170 0 1450
Y H 1451 5838 54
Y W 22 1
Y W 28 1
Y W 20 1
Y Y 12 2 1
Y S 100 9
Y Y S 1500 20000
Y Y A+ 1500 5838 54 0
Y Y A+ 1500 0 240 1500
Y Y A+ 1550 1500 1500
Y Y A+ 1552 1500 1502
Y Y - 1554 1500 1503
Y Y - 160 0
Y Y L 1550 0 1550 20000
Y Y A+ 1551 1500 1555
Y Y A+ 1552 1500 1555
Y Y A+ 1553 1500 1555
Y Y A+ 1554 1500 1555
Y Y A+ 1555 1500 1555

STACK 53

Variable 2003 gets 1st cursor frequency

Cursor on. parameters to 2000 - 2002
Cursor off
Variable 2004 gets 2nd cursor frequency

BLOCK 1358

IF 1st cursor frequency > 2nd
then swap then

END OF BLOCK 1350

Variable 20 gets zoom bandwidth

Variable 19 gets zoom center frequency

BLOCK 1400

IF bandwidth not < ctr freq
then bandwidth = ctr freq - 1

END OF BLOCK 1400

END OF BLOCK 1300 (from stack 52)

BLOCK 1450 (ends in stack 54)
Branch per variable 19

to L1451, 1452 or 1453

Pause for operator action

Parameter entry
Range: 1 - 2
Variable: 24

BLOCK 1500 (ends in stack 54)

Branch per variable 24

to L1501 or 1502

Parameter entry
Range: 1 - 3
Variable: 2006 (temporary)

BLOCK 1556 (ends in stack 54)

Branch per variable 2006

to L1551, 1552 or 1553

Pause & loop to here--abort

139
Variable 22 gets default start track
Variable 21 gets default # of records

Parameter entry
Range: 1 - 197
Variable: 22

Variable 22 gets user's start track
variable 21 gets # of records left

END OF BLOCK 1550 (from stack 3)

Position throughput file to start track

Pause for operator action
Variable 13 gets current block size
Set block size to max for throughput
Perform ADC throughput
Restore block size

END OF BLOCK 1590 (from stack 53)

END OF BLOCK 1450 (from stack 53)

BLOCK 1590

Save variable parameters

Pause for operator action
Loop to repeat transfer function

END OF TRANSFER FUNCTION (except subroutines)
STACK 55

BASEBAND TRANSFER FUNCTION SUBROUTINE

Check ABC input selector. If not 2 channel
inform operator
pause for operator to correct it and
loop to check again.
Clear blocks needed for averaging

BLOCK 3050

Branch per variable II
to L3051, 3052 or 3053
L3051 & 3052 equivalent—non-transient

Analog input
Transform
Clear DC

Average spectra
Loop for specified # of averages
Subroutine call—clear 2nd half block
Compute transfer function
Correct amplitude for hanning

L3053 transient signals
Instruct operator to impact on cyc
See with "beep"
Analog input
Check for overload—if so, see again
Transform
Clear DC

Average spectra
Loop for specified # of averages
Subroutine call—clear 2nd half block
Compute transfer function
END OF BLOCK 3050
Take logs of spectra

Return to calling program
SUBROUTINE FOR ON LINE ZOOM TRANSFER FCN

STACK 56

55
6
31
4000
7
8
9
10
4000
45
5
1
3
4015
45
5
2
2
4200
4210
4220
4500
7
9
10
4510
45
6
1
2
4550
45
5
1
1
4590
5
4
2
2
4515
5
150
0
4200
2
2
3
4516
SUBROUTINE FOR OFF LINE ZOOM TRANSFER FCN

Clear blocks needed for averaging

SUBROUTINE TO MOVE DATA BLOCKS

Move data blocks from zoom area to area for operator viewing

Take logs of spectra

Return to calling program

SUBROUTINE TO MOVE DATA BLOCKS

Move data blocks from zoom area to area for operator viewing

Clear blocks needed for averaging

3
2
2
3
4000
Check ADC input selector if not 2 channels inform operator pause for operator to correct it and loop to check again Clear blocks needed for averaging

Zoom, displaying Channel 3 power spectrum Average spectra Loop for specified # of averages Compute transfer function Subroutine call to move data Return to calling program

Zoom, displaying Channel 3 power spectrum Average spectra Loop for specified # of averages Compute transfer function Subroutine call to move data Return to calling program
<table>
<thead>
<tr>
<th>L</th>
<th>-57</th>
</tr>
</thead>
<tbody>
<tr>
<td>MS</td>
<td>37</td>
</tr>
<tr>
<td>MS</td>
<td>17</td>
</tr>
<tr>
<td>V</td>
<td>W538</td>
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<td>4</td>
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</tr>
<tr>
<td>Y</td>
<td>1109</td>
</tr>
<tr>
<td>Y</td>
<td>7</td>
</tr>
<tr>
<td>Y</td>
<td>5838</td>
</tr>
<tr>
<td>Y</td>
<td>52</td>
</tr>
<tr>
<td>Y</td>
<td>10</td>
</tr>
<tr>
<td>Y</td>
<td>2000</td>
</tr>
<tr>
<td>Y</td>
<td>1</td>
</tr>
<tr>
<td>T</td>
<td>143</td>
</tr>
</tbody>
</table>

STACK 57

POWER SPECTRAL PROGRAM
Recall previous variable parameters

Read text buffer #51
Print messages 4 & 5

Block size 512-user may change after pause
Pause for operator action

Parameter entry
Range: -4
Variable: 14

BLOCK 1100

Set 2 filters-auto select mode
END OF BLOCK 1100
Pause for operator action

Analog input in "REPEAT" to monitor inputs
Power spectrum (log) for user reference
Jump to next stack and continue
STACK 58
LOOP POINT FOR POWER SPECTRUM

Parameter entry
Range: 1 - 32767
Variable: 15

Parameter entry
Range: 1 - 2
Variable: 16

BLOCK 1250 (ends in stack 58)
Branch per variable 16
  to L1251 or 1252

Pause for operator action
Subroutine call - baseband measurement
  then go to end of block 1250

Parameter entry
Range: 1 - 3
Variable: 17

Parameter entry
Range: 1 - 2
Variable: 18

BLOCK 1300 (ends in stack 59)
Branch per variable 18
  to L1301 or 1302

Parameter entry
Range: 1 - 32767
Variable: 19

Parameter entry
Range: 1 - (value of var 19 - 1)
Variable: 20
STACK 59

L -S9
STACK S9
L 1303
/.
Y W 2000
Y B 2003 2001D
Y B 19 1

Corser on. Cursor parameters to
variables 2000 - 2002 (2001=frequency)
Variable 2003 gets 1st cursor frequency

Y Y 2004 2001D
Y / -1
Y L 1350
Y IF 2003 2004D 2 -2
Y IF 2004 2003D 2
Y Y - -2
Y Y - 2003 2001D
Y L 1359
Y Y - -2
Y Y - 2000
Y Y + 2001 2003D 2000D
Y Y - 19 2001D
Y Y - 1400
Y IF 20 190 1 -2
Y IF 20 190 1
Y L 1409
Y L 1389
Y L 1450
Y Y + 0 1450 17D
Y Y + 1450 54
Y Y + 1451
Y Y + 5838 54
Y Y + 22 1
Y Y + 1459 3
Y Y + 1452
Y Y + 1453
Y Y + 20 1
Y Y Y - 19 0 2
Y Y Y - 19 24 2000D
Y Y + 1500
Y Y + 5838 54
Y Y + 0 1500 24D
Y Y + 0
Y Y + 1501
Y Y + 1509
Y Y + 1502
Y Y Y - 28 1
Y Y Y - 28 1
Y Y Y - 100 0
Y Y + 0 1550 2000D
Y Y + 0
Y Y + 0
Y Y + 1551
Y Y + 1552
Y Y + 1554
Y Y + 1553
Y Y + 1555
Y Y Y - 1
Y Y Y - 1

Pause for operator action

Parameter entry
Range: 1 - 2
Variable: 24

BLOCK 1500 (ends in stack 60)

Parameter entry
Range: 1 - 3
Variable: 2000 (temporary)

BLOCK 1550 (ends in stack 60)

Pause & loop to here—abort

145
Variable 22 gets default start track
variable 21 gets default # of records

Parameter entry
Range: 1 - 197
Variable: 22

Variable 22 gets user's start track
Variable 21 gets # of records left

Position throughput file to start track

Pause for operator action
Variable 13 gets current block size
Set block size to max for throughput

Restore block size

END OF BLOCK 1500 (from stack 59)
END OF BLOCK 1450 (from stack 59)

Branch per variable 17
to L601, 1602 or 1603

Initialize zoom—on line preprocessor
Subroutine call—zoom measurement
Reset zoom to baseband
Go to end of block 1600

Initialize zoom—off line preprocessor
Subroutine call—zoom measurement
Reset zoom to baseband
Go to end of block 1600

Initialize zoom—off line software
Subroutine call—zoom measurement
Reset zoom to baseband
END OF BLOCK 1600
END OF BLOCK 1250 (from stack 58)

Power spectrum complete
Print messages to operator
Save variable parameters
Pause for operator action
Loop to repeat power spectrum
END OF POWER SPECTRUM (except subroutines)
BASEBAND POWER SPECTRUM SUBROUTINE

Check ADC input selector. If not 1 channel
inform operator
pause for operator to correct it and
loop to check again
Clear block needed for averaging

Analog input, displaying average
Transform
Clear DC
Average spectrum
Loop for specified # of averages
Subroutine call - clear 2nd half block
Hanning correction for broadband noise

Load average to block 0
Take log
Return to calling program

SUBROUTINE FOR OFF LINE ZOOM POWER SPECTRUM

Clear block needed for averaging

Zoom
Display average (1 sweep)
Average
Loop for specified # of averages
Load average to block 0
Take log
Return to calling program
STACK 62
CLEAR 2ND HALF BLOCK SUBROUTINE
Set return label value

Return label

Clear last half block 4

Clear last half block 5

Entry pt. Pur Spec pgm
Set return label value

Return label

Clear last half block 1

Was HP Filter selected?

No, Gto 9051

Yes

Get current blocksize
Store B0s/2 in variable 26
Store B0s/4 in variable 25

Indirect return

End of pgm 9050-9055
Return to calling program

Text buffer messages for Transfer Function and Power Spectrum programs

BUFFER 0  MESSAGE 0

51
HP TRANSFER FUNCTION PROGRAM

51
SELECT EXCITATION TYPE
51-RANDOM - BASEBAND OR ZOOM
2-HP DAC - BASEBAND ONLY
3-TRANSIENT - BASEBAND ONLY

51
INPUT DESIRED DAC OUTPUT IN MV

51
HP POWER SPECTRUM PROGRAM

51
SET ADC FREQUENCY RANGE AS DESIRED
(SAMPLE MODE & MULTIPLIER)
SET ADC TRIGGER TO "FREE RUN"
CHANGE BLOCK SIZE IF DESIRED
PRESS "CONTINUE" WHEN READY

51
ARE HP FILTERS INSTALLED?
0=NO, 51=YES
SET KEYBOARD REPEAT/SINGLE SWITCH TO "REPEAT"  
PRESS "CONTINUE" WHEN READY

UPPER LIMIT = 99

ENTRY OUT OF RANGE—PLEASE REENTER  
LOWER LIMIT = 0  
PRESS CONTINUE WHEN READY

TURN ON RANDOM EXCITATION SOURCE

SET TRIGGER SOURCE AS DESIRED  
IMPACT STRUCTURE REPEATEDLY

SET OVERLOAD VOLTAGES AND TRIGGER  
LEVELS FOR SIGNAL AMPLITUDES  
MOVE REPEAT/SINGLE SWITCH TO  
"SIMPLE" WHEN READY. IF SOURCE  
IS NOT IN FREE RUN, TRIGGER THE  
SYSTEM AGAIN TO CONTINUE.

ENTER NUMBER OF AVERAGES DESIRED

ENTER MEASUREMENT TYPE  
1=BASEBAND  
52=ZOOM

ENTER ZOOM MEASUREMENT MODE  
1=ON LINE, PREPROCESSOR  
52=OFF LINE, PREPROCESSOR  
3=OFF LINE, SOFTWARE
ZOOM NOT APPROPRIATE WITH HP DAC
BASEBAND MEASUREMENT WILL BE MADE

HOW WILL YOU SPECIFY ZOOM BANDWIDTH?
1=NUMERIC ENTRY - CTR FREQ & BW
2-CURSOR - ON PRIOR MEASUREMENT

ENTER CENTER FREQUENCY

ENTER BANDWIDTH

MOVE CURSOR TO START FREQUENCY
PRESS "VALUE" (SWITCH REGISTER 11)

MOVE CURSOR TO END FREQUENCY
PRESS "VALUE"

ANALYZE OLD OR NEW DATA?
1=OLD (FROM THROUGHPUT FILE)
2=NEW

THROUGHPUT COMPLETED

PRESS "CONTINUE" FOR MEASUREMENT

MEASUREMENT COMPLETE

TO DISPLAY RESULTS, PRESS:
"DISPLAY" 0 LOG TRANSFER FCN
"DISPLAY" 1 COHERENCE
"DISPLAY" 2 INPUT POWER SPECT
"DISPLAY" "3" OUTPUT POWER SPECT
"DISPLAY" "54" CROSS POWER SPEC

54  25
TO COPY DISPLAY ON TERMINAL:
PUT TERMINAL IN GRAPHICS MODE
PRESS "GOLD KEY" "PLOT"

54  26
TO MAKE ANOTHER MEASUREMENT:
PUT TERMINAL IN ASCII MODE
PRESS "CONTINUE"

54  27
ENTER STARTING TRACK FOR THROUGHPUT

54  28
THROUGHPUT WILL USE TRACKS
135 THROUGH 198 ON THE LOWER
(FDS) DISC. IS THIS OK?
1=NO - ABORT
2=YES - PROCEED
3=NO - ASK ME FOR TRACK #

55  29
IMPACT STRUCTURE ON CUE (BEEP)
FOR EACH AVERAGE

55  30

55  31
SET ADC INPUT SELECTOR TO "AB"
PRESS "CONTINUE" WHEN READY

55  32
SET ADC INPUT SELECTOR TO "A"
PRESS "CONTINUE" WHEN READY

Beeps to cue operator
Beeps to cue operator
Beeps to cue operator
APPENDIX C

CALIBRATION DATA

ACCELEROMETER MOUNTING WAX

Model 080A24

Petro-Wax functions to transfer motion from the test object to the sensor. It is used to couple the sensor directly to the test object. It is a convenient, temporary mounting replacement for studs and permanent adhesives.

An elastic interface structure with heavy damping, it forms quickly to irregular surfaces to facilitate easy mounting of sensors. This pliable wax enables the sensor to be mounted in nearly any convenient spatial coordinate system.

In conjunction with the sensor, the wax forms a heavily damped spring-mass system. The frequency response is a function of transducer mass, mounting area, depth of wax, and test temperature. As the amount of wax at the interface of the sensor and test object increases, the first resonance of the system decreases, which limits the frequency response of the fixtureing (see graphs). This mounting technique is primarily for use at room temperature. The wax cannot be used effectively at high or low temperatures.

How to use: (1) To insure a secure and valid fixture, be certain all surfaces are free of oil and dirt.

(2) Apply the wax directly to the base of the transducer or to the adhesive mounting base. The amount of wax used depends on your individual application. Use of the adhesive mounting base helps keep the transducer clean.

(3) Press the sensor firmly against the test structure to insure secure mounting and as little wax at interface as possible. This provides the best frequency response.

(4) Proceed with measurement.

PETRO-WAX is available in quantity from:
Katt & Associates
P.O. Box 98269
Pittsburgh, PA 15227
(412) 885-5727

MODE NO. 3080

Horizontal Scale 5 000 Hz
Magn. : vert. scale 10 db/div
Phase: vert. scale 50°/div

<table>
<thead>
<tr>
<th>FREQ (Hz)</th>
<th>Dev (%)</th>
<th>Dev (%)</th>
<th>Dev (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>-1</td>
<td>-1</td>
<td>-1</td>
</tr>
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<td>0</td>
</tr>
<tr>
<td>100</td>
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<td>0</td>
<td>0</td>
</tr>
<tr>
<td>200</td>
<td>0</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>500</td>
<td>1</td>
<td>2</td>
<td>3</td>
</tr>
<tr>
<td>1 000</td>
<td>1</td>
<td>3</td>
<td>11</td>
</tr>
<tr>
<td>2 000</td>
<td>2</td>
<td>10</td>
<td>274</td>
</tr>
<tr>
<td>5 000</td>
<td>10</td>
<td>74</td>
<td>-27</td>
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<tr>
<td>10 000</td>
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<td></td>
<td></td>
</tr>
<tr>
<td>20 000</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

PCB PHEZOTRONICS INC. 3425 WALDEN AVENUE DREXEL NEW YORK 10043-2495 TELEPHONE 718-584-0001 TWX 710-263-1377
MODEL NO. 302A
Horizontal Scale 10 000 Hz
Mag.: vert. scale 10 dB/div
Phase: vert. scale 30°/div

<table>
<thead>
<tr>
<th>FREQ (Hz)</th>
<th>DEV (%)</th>
<th>DEVI (%)</th>
<th>DEVI (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>-1</td>
<td>-1</td>
<td>-1</td>
</tr>
<tr>
<td>50</td>
<td>0</td>
<td>0</td>
<td>-1</td>
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<tr>
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<td>0</td>
<td>0</td>
</tr>
<tr>
<td>200</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>500</td>
<td>1</td>
<td>1</td>
<td>2</td>
</tr>
<tr>
<td>1000</td>
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<td>7</td>
<td>41</td>
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<tr>
<td>5000</td>
<td>62</td>
<td>62</td>
<td>35</td>
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<td>10 000</td>
<td>56</td>
<td>56</td>
<td>-450</td>
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<tr>
<td>20 000</td>
<td></td>
<td></td>
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MODEL NO. 303A
Horizontal Scale 25 000 Hz
Mag.: vert. scale 10 dB/div
Phase: vert. scale 50°/div

<table>
<thead>
<tr>
<th>FREQ (Hz)</th>
<th>DEV (%)</th>
<th>DEVI (%)</th>
<th>DEVI (%)</th>
</tr>
</thead>
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<td>20</td>
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<td>0</td>
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<tr>
<td>100</td>
<td>0</td>
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<td>0</td>
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<tr>
<td>200</td>
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</tr>
<tr>
<td>500</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>1000</td>
<td>1</td>
<td>1</td>
<td>2</td>
</tr>
<tr>
<td>2000</td>
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<td>3</td>
<td>8</td>
</tr>
<tr>
<td>5000</td>
<td>0</td>
<td>11</td>
<td>48</td>
</tr>
<tr>
<td>10 000</td>
<td>1</td>
<td>90</td>
<td>26</td>
</tr>
<tr>
<td>20 000</td>
<td>15</td>
<td>-16</td>
<td>-546</td>
</tr>
</tbody>
</table>
**CALIBRATION CERTIFICATE**

**IMPACT FORCE HAMMER**

Model No. 086403
Transducer Model No. __
Transducer Serial No. __
Hammer Calibrator: PCB Model __
Pendulous Mass 0.72 lb (326 gm)
Date: 2-23-83
Initials: __

Customer: __
Invoice No: __

Basic Force Transducer Sensitivity 10.60 mV/lb (reference)
(for stationary installations)
Reference Accelerometer Sensitivity 10.12 mV/g (302±5/N, 0.253)
Ratio: Force Transducer Sensitivity/ Accelerometer Sensitivity 1.05 (1.03)
(from test of mass impacting stationary hammer)

**HAMMER SENSITIVITY (3)** (with 2.7 gm Al/Plastic, Al/Steel, Al/Rubber Tips)

<table>
<thead>
<tr>
<th>Configuration</th>
<th>with Steel Extender</th>
<th>with Al. Extender</th>
<th>with No Extender</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ratio: Force/Acceleration Sensitivity (2)</td>
<td>1.026</td>
<td>6.972</td>
<td></td>
</tr>
<tr>
<td>Hammer Sensitivity mV/lb</td>
<td>10.38</td>
<td></td>
<td>9.84</td>
</tr>
<tr>
<td>Difference: (1) (%)</td>
<td>-2.1%</td>
<td></td>
<td>-7.2%</td>
</tr>
</tbody>
</table>

**NOTES:**

(1) Difference from reference sensitivity of force transducer
(2) In transfer function testing, the important factor is the ratio of sensitivities.
(3) Because of normal behavior of the hammer structure, the apparent sensitivity of the hammer in motion differs from the stationary calibration of the force transducer. It is less by a factor proportional to the ratio of the mass of the impact cap and seismic distributing mass in the transducer to the total mass of the hammer structure. Using a heavier hammer head or installing the force transducer on the structure only changes the problem. A heavier hammer head tends to penetrate the test object or cause multiple bouncing. When mounted on the test object, the inertial mass in the transducer causes it to act as an accelerometer sensing motion of the test object.
1. VOLTAGE SENSITIVITY       \( 14.3 \text{ mV/g} \) @ 100 Hz, 8 g's PK
2. MAXIMUM TRANSVERSE SENSITIVITY       1.7 percent
3. RESONANT FREQUENCY       26.0 \( \text{KHz} \)
4. DISCHARGE TIME CONSTANT       4.1 seconds
5. OUTPUT BIAS LEVEL       8.6 Volts

6. FREQUENCY RESPONSE:

<table>
<thead>
<tr>
<th>Freq, Hz</th>
<th>10</th>
<th>30</th>
<th>50</th>
<th>100</th>
<th>300</th>
<th>500</th>
<th>1000</th>
<th>2000</th>
<th>5000</th>
<th>10000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Deviation %</td>
<td>-3.3</td>
<td>2.0</td>
<td>-0.8</td>
<td>0</td>
<td>0.7</td>
<td>-1.0</td>
<td>-1.2</td>
<td>-2.0</td>
<td>-2.4</td>
<td>-2.1</td>
</tr>
</tbody>
</table>

Calibration traceable to NBS through project no. 237/221322

PB PIEZOTRONICS INC
3425 WALDEN AVENUE
DEPEW, NEW YORK 14043

Customer Reference: [Redacted]
Order No. [Redacted]

---

PCB PIEZOTRONICS INC
3425 WALDEN AVENUE
DEPEW, NEW YORK 14043

1. VOLTAGE SENSITIVITY       \( 10.12 \text{ mV/g} \) @ 100 Hz, 8 g's PK
2. MAXIMUM TRANSVERSE SENSITIVITY       0.5 percent
3. RESONANT FREQUENCY       235 \( \text{KHz} \)
4. DISCHARGE TIME CONSTANT       0.8 seconds
5. OUTPUT BIAS LEVEL       11.8 Volts

6. FREQUENCY RESPONSE:

<table>
<thead>
<tr>
<th>Freq, Hz</th>
<th>10</th>
<th>30</th>
<th>50</th>
<th>100</th>
<th>300</th>
<th>500</th>
<th>1000</th>
<th>2000</th>
<th>5000</th>
<th>10000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Deviation %</td>
<td>-1.0</td>
<td>-0.6</td>
<td>0.3</td>
<td>0</td>
<td>0.0</td>
<td>-0.3</td>
<td>-0.6</td>
<td>-1.2</td>
<td>-1.6</td>
<td>-1.6</td>
</tr>
</tbody>
</table>

Calibration traceable to NBS through project no. 237/229322

PB PIEZOTRONICS INC
3425 WALDEN AVENUE
DEPEW, NEW YORK 14043

---

Date: 3-9-83

By: [Signature]
ICP TRANSDUCER DATA

Model: 086823
SN: 269
Gain: 10.6
Linearity: ±2%

Cal. Range: 0-526 lbs
Input Time Constant: 2000 sec
Response Time: 10 mSec
Natural Frequency: 31 kHz
Output Impedance: <1000 Q

Date: 7-23-67

BUFFALO, NEW YORK 14225

P.O. BOX 33
ZIEZOTRONICS LTD.

156
# Calibration Certificate

**Impulse Force Hammer**

**Model No.:** 090600  
**Serial No.:** 235  
**Kit No.:** 6142100  
**Customer:** 

**Date:** 4/18/89  
**Invoice No.:** A30272

**Range:** 0 - 50 lb  
**Linearity error:** ±2.0%  
**Discharge Time Constant:** 100 ms  
**Output Impedance:** 100 ohms  
**Output Bias:** 12.2 volts  
**Traceable to NBS through:** 737/22932

**Hammer Sensitivity:**

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Tip Extender</th>
<th>Steel</th>
<th>Vinyl</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Scaling Factor (Sensitivity Ratio)</strong></td>
<td></td>
<td>0.198</td>
<td>0.124</td>
</tr>
<tr>
<td><strong>Hammer Sensitivity</strong></td>
<td>m/s²</td>
<td>113.3</td>
<td>70.9</td>
</tr>
<tr>
<td></td>
<td>(m/s²)</td>
<td>25.5</td>
<td>15.9</td>
</tr>
</tbody>
</table>

**Notes:**

1. The sensitivity ratio (Sa/Sf) is the scaling factor for converting structural transfer measurements into engineering units. Divide results by this ratio.

2. Each specific hammer configuration has a different sensitivity. The difference is a constant percentage, which depends on the mass of the cap and tip assembly relative to the total mass of the head. Calibrating the specific hammer structure being used automatically compensates for mass effects.

Effective mass with X02407 attached and vinyl capped plastic tip.
# Calibration Data

**Model No.:** 3090  
**Serial No.:** J28  
**Range:** 1000 g's  
**Max Input:** 1000 g's  
**Max Temp:** 150 °F

---

### I.C.P. Accelerometer (per ISA S37.2)

1. **Voltage Sensitivity**  
   - 56.8 mV/g  
   - @ 100 Hz, 8 g's PK

2. **Maximum Transverse Sensitivity**  
   - 2.8 percent

3. **Resonant Frequency**  
   - 2130 KHz

4. **Discharge Time Constant**  
   - 0.2 seconds

5. **Output Bias Level**  
   - 8.7 Volts

6. **Frequency Response:**

<table>
<thead>
<tr>
<th>Freq, Hz</th>
<th>10</th>
<th>30</th>
<th>50</th>
<th>100</th>
<th>300</th>
<th>500</th>
<th>1000</th>
<th>3000</th>
<th>5000</th>
<th>7000</th>
<th>9000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Deviation %</td>
<td>4.8</td>
<td>2.6</td>
<td>1.1</td>
<td>0</td>
<td>+0.9</td>
<td>+1.4</td>
<td>+1.9</td>
<td>+3.2</td>
<td>+3.8</td>
<td>+5.0</td>
<td></td>
</tr>
</tbody>
</table>

Calibration traceable to NBS through project no. 737/22/16

---

**Model No.:** 303403  
**Serial No.:** J28  
**Range:** 500 g's  
**Max Input:** 2000 g's  
**Max Temp:** 200 °F

---

### I.C.P. Accelerometer (per ISA S37.2)

1. **Voltage Sensitivity**  
   - 1.06 mV/g  
   - @ 100 Hz, 8 g's PK

2. **Maximum Transverse Sensitivity**  
   - 1.9 percent

3. **Resonant Frequency**  
   - 28 KHz

4. **Discharge Time Constant**  
   - 0.10 seconds

5. **Output Bias Level**  
   - 8.6 Volts

6. **Frequency Response:**

<table>
<thead>
<tr>
<th>Freq, Hz</th>
<th>10</th>
<th>30</th>
<th>50</th>
<th>100</th>
<th>300</th>
<th>500</th>
<th>1000</th>
<th>3000</th>
<th>5000</th>
<th>7000</th>
<th>9000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Deviation %</td>
<td>2.0</td>
<td>1.5</td>
<td>1.1</td>
<td>0</td>
<td>+0.7</td>
<td>+1.1</td>
<td>+1.8</td>
<td>+2.4</td>
<td>+2.4</td>
<td>+3.1</td>
<td></td>
</tr>
</tbody>
</table>

Calibration traceable to NBS through project no. 737/22/16
I.C.P. TRANSDUCER DATA

Model: 222A
S/N: 267

Col. Range: 0-600 psi
Input Time Constant: 2 sec
Rise Time: 10 µsec
Sensitivity: 0.99 mV/psi
Natural Frequency: 50 kHz

Linearity: ±1% FS
Output Impedance: 100 Ohms

Date: 5-27-71

By: [Signature]

*By comparison with reference standard per ISA 5.37.10

---

Graph depicting the relationship between input and output in millivolts.
APPENDIX D

LOCALLY GENERATED USER KEYBOARD PROGRAMS

The locally generated keyboard programs presented in the next six pages allow the use of impulse technique or random excitation while utilizing the Modal package of the HP-5451C Fourier Analyzer. Each program and subprogram is identified by the first line of the program listing. Program number 1 (identified by the -1 on line 1) is the control program, and from this program subprograms 50, 51, 52, 53, 54, 58 and 59 are called as needed.

All eight programs are automatically loaded into the computer memory when the following command is executed, as discussed in Chapter V, step number 35,

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47 CL  4
51 CL  5
55 CL  6
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63 Y  45  2  1  2
70 Y  45  1  1  1
77 D  5  4
82 SP  1  2  2
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94 CH  1  2  2
100 X#  1
104 X#  6
108 X#  2
112 X#  1
116 X#  3
129 X#  2
129 X#  4
128 X#  3
132 X#  5
136 X#  4
140 X#  6
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150 .

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31 Y  2  41D  2
38 J  100  59
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# APPENDIX E

## TEST CHAMBER CHARACTERIZATION DATA

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**LAB AIR, FRONT REMOVED**

**Temp: 23°C**
## Tank Characterization

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**LAB AIR, FRONT REMOVED**

**TEMP: 24°C**
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LAB AIR, FRONT REMOVED
TEMP: 24°C
### Tank Characterization

**Average Modal Frequencies and Damping**

- **Centre Frequency (Hz)**: 281 Hz
- **Delta Frequency (Hz)**: 13 Hz
- **Set Up No.**: EL4.21 L.I. 696 P.L. L.D.

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<th>Damp. Coeff. (Rad/Sec)</th>
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</thead>
<tbody>
<tr>
<td>1</td>
<td>691.0089</td>
<td>0.9304</td>
<td>40.3972</td>
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<tr>
<td>2</td>
<td>772.3085</td>
<td>0.4990</td>
<td>24.2164</td>
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<td>3</td>
<td>1001.7299</td>
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<td>21.3372</td>
</tr>
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</table>

**Remarks:**

- Damping Factor: Mean 0.4667, SDEV 0.3689
- Over All 66 Modes: Mean 0.4667, SDEV 0.3689
### Tank Characterization

**Average Modal Frequencies and Damping**

Center Freq 122.1 Hz  Delta Freq 0.977 Hz  
Set up No. 49  I.L. 6034  P.L. 4951

<table>
<thead>
<tr>
<th>Mode</th>
<th>Nat. Freq (Hz)</th>
<th>Damp. Fact. (%)</th>
<th>Damp. Coeff. (Rad/Sec)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1154.2720</td>
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<tr>
<td>2</td>
<td>1209.6404</td>
<td>0.3617</td>
<td>27.4907</td>
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**Remarks:**

Mean 0.3642  
S.D. 0.0035

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**TANK CHARACTERIZATION**

**AVERAGE MODAL FREQUENCIES AND DAMPING**

**CNTR FREQ 1915 Hz**  **DELTA FREQ 1438 Hz**

**SET UP NO. 23 I.L. 3046 P.L. 3527**

<table>
<thead>
<tr>
<th>NODE</th>
<th>NAT. FREQ (Hz)</th>
<th>DAMP. FACT. (%)</th>
<th>DAMP. COEFF. (RAD/SEC)</th>
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<tbody>
<tr>
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<td>2151.9443</td>
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</table>

**REMARKS:**

- MEAN 0.7918
- SDEV 0.2458

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### Tank Characterization

**Average Modal Frequencies and Damping**

- Center freq: 248 Hz
- Delta freq: 6.22 Hz
- Set up no.: L.L. 6086, P.L. 2.051

<table>
<thead>
<tr>
<th>Mode</th>
<th>Nat. Freq (Hz)</th>
<th>Damp. Fact. (%)</th>
<th>Damp. Coeff. (Rad/sec)</th>
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<tbody>
<tr>
<td>1</td>
<td>2333.2114</td>
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<tr>
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<td>4</td>
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**Remarks:**
- Mean: 0.7375
- Std dev: 0.5084
### Tank Characterization

#### Average Modal Frequencies and Damping

CNTR FREQ 3.024 Hz DELTA FREQ 1.628 Hz
SET UP NO. 1.5 I.L. 1.084 P.L. 1.081

<table>
<thead>
<tr>
<th>MODE</th>
<th>NAT. FREQ (Hz)</th>
<th>DAMP. FACT. (%)</th>
<th>DAMP. COEFF. (RAD/SEC)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2818.4990</td>
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<td>123.0181</td>
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<td>2</td>
<td>3024.3071</td>
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<td>3</td>
<td>3040.4941</td>
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<tr>
<td>4</td>
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<td>0.4668</td>
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<td>6</td>
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<td>0.0932</td>
<td>18.3111</td>
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<td>7</td>
<td>3165.7607</td>
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<td>9</td>
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<td>0.4503</td>
<td>91.8050</td>
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<td>10</td>
<td>3298.6997</td>
<td>0.2684</td>
<td>55.6223</td>
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**Remarks:**

Mean 0.3601
SDev 0.1934
## Tank Characterization

**AVERAGE MODAL FREQUENCIES AND DAMPING**

- **Center Frequency**: 3525 Hz
- **Delta Frequency**: 0.977 Hz
- **Set Up No.**: L...I.L. 4263 P.L. L971

### Modal Properties

<table>
<thead>
<tr>
<th>Mode</th>
<th>Nat. Freq (Hz)</th>
<th>Damper Factor (%)</th>
<th>Damper Coeff. (Rad/Sec)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3473.6230</td>
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<td>2</td>
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<td>4</td>
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<td>3689.9521</td>
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</tr>
</tbody>
</table>

### ReMARKS:

- **Mean**: 0.2763
- **SDEV**: 0.1065
## Tank Characterization

### Average Modal Frequencies and Damping

- Center freq.: 4102 Hz
- Delta freq.: 1989 Hz
- Set up no.: E7 LE9 P.L. 6581

<table>
<thead>
<tr>
<th>MODE</th>
<th>NAT. FREQ (Hz)</th>
<th>DAMP. FACT. (%)</th>
<th>DAMP. COEFF. (RAD/SEC)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3708.0176</td>
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<td>36.4081</td>
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<tr>
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<td>3739.4619</td>
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<tr>
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<td>3978.8354</td>
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</tr>
<tr>
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<td>80.1305</td>
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<td>5</td>
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<td>0.3084</td>
<td>83.4207</td>
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<td>0.7869</td>
<td>211.0450</td>
</tr>
<tr>
<td>7</td>
<td>4373.2285</td>
<td>0.0283</td>
<td>7.7682</td>
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<tr>
<td>8</td>
<td>4397.8242</td>
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</table>

**Remarks:**

- Mean: 0.3115
- SD: 0.2176
<table>
<thead>
<tr>
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<th>NAT. FREQ (Hz)</th>
<th>DAMP. FACT. (%)</th>
<th>DAMP. COEFF. (RAD/SEC)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
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<td>48.3874</td>
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<tr>
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</tr>
<tr>
<td>3</td>
<td>4600.7148</td>
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</table>

**REMARKS:**

MAIN 0.2219
SDEV 0.0627
## Mode Characterization

### Average Modal Frequencies and Damping

<table>
<thead>
<tr>
<th>Mode</th>
<th>Nat. Freq (Hz)</th>
<th>Dam. Freq (Hz)</th>
<th>Set Up No.</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4.0328, 9.8633</td>
<td>4.9344, 22.27</td>
<td>0.1722</td>
</tr>
<tr>
<td>2</td>
<td>5.0377, 24.15</td>
<td>5.0934, 20.27</td>
<td>0.0785</td>
</tr>
<tr>
<td>3</td>
<td>5.1666, 7.38</td>
<td>5.2244, 5.25</td>
<td>0.1159</td>
</tr>
<tr>
<td>4</td>
<td>5.5041, 0.94</td>
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<td>0.2353</td>
</tr>
<tr>
<td>5</td>
<td>5.9045, 0.94</td>
<td>5.9045, 0.94</td>
<td>0.1159</td>
</tr>
<tr>
<td>6</td>
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<td>6.4893, 1.92</td>
<td>0.1159</td>
</tr>
<tr>
<td>7</td>
<td>4.1441, 7.67</td>
<td>4.1441, 7.67</td>
<td>0.1159</td>
</tr>
<tr>
<td>8</td>
<td>4.8930, 1.92</td>
<td>4.8930, 1.92</td>
<td>0.1159</td>
</tr>
<tr>
<td>9</td>
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<td>5.0752, 1.92</td>
<td>0.1159</td>
</tr>
<tr>
<td>10</td>
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<td>5.3413, 1.92</td>
<td>0.1159</td>
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</tbody>
</table>

### Remarks:
- SPEV 0491a7 179
## Tank Characterization

### Average Modal Frequencies and Damping

**Center Frequency (Hz)**: 5587 Hz  
**Delta Frequency (Hz)**: 0.91 Hz  
**Set Up No.**: 210  
**I.L.**: L686  
**P.I.**: L957

<table>
<thead>
<tr>
<th>Mode</th>
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<th>Damp. Fact. (%)</th>
<th>Damp. Coeff. (RAD/SEC)</th>
</tr>
</thead>
<tbody>
<tr>
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</table>

**Remarks:**

- Mean: 0.1660
- SDEV: 0.0525

180
<table>
<thead>
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<th>DAMP</th>
<th>AMPL.</th>
<th>PHS</th>
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</table>

**Remarks:**
## Tank Characterization

PRELIMINARY MODE IDENTIFICATION

Center freq. 1221 Hz, Delta freq. 0.977 Hz
Set up no. 1221, I.L. loss .001, P.L. loss .001

<table>
<thead>
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<th>ITER.</th>
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<td>AMPL.</td>
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Remarks:
**Tank Characterization**  

**Preliminary Mode Identification**

Center freq. 1975 Hz, Delta freq. 1628 Hz  
Set up no. E3, I.L., O.L., P.L., L.O.

<table>
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<th>Iter.</th>
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<tbody>
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<table>
<thead>
<tr>
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<th>Damp</th>
<th>Ampl.</th>
<th>PHS</th>
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</table>

**Remarks:**
**PRELIMINARY MODE IDENTIFICATION**

**CHTR FREQ 2781 Hz, DELTA FREQ 6221 Hz**

**SET UP NO. Z.T., I.L. 2850, P.L. 2851**

<table>
<thead>
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<th>FREQ.</th>
<th>DAMP</th>
<th>AMPL.</th>
<th>PHS</th>
<th>INTERVAL</th>
<th>LEVEL</th>
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**REMARKS:**

---

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### Tank Characterization

**PRELIMINARY MODE IDENTIFICATION**

CNTR FREQ: 30.23 Hz, DELTA FREQ: 1.627 Hz

SET UP NO.: X__, I.L. I.A., I.L. I.D.

<table>
<thead>
<tr>
<th>MODE</th>
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<th>LEVEL 10.0</th>
<th>ITER. 5</th>
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<tbody>
<tr>
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<td>8</td>
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**REMARKS:**
## Tank Characterization

### Preliminary Mode Identification

**Center Frequency:** 352.5 Hz, **Delta Frequency:** 9.77 Hz
**Set Up No.:** 236, **I.L.:** 2066, **P.L.:** 2066

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**Remarks:**
## Tank Characterization

**Preliminary Mode Identification**

Center Frequency \(40.02\) Hz, Delta Frequency \(1.95\) Hz; Set Up No. \(27\); I.L. \(106\), P.L. \(2.05\)

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**Remarks:**
## Tank Characterization

**Preliminary Mode Identification**

- **Center Frequency**: 46.39 Hz, Delta Frequency: 0.38 Hz
- **Set Up No.**: \( \text{SETUP}_7 \), I.L.: \( \text{L}_{1666} \), P.L.: \( \text{L}_{1825} \)

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<th>PHS</th>
<th>INTERVAL LEVEL</th>
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**Remarks:**
## Tank Characterization

### Preliminary Mode Identification

Center Frequency: 5709 Hz, Delta Frequency: 1428 Hz

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Remarks:

189
### Tank Characterization

**Preliminary Mode Identification**

- Center Frequency: 5367 Hz
- Delta Frequency: 0.937 Hz
- Set Up No.: 210
- I.L.: 1.000
- P.I.: 2.001

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<th>AMPL.</th>
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**Remarks:**
### APPENDIX F

**SPECIMEN DAMPING MEASUREMENT DATA**

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<th>(Fixed)</th>
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<th>TYPE</th>
<th>RANGE</th>
<th>INPUT LOC.</th>
<th>RESPONSE</th>
<th>FILTER</th>
<th>STORE</th>
<th>DATA</th>
<th>LOC.</th>
<th>(GRID NO.)</th>
<th>(CF, DF)</th>
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LAB AIR, FRONT REMOVED

SPECIMEN IN TEST CHAMBER

Temp: 24.5°C
THE DESIGN OF A TEST PROCEDURE FOR THE MEASUREMENT OF ACOUSTIC DAMPING OF MATERIALS AT LOW STRESS

R A HEIDGERKEN SEP 83

UNCLASSIFIED

END

DATE

D 83

DIE
### SPECIMEN (FIXED) p. 2

<table>
<thead>
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<th>RESPONSE</th>
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**LAB AIR, FRONT REMOVED**

**SPECIMEN IN TEST CHAMBER, TEMP 24.5°C**

192
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<th>SETUP NO.</th>
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<th>RANGE</th>
<th>INPUT LOC.</th>
<th>RESPONSE</th>
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**SPECIMEN (FIXED) A \( \frac{1}{2} \)**

**SPECIMEN IN TEST CHAMBER**

**Temp:** 24.5°C

193
**SPECIMEN (FIXED)**

**AVERAGE MODAL FREQUENCIES AND DAMPING**

**CNTR FREQ:** 673 Hz  **DELTA FREQ:** 795 Hz

**SET UP NO.:** E1  **I.L.:** A  **P.L.:** X

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<th>DAMP. COEFF. (RAD/SEC)</th>
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**REMARKS:**

**OVERALL - 26 NODES**

**MEAN:** 0.1112  **SDEV:** 0.0601

**DAMPING FACTORS**

194
**SPECIMEN (FIXED)**

**AVERAGE MODAL FREQUENCIES AND DAMPING**

<table>
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<th>Nat. Freq (Hz)</th>
<th>Damp. Fact. (%)</th>
<th>Damp. Coeff. (Rad/Sec)</th>
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**REMARKS:**

Damping Factor

Mean 0.2173
SDEV 0.1089

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Specimen (Fixed)  p. 3

Average Modal Frequencies and Damping

Center freq: 2969 Hz  Delta freq: 1.95 Hz  
Set up No.: 23  I.L.: A  P.L.: X

<table>
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<th>Nat. Freq (Hz)</th>
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<th>Damp. Coeff. (Rad/sec)</th>
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Remarks:

Damping Factor

Mean 0.0559
SDev 0.0063

196
### AVERAGE MODAL FREQUENCIES AND DAMPING

**Specimen (Fixed)**  

AVERAGE MODAL FREQUENCIES AND DAMPING  

<table>
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<th>Damp. Fact. (%)</th>
<th>Damp. Coeff. (RAD/SEC)</th>
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**Remarks:**  

**DAMPING FACTOR**  

Mean 0.0983  
SD 0.0025
### Average Modal Frequencies and Damping

**Specimen (Fixed)**

**Average Modal Frequencies and Damping**

- **Center Frequency**: 514 Hz
- **Delta Frequency**: 1953 Hz
- **Set Up No.**: 23

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**Remarks:**

- **Damping Factor**
  - Mean: 0.0958
  - SDev: 0.0576

---

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### Specimen (Fixed)

**Average Modal Frequencies and Damping**

- **Central Frequency (Hz):** 6494 Hz
- **Delta Frequency (Hz):** 4.069 Hz
- **Setup No.:** 32
  - I.I.: A
  - P.L.: X

<table>
<thead>
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**Remarks:**

- **Damping Factor**
  - **Mean:** 0.1310
  - **Sdev:** 0.0542
### Specimen (Fixed)

AVERAGE MODAL FREQUENCIES AND DAMPING

CNTR FREQ 3448 Hz DELTA FREQ 6.01 Hz
SET UP NO. E5 I.L. A P.L. X

<table>
<thead>
<tr>
<th>MODE</th>
<th>NAT. FREQ (Hz)</th>
<th>DAMP. FACT. (%)</th>
<th>DAMP. COEFF. (RAD/SEC)</th>
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**Remarks:**

DAMPING FACTOR

MEAN 0.0917
SDEV 0.0541
**SPECIMEN (FIXED)**

AVERAGE MODAL FREQUENCIES AND DAMPING

**CNTR FREQ 10.985 Hz**  **DELT A FREQ 4.82 Hz**

**SET UP NO.**

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**REMARKS:**

**DAMPING FACTOR**

- **MEAN** 0.1000
- **SDEV** 0.0396
**PRELIMINARY MODE IDENTIFICATION**

**SPECIMEN (FIXED)**

**MODE** | **FREQ.** | **DAMP** | **AMPL.** | **PHS**  
---|---|---|---|---
1 | 51.26 | 8.24 | 38.26 | 184
**PRELIMINARY MODE IDENTIFICATION**

**CNTR FREQ 1758 Hz, DELTA FREQ 9.83 Hz**

**SET UP NO. 2, I.L. A, P.L. X**

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**REMARKS**
### Preliminary Mode Identification

**Center Frequency (Hz):** 2969 Hz, **Delta Frequency (Hz):** 185.3 Hz, **Setup Number:** 23, **In.L.:** A, **P.L.:** B

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**Remarks:**
## PRELIMINARY MODE IDENTIFICATION

CNTR FREQ $437.7$ Hz, DELTA FREQ $1.221$ Hz  
SET UP NO. 27, I.L. __, P.L. __

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### REMARKS:

205
## Preliminary Mode Identification

Center frequency $550$ Hz, Delta freq $1.953$ Hz

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Remarks:
PRELIMINARY MODE IDENTIFICATION

**SPECIMEN (FIXED)**

**SET UP**
- **CNTR FREQ:** 6.91 Hz
- **DELTA FREQ:** 4.69 Hz

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**REMARKS**

207
PRELIMINARY MODE IDENTIFICATION

CENTER FREQUENCY 844.9 Hz, DELTA FREQUENCY 6.014 Hz
SET UP NO. X7, I.L. A, F.L. X

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**LARGE HAMMER (0 → ~6000 Hz)**

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**SPECIMEN ON FOAM RUBBER**

**LAB AIR, Temp: 21°C**
## SPECIMEN (FREE) P. 2

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**SMALL HAMMER (0 - 12,000 Hz)**

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**SPECIMEN on FOAM Rubber**

**Labs Air, Temp: 21°C**
### Specimen (Free)

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<td>CF 8,367</td>
<td>DF 4.883</td>
<td>A</td>
<td>X</td>
<td>Y</td>
<td>Y</td>
<td>25</td>
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<td></td>
<td></td>
<td>27</td>
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Specimen on Foam Rubber

Lab Air, Temp: 21°C

212
**SPECIMEN (FREE)**

**AVERAGE MODAL FREQUENCIES AND DAMPING**

**CNTR FREQ 152 Hz**
**DELTA FREQ 2.4 Hz**
**SET UP NO. 31**

<table>
<thead>
<tr>
<th>MODE</th>
<th>NAT. FREQ (Hz)</th>
<th>DAMP. FACT. (%)</th>
<th>DAMP. COEFF. (RAD/SEC)</th>
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<tbody>
<tr>
<td>1</td>
<td>1415.4900</td>
<td>0.084</td>
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<td>2</td>
<td>1761.7776</td>
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<td>10.3001</td>
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**REMARKS:**

**MEAN 0.0885**

**SDEV 0.0064**

**SPECMEN on FOAM RUBBER**

**LAB AIR T = 21°C**

OVERALL - 28 MODES

MEAN 0.0500

SDEV 0.0504

213
### Specimen (Free)

**Average Modal Frequencies and Damping**

**Center Freq.** 2735 Hz  **Delta Freq.** 4.863 Hz  
**Set Up No.** [Blank]  **I.L.** [Blank]  **P.L.** [Blank]

<table>
<thead>
<tr>
<th>Mode</th>
<th>Nat. Freq. (Hz)</th>
<th>Damp. Fact. (%)</th>
<th>Damp. Coeff. (Rad/sec)</th>
</tr>
</thead>
<tbody>
<tr>
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<td>1762.5181</td>
<td>0.1202</td>
<td>13.3060</td>
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<td>2</td>
<td>2906.4292</td>
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<td>3</td>
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<tr>
<td>4</td>
<td>2956.5000</td>
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<td>15.8496</td>
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<tr>
<td>5</td>
<td>3578.7559</td>
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<td>16.7333</td>
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**Remarks:**

\[ T = 21°C \]

**Mean:** 0.1264  **SDev:** 0.0774  
**Specimen on Foam Rubber**

214
### Average Modal Frequencies and Damping

**Center Frequency (Hz):** 4092 Hz  
**Delta Frequency (Hz):** 3.85 Hz  
**Set Up No.:** 23  
**I.L.:** 8  
**P.L.:** X

<table>
<thead>
<tr>
<th>Mode</th>
<th>Nat. Freq (Hz)</th>
<th>Damp. Fact. (%)</th>
<th>Damp. Coeff. (RAD/SEC)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4322.3330</td>
<td>0.0339</td>
<td>9.2100</td>
</tr>
</tbody>
</table>

**Remarks:**  
Specimen on Foam Rubber  

T = 21°C
### Average Modal Frequencies and Damping

**Center Frequency (Hz):** 495.1
**Delta Frequency (Hz):** 1.25

<table>
<thead>
<tr>
<th>Mode</th>
<th>Natural Frequency (Hz)</th>
<th>Damping Factor (%)</th>
<th>Damping Coefficient (RAD/SEC)</th>
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<tbody>
<tr>
<td>1</td>
<td>5043.016</td>
<td>0.0271</td>
<td>8.6007</td>
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<tr>
<td>2</td>
<td>5149.8730</td>
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<td>9.0214</td>
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<td>3</td>
<td>5172.4209</td>
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<td>8.0086</td>
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**Remarks:**

Specimen on foam rubber, T=21°C

Mean: 0.0265
SDEV: 0.0017
**SPECIMEN (FREE)**

AVERAGE MODAL FREQUENCIES AND DAMPING

<table>
<thead>
<tr>
<th>Mode</th>
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<th>Damp. Fact. (%)</th>
<th>Damp. Coeff. (Rad/sec)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>5173.5977</td>
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<td>2</td>
<td>5677.7148</td>
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<td>11.0148</td>
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</table>

**Remarks:**
- Specimen on foam rubber
- Mean: 0.0295
- Sdev: 0.0020

Center freq: 573 Hz
Delta freq: 469 Hz
Set up no.:

217
**Specimen (Free)**

### Average Modal Frequencies and Damping

**Center Frequ.:** 6738 Hz  
**Delta Frequ.:** 2.49 Hz  
**Set Up No.:** N/A  
**I.L.:** N/A  
**P.I.:** N/A

<table>
<thead>
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<th>Damp. Fact. (%)</th>
<th>Damp. Coeff. (rad/sec)</th>
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<tr>
<td>1</td>
<td>6656.9732</td>
<td>0.0313</td>
<td>13.0864</td>
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<tr>
<td>4</td>
<td>7117.5123</td>
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</table>

**Remarks:** Specimen on Foam Rubber  
**T:** 21°C  
**Mean:** 0.0250  
**S.D.E.V:** 0.0080

218
<table>
<thead>
<tr>
<th>NODE</th>
<th>NAT. FREQ (Hz)</th>
<th>DAMP. FACT. (%)</th>
<th>DAMP. COEFF. (RAD/SEC)</th>
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</thead>
<tbody>
<tr>
<td>1</td>
<td>717.5342</td>
<td>0.0184</td>
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<td>719.0869</td>
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<td>3</td>
<td>779.1553</td>
<td>0.0168</td>
<td>8.2050</td>
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<td>4</td>
<td>790.3027</td>
<td>0.0509</td>
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<td>811.0342</td>
<td>0.0337</td>
<td>17.1760</td>
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REMARKS:
Specimen on foam rubber, 721°C

MEAN 0.0331
SDEV 0.0195
### Modal Frequencies and Damping

<table>
<thead>
<tr>
<th>Mode</th>
<th>Natural Frequency (Hz)</th>
<th>Damping Factor (%)</th>
<th>Damping Coefficient (RAD/SEC)</th>
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</thead>
<tbody>
<tr>
<td>1</td>
<td>8678.7637</td>
<td>0.0350</td>
<td>19.0990</td>
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<td>2</td>
<td>8909.0447</td>
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<td>3</td>
<td>9140.6094</td>
<td>0.0145</td>
<td>8.2993</td>
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</table>

**Remarks:**

- Specimen on Foam Rubber
- $T = 21.9^\circ C$
- Mean: 0.0209
- SDEV: 0.0123
Specimen (Free)  P. 9

Average modal frequencies and damping

Center freq: 10.94 Hz  Delta freq: 4.17 Hz
Set up no. 24  I.L.  B  P.L.  X

<table>
<thead>
<tr>
<th>Mode</th>
<th>Nat. Freq (Hz)</th>
<th>Damp. Fact. (%)</th>
<th>Damp. Coeff. (Rad/Sec)</th>
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<tbody>
<tr>
<td>1</td>
<td>9.775.7812</td>
<td>0.0356</td>
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<td>9.973.4197</td>
<td>0.0224</td>
<td>14.0266</td>
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<td>3</td>
<td>11.040.918</td>
<td>0.0323</td>
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Remarks:

Specimen on foam rubber  T = 21°C

Mean 0.0301
Sdev 0.0069

221
### Preliminary Mode Identification

**Center Frequency (Hz)**: 152.4 Hz  
**Delta Frequency (Hz)**: 2.4 Hz  
**Set Up No.**: 21  
**I.L.**: 8  
**P.L.**: X

<table>
<thead>
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<th>Interval</th>
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<th>Iter.</th>
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<tbody>
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<table>
<thead>
<tr>
<th>Mode</th>
<th>Freq.</th>
<th>Damp</th>
<th>Ampl.</th>
<th>PHS</th>
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</thead>
<tbody>
<tr>
<td>1</td>
<td>41.39</td>
<td>7.93</td>
<td>121.58</td>
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<td>69.19</td>
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**Remarks:**
### Preliminary Mode Identification

**Centre Freq. 2735 Hz, Delta Freq. 54.23 Hz**

<table>
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<th>Freq.</th>
<th>Damper</th>
<th>Amplitude</th>
<th>Phase</th>
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<td>11.12</td>
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<td>31.12</td>
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<td>242</td>
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<td>2</td>
<td>51.94</td>
<td>17.87</td>
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<td>59.96</td>
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<td>81.15</td>
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**Remarks:**
PRELIMINARY MODE IDENTIFICATION

CNTR FREQ 509.2 Hz, DELTA FREQ 3.225 Hz
SET UP NO. 23, I.L. 6, P.L. X

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<th>DAMP</th>
<th>AMPL.</th>
<th>PHS</th>
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<tr>
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<td>63.97</td>
<td>6.95</td>
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REMARKS:
PRELIMINARY MODE IDENTIFICATION

CMTR FREQ $1751\text{ Hz}$, DELTA FREQ $6.53\text{ Hz}$
SET UP NO. 277, I.L. 6, P.L. X

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<tr>
<td>3</td>
<td>59.26</td>
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<td>72.31</td>
<td>3.89</td>
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REMARKS: I
### Preliminary Mode Identification

**CNTR FREQ:** 3.793 Hz, **DELTA FREQ:** 0.67 Hz  
**SET UP NO.:** 25, **I.L.:** A, **P.L.:** X

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**REMARKS:**
**PRELIMINARY MODE IDENTIFICATION**

**CNTR FREQ 6733 Hz, DELTA FREQ 2.44 Hz**

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<th>FREQ.</th>
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<tr>
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<td>43.58</td>
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**REMARKS:**

- 227
**PRELIMINARY MODE IDENTIFICATION**

**CNTR FREQ 2935 Hz, DELTA FREQ 3.98 Hz**

**SET UP NO. 97, I.L. 5, P.L. X**

<table>
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<td>3785</td>
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<td>2</td>
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**REMARKS:**

---

228
**PRELIMINARY MODE IDENTIFICATION**

**CNTR FREQ** 9009 Hz, **DELTA FREQ** 2.44 Hz

**SET UP NO.** 2, **I.L.** 0, **P.L.** X

<table>
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<table>
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<tr>
<th>MODE</th>
<th>FREQ.</th>
<th>DAMP</th>
<th>AMPL.</th>
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<tbody>
<tr>
<td>2</td>
<td>24.31</td>
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**REMARKS:**
**PRELIMINARY MODE IDENTIFICATION**

**CNTR FREQ 10307 Hz, DELTA FREQ 488.3 Hz**

**SET UP NO. 17**

<table>
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<td>79.47</td>
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**REMARKS:**
LIST OF REFERENCES

1. University of Dayton Research Institute, Vibration Damping Short Course Notes, Course Director: Drake, M.L., 1981.


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Cameron Station  
Alexandria, Virginia  22314 |
| 2.  | 2          | Library, Code 0142  
Naval Postgraduate School  
Monterey, California  93943 |
| 3.  | 2          | Mr. E.J. Czyryca, Code 2814  
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Annapolis, Maryland  21402 |
| 4.  | 2          | Mr. V.J. Castelli, Code 2844  
David W. Taylor Naval Ship R&D Center  
Annapolis, Maryland  21402 |
| 5.  | 1          | Department Chairman, Code 69  
Department of Mechanical Engineering  
Naval Postgraduate School  
Monterey, California  93943 |
| 6.  | 5          | Professor Y.S. Shin, Code 69Sg  
Department of Mechanical Engineering  
Naval Postgraduate School  
Monterey, California  93943 |
| 7.  | 1          | Professor A.J. Perkins, Code 69Ps  
Department of Mechanical Engineering  
Naval Postgraduate School  
Monterey, California  93943 |
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Cedar Rapids, Iowa  52404 |