ISA REPRINT

MODALAB—A COMPUTERIZED DATA ACQUISITION AND ANALYSIS SYSTEM FOR STRUCTURAL DYNAMIC TESTING

Strether Smith, Richard C. Stroud,
George A. Hamma, William L. Hallauer
and Richard C. Yee

LOCKHEED MISSILES & SPACE COMPANY
A GROUP DIVISION OF LOCKHEED AIRCRAFT CORPORATION
SPACE SYSTEMS DIVISION · SUNNYVALE, CALIFORNIA

Presented at the 21st
International Instrumentation Symposium
held in Philadelphia, Pennsylvania
May 19-21, 1975

Instrument Society of America
Instrumentation · Systems · Automation
400 Stanwix Street
Pittsburgh, Pa. 15222

MODALAB--A COMPUTERIZED DATA ACQUISITION AND ANALYSIS SYSTEM FOR STRUCTURAL DYNAMIC TESTING

Strether Smith
Research Scientist
Lockheed Palo Alto Research Lab
Palo Alto, Calif.

Richard C. Stroud
Research Specialist
Lockheed Missiles & Space Company
Sunnyvale, Calif.

George A. Hamma
Scientist, Senior
Lockheed Palo Alto Research Lab
Palo Alto, Calif.

William L. Hallauer
Assistant Professor
Virginia Polytechnic Institute
Blacksburg, Va.

Richard C. Yee
Lab Analyst
Lockheed Palo Alto Research Lab
Palo Alto, Calif.

ABSTRACT

MODALAB (MObile Dynamic Analysis LABoratory) is a new system for collection and analysis of structural dynamic data. Its introduction makes possible the practical application of several test/analysis methods that were previously impractical due to their demanding data acquisition/processing requirements.

INTRODUCTION

Modal testing is the most sophisticated and demanding form of experimental dynamic structural analysis. The objective of a modal test is to determine the modal parameters (modal frequency, damping coefficient, and mode shape) so that structural loads, system performance, and control system response can be predicted. The experimental difficulty and cost of these experiments vary according to structural complexity as well as the number and accuracy of modal characteristics required, but tests with more than 100 transducers and costing more than $250,000 are becoming common. In the past few years there has been increasing emphasis on developing modal testing methods that are more accurate, faster, and less expensive.

Many methods for determining modal parameters have been proposed in the literature. For purposes of discussion here they will be divided into two categories: 1) the multiple-driver, tuned-dwell method and 2) the spectrum-analysis approach.

Almost every large-scale modal test since 1950 has been performed using variations on the multiple-driver, tuned-dwell technique of Lewis and Wrisley(1). The objective of this approach is to "produce, in a complex structure, by means of adjustable forces, oscillations that consist essentially of one natural mode of motion". The procedure used is to iteratively adjust several shakers until a) the response phases are all coherent (all in phase or 180° out of phase) and b) beat free decays are obtained. If this can be accomplished, the responses measured will be due only to a single mode. Difficulties arise in adjusting a large number of shakers to isolate modes in frequency ranges of high modal density.

Despite the multiple shaker method's shortcomings, it was the only viable method for modal testing of complex structures characterized by modes with similar natural frequencies. For such testing, a large number of response measurements have generally been required and the limitations of available data-acquisition systems dictated a dwell approach. With this method the amount of data collected and analyzed was relatively small and within the capabilities of the data-acquisition and analysis equipment then available. Emphasis was placed on the skill of the system operators to tune pure modes rather than on data analysis to extract modal data.

As modal-test requirements have become more demanding (i.e., more accurate measurement of more complicated modes in higher modal density), the Lewis and Wrisley approach has become less tenable, and the need for more advanced methodology has been established.

A wide variety of analytical and experimental/analytical separation procedures, which offer the promise of more accurate and faster modal testing, have been proposed during the last 25 years. For the purposes of this discussion it suffices to note that all of these methods rely on analysis of some form of transfer-function spectra. To apply them it is necessary to determine the amplitude and phase relationships between the system forcing function(s) and the system responses as a function of frequency. The immediate implication is that between one and two orders of magnitude more data per channel must be collected and analyzed than with the dwell approach. For large systems (more than 100 channels) it is necessary to apply essentially real-time digital-filtering techniques to satisfy data-storage limitations and to achieve reasonable testing times. MODALAB is a 256 channel system built to provide this capability.

MODALAB CONCEPTS

Basic design objectives of MODALAB were:

1. Exploit recent advances in data-acquisition and processing hardware.

2. Use this capability to implement previously "theoretical" test methods as practical operating procedures.

3. The system should be flexible and easily modified to develop and apply new methodology.

4. The system must be capable of performing a full modal test without outside computational support.

5. The system should be mobile.

These requirements were satisfied by a system incorporating a modern medium-sized computer with a disk operating system combined with a powerful, high-speed data-acquisition system.

Since the modal-analysis techniques are spectrum oriented, it is desirable that the experimental determination of spectra be made in as many ways as possible. Several practical methods are available, the most popular being the Fast Fourier Transform for generalized test input functions, and single-frequency filtering for sine-wave testing. MODALAB is capable of collecting data and analyzing it by both of these methods.

The Fast Fourier Transform system implemented on MODALAB is a pure software system, but because of the fast and powerful computer used, it is nearly as fast as a hardware analyzer. When configured for low-frequency modal analyses, the system can collect data for a 1024-point transform from up to 32 channels at once at a data collection rate of 475 samples per channel per second. As such, it may be used to collect and analyze data from transient or stationary random excitations up to about 120 Hz. The spectra may be post processed to determine modal parameters. This approach, however, has limits of applicability which are important in large-scale modal testing. First, the relatively few channels that can be measured requires that the experiment be repeated a number of times. This, in turn, requires an assumption of test repeatability. Second, using the Fourier Transform to relate arbitrary time histories to spectra that will be used for modal analysis requires a high degree of structural linearity. In many systems this assumption is not realistic. The Fourier Transform approach is being studied to overcome these problems.

With these restrictions in mind, MODALAB has been optimized to perform in the sine-sweep mode. To do this the following special purpose capabilities have been designed into the system.

1) Real-time digital filtering using a hardware/software implementation of the single-frequency Fourier Transform, and

2) Shaker Control System consisting of:

    a) A computer-controlled sine-wave generator with frequency resolution of at least 1 part in 10^4 for frequencies between .1 and 200 Hz, and

    b) Computer- or manually-controlled amplitude and polarity adjustment for up to 15 exciters.

With this capability, the system collects, filters and stores the complex spectra from up to 256 input signals. The data may be taken at any density (in frequency space) up to greater than one part in 10^4. Thus, an essentially continuous transfer function may be obtained. The filtering bandwidth may be software controlled to essentially any value. Testing time is determined by data density (in frequency space) and filter bandwidth, but is independent of the number of channels. Once the response and forcing-function data are collected they may be analyzed in a variety of ways, some of which are subsequently discussed.

MODALAB has been designed to be a flexible, software-oriented system. To provide this a powerful hardware system is required which is described in the following section.

HARDWARE DESCRIPTION

This discussion of the MODALAB hardware system includes a description of the data collection/analysis system as well as a typical excitation and instrumentation system.

The system block diagram is shown in Figure 1. For purposes of discussion it may be broken into nine basic subsystems.

1) The Computer/Data Storage System includes the following devices:

    a) Central Processor: Digital Equipment Corporation PDP-11/45 with the following important features:

        i) 24K 16-bit words of core memory (960 nsec)

        ii) 4K words of MOS memory (450 nsec)

        iii) Hardware floating-point processor

This fast and powerful processor system is essential to the real-time digital-filtering tasks, the FFT processors, and to other heavily used arithmetic routines.

    b) Disk Storage: Two 1.2-million word cartridge disks; one used for the operating system and program storage, and the second for temporary data storage.

    c) Magnetic Tape Storage: One 11-million word digital magnetic-tape system for permanent data storage and for communication with other systems.

    d) Paper-tape reader/punch for communication with other systems.

2) The Analog Data System is made up of two Datel 256 multiplexer/analog-to-digital converter units of 128 differential-input channels each.

The analog-to-digital converters have 12 bit (1 part in 4096) resolution and a range of $\pm$ 2.5 V. The units are sampled alternately to provide a maximum system sampling speed of 200,000 channels per second. The analog system also includes 16 digital-to-analog converters used for computer control of the Shaker Control System.

3) The Interface System is a special purpose device that provides, in addition to conventional data-acquisition system control, the following functions for sine-wave testing:

   a) sine-wave oscillator frequency control,

   b) Analog Data System control,

   c) hardware implementation of the real-time digital filtering algorithm, and

   d) Shaker System control through the Analog Data System.

4) The Shaker Control System provides amplitude-, phase- and frequency-controlled stimuli to the power amplifiers for multiple-shaker sine-wave testing. These variables may be manually or computer controlled. In addition, an analog servo system is used to control the excitation level.

5) The Analog Monitor System includes the following:

   a) a 32-channel multiplexer/signal conditioning system to construct a 16 Lissajous-pattern display on the monitor oscilloscope,

   b) frequency display with .001 Hz resolution, and

   c) patch panel allowing monitoring access to any system channel by manual patching.

6) The Signal Conditioning System provides transducer power, amplification, a-c coupling, and adjustable low-pass filtering.

7) The Operator Interaction System includes a conventional keyboard terminal and an alphanumeric/graphics terminal with copier. The keyboard terminal is used for program entry and editing, and the alphanumeric/graphics system is used for high-speed data output and for graphical presentation of results.

The items described above are shown in Figure 2. The compactness and transportability of the system are evident.

8) The Instrumentation System consists of:

   a) Accelerometers: 250 Unholtz-Dickie 1000-PA devices. These are amplifier-followed piezoelectric instruments whose significant characteristics are:

      Sensitivity 1 v/g

      Frequency Response $\pm$ 5% from .1 Hz to 2KHz

      Self-induced broadband noise equiv. to 20 μg rms

   b) Load cells: Strain-gage units with internal amplification having a sensitivity of .1 v/g. The frequency response matches the nominal accelerometer characteristic.

9) The Excitation System consists of Acoustic Power Systems Model 113 shakers and Model 114 power amplifiers. They provide up to 1.56 N (35 force-lb) with a stroke of 16 cm (6.25 in). The amplifiers are d-c coupled and have the choice of voltage or current feedback control. They normally operate with current feedback.

A typical shaker/load-cell/accelerometer installation is shown in Figure 3.

THEORETICAL CONSIDERATIONS

This section describes the current experimental and analytical procedures used for modal testing with MODALAB. The method is a combined experimental/analytical procedure that uses a spectrum-analysis approach with multiple-shaker modal tuning for mechanical and analytical separation of modes.

The steps in a modal test are as follows:

1) Determine wide-band single-shaker complex-admittance spectra for each shaker and the associated responses.

2) Analyze this data by the method of Asher[2] to determine modal frequencies and the appropriate force distributions for modal tuning.

3) Determine narrow-band multiple-shaker complex-admittance spectra using the force distributions found in step 2.

4) Analyze this data with the method of Smith and Woods[3] to determine the modal parameters.

These steps provide an objective procedure for the performance of a modal test. The combination of an analytical method for determination of shaker force distribution with a method for modal analysis of multiple-shaker sweeps allows the performance of a modal test with relatively little operator judgment required.

The basis for the modal-analysis method is the complex energy-admittance method described in Ref. 3. The method extends the technique of Kennedy and Pancu[4] to allow the analysis of modes that have been reasonably well tuned by using multiple shakers. If it is assumed that the mass of the test structure may be lumped at the experimental degrees of freedom, the kinetic energy associated with the motion of the jth degree of freedom, T_j, can be expressed as

$$T_j = \frac{1}{2} m_j \dot{x}_j^2 \tag{1}$$

where m_j = mass associated with the jth response

and $\dot{x}_j$ = the measured velocity of the jth mass.

The total kinetic energy of the entire structure, T_T, may be represented by

$$T_T = \sum_{j=1}^{N} T_j \tag{2}$$

Now define the power input to the specimen (P_k) at the kth location as the complex product

$$P_k = f_k \dot{x}_k \tag{3}$$

where f_k = force input at the kth location.

The total power input from K forces may be expressed as

$$P = \sum_{k=1}^{K} P_k \tag{4}$$

The local energy admittance of point j is then defined as the complex ratio

$$A_{Ej} = \frac{T_j}{i\omega P} \tag{5}$$

where $i = \sqrt{-1}$

and ω = excitation frequency.

The total energy admittance (defined as)

$$A_{ET} = \frac{T_T}{i\omega P} \tag{6}$$

is a global property representing the behavior of the specimen as a whole.

For a single-mode response with structural damping, the velocity at location j due to excitation at location k is

$$\dot{x}_j = \omega F_k A_{jk} \frac{g + i(1-\beta^2)}{(1-\beta^2)^2 + g^2} \tag{7}$$

A_{jk} = transfer function from point k to point j at resonance

g = structural damping coefficient

β = excitation frequency/modal frequency

If the forces are coherently phased and remain proportional throughout the sweep and if responses are coherently phased (i.e., if the mode is well tuned), it may be shown (from Eq. 5 and Eq. 7) that

$$A_{Ej} = m_j b_j \frac{(1-\beta^2) - i g}{(1-\beta^2)^2 + g^2} \tag{8}$$

where b_j is a constant.

Similarly, from Eq. 6 and Eq. 7 we get

$$A_{ET} = \sum_{j=1}^{N} A_{Ej} \tag{9}$$

Both of these relationships are of the same form as the displacement response admittances which Kennedy and Pancu[4] exploited to determine modal properties. The same techniques are used here with the following exceptions:

1) The total-energy admittance, being a global property, is used to determine the global parameters (i.e., natural frequency and damping).

2) The local-energy admittance is used to determine the eigenvector by fitting a circle to the complex representation of the admittance.

The total energy admittance also provides a single concise indication of modal purity (or the success of modal tuning). Deviations from the classical resonance circle (Fig. 4) are indications of a lack of purity.

Among the assumptions made in the energy-admittance derivation is that the responses are coherently phased. The consequence of this assumption is that the response must be relatively pure (i.e., responding primarily in one mode). To do this, it is required that the relative amplitudes and phases of the shakers be adjusted to tune the mode of interest and suppress all others. In the past, this tuning procedure has been performed subjectively by skilled operators using tedious, time consuming, and often unsuccessful procedures. A more objective approach for obtaining the appropriate shaker-force adjustments was proposed by Asher[2].

A generally accepted criterion for resonance of a purely excited mode is that all admittance be in quadrature with the excitation or, conversely, devoid of coincident admittance. The objective of tuning, therefore, should be to select a combination of forces that produces an admittance with no coincident component.

Consider the expression

$$\vec{x}(\omega) = [A(\omega)] \vec{F} \tag{10}$$

where $\vec{x}(\omega)$ is the complex-displacement vector,

$[A(\omega)]$ is the complex-admittance matrix, and

$\vec{F}$ is the excitation-force vector.

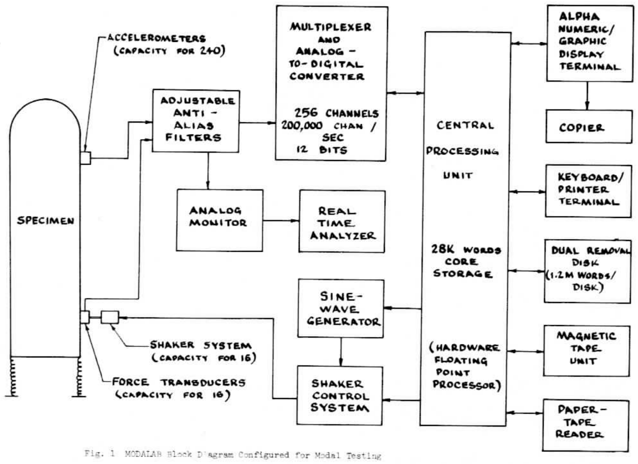

Fig. 1 MODALAB Block Diagram Configured for Modal Testing

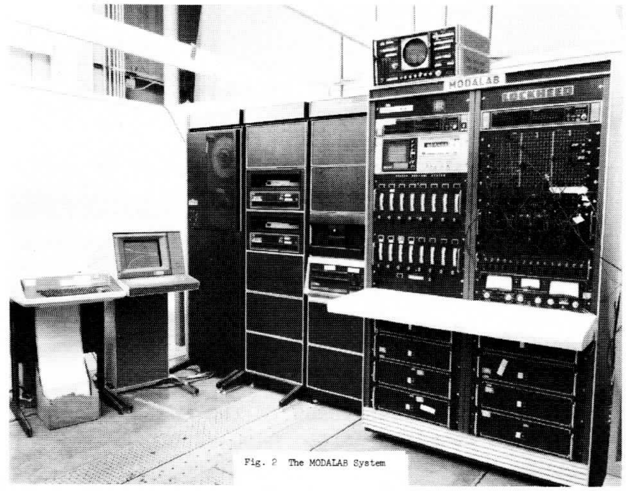

Fig. 2 The MODALAB System

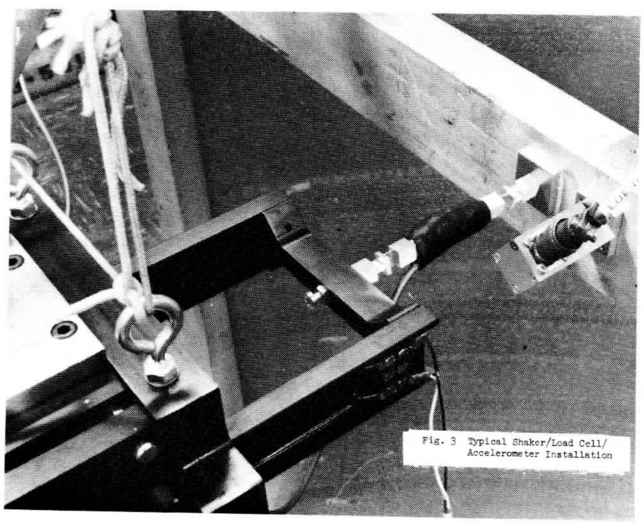

Fig. 3 Typical Shaker/Load Cell/ Accelerometer Installation

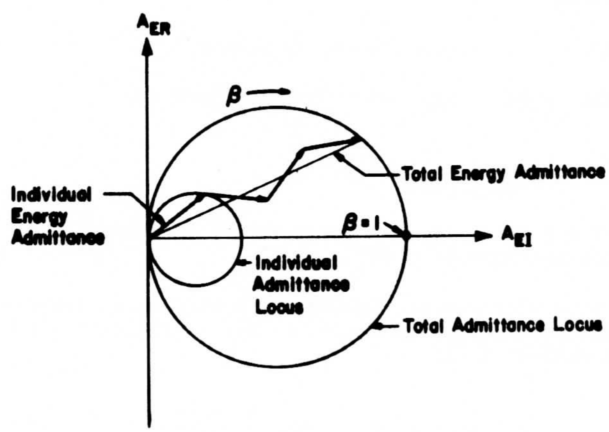

Fig. 4 Classical Resonance Behavior

The columns of the admittance matrix are the displacement-response vectors for a unit excitation by a single shaker. This matrix can be constructed experimentally by means of a series of wide-band, single-shaker, sinusoidal sweeps or by means of Fourier transform techniques applied to response data from impulse, random or rapid sinusoidal-sweep data.

Eqn. 10 may be separated into its real, $C(\omega)$, and imaginary, $Q(\omega)$, parts. It was stated in the preceding paragraph that the objective of tuning is to achieve a condition of null coincident response; i.e.,

$$[C(\omega)]\ \vec{F} = 0 \tag{11}$$

If a solution for Eqn. 11 exists, the columns of $[C(\omega)]$ must be linearly dependent and the determinant of $[C(\omega)]$ will vanish.

Therefore $|C(\omega_n)| = 0$ implies that ω_n is a resonant frequency and solution of the corresponding set of homogeneous equations

$$[(C(\omega_n)]\ \vec{F} = 0 \tag{12}$$

yields the force distribution required to excite the nth mode exclusively.

By using the two methods discussed above, modal testing can become a straightforward, objective procedure. Difficulties that arise in the use of this method are:

1) selection of appropriate shakersto assure that modes of interest can be tuned (i.e., there is a solution of Eqn. 11 for each mode of interest), and

2) determination of how many and which shakers should be used to tune each mode (see Refs. 5 and 6).

SYSTEM APPLICATION

A modal test has been performed on the specimen shown in Fig. 5. It is 9.1 m (30 ft) long, weighs 6000 Kg (13,000 lbs), and structurally simulates the Large Space Telescope. The specimen is made up of semi-monocoque shells with mass simulators for the optical elements and the control moment gyros.

In the test described here it was desired to obtain the modal behavior of the specimen at very low vibration amplitudes (of the order of .00025 mm (.000010 inches). Thus, the accelerometer outputs were amplified giving a nominal calibration of 100 v/g (or 25×10^{-3} g full scale).

The excitation system was made up of ten 4.5N (1-lb) shakers. The specimen was supported by an air bag suspension system which provided nominal "free free" support for frequencies above 2 Hz.

The normal procedure described in the previous section was followed for each of the frequency ranges of interest. The data to be shown here demonstrate the successful separation of two modes at about 34.5 Hz.

Figure 6 shows a set of complex admittance spectra for a 3x3 array of shaker/response locations that were selected as having high admittances at about 34.5 Hz. The real component of the admittance was entered into the Asher analysis, and the resulting plot of the coincident-admittance determinant is shown in Fig. 7. There are two zero crossings indicating the presence of two modal frequencies, at about 37.5 and 37.75 Hz. The force-distribution determinations for the three selected shakers were made and narrow-band sweeps were performed for each mode. The total-energy admittance for each mode is shown in Fig. 8. As may be seen, both modes are nearly perfectly tuned and modal analysis is straightforward. Modal frequency and damping are determined from the total-energy admittance and the mode shape is found by fitting circles to the local-energy admittance.

The two modes separated are characterized by rocking motions, about perpendicular axes, of one of the principle optical elements. In theory, these modes have identical frequencies but minor eccentricities in the actual structure caused the small difference. Without mode separation, these two modes would probably be interpreted as a single, wide (high-damping) mode, an entirely erroneous result. Using this procedure, it has been possible to acquire and analyze modes in frequency ranges of high modal density at the rate of about two hours per mode for 240 response channels.

CONCLUSIONS

This paper has described the hardware and software that presently make up the MODALAB dynamic-analysis system. The system uses a combination of hardware and software to provide rapid data acquisition and analysis of single- and multiple-shaker sine-sweep data for modal analysis.

By virtue of its software-oriented nature, the system is being upgraded to apply new techniques and improve previously used ones.

Work presently in progress includes:

1) Generalization of the method of Smith and Woods to allow the analysis of poorly tuned modes.

2) Generalization and improvements to the method of Asher.

3) Data-smoothing algorithms to improve noisy (low level) data.

4) Analytical mode-separation routines for untuned transfer functions.

MODALAB has sufficient hardware power to apply any dynamic analysis procedure presently envisioned. The software-oriented approach used in designing the system will allow efficient research into and development of new techniques and methodology.

REFERENCES

(1) Lewis, R.C., and Wrisley, D.L., "A System for the Excitation of Pure Natural Modes of Complex Structures," J. Aeronaut. Sci., Vol. 17, No. 11, 1950, pp 705-722

(2) Asher, G.W., "A Method of Normal Mode Excitation Utilizing Admittance Measurements," Proc. National Specialists' Meeting on Dynamics and Aeroelasticity, Ft. Worth, Institute of Aero. Sci., 1958, pp 69-76

(3) Smith, Strether, and Woods, A.A., Jr., "A Multiple Driver Admittance Technique for Vibration Testing of Complex Structures," The Shock and Vibration Bulletin, Bulletin 42, Part 3, p 15, January 1972

(4) Kennedy, C.C., and Pancu, C.D.P., "Use of Vectors in Vibration Measurement and Analysis," J. Aerospace Sci., Vol. 14, No. 11, 1947, pp 603-625

(5) Asher, G.W., "A Note on the Effective Degrees of Freedom of a Vibrating Structure," AIAA Journ., Vol. 5, No. 4, April 1967, p 822

(6) Craig, R.R., Jr., SU, Y.-W.T., "On Multiple-Shaker Resonance Testing," AIAA Journ., Vol. 12, No. 7, July 1974, pp 924-931

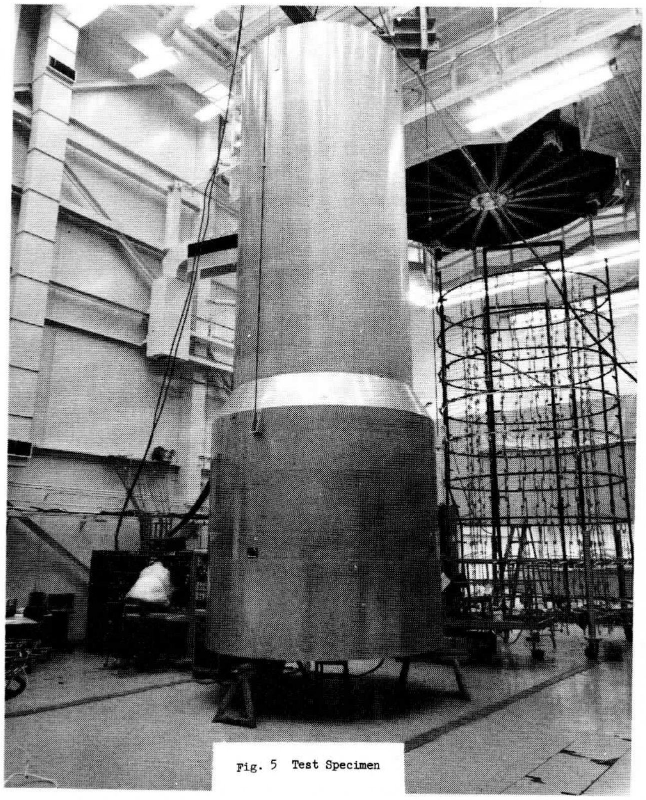

Fig. 5 Test Specimen

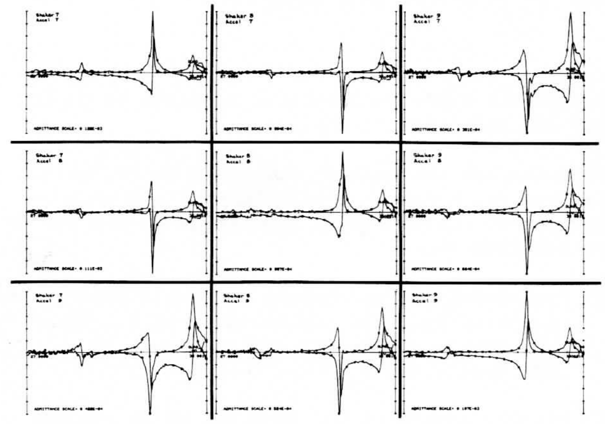

Fig. 6 Complex Admittance Array

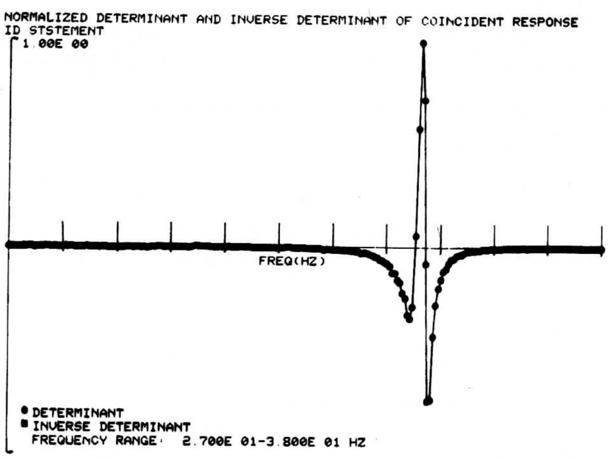

Fig. 7 Coincident Admittance Determinant

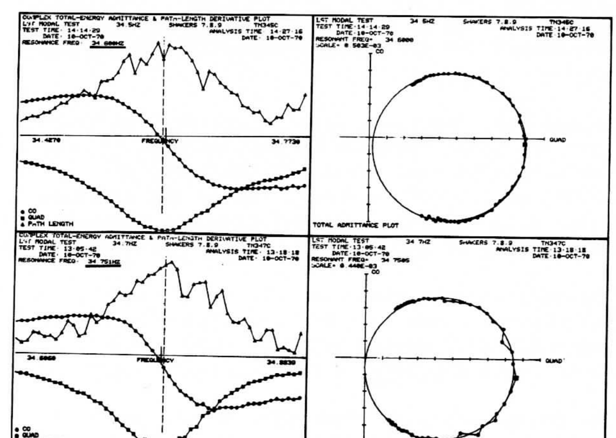

Fig. 8 Total Energy Admittance for Tuned Modes