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(Part 1)

THE SHOCK AND VIBRATION BULLETIN

Part 1
Summaries of Presented Papers

OCTOBER 1974

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FOREWORD

This part of the Shock and Vibration Bulletin is published prior to the meeting. Actual papers will be published in later parts.

Several reasons exist for so doing:

(1) The summaries in this part are delivered to registrants in advance of the meeting to enable them to choose more wisely the papers which they wish to hear.

(2) It is our intent that these advance summaries be used to formulate pertinent questions and comments to be posed during the question and answer sessions at the meeting.

(3) Some of the papers to be presented are not offered for publication in the bulletin.

(4) Papers covering a very useful piece of work may fail to survive the refereeing procedure for reasons unrelated to the work itself.

In the latter two cases, the summaries provide information not otherwise available, and if one is interested in further details, he may contact the authors. Thus, it becomes important to publish this information where it may be referenced in other papers and becomes a part of the useful body of literature.

In particular, we request you to use these summaries to prepare questions for the discussion periods following the paper presentation. Nothing adds to a symposium quite as much as an interesting and informative question and answer session. In controversial areas, it is quite useful to have other views presented. We encourage you to prepare for an active participation in the discussions. Rest assured that your questions and comments will be appreciated.

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IMPACT

EXPLOSIVELY PROPELLED ROTATING PLATES FOR OBLIQUE IMPACT EXPERIMENTS

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Detonating solid explosive is employed to accelerate massive plates to high velocity. The plate is launched intact with a low rotational velocity. The device being studied is positioned along the flight path, allowing distance for plate rotation before impact. Thus, any angle between the plate surface and plate velocity vector can be obtained. Either plastic or metal plates are employed to study impact response against either hard or soft materials.

This technique is employed as an impact simulator to study the performance of full-scale fuzing hardware at velocities from 1500 to 3500 meters/second. Experiments are described in which 160 kg of explosive are employed to propel 15 kg plates achieving impact at angles ranging from near normal to oblique conditions. Procedures for explosive system design and studies of experiment repeatability are described. Plate thickness requirements are estimated from normal shock relationships. Carefully designed experimental arrangements permit recovery of pulse X-ray impact photographs.

It is concluded that explosively accelerated rotating plates provide an accurately controlled and relatively inexpensive impact simulator for fuzing studies at high velocity.

IMPACT TESTING USING A VARIABLE ANGLE ROCKET LAUNCHER

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One of the several facilities at Sandia Laboratories for environmental shock testing is the so-called Rocket Launcher Facility. This is probably a misnomer because rockets are not actually launched at the facility, but instead are used to propel a sled and test unit along a guide rail until a predetermined velocity is reached, at which time the test unit is separated from the sled and allowed to fly free to impact with a target. The relative impact angle between the line of flight and the target face can be varied from 90° to near 0°.

The main feature of the facility is the guide rail. This guide rail consists of a 16-inch deep wide-flange beam, approximately 73-feet long. One end of the beam is welded at right angles to the flange of an 8-foot long similar wide-flanged shape. The ends of this short beam are attached to two upright frame supports by means of hinges. The free end of the long guide rail can then be elevated by a cable passing over pulleys between upright columns on either side of the rail.

The sled which is propelled along the beam is a welded steel structure designed to ride on the lower flange of the guide rail. The test unit is suspended by special hangers below the bottom of the sled. Propulsion is usually provided by 5-inch diameter HVAR solid propellant rocket motors. The sled has provisions for mounting up to three of these motors on each side of the center web and between the upper and lower flanges of the guide rail. These motors have a nominal burn time of 1.1 seconds and an impulse of 5200 lb-sec.

A computer program is used to determine the length of travel, velocity and acceleration based upon weight of the test unit and the number of motors used. The maximum test unit weight is 2500 lbs. and six HVAR motors can propel this weight to a free flight velocity of approximately 200 feet per second. Similarly, a 50-lb. test unit can be propelled to approximately 400 feet per second.

In use, the length of travel is adjusted so that when the desired velocity is reached the sled contacts a honeycomb braking material on the guide rail. When this contact is made the sled begins to slow down and the test unit separates from it and continues in free flight to impact. The impact target can be varied to suit the test requirements, but normally concrete slabs or dirt banks are used.

Due to the relatively short travel distance (80-100 feet) of the test unit, it is possible to hardwire between transducers in the unit and a bunkered conditioning and recording station several hundred feet away. High speed motion picture cameras are also used on nearly all tests to record impact motions.

EVALUATION OF THE SHOCK PULSE TECHNIQUE TO THE UH-1 SERIES HELICOPTER

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The U.S. Army Aviation Systems Command (AVSCOM) has an on-going program to develop a system which will automatically accomplish inspection, diagnostic and prognostic maintenance funtions on related subsystems of the UH-1 helicopter. Past efforts have included the collection of vibration data with a subsequent analysis of the resulting power spectral densities to determine the condition of the helicopter power train. Another approach, particularly in determining bearing conditions, is to use shock pulse techniques. An evalution of this technique to the UH-1 series helicopter was made by Parks College under contract to AVSCOM. Initially, the effort concentrated primarily on the hanger bearings of the tail rotor assembly and the 42° gearbox on operational helicopters of a Reserve Army Aviation Unit. Subsequently, data was collected from the AIDAPS helicopters at Fort Rucker with implants in the 42° gearbox. A standard, off-the-shelf, SKF Industries MEPA-10A Shock Pulse Meter was employed to construct shock emmission envelopes of shock rate versus shock level.

In constructing the shock emission envelope, the first point plotted on the ordinate axis is the rate level at a value level of one. A threshold varying dial on the MEPA-10A meter housing ranges from a level of one to ten thousand in a logarithmic scale. As the threshold is increased, successive rates were plotted until the wave crossed the abscissa axis. The value at the intercept becomes the highest potentiometer level at which at least one shock pulse per second can be measured. The curves drawn were then compared to the general curve forms of different types of damage and, coupled with the rate and level values, an assessment of bearing condition was made. Selected hanger bearings and 42° gearboxes were removed for teardown analysis.

The hanger bearings had shock rates ranging from 55-340 pulses per second potentiometer levels varying from 45 to 6000 units. Those hanger bearings which were deemed to have excessive levels were removed and new bearings installed. The range of levels of the new bearings was markedly reduced from 50–100 units. Teardown analysis showed damage, primarily pitting and corrosion, to vary from slight to severe. The data has been summarized in a single shock emission envelope which can be used to separate hanger bearings of normal wear or the onset of damage from those with severe damage.

The tests conducted at Fort Rucker were on 42° gear boxes of known condition. Data was collected on good gear boxes and also from those implanted with damaged bearing elements. Of particular interest was the observation of progressive damage while a test was in progress. The shape of the shock emission curve changed continuously over a period of minutes in both rate and shock level. The shock level stabilized at a factor of 10 higher than the initial readings. Subsequent teardown analysis verified that a ball in the duplex bearing on the input quill had developed a spall possibly from metal fatigue. A new spall was also developing on the outer race.

The use of shock pulse techniques has proven successful in the limited applications to date. Further work is in progress on the other elements of the UH-1 power train.

STRUCTURAL RESPONSE MODELING OF A FREE-FALL MINE AT WATER ENTRY

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The free-fall mine concept applies to aircraft planting of underwater mines without air retardation devices such as parachutes. The planting of a minefield with retarded mines has certain advantages over conventional retarded mines. These advantages include higher placement accuracy and less possibility of the enemy being able to observe and plot the placement location. The high speed at water entry of the free-fall mine, however, imposes severe structural loading on the mine at water impact as well as generating bottom burial problems. This study was directed toward defining the water-entry pressures and impulse, the structural response of the mine, the failure modes, and the bottom burial characteristics.

A test vehicle was designed by the Naval Ordnance Laboratory to test the free-fall delivery concept. This vehicle was a 21-inch-diameter, 0.5-inch-thick right circular steel cylinder, 113 inches long, with a 1.5-inch-thick flat steel plate nose and a blunt after end with four fins. It was ballasted with concrete to a weight of 2,060 pounds. Six of these units were built in full-scale size and test dropped. Three of them contained copper ball and mechanical oscillator peak shock recorders. Structural deformation in the form of dishing of the nose due to hydrodynamic pressures and dishing of the tail due to inertial forces occurred at entry velocities of 500 feet per second and above at a nominal vertical entry angle. Bottom impact shocks were considerably lower than water-entry shocks and burial was not a problem.

It was decided to conduct subscale model testing of the water-entry phase to gain a better understanding of the hydrodynamic forces and resulting vehicle behavior, and also to develop and validate modeling techniques as a less expensive means than a full-scale prototype testing to obtain necessary design data for water-entry vehicles. Since prototype data were available for the Free-Fall Mine Test Vehicle, this design was modeled geometrically to 0.231 scale using materials the same as those of the prototype. The model was similarly instumented with copper ball and mechanical oscillator peak shock recorders. Tests were made at velocities of 50 to 300 feet per second with 85 and 90 degree water-entry angles. Data from the recorders were highly reproducible.

Peak shock data from the model and prototype were compared using modified Froude number scaling. Good agreement resulted.

The mine design was programmed into the NASTRAN structural response computer code. A water-entry pressure pulse was inferred for the model so the computer-calculated peak shock values resulting from the input of that pulse matched the experimental data. Scaling of this water-entry pulse to prototype values and inputing it to the computer resulted in peak shock values in good agreement with prototype data. This agreement supports confidence in the accuracy of the computer modeling program which includes outputs of stress, frequency, and acceleration.

As a result of the tests made in this program, it is felt that subscale models can be used to accurately simulate the structural response of vehicles at water entry and that subscale models can also be used to generate data in the form of the pressure pulses experienced by bodies for input to structural response computer programs.

PACKAGING AND SHIPPING

HIGHWAY SHOCK INDEX (SI) PROCEDURE FOR DETERMINING (SI)

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Representatives of the United States Army, Navy, Air Force, and Marine Corps agreed that it should be possible to establish shock indices which would be representative of the cargo environment for the various transport modes. The Services formed a steering Committee to initiate and guide the development of a highway shock index. The highway mode was selected because of the relative ease in controlling the environment and related variables. Exploratory work was accomplished on a contractual basis and reported on in papers which were presented at the shock and vibration symposium and annual meeting of the Highway Research Board in 1972. The final work, which consisted of a series of instrumented field tests on three typical highway cargo vehicles was conducted by MTMTSTEA Engineers at Fort Eustis, Virginia. This work resulted in the development of a simple, practical procedure for determing the shock index of a cargo vehicle, and is presented in the paper "Highway Shock Index (SI), Procedure for Determining (SI)."

The purpose of the Highway Shock Index (SI) is to provide a means for selecting highway cargo vehicles on the basis of their "rough" riding characteristics. The selection is based not on the vehicle configuration but on the combined payload spring rate "K" of the springs and tires on an axle. SI makes it possible to select a relatively "soft" riding vehicle for fragile cargo and thus minimize the possibility of damage to the cargo. The shock index rating system applies to restrained cargo only.

The shock index for a cargo vehicle should be representative of the roughest ride area on the truck cargo bed. Previous tests have shown, and recent tests have confirmed that for normal operating conditions the roughest ride on a truck cargo bed is found over the rear axle for a two axle cargo truck; or, for a truck tractor, semitrailer combination either near the rear axle of the trailer or over the fifth wheel of the truck tractor depending on which axle has the higher payload spring rate.

Under normal operating conditions maximum shocks on the cargo bed will occur in the vertical direction; and, based on extensive tests by MTMTSTEA and other organizations a maximum shock of 10g is considered reasonable for a very rough road surface. Consequently, the Highway Shock Index is based on a scale of 0 to 10g's; the numerical values of shock index vary from 5 to 0 with 5 corresponding to 0g's, representing the softest "ride."

The dead weight of the vehicle is not involved in the determination of shock index; the unloaded weight of the vehicle is already in place and therefore is not involved in the determination of the *payload* spring rate. The tests have shown that of the three major variables percent maximum payload, tire pressure and speed, percent maximum payload has a major effect on shock index whereas tire pressure in the practical range and speed cause relatively minor changes.

Since percent maximum payload has the most effect on the "ride" on the truck cargo bed, a graph relating the payload axle spring rate, axle payload and shock index was developed. In order to develop the graph tests were conducted on a range of cargo vehicles. The payload capability of these vehicles varied from 13,000 pounds on a two-axle truck; to 24,000 pounds on a two-axle truck tractor, single axle trailer combination; to 40,000 pounds on a three-axle truck tractor, two-axle semitrailer combination.

The vehicles were instrumented to measure shock on the cargo bed and were driven over fixed, unyielding bumps at various speeds at different tire pressures, and with different payloads.

The repeatability of data measurements recorded on the test course was satisfactory in spite of the many variables that affect a dynamic test of this type. Approximately eighty percent of all data recorded over the axles of the trucks was used in the preparation of the graph and table which can be used to determine the shock index of a highway cargo vehicle.

THE DYNAMIC ENVIRONMENT OF FOUR-INDUSTRIAL FORKLIFT TRUCKS

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The forklift truck is in general use throughout the industrial complex as a means of short, in-plant transport; but little work has been done to determine the frequency and amplitude of accelerations to which cargo is subjected during the virtually inevitable trips on the forks.

The purpose of this study was to determine the dynamic input to cargo during carriage on various types of forklift trucks. Among the variations examined were trucks with capacities of 2000 pounds, 3000 pounds, 4000 pounds and 7000 pounds, trucks with pneumatic and solid tires and trucks powered by gasoline engines and electric motors.

The cargo used was a simulated bomb configured to correspond in size, shape, weight and center of gravity to an existing weapon. The shape was mounted on a cradle-like rack which is used to transport and handle weapons. This configuration was, in turn, picked up by a forklift truck and transported over a prescribed route. Accelerometers were mounted at the base of the rack near the point of contact with the forks.

It was found that there was little steady-state continuous excitation transmitted through the forks to the load. Many discrete excitations were present. Data reduction in the form of shock response spectra showed responses up to 10g in amplitude below 20 Hz and up to 40g at 100 Hz.

A STATISTICALLY BASED PROCEDURE FOR TEMPERATURE SENSITIVE DYNAMIC CUSHIONING CURVE DEVELOPMENT AND VALIDATION

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SUMMARY

Foamed thermoplastic materials are particularly attractive for use as lightweight, low-cost, easily fabricated cushioning system in shipping containers. However, foamed materials have found limited use in military containers, due primarily to the difficulty the container designer encounters in attempting to predict the dynamic response of the packaged item when a container is exposed to the rigors of worldwide distribution. Military containers encounter widely varying temperature extremes in worldwide distribution. However, current design practice utilizing available dynamic cushioning curves completely ignores the effect of temperature, due to the absence of temperature sensitive dynamic cushioning design curves.

In an attempt to develop statistically significant design curves which consider temperatures in addition to the classical ambient value (70°F), an experiment was designed to investigate the change in cushion impact absorption characteristics at specified extreme temperature values of −65° and 160°F. The basic concept of the experiment was the premise that an actual drop test program was required to establish the validity of a temperature effect hypothesis. It was highly desirable that any experimental data be acquired within the framework of randomization. Since a reasonable number of drops were required to achieve statistical validity, the physical limitations of the drop test program were scrutinized, and it was found that the factors which affected the results were static stress level, material type, material density, material thickness, drop height, and temperature. Consequently, predetermined stress levels were set for each material type, density, and drop height, since it was a relatively time consuming process to vary the stress level after each drop. With the existing restrictions on randomization, a split-split plot experimental design was determined to be most pertinent in testing whether temperature does significantly affect the performance of cushioning materials.

The material chosen for the initial drop tests was Hercules Minicell, a cross-linked, closed cell, polyethylene foam. This material is a relative newcomer to the container cushioning material market.

Prior to initiating full scale drop testing, a pilot test program was conducted to verify the adequacy of the experimental design, and to also familiarize the drop equipment operators with the test procedures and equipment. A computerized data form was developed which included a randomization scheme for identifying the order of experimentation.

The drop test program included drop heights of 12, 18, 24, and 30 inches. Stress levels were selected for each drop height based upon the expected location of the developed design curve for the specific characteristics under consideration. Consequently, stress levels are not necessarily common for all drop heights.

The data analysis consisted of: 1) the development of an outlier test based on the variance associated with the three replications of each experimental condition; 2) the development of design curves for the various temperatures, drop heights, and material thicknesses based upon a second order polynomial regression of the logarithm of stress versus deceleration at impact; 3) conducting an analysis of variance for each drop height to determine the effect of temperature differences.

The developed outlier test identified some data values which were statistically invalid; consequently, these values were not utilized in the development of the design curves. Thirty-six design curves were developed, of which thirty-three exhibited a statistically significant parabolic shape at the .05 alpha level. The analysis of variance concerning the temperature effect hypothesis was significant for all drop heights at the .05 alpha level.

Experimental results clearly indicate that a temperature effect does, in fact, exist, and should be taken into account by the container cushioning system designer. Thus, he now has available the necessary temperature sensitive design curves for Hercules Minicell.

A COMPARISON OF PACKAGING CUSHION MATERIAL SHOCK PERFORMANCE EVALUATION TECHNIQUES

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For approximately 30 years, packaging cushioning materials have been scientifically evaluated in order to quantitatively describe their dynamic performance when subjected to externally applied mechanical shock. In the main, this testing has been done as mass impacting cushion. The American Society for Testing and Materials has adopted a standard for this type of testing, designated D-1596. Much data on a wide variety of materials is currently available.

A more recent use of shock test systems to evaluate products and packaging has generated interest in using this vehicle to evaluate cushioning materials. The method under consideration utilizes a weighted, instrumented test block in a simulated package to determine dynamic performance.

This paper compares these two evaluation techniques first from the test design standpoint. They are then examined from an operational orientation. Finally, the two methods are compared in their generation of data. Several materials, both synthetic and non-synthetic are evaluated, the data presented comparatively and graphically. Differences are analyzed statistically for significance of variation. In a discussion section, the two evaluation techniques are compared for general suitability when used to generate engineering data for use in design of protective packaging. The paper is concise with conclusions heavily supported by empirical data.

AEROSPACE VEHICLES

AN ANALYTICAL/EXPERIMENTAL DETERMINATION OF TRANSPORTER LOADS ON THE VIKING SPACECRAFT

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The Viking Spacecraft are to be launched from the John F. Kennedy Space Center in 1975. The two space vehicles will be assembled in the Spacecraft Assembly and Erection Facility and transported to Launch Complex 41. The transportation is over 13 miles of paved roads of varying texture including a draw bridge and several railroad crossings. The basic requirement on this ground handling phase, with respect to loads, is that handling or transport loads shall not impose critical loading conditions or cause damage to the space flight structure.

That transportation loads are not critical for the Viking Spacecraft was demonstrated in November and December of 1973 during transportation testing of a Viking Dynamic Simulator. This paper presents the experimented and analytical methods used in this demonstration. The manner in which spacecraft modal data was used to establish allowable loads and to monitor actual loads is felt to be novel and of general interest.

An accurate mathematical model of the Viking Dynamic Simulator had been previously developed and verified through modal survey testing. From this model six members of primary structure were identified as the critical members which would define the loading conditions in the total spacecraft. That is, by assuming that the loads of interest will be due to response of the spacecraft in its normal modes, it is possible to define modal loads at any point when the load in the six members are known. Working on this basis, allowable loads by modes were established for each of the six structural members.

Strain gages were installed and calibrated in the six structural members, and loads were monitored during the transportation test. The paper presents the development of the allowable load criteria, the loads measurements made in transportation of the Viking Dynamic Simulator and results of comparison. An analytical extension of the results to predict loads in the Viking Spacecraft is also reviewed.

DETERMINATION OF PROPELLANT EFFECTIVE MASS PROPERTIES USING MODAL TEST DATA

Jay C. Chen and John A. Garba

The development of finite elements models for dynamic analyses of spacecraft and launch vehicle systems requires proper dynamic representation of fluid masses.

Proper mass representation is required for the prediction of loads in structural members as well as in the data reduction for the modal test. A limited number of effective propellant mass properties can be obtained analytically from the derivation of fluid slosh models using classical fluid mechanics. Full scale propellant tank testing provides a means for obtaining complete fluid mass properties accounting for such effects as geometric tank irregularities, baffles, and propellant management devices. Due to requirements of elaborate fixturing and instrumentation such testing tends to be expensive.

Modal vibration testing is required on most spacecraft projects as a means of verification of the mathematical model. This paper presents a method for the evaluation of the effective mass of the fluid from modal test results. Overall cost is reduced by elimination of a special effective mass test and substitution of a slight extension of a required modal test.

First, the effective mass matrices of the fluids are derived analytically from the measured system frequencies and mode shapes. While the derivation is intended for the determination of effective fluid mass the method is equally applicable to other unknown masses.

Next the procedure is demonstrated by computer simulation. Feasibility of the method is established by applying it to an example problem wherein analytical data is presumed to be derived from modal test. The fluid effective mass matrix is thus reconstructed and compared to known results. The agreement is found to be very good.

To further illustrate the feasibility of the method it is applied to the Viking Orbiter. In using Viking modal data it was found that the required modal matrix formulated from test data can be ill-conditioned if test inaccuracies exist in the test data. This conditioned modal matrix violated certain assumptions made in the derivation of the method. Based on the past experiences, it is likely that the inaccuracies existed in the mode shape measurements rather than the natural frequency measurements. Therefore, a perturbation technique was developed to "treat" the raw test data to avoid this ill-conditioning and perturb the data to result in a solution with physical reality. The results of the fluid effective mass obtained by this method are compared to that obtained from the full scale tank tests.

The Viking Orbiter data is presented to illustrate the application of this method to a spacecraft containing large fluid masses. The advantages of the method are the elimination of a full scale fluid slosh test and the ability of obtaining the cross-products of the inertia which in general are not measured from the full scale slosh test. This information might be of value to future projects such as the space shuttle.

UNIQUE FLIGHT INSTRUMENTATION/DATA REDUCTION TECHNIQUES EMPLOYED ON THE VIKING DYNAMIC SIMULATOR

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The Viking Spacecraft will be launched aboard a new launch vehicle system, the Titan IIIE/Centaur in August 1975. The Jet Propulsion Laboratory (JPL) is responsible for the

Viking Orbiter System, which is part of the overall Viking Project managed by the Viking Project Office at Langley Research Center (LRC) for NASA.

Although the Viking Spacecraft is not the first hardware designed to fly aboard a new launch vehicle system, areas exist where somewhat unique approaches have been taken to engineering problems. These areas include the basic approach to design of the primary structure; the methods employed in the verification of the design, in particular the flight of a dynamic simulation of the Viking Spacecraft (referred to as the Viking Dynamic Simulator (VDS)); and the flight instrumentation and data reduction techniques employed.

This paper will emphasize the somewhat unique aspects of flight instrumentation and post-flight reduction techniques used on the VDS data obtained in February 1974. The Viking Spacecraft design approach is briefly discussed to provide necessary background information.

The Viking Spacecraft is one of the few projects in which, from inception, the basic design philosophy has been to use predicted transient flight loads to design the primary load carrying structure. This approach has produced some far ranging ramifications in both the test related and analytical aspects of the program. The testing has been directed towards verification of the analytical models and techniques through extensive modal test programs, and, for high level tests, using structural response limitations based on the predicted flight induced loads. This is significantly different from the generally more conservative approach of designing and testing for simulated flight environments. The analytical aspects of the program have emphasized more detailed coupled launch vehicle shill spacecraft dynamic models and responses to external forcing functions. The "closed-loop" aspects of designing for analytically predicted flight loads, which are functions of the dynamic characteristics of the structures being designed, provided impetus for complete verification of the design techniques.

A flight of the new launch vehicle system provided an unusual opportunity to actually fly a dynamic simulation of the spacecraft, the VDS. The main objectives of the VDS were to verify the mathematical modeling techniques, the methodology employed in coupling the spacecraft and launch vehicle models, and the loads prediction techniques themselves. To meet the objectives, the VDS employed a unique set of flight instrumentation, a mixture of accelerometers and strain gauges, as well as a specially designed data reduction facility that included sets of phase and amplitude matched filters and analog computers. The special instrumentation and data reduction facilities made it possible to obtain significantly more information than normally possible with fifteen channels of FM/FM multiplexed instrumentation.

Six pin-ended struts referred to as the Proof Flight Lander Adapter (PFLA) were instrumented with strain gauges. These struts were then individually loaded prior to assembly and calibrated as axial load cells. The flight data obtained from the six struts were used to (1) provide direct data for verification of the loads prediction techniques, (2) provide a complete time history of the forces and moments across an interface by combining the loads with an analog computer and special filters in demultiplexing equipment, and (3), completely define the accelerations of the structure above the PFLA with processing similar to that used in (2). The combination and location of thirteen channels of accelerometers and load cells could, with some assumptions, yield over thirty response parameters when completely reduced and manipulated with the specialized playback equipment.

The use of calibrated load cells which are an integral part of the primary spacecraft structure was demonstrated as a feasible approach to flight instrumentation on the VDS. The advantage of using load cells was that a complete force time history was obtained across an interface. From this, the complete acceleration time history of the structure above the interface was derived. In the past, attempts to determine a complete acceleration time history across an interface, with accelerometers, resulted in limited success because of "local" reasonances. The VDS results will be of interest to engineers selecting flight instrumentation and reduction of data for verification of spacecraft loads and environments. A set of load cell instrumentation similar to that used on the VDS will be employed as the primary response transducers on two Viking Spacecraft to be launched in 1975.

ANALYTICAL PREDICTION AND CORRELATION FOR THE ORBITER DURING THE VIKING SPACECRAFT SINUSOIDAL VIBRATION TEST

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ABSTRACT

The Jet Propulsion Laboratory is responsible for the Viking Orbiter System which is part of the overall Viking Project managed by the Viking Project Office at Langley Research Center for NASA. The Spacecraft will be launched on a Titan IIIE/Centaur Launch Vehicle in August 1975.

The Viking Spacecraft (V-S/C) consists of two major subsystems, the Viking Orbiter (VO) and the Viking Lander Capsule (VLC), built by Martin Marietta Aerospace (MMA).

The Viking Orbiter primary structure was designed for flight loads established by loads analysis. Loads analysis is a procedure for obtaining member design loads from transient response analyses for various flight events.

Qualification testing for the Viking Spacecraft System was divided into three categories, the low frequency, the middle frequency, and the high frequency regimes.

Structural integrity of the VO system in the low frequency range from 0 to 40 Hz was verified by a static test.

The middle frequency range, 20 to 200 Hz overlaps the low frequency range at the lower end. In this range a sine vibration test was used to qualify the Orbiter system.

Testing in the high frequency range, above 200 Hz, consisted of acoustic excitation and pyrotechnic firings.

All VO test models used in the qualification test program had only a mass simulation of the VLC. Thus these tests did not allow for the evaluation of the VO/VLC dynamic interaction.

Developmental type testing using representative elastic VO and VLC hardware was undertaken to study such interaction. This testing consisted of sinusoidal excitation of the

spacecraft in three orthogonal axes. Other objectives of this test were (1) the evaluation of secondary structure, (2) a check of the adequacy of component test levels, (3) the comparison of analytical predictions with test results and (4) to serve as a precursor for the Orbiter qualification test.

The Spacecraft test was implemented such that the responses were limited not to exceed design loads in primary structural members.

Pre-test analysis was required to establish the fixture requirements to prevent a large rotary motion of the V-S/C fixture system that would result in a rotation of the armature that may terminate the test.

Additional requirements on the pre-test analyses were to aid in:

1. Location and number of control accelerometers and strain gages.

2. Location and number of peak limit accelerometers and strain gages (accelerometers and strain gages that would shut down the test if preselected values are exceeded).

The following instrumentation and test facility restrictions had to be used:

1. Control channels = 36

2. Peak limit channels = 59

3. Channels for recording data = 274

4. Number of strain gages available for monitoring $\simeq$ 700

5. Accelerometer locations = 113

6. Minimize the cyclic load on the structure

7. Schedule

8. Data analyses cost.

Computer simulation of the test setup was run to pre-select the parameters and levels for control to:

- not exceed load in other members

- estimate the fatigue life used during a sine test

- minimize probability of test termination because of an exceedance of a limit.

Experimental selection of the proper combination of parameters for control (by trial) would have consumed resources in test time, test data analyses, test data evaluation; increased probability of test malfunction during numerous shutdowns; as well as consume fatigue life of the structure.

As a result of a good computer simulation that includes a complex finite element model of the V-S/C, test fixture/shaker dynamic model, and the control system the V-S/C sine vibration test was completed ahead of schedule and all the objectives were met. The close interaction between the analysis and the test directly contributed to a successful program.

This paper deals with the comparison of analytically calculated frequency responses for member forces and accelerations to test responses obtained from strain gauge and accelerometer data.

A summary is presented of the method used in modeling the system, verification of the components by test, the selection of damping, the method of calculating controlled responses and the test implementation.

Since the correlation of modal test results and the mathematical model was considered to be excellent for Viking Orbiter the comparison of analytical and experimental response data produces a measure to which such correlation can be achieved with good data. It also provides insight into the importance of other parameters such as damping.

FAIL SAFE FORCED VIBRATION TESTING OF THE VIKING 1975 DEVELOPMENTAL SPACE CRAFT

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This paper describes the structural developmental Forced Vibration Test Program accomplished on the Viking 1975 Spacecraft. Test conduct was complicated since Viking hardware is furnished by three separate agencies: Jet Propulsion Laboratory (Viking Orbiter), Martin Marietta Aerospace (Viking Lander Capsule), and General Dynamics/Convair Aerospace (Interface Adapters to Centaur). The Viking Project Office at the Langley Research Center is the Project Manager for the Viking Project.

Considerable doubt existed as to the feasibility of mounting such a large structure on a shaker and maintaining adequate control of the test. The height of the spacecraft coupled with its C.G. offset would cause large overturning moments. In addition, control system limitations observed on previous testing of another spacecraft with similar response characteristics (narrow bands with high amplitudes) had resulted in damage to the structure. Accordingly, a response analysis of the entire test set-up and control system experiments preceded actual test implementation.

The effects of test philosophy on the development of the vibration control system are presented. Structural design of the spacecraft was accomplished using loads analyses based on statistical evaluation of previous Titan Booster flights. Since the Viking was "tailored" to meet a specific, limited launch environment, the standard practice of vibration testing the structure to a large number of loading cycles was rejected. This approach was reinforced by the fact that portions of the Orbiter structure were scheduled for static testing to ultimate load levels following the vibration test.

The two main criteria used to conduct a fail-safe vibration test of the spacecraft were:

1. Vibration input to the test specimen would be controlled so that loads in primary structure members would not exceed analytically predicted flight loads.

2. The number of loading cycles on selected primary structure would be monitored for cumulative damage ratios, a measure of possible fatigue damage to the structure.

To meet these criteria, a 36 channel control and 60 channel limit system were utilized along with over 200 other dynamic signals. Control system limitations, including switching capability and overshoot, are described. An analog/digital process was used for computation of member loading, bending moments, and cumulative damage ratios.

MODALAB A
A NEW SYSTEM FOR STRUCTURAL DYNAMIC TESTING

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I. BASIC CONCEPTS AND HARDWARE

The art of ground vibration testing has been advancing steadily since its inception about 40 years ago. Several major systems have been constructed to apply variations of the multiple shaker dwell technique of Lewis and Wrisley (Ref. 1) and equipment sophistication has reached a high level as shown in Ref. 2. However, in many cases the results of the tests performed have been less than satisfactory and good results have generally been restricted to frequency ranges of low modal density.

For the past 20 years various researchers have developed techniques based on the sine sweep approach (Refs. 3-5), but these have not been applied in practical large scale testing because of the vast quantity of data that must be acquired, stored, and analyzed. Until very recently the hardware required to perform tests using this approach has not been available. MODALAB (MObile Dynamic Analysis LABoratory) is the first system capable of performing large scale multiple channel testing using these methods.

Use of sine sweep techniques provides the following capabilities that are not feasible with dwell techniques:

1. Determination of appropriate multiple shaker force distributions for modal tuning by analytical means (Ref. 4).

2. Determination of mode shape from analysis of complex response (Refs. 3, 5).

3. Determination of modal frequency and damping on the basis of overall structure behavior (Ref. 5).

The result of using this approach is considerable improvement in data accuracy and reliability and a great reduction in time and cost required for testing.

The MODALAB system includes the following features that are necessary for implementation of these advanced techniques:

1. Real-time digital filtering.

2. Permanent high speed disk storage of data with immediate accessibility for processing.

3. A highly flexible software oriented system that allows rapid technique development and program modification.

4. Rapid graphic output for inspection of test progress by real time display and for evaluation of reduced data.

5. The system is capable of standing alone from data collection to comparison of experimental and analytical data.

MODALAB will also perform as a conventional high speed data acquisition system. The optimization for sine wave testing (specifically the real time filtering process) does not degrade general purpose operation.

At present software has been written to perform and analyze the following types of test:

1. Wide-Band Sine Sweep
   a. Transfer function (complex or total)
   b. Transmissibility (complex or total)
   c. Linear sums of responses
   d. Analysis to determine multiple shaker force distributions by Ref. 4

2. Narrow-Band Sine Sweep (Single Shaker)
   a. Modal analysis by Ref. 3

3. Narrow-Band Sine Sweep (Multiple Shaker)
   a. Modal analysis by Ref. 5

4. Transient Recording (Short Period)
   a. Kinetic energy decay analysis
   b. Fast Fourier transform

5. Random Signal Recording (Long Period)
   a. Fast Fourier transform

Performance of these tasks using software oriented concepts requires a powerful hardware system. MODALAB is made up of seven basic subsystems:

1. Computer/storage - disk operating system on a medium sized computer.

2. Operator communication through graphic and terminal input/output.

3. Analog data acquisition—256 channels at 200,000 ch/sec.

4. Interface system including part of the digital filtering algorithm in hardware.

5. Modal testing shaker control system including amplitude, phase and frequency control of up to 15 shakers by manual or computer control.

6. Instrumentation—240 high resolution accelerometers and 15 load cells and associated signal conditioning.

7. Shaker system with fifteen 30-lb 6-1/4" stroke shakers, and current controlled D.C. coupled amplifiers.

The hardware system will be detailed in the presentation.

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MODALAB R
A NEW SYSTEM FOR STRUCTURAL DYNAMIC TESTING

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II. ANALYSIS CONCEPTS AND PROCEDURES

It has long been recognized that accurate experimental definition of modal characteristics of large structures is enhanced by simultaneous excitation with several coherently phased shakers (Ref. 1). However, many factors have precluded a satisfactory, systematic procedure for multi-point excitation modal testing. The two most significant problems have been 1) the lack of efficient and reliable criterion for identifying modal purity and 2) the absence of a method for determining the proper force distribution to excite a pure modal response. Theoretical data-analysis procedures capable of overcoming these obstacles have existed but gone unused because of stringent data acquisition/processing requirements. The development of MODALAB permits the practical application of many of these techniques. Two of these are of particular importance.

A method first proposed by Asher (Ref. 4), but not previously used in a practical application, is used to identify resonances in high modal density situations and to determine the appropriate force distributions to tune the associated modes. This technique uses the data acquired during a series of wideband, single-shaker sinusoidal sweeps. These data are assembled into a coincident-response matrix, the determinant of which is examined as a function of frequency. Those frequencies for which this detriminant vanishes are resonant frequencies and the solution of the corresponding set of homogeneous equations yields the force distributions required to tune the respective modes. Modal tuning by this method is a purely objective procedure.

The energy admittance method of Smith and Woods (Ref. 5) provides a single concise measure of modal response for an entire structure. The total-energy admittance is defined as the complex ratio of kinetic energy of the entire structure to the power input by shakers. This quantity is analogous to the complex displacement/force admittance first discussed by Kennedy and Pancu (Ref. 3). Consequently, if the real and imaginary components of the total-energy admittance are plotted against each other, the locus of data points will, for a well-tuned mode, form a circle which is centered on the imaginary axis and tangent to the real axis. Deviations from this pattern indicate a lack of modal purity. The total-energy admittance is used to determine the modal frequency and damping based on overall structure behavior. Individual-energy admittance data provides the basis for accurate determination of the mode shape.

The analytical techniques implemented in MODALAB depend heavily on the computing power of the system. In addition to the basic software function of processing data, MODALAB has the hardware required to permit software control of many data acquisition and system checkout tasks. The computer is a modern, medium-sized unit with an easily used disk operating system. Programs are constructed on a modular basis and coded almost entirely in FORTRAN. The software system provides exceptional flexibility and ease of modification ideally suited for structural dynamic testing.

The test-support programs can be classified into four categories: (i) header assembly r tines to establish test parameters and to allocate disk storage space for data; (ii) real-time processors to acquire data and concurrently display selected parameters on a graphics terminal; (iii) post-processors to edit and reduce data to final form; and (iv) equipment setup/checkout routines. These programs will be described in the presentation.

The steps in a modal survey conducted with MODALAB are:

1. A series of wide-band, single-shaker, sinusoidal sweeps.

2. Resonance determination and force-distribution analysis.

3. Interactive refinement of modal tuning.

4. Decay test.

5. A narrow-band, multiple-shaker sinusoidal sweep.

6. Immediate postprocessing to determine modal properties and to critique results.

7. Momentum/orthogonality check.

The procedure is almost entirely objective and results in a very rapid test with modal analysis completed within one hour of starting the narrow-band sweep. Average testing time is about 2-1/2 hours per mode, including all phases of the test in frequency ranges of moderate modal density.

The presentation will include data measured on a complex structure having high modal density.

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A METHOD FOR DETERMINING TACTICAL MISSILE JOINT COMPLIANCES FROM DYNAMIC TEST DATA

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The modular configuration approach commonly employed in designing tactical missiles results in the use of numerous separable airframe joints. The compliance of these joints, and in particular the flexural compliances, play an important role in missile structural dynamic response characteristics. Exploitation of the modular missile design concept often results in the need to determine joint compliances for subsequent dynamic response modeling for new module combinations or configurations. However, the behavior and complexity of most airframe joint designs is such that they are not amenable to simple analytical modeling techniques. Experimental determination of joint compliances is therefore commonly used. Two experimental approaches to determining joint compliances are discussed.

The first method of determining joint compliances involves elastic mode testing of special hardware that embodies one type of joint at a time. The design of the experiment is discussed from the standpoint of placement of the joint, control of boundary conditions and minimization of experimental errors. The application of this technique to three different joint designs is presented. The three designs are a threaded coupling ring, a marmon clamp and a radial shear bolt joint. The advantage of this experimental approach is that it involves only a single unknown joint compliance that can be readily deduced with the aid of a beam elastic model. The major limitation of this method is that it requires fabrication and test of a set of special hardware for each different joint.

The second method of experimentally determining airframe joint compliances involves elastic mode testing of the entire tactical missile. This circumvents the problem of designing and testing special hardware but it leaves one with the problem of extracting the airframe joint compliances from the measured mode shapes and frequencies. This can be a formidable undertaking when using trial and error procedures involving the variation of joint compliances in a beam elastic mode model of the test configuration. To simplify the task of determining compliances from elastic mode test data, a method referred to as the joint compliance extraction technique has been developed. This technique is based on the premise that the elastic mode characteristics of a tactical missile can be represented adequately with a beam model and that the mass distribution and, with the exception of the joints, the stiffness distribution are known. Thus any errors between the measured and computed mode shapes and measured and computed mode frequencies are attributed to the compliance of the joints.

The technique is formulated as a non-linear programming problem involving a cost or penalty function that is to be minimized. The cost function consists of the sum of weighted squares of the differences between measured and computed mode shapes and frequencies. The technique is derived from a general method developed by Hall, Calkin and Sholar.[1]

The minimization of the cost function is achieved by a modified steepest descent method in which the second order terms are approximated by differences. This and other features designed to insure and speed convergence are presented. The joint compliance extraction technique has been coded in FORTRAN and applied to actual tactical missile elastic mode test data. Four flexural joint compliances were extracted from a set of three measured mode shapes and frequencies. The results of this application agree well with earlier results that were obtained by a laborious trial and error procedure.

[1] Hall, B.M., Calkin, E.D. and Sholar, M.S., "Linear Estimation of Structural Parameters from Dynamic Test Data", AIAA/ASME 11th Structures, Structural Dynamics, and Materials Conference, Denver, Colorado, April 22-24, 1970.

BLAST AND IMPULSIVE LOADING

DESIGN STUDY OF A NEW BRL EXPERIMENTAL BLAST CHAMBER

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This paper reports the results of a design study of a blast chamber intended for repetitive firings of explosive charges up to 30 lb weight with minimal disturbance to personnel in its vicinity. The study includes a survey of past work in blast chamber design, evaluation of several alternate design concepts and analyses to establish chamber material, spherical or cylindrical shape, size and thickness. Responses to initial and reflected blast waves were predicted, as well as stresses from internal static pressures. Effects of chamber evacuation on modification of loading and chamber stresses were considered. Other factors considered were chamber venting for unevacuated chambers, responses to charges detonated off-center, lining with shock-absorbing materials, and spalling effects.

The chamber design was found to be controlled by the initial and reflected blast loading, rather than static pressure. Partial evaluation to about 1/3 atmosphere markedly reduced blast loading and resultant chamber stresses, while off-center charge detonation increased stresses considerably. Reflected shocks were nearly in phase with chamber natural periods and therefore amplified stresses caused by initial blast loading. Stresses from static pressure rise were insignificant compared to those from blast loading, and no venting was necessary. Recommended construction material was mild steel, and recommended geometry was spherical. A steel vessel required no shock-absorbing lining and could not be spalled by any explosion other than a contact explosion. Chamber thicknesses were determined as a function of radius, and combinations for safe design were recommended.

X-RAY SIMULATION WITH LIGHT-INITIATED EXPLOSIVE

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When a weapon structure is exposed to X rays, the energy deposited in the surface material may result in its vaporization and subsequent blowoff. This blowoff imparts an impulse to the structure within a few microseconds. To furnish information about structural behavior caused by this loading, previous laboratory experiments have employed electron-beam machines, magnetically accelerated flyer plates, and strip explosives to simulate blowoff impulse. In the method described here, the weapon structure is spray-painted with a coating of silver acetylide-silver nitrate (SASN), a light-sensitive explosive. This explosive combines initiation sensitivity and loading simultaneity with the ability to detonate in thin layers. When detonated by an intense flash of light, the explosive delivers a pressure

load that simulates X-ray impulse effects on a test structure. This technique is particularly useful for structures with irregular surfaces and is employed with other simulation techniques and with underground experiments to study weapon vulnerability.

This paper will discuss the facility used for spray-painting SASN. The explosive formulation process as well as the spray techniques will be presented. A discussion of the explosive behavior is included.

The results of a test on a stainless-steel ring are presented. Strain responses are compared with predicted responses from a well-known (Humphrey-Winter) theory.

This testing technique is currently being used to conduct impulse tests on various types of structures including reentry vehicle nosetips and aft ends.

STRUCTURAL DYNAMIC RESPONSE TO HEIGHT-OF-BURST AIR BLAST LOADING

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With advances in the design of nuclear hardened weapon systems, it has become increasingly more important to accurately assess the hardness of protective structures to air blast loads. The traditional technique[1] uses equivalent triangular pressure-time histories to represent the air blast which introduces a degree of approximation which can result in hardness overestimates or underestimates, depending on the method for determining the load duration time. This paper describes the development of structural dynamic response factors for air blast loading which considers the complete pressure history. The loading is based on the most recent advances in air blast phenomenology[2] which provides a calculational procedure for the determination of the waveform shape as a function of weapon yield, peak pressure and detonation height-of-burst.

The analysis modeled the structure as a single degree of freedom system with an elastic-perfectly plastic spring and no viscous damping. In developing dynamic response charts, a series of code calculations were performed to determine the maximum deflection for various values of frequency and structure yield resistance force for each of a series of air blast waveforms. For hardened structure frequencies of interest, the calculation results essentially collapsed into one set of parametric curves which relate the following key variables: structural frequency, weapon yield, pressure waveform time scale, allowable ductility ratio, and dynamic load factor. These curves provide a technique for hardness evaluation which is no more complex than using the dynamic load chart for triangular pulses.[1]

A comparison of these waveform results with those for the triangular pulse with the same peak pressure shows that the use of equivalent triangles which preserve the initial slope of the pressure decay produces an unconservative hardness evaluation. On the other hand, preserving the total specific impulse with a triangular pulse produces results which are too conservative.

1. Newark, N. M., *et. al.* "Air Force Design Manual," AFSWC-TDR-62-138, December 1962.
2. Brode, H. L., "Height-of-Burst Effects at High Overpressures," DASA 2506, July 1970.

It is recommended that the technique based on the use of the actual pressure waveform be used for nuclear hardness assessment of protective structures.

RESPONSE OF FLAT PLATES SUBJECTED TO MILD IMPULSIVE LOADINGS

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This paper presents the results of an effort to determine failure of flat plates subjected to mild impulsive loadings generated by a fuel air explosive. Mechanisms by which flat plates deform under impulsive loads are usually divided into the two areas, i.e., a static mode in which the plate deforms in a continuous pattern involving the entire plate and a plastic hinge mode in which the deformation takes place in a discontinuous pattern by travelling plastic hinges. The static mode is associated with quasistatic loads of long duration and low pressure whereas the travelling plastic hinge mode is associated with reasonably high pressures of short duration. The dividing region between these two modes is usually defined in terms of the characteristic period of the structure and the positive pressure phase of the loading function. Plates whose characteristic periods are approximately equal to or greater than four times the positive pressure phase of the loading are dependent on the shape of the pressure time curve and not just the total impulse. The positive pressure phase durations resulting from a reflected detonation wave of a fuel air explosion are in the millisecond range and are approximately equal to or greater than the characteristic periods of many thin flat plates found in metal buildings, aircraft, radar vans, and other related equipment.

In the case of ductile metals such as some aluminum alloys, mild steel and some stainless steels, the elastic portion of the stress-strain curve is quite small and the assumption of a rigid-strain-hardening constitutive relation appears to be quite realistic. A membrane model based on a total plastic strain energy function was developed using this type of constitutive relation and an assumed final deformed shape. Using a forcing function based on a measured pressure time history for a fuel air explosion, equations of motion for a flat plate were formulated and the resulting non-linear differential equation was solved using an analog simulation program.

Impulsive loads from a fuel air explosion were applied to square aluminum panels using a gas bag technique. The panels were mounted on a rigid frame at one end of a 4′ x 4′ x 20′ plastic bag and a detonation wave was generated at the other end of the bag by use of a small explosive charge. Both side on and reflected pressures were recorded.

Plate failure began at the midpoint of one of the edges of the plate and progressed along the edges of the clamped plate. Examination of the failed surface reveals a simple tensile type failure. The deformation mode was established by observing dynamic Moire' fringes using a parallel lined grid and high speed photography. Plate deformation was found to occur by propagation of a plastic hinge, however, the center point deflection at failure agreed very well with the analytical predictions and for the plates tested was found to be independent of thickness as indicated by the membrane assumption.

A MATRIX STRUCTURAL DYNAMIC MODEL OF PARACHUTE THERMAL COVER EJECTION BY PYROTECHNIC DEVICES

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A method was developed for predicting ejected velocities and structural loadings of a parachute thermal cover ejected by four cartridge type pyrotechnic devices.

Observation of ground test firings of the thermal cover assemblies indicated velocity losses were as great as 35 to 40 percent of the velocities obtainable in test firings of the pyrotechnic charge adaptors ejecting rigid bodies of equal weight. It was hypothesized that these velocity losses were attributable to portions of the available energy from the charge adaptors being lost to thermal saver structural deformation, acceleration of the re-entry vehicle forebody, local deformation or springing of the charge adaptor-forebody mount ring, and parachute bridle extraction and other frictional or drag losses. It was also observed that non-simultaneous firing of the charge adaptors caused velocity losses that increased in severity as the time of non-simultaneity increased. Attempts to assign percentage values to the velocity losses attributed to each of these parameters by empirical method failed to yield accurate pretest predictions of thermal cover ejection performance. A second hypothesis was that these parameters affected performance not only by utilizing available energy, but also by reducing the total energy produced by the pyrotechnic devices during the ejection event, by decreasing the inertial load on the ejector pistons through structural response of the thermal cover.

To test the latter hypothesis, and to provide a means of obtaining improved predictions of ejection performance, a computer program was written in Fortran IV utilizing a matrix structural dynamic model of the thermal cover and ejection system which included all of the parameters cited. A unique problem in the development of the dynamic model was the simulation of the pyrotechnic forcing function, since it was necessary that the model predict the effect of the inertial loading of the ejector piston on the pressure time history produced by the charge adaptor. The overall problem was therefore the development and integration of the equations of motion of a system subjected to a highly non-linear function. The methods used, and the results achieved, are described in the following sections.

THERMAL COVER EJECTION EQUATION OF MOTION

The dynamic model is simulated as a 6 axial degree of freedom matrix structural model. Five degrees of freedom are related to the structural model of the thermal cover, and one degree of freedom is assigned to the lumped forebody mass. The five thermal cover degrees of freedom consist of one each assigned to the four corners at piston attachments, and one for the top center of the parachute cover box structure. Each corner is forced by the piston charge adaptor after charge ignition. The charge adaptor forcing function is simulated as described in the following section. Non-simultaneous firing events are accommodated by specifying the firing time of each of the four pistons individually.

Parameters that are input into the computer program are forebody weight, thermal cover weight and stiffness matrices, charge adaptor firing times, charge adaptor to forebody local stiffness, thermal cover structural damping and piston loss damping attributed to the various drag and friction effects. Logic related to each of the force contributors is contained in separate subroutines which facilitates logic changes without major reprogramming.

PISTON CHARGE ADAPTOR FORCING FUNCTION

The piston forcing function is a three-dimensional function in tabular form giving piston force as a function of piston displacement and time since ignition. This was derived from pressure data recorded from rigid body firings of single pistons ejection 1, 2, 3, 5, and 10 lb. weights. The average pressure-time history for each ejected weight was integrated twice with respect to time to obtain piston displacement time histories. The forcing function table is a tabulation of these time histories. For integration of the equations of motion, a table look up and bilinear interpolation scheme is used.

THERMAL COVER STRUCTURAL MODEL

Thermal cover stiffness and mass matrices are included in the ejection velocity prediction model to simulate the effect of structural response on the energy produced by the charge adaptors, and to simulate the energy lost to structural deformation. They also enable the user to obtain structural loadings of the thermal cover during ejection.

The thermal cover structural model is formulated as a multi-degree of freedom freedom system which is reduced to a five degree of freedom system using the Guyan transformation. Stiffness and mass matrices are refined to obtain eigenvalue-eigenvector solutions which are calibrated to agree with vibration test data.

CHARGE ADAPTOR TO FOREBODY LOCAL STIFFNESS

Charge adaptor to forebody local stiffness is included to simulate the structural response of the mounting due to charge adaptor firing forces. Measurements of local stiffness were obtained from impedance vibration testing.

MOVING FOREBODY

The moving forebody is included to simulate the loss of energy to accelerating the forebody, and the effect on total energy produced. The forebody is modelled as a lumped mass free to move in its axial degree of freedom under the sum of the forces of the four charge adaptors.

DAMPING

Damping includes all forces which are functions of mass point velocity. Included are structural damping critical damping of the cover center mass point, and losses due to bridle extractions and piston and baffle drag. Drag type damping is modelled as a retarding force on the corner mass points which is proportional to velocity.

VIBRATION TESTING

Vibration tests were conducted on various thermal cover assemblies. The purpose was to obtain data for refinement of the structural model for better simulation of response during ejection.

RESULTS

The simulation of the piston forcing function was verified when the model demonstrated the ability to predict ejected velocities of rigid body single piston firings of various ejected weights.

Initial parametric studies of thermal cover structural characteristics utilizing the dynamic model proved the hypothesis that structural response affected the total energy generated by the charge adaptors. After calibration and refinement of the structural model, the dynamic model demonstrated the ability to make pretest predictions of velocity accurate to 5 percent.

Extensive parametric studies of thermal cover design has lead to a major redesign of the entire ejection system. The modelling technique and program have proven to be useful accurate tools for this effort.

EFFECT OF SHOCK SPECTRUM-SIMILAR SHOCKS ON A LARGE COLLECTION OF COMPLEX STRUCTURES

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The problem of two shock tests involving shocks which have different time histories but essentially the same shock spectrum is considered. This problem invariably arises in any discussion dealing with a shock test specification stated in terms of a shock spectrum requirement. A technique, based on the effect of such spectrum-similar shocks on a class of ideal systems more general than those involved in the definition of a shock spectrum, is employed.

A set of more than 6000 ideal, four-degree-of-freedom, linear oscillators are considered. The maximum response induced in these systems by a classic half-sine acceleration shock, and by a spectrum-similar complex shock are compared. The complex shock is made up of sinusoidal acceleration components whose envelopes first grow and then die out exponentially. The effect of system damping is examined. The half-sine shock is characteristic of a drop type tester, the complex shock of an electrodynamic shaker type tester.

The results are presented in terms of distributions of maximum response differences. These are compared with a set of distributions based on shock spectrum bounding techniques developed previously by the authors. Expected pass/fail statistics for idealized shock tests performed with the two spectrum-similar shocks are discussed.

STRUCTUREBORNE GUN BLAST SHOCK TEST USING AN ELECTROHYDRAULIC VIBRATION EXCITER

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The paper describes a method of simulating the dynamic effects of a specified structureborne gun shock. The shock test levels were derived by others and the validity of the levels are not reviewed, except that the levels are defined as being applicable to equipment mounting points which are not exposed directly to the air blast wave. The tests were conducted to determine the effect on the operation of sophisticated electronic equipment during and after a series of shocks. The equipment under test was designed for MIL-S-901 shock environment. No physical damage was expected from the gun shock test and none has been observed to date.

The shock requirement was specified in terms of velocity and acceleration levels. The requirements were interpreted as a nominal shock response spectra and reasonable tolerance limits were assigned. The tolerance limits were adjusted to allow the use of a shock pulse having zero average velocity and approximately one half sine waveform. The choice of waveform was made to simplify synthesis and any of a variety of other waveforms would yield an "in-tolerance" response spectra.

The nominal horizontal response spectra is

Frequency, Hz	Nominal Level	Upper Limit	Lower Limit
10-80	9.0 in./sec.	18 in./sec.	-
10-20	-	-	0.05 in.
20-80	-	-	6.0 in./sec.
80-500	12.0 g's	24.0 g's	8.0 g's

Response of frequencies less than 10 Hz was undefined and was nominally the 0.1 inch displacement line. Response of frequencies greater than 500 Hz was uncontrolled and was nominally the peak acceleration 12 g's, but most tests are 10 g's.

An electro-hydraulic vibration exciter was available having performance parameters of:

- peak vector force (average) — 50,000 lbs.
- peak velocity — 17 in./sec.
- frequency response — >250 Hz

Vibration tables are provided for vertical and horizontal motion and restrained on hydrostatic bearings.

A waveform synthesizer and low pass filter was calibrated to give the required response spectra with the test specimen on the table. The output of control and monitor accelerometers were recorded on:

- direct writing oscillograph
- magnetic tape
- storage oscilloscope
- real time response spectrum analyzer (control only)

for control and analysis of the shock pulse. Ten shock pulses were applied in each direction along each of 3 axes (60 total shocks). All pulses were recorded and the first and last were analyzed to ensure uniformity of the shock input.

Results are presented from tests of cabinets and more complex assemblies, including one antenna pedestal assembly. The shock response spectra specified was readily synthesized, calibrated and applied to the test specimens which ranged in weight from 500 to 4,500 lbs. and included specimens mounted on shock isolators.

It is concluded that the test method described here is useful for detecting nuisance malfunctions in complex electronic and electrical equipment when the equipment is subjected to structureborne shocks at levels characteristic of gun firing. The shock levels are at levels of approximately 1/20 of the level of MIL-S-901C and consequently give a measure of circuit fragility of levels roughly equivalent to the response to vibration. Vibration per MIL-STD-167 extends through the frequency range of 4 to 33 Hz and does not yield data relative to the response in the 33 to 150 Hz region of the spectrum, which is strongly excited by structureborne gun shock of the type specified.

MEASUREMENTS OF THE DYNAMIC IMPACT LOADING OF CONCRETE

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As a structural material, concrete is often used as a protective and/or containment medium for impact and blast loadings. For such applications, an assessment of material degradation and damage are important failure criteria to establish. Specifically, failure types associated with material removal and spall characterization are very dependent upon the particular loading conditions. These loadings may be classified as air burst, surface contract, or of the buried blast types. The latter loadings may also be considered static or dynamic in classification depending upon whether the loading charge has been affixed to or dropped onto the specific material medium.

A search of the open literature has evidenced a paucity of data available on the dynamic and impact loading of concrete. Some previously reported studies sealing with the dynamic loading and response of rock, loose and compacted granular media are reported in references [1-3]. Since rock aggregates are brittle in nature, their response is qualitatively similar to concrete. Except for empirical data such as reported on in [4], and laboratory studies conducted in [5, 6], less information on the macroscopic behavior of concrete is available.

Based upon previously reported studies of material elements of finite size and taking into account such effects as charge geometry, orientation, stand off, embedment, and coupling materials, it appears that three distinct loading failure/fracture regions exist for the impacted concrete media. The first is the pulverized or crater region, the second contains crushed aggregate, while the third region consists of numerous macrocracks. The latter region may be classified as a region in which elastic waves can be considered important.

In order to obtain more information relative to the macrocrack region, an experimental investigation of the dynamic response of concrete bars has been conducted. Specifically, a series of long bar test specimens, approximately eighteen inches in length, have been fabricated and instrumented for impact type studies. The concrete aggregate specimens have been impacted at stress levels well below the compressive failure strength of the specimens by bars of varying pulse length in order to investigate the extent and nature of the tensile fracture phenomenon occurring within the specimens. The bars have been instrumented with surface strain gauges positioned so that the strain pulse propagation characteristics in the various bar specimens can be observed. Pulse shape change, pulse wave speed characteristics, and amplitude attenuation due to the impact loading have been catalogued.

In addition to the above studies, changes in the particle aggregate size of the concrete mixture have been examined for potential scaling purposes. Such information should provide a basis for predicting the failure/fracture boundaries of concrete specimens subjected to impulsive type loads including both the spall and corner fracture phenomena.

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VIBRO-ACOUSTICS

DYNAMIC STRAIN MEASUREMENT TECHNIQUES AT ELEVATED TEMPERATURES

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The measurement of strain during sonic fatigue tests of aircraft structures is essential to the determination of structural response and the estimation of fatigue life. Of the many tests that have been performed at the Air Force Flight Dynamics Laboratory (AFFDL) Sonic Fatigue Facility, those that presented a difficult strain measurement situation were the tests run in conjunction with structural heating. It is in such an environment that high thermal stresses and the dynamic strains of acoustically induced vibration combine to reduce the life of the most carefully chosen and installed strain gage system. In a typical high temperature acoustic test in the AFFDL, the test article is subjected to a high intensity sound field and simultaneously heated to its operational temperature. Many of these tests have been on aircraft structures that in service are exposed to the noise and heat of jet engine exhaust. In the test facility, the sound field is simulated using sound generators and heat is induced by means of reflective quartz lamps.

The objective of the program was to develop and demonstrate strain and temperature measurement techniques for strain levels approaching 1000 micro inches/inch and temperatures up to 1400°F. The reliability requirement was specified as one hour of operational life at these limits of thermal and dynamic loading. Three temperature ranges were selected for investigation, 200°F to 500°F, 500°F to 800°F, and 800°F to 1400°F. For the temperature range, 200°F to 500°F, commercially available gages were selected. For higher temperatures, many special alloyed gages were fabricated primarily to meet static strain requirements of the program. Gage attachment methods include organic cements, ceramic cements, and flame sprayed ceramics. Leadwire materials included Nichrome V wire, Nichrome V ribbon, constantan wire, Nickel clad copper wire, and stainless clad copper wire. The assumption was made that strain gage thermocouple installation techniques used in testing gas turbine engines should also work in a high temperature, high intensity acoustic noise environment. This was verified early in the program in a test conducted in the Sonic Fatigue Facility.

Three panels were prepared for testing in the progressive wave test section of the Small Facility. The 1200°F panel consisted of a 14 by 20 in 0.023 in thick sheet of Inconel X alloy. Platinum, 8% tungsten and Nichrome V alloy gages were installed using ceramic cement. High intensity sound was generated by a low-frequency pure tone siren at a sound pressure level (SPL) of 160 dB. Thermocouple readings were used to control the output of the quartz lamp heaters.

The 1400°F panel, to be tested next, was cut from a sheet of 0.025 Rene 41. Dynamic strain gages of platinum, 8% tungsten, Karma, and Nichrome V alloy were installed using the flame spray technique. For this test the sound source was changed over to an air modulator type capable of producing a wideband spectrum.

The lower temperature measuring methods were tested on the 500°F panel, a 0.020 thick sheet of 321 stainless. Glass-fiber reinforced epoxy and phenolic backed gages were installed using M-Bond 610 adhesive.

An analysis of the test results pointed out gaging techniques that should be considered when testing in a high temperature and intense acoustic environment. The ceramic cement and flame spray installed gages on the 500°F and 1200°F test panels had failed because the strains had exceeded the limits for these materials. Ceramic cement are suitable for high temperature acoustic tests if the static strains do not approach the yield point of the base material. Flame sprayed ceramic is also highly suitable and should be used where the strain may exceed the yield point. Analysis of the 500°F panel indicated that all of the epoxy adhesive bonded gages were functional and all failures had occured in the lead wire system.

AN ACTIVE LINEAR BRIDGE FOR STRAIN MEASUREMENT

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The Air Force Flight Dynamics Laboratory, Sonic Fatigue Facility, located at Wright-Patterson Air Force Base, operates three acoustic test chambers for experimentation on flight vehicles, missiles, and equipment in a high intensity sound environment.

Effects of the high intensity sound on the test specimen are determined through the use of transducers on or near the specimen and connected to the data acquisition system. The data acquired from most tests consists of the outputs of microphones, strain gages, accelerometers and displacement transducers. Of particular importance in most experiments and tests are the stress levels and location thereof in the test specimens. As a consequence many strain measuring channels are provided in the AFFDL Sonic Fatigue Facility data acquisition system.

These strain measuring channels consist of constant voltage strain gage bridge modules of standard configuration. Almost 100% of the strain measurements made in the facility use the quarter-bridge circuit. This particular bridge circuit is one of the most commonly used circuits in strain measurement work. Although other bridge circuits are more advantageous under certain conditions in regard to linearity and sensitivity, they are usually more difficult and often impossible to implement on real structures. This is due to non-accessibility to the reverse side of a structure caused by a brace or support member covering the area of interest. For these and other reasons the quarter-bridge configuration is the only one discussed here.

The output from the strain bridge is a function of the bridge excitation voltage, gage factor, strain amplitude and the condition of balance. The bridge output equation is composed of two terms, one for the active leg and one for the balancing leg. Examination of the two terms of the equation reveals the active term to be a non-linear function of the active element or ΔR and the second term to be a constant. This shows that the bridge is non-linear regardless of the state of d-c balance. It is also pointed out that in a bridge circuit of this kind there can be two types of balance; a strictly d-c balance that only eliminates the d-c offset and nothing more, and what might be called sensitivity balance.

It is this latter type balance that is of primary interest to those whose principal concern is dynamic measurement. Most bridge designs do not provide for maintaining this type of balance. Sensitivity balance usually is taken to be the condition where the two elements of the active leg are equal but more specifically it is the ratio of these two elements at the time calibration was performed which should not change. This equality or ratio is only established or maintained at the initial set up time and not, of course, during the strain measurement.

The testing of specimens at elevated temperatures to 1400°F is often performed at the Sonic Fatigue Facility. Such high temperature causes large changes in the strain gage static resistance and accompanying sensitivity balance. This results in undesirable changes in bridge calibration. An auto-balancing circuit was designed in an attempt to solve the problem of sensitivity unbalance and restore room temperature calibration. This circuit successfully eliminated the d-c offset problem resulting from high temperature operation but the sensitivity unbalance problem remained.

The sensitivity unbalance problem was then corrected by a modification of the auto-balance bridge circuit which made the output a linear function of ΔR or strain. This circuit uses one leg composed of two resistors in a divider for one input to a differential operational amplifier and a three element leg of which the center element is the strain gage. Feedback input to the amplifier is then taken from the junction of the middle and bottom elements. This circuit is analyzed to show that a constant current in maintained in the gage element to a high degree of accuracy. An additional feature of this circuit is, that by properly choosing the values and ratios of the bridge elements and/or excitation voltages, the output at zero strain can be made zero, which allows the circuit to be used for static as well as dynamic strain measurements.

LABORATORY VIBROACOUSTIC PROFILE TESTING FOR IMPROVED AVIONICS EQUIPMENT RELIABILITY IN A GROUND LAUNCHED RPV

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Modification of an existing operational air launched RPV (Remotely Piloted Vehicle) to obtain a rocket assisted ground launch capability required reasonable pre-launch assurance that critical on-board flight control subsystems would continue to function normally during the severe launch vibroacoustic environment.

The ground launch environment was analytically predicted and a novel testing technique was devised which allowed the on-board avionics equipment to be "flown" in the laboratory through this severe environment. This unique approach permitted the test item to be exposed to its predicted operational environmental profile during which the operational performance of the test item was checked. The use of realistic environmental test levels provided reasonable assurance of satisfactory flight test performance without unnecessarily stressing the test item which was scheduled for use in the flight test program.

Minor failures which occurred during laboratory testing were identified and repaired; thus, giving increased assurance of environmental reliability during launch. Included in this paper are discussions of the predicted technique, test philosophy, testing method, and test results.

ANALYSIS AND FLIGHT TEST CORRELATION OF VIBROACOUSTIC ENVIRONMENTS ON A REMOTELY PILOTED VEHICLE

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The requirement to specify environmental criteria for the design and testing of airborne equipment installed in a new aircraft has provided considerable challenge to environmental engineers. To satisfy this requirement in terms of a realistic set of specifications calls for the availability of extensive flight test data encompassing the full range of service conditions encountered by the equipment. Whenever this is not possible, environmental flight data from a similar vehicle are often used for extrapolation to the new design. In both cases, however, the requirement for an early or parallel effort to analytically predict the expected service environments and the corresponding equipment response levels is essential to the proper establishment of the objectives specifications.

This paper presents the results of a comprehensive analysis and flight test program conducted to evaluate the vibroacoustic environment on the AQM-34L model jet-powered Remotely Piloted Vehicle (RPV). The reported activity was performed by Northrop Corporation, Electronics Division, under contract to the USAF Aeronautical Systems Division for the specification of vibration, thermal, and climatic environments for airborne equipment installed in a general class of air-launched RPV.

The primary source of structural vibration on a jet-powered flight vehicle is the acoustic noise field generated by the engine exhaust and the aero-dynamic boundary layer surrounding the vehicle surfaces. Another important source of acoustic excitation for an air-launched RPV such as the AQM-34L is the power plant noise of the carrier aircraft. The vehicle primary structure response to these types of excitation is expressed in broadband PSD levels and are used as a basis for establishing the environmental vibration specifications for equipment payloads.

The analytical prediction of the induced vibration levels was accomplished by partitioning the vehicle fuselage stations into representative equipment zones where the external acoustic excitation remains essentially constant. The corresponding octave band sound pressure level for each zone was calculated on the basis of the worst case flight condition. These levels were subsequently converted into wideband acceleration spectral density data by an empirical procedure devised by P. T. Mahaffey and K. W. Smith of Convair/Fort Worth.

The analysis of the equipment response levels within the various zones was performed as if each equipment and its associated mounting structure were driven by the vehicle primary structure at a set of fixed boundary points. Such a procedure, while allowing for the motion of the secondary structure and the mounted equipment to be accurately described by analysis, does not account for the dynamic coupling which exists between the vehicle structure and the payload equipment weights. The resulting prediction is therefore analogous to the solution of a linear, multi-degree freedom system subjected to a prescribed base motion. This type of analysis represents a baseline procedure directed at evaluating the vibration levels induced at the various equipment locations while providing the necessary flexibility for accommodating future modifications of the equipment payloads. The required vibration transmissibility data for the specified equipment/structure subassemblies were obtained by constructing the appropriate finite-element dynamic models which were analyzed by means of proprietary computer program. The viscous damping constant for each dynamic model was estimated to vary between 4% to 10% of critical. The equipment response levels were obtained by multiplying the primary structure PSD's by the square of the transmissibility along each of the three orthogonal axes and enveloping the resulting curves in the form of tight straight line PSD's.

The scope of the flight test measurement program was planned to reflect both normal as well as extreme operational environments of the RPV and its equipment payloads. These included the ground run-up to the RPV turbojet engine; takeoff, and landing in the captive mode; and cruise at various altitudes in the free-flight mode. Vibroacoustic environmental data were recorded during each of these operational modes by flush-mounted microphones and 29 accelerometers. Location of certain accelerometers were selected to coincide identically with the equipment locations considered in the analysis to facilitate direct comparison of the results. A real-time spectrum analyzer which incorporated a 6.4 Hz constant bandwidth filter was used for data reduction.

The satisfactory correlation between predicted and measured sound pressure levels on the RPV confirmed the assumptions on the primary sources of acoustic excitation and their respective intensities at the various fuselage stations. Thus the prediction technique used in the analysis provides a valid basis to estimate the acoustic levels on a general class of jet-powered RPV.

The primary structure vibration levels derived from the sound pressure levels using the Mahaffey-Smith 90% confidence data compared favorably with the measured levels except at the low and high frequency extremes of the PSD envelopes where discrepancy ranged from +10 db to -10 db, respectively. This result demonstrated the need to introduce a correction term to the computed PSD levels the using Mahaffey-Smith approach to achieve better correlation between the predicted and measured data.

Flight test correlation of equipment vibration levels proved to be considerably more difficult to obtain on an individual equipment basis due to differences in the physical location of the vibration transducers as well as limitations on the validity of the preliminary analysis results. It was therefore concluded that some of the flight test data could be incorporated into an updated version of the equipment dynamic models so that a new set of analyses could be performed. This was accomplished and the results demonstrated an improved correlation between the equipment PSD's and the measured levels. These were subsequently used in the specification of the equipment vibration environments on the RPV.

It is concluded on the basis of the foregoing correlations that analytical prediction methods can be utilized satisfactorily to establish baseline environmental criteria for future

RPV and other flight vehicles. The opportunity to verify the predicted vibroacoustic environments on th AQM-34L vehicle through a successful flight test program justifies this conclusion.

AEROACOUSTIC ENVIRONMENT OF RECTANGULAR CAVITIES WITH LENGTH TO DEPTH RATIOS IN THE RANGE OF FOUR TO SEVEN

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Ever increasing flight conditions produce high dynamic loadings on both the aircraft and its internally carried stores. The intense oscillating pressure environment encountered in open weapons bay cavities have resulted in severe vibration and buffeting problems during subsonic, transonic, and supersonic speeds. The oscillatory loads on the store, which are a result of free stream flow over the open cavity, may result in weapon breakage, while at the same time produce initial loads on the stores prior to separation that causes uncertain weapons trajectories.

This paper describes a series of flight tests in which the aeroacoustic environment of five different cavity configurations was determined. The configurations tested included: three empty cavities with length-to-depth (L/D) ratios of 4, 5, 7; one cavity with a L/D ratio of 7 with an ogive store symmetrically mounted in it; and one closed cavity configuration. The cavities were mounted in a modified munitions dispenser pod which was flown on a RF-4C test aircraft. The test altitudes were 3,000 ft, 20,000 ft, and 30,000 ft and the Mach number ranges were 0.61–0.93 for the 3,000 ft altitude and 0.61-1.30 for 20,000 ft and 30,000 ft. Data were recorded continuously as the aircraft slowly accelerated from the lowest to the highest Mach number.

The cavities were instrumented with nine microphones, three static pressure ports, one accelerometer, and one thermocouple. The aeroacoustic pressure environment in the cavities was defined from these measurements, and typical results are presented. The results were correlated with empirical wind tunnel predictions and formed the basis for a modified prediction method. The prediction scheme includes methods to predict the first three resonant frequencies, the fluctuating pressure amplitudes of these frequencies, and the one-third octave broadband spectrum. These are predicted as a function of L/D ratio, longitudinal cavity location, free stream Mach number, and free stream dynamic pressure. The prediction scheme can be applied to nearly rectangular cavities with L/D ratios in the range of 4 to 7 and for a Mach number of 0.6 to 1.3.

Also included in the paper is an example illustrating the application of the prediction scheme. The result of the example are compared with results obtained from a prediction scheme presented in the literature.

ACOUSTICALLY INDUCED VIBRATION PREDICTION IN TRANSPORT AIRCRAFT

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In transport category aircraft, vibration may be induced by a variety of independent causes, each sufficiently different to require separate consideration in the vibration prediction process. The success of the analyst in predicting the complete vibration environment will, in large part, hinge on the quality of the methods, data, and information he can assemble to account for these various independent sources. This investigation was conducted in response to that need for additional and improved methods. The objective was to provide a method for estimating acoustically induced structural vibration in jet-powered transport category aircraft. A number of magnetic tape recordings of noise and vibration level which were made recently on two contemporary military transport aircraft were reassembled for use in this investigation, and processed in a more sophisticated and automated fashion than accomplished previously. Additionally, a total of 30 flat and curved structural panels were laboratory tested to obtain recordings of noise and vibration data. The panel test data and the transport aircraft data formed the basis for this work, and the results therefrom. The familiar noise-vibration correlation plot was a primary investigative tool in this effort. The measured data were available in sufficient quantity and quality to allow correlation of noise and vibration levels in third-octave frequency bands, with confidence lines statistically fitted to the data. These correlations were established with computerized sorting and plotting techniques for numerous ensembles of data. each having some common trait. The vibration data in these correlations were then operated upon in various ways and recorrelated to obtain indications of: (a) the relative response in three different axes or directions, (b) the relative difference between on-ground operation and pressurized cruise flight, and (c) the effect of structural mass and rigidity on vibration level.

It was found that a given noise level excites consistently higher vibration levels in laboratory test panels than it does in aircraft structure of similar mass, rigidity, and configuration. Laboratory test panels are suitable for exploring the relative influence of structural and environmental parameters, but should not be used to predict aircraft vibration level. Until further work has been done to define the effect of various factors peculiar to aircraft structure and its environment, only aircraft noise and vibration data should be used as a basis for vibration prediction. Vibration levels on the frames and stringers of shell type structure were found to be of similar magnitude in the normal and lateral directions. In some frequency bands, normal levels are slightly higher, while in other bands lateral levels are higher. Vibration in the tangential direction is consistently lower. The ratio of lateral-to-normal and tangential-to-normal vibration level was determined statistically and found to vary also with sound pressure level. Vibration in the pressurized fuselage was found to be generally lower, at any given noise level, during pressurized cruise flight than during engine run-up on the ground. The ratio of flight-to-ground vibration level was determined statistically and found to vary with confidence level. At high confidence levels this trend reverses in the high frequency bands, where vibration during pressurized cruise flight is higher. An empirical expression for mass and rigidity effects was determined, which showed the effects to be offsetting. The mass and rigidity of the structure used herein is thought to be typical of most conventional aircraft structure, whereby the effects of mass and rigidity on vibration would typically be small. Exceptions are not uncommon, however, and instances are conceivable where mass and rigidity could deviate greatly from the trend found here (e. g., in high strength-to-weight structure such as honeycomb and composities, and in low strength-to-weight structure such as used in high temperature environments). In such instances the effect of mass and rigidity on vibration level could be substantial. The data showed considerable scatter which the normalizing equation, being limited to mass and rigidity effects, does not detect. Much of this scatter is thought attributable to variation in other factors such as damping, accelerometer positioning on the structure, unconventional vibratory response, spatial correlation and autocorrelation of sound pressures, and sound incidence. The outcome of this work was an assembly of charts and graphs for use in predicting acoustically induced structural vibration in transport aircraft.

SIMPLIFIED TECHNIQUES FOR PREDICTING VIBRO-ACOUSTIC ENVIRONMENTS

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Research in seeking practical techniques or methods for predicting vibration environments of flight structures subjected to acoustic environments has gained considerable attention in recent years. In this effort, mechanical impedance techniques were adapted to tailor dynamic tests of highly critical and expensive components to ensure a highly accurate control of response loads.

A method, based on Thevenins' and Norton's theorems, has been developed to predict broad frequency range vibration criteria which account for both primary and component load impedance for structures excited by random acoustic excitations. The force responses were predicted by a one-dimensional equation which utilizes four types of parameters at equipment mounting locations. These parameters consist of input impedance of support structure, acoustic mobility of structural system, input impedance of component package, and blocked pressure spectrum. The accuracy of the equation in predicting force responses has been verified satisfactorily and the method is currently being used for force-control environments of complex structures.

However, exact analytical computation of these above four parameters are not practical for performing quick estimates with adequate accuracies. In order to simplify computational procedures of this method and to gain a wider application by engineers, a series of design charts and nomograms was developed to eliminate the need for computer requirements and to provide adequate information with acceptable tolerance limits. This is accomplished by the use of simplified equations derived from existing analytical solutions. The computation of structural impedances is evaluated in three different frequency ranges, namely: low frequency range or frequencies below the fundamental frequency of the shell, intermediate frequency range, and high frequency range or frequencies above the ring frequency of the shell. The resulting impedance equation generally contains a frequency dependent and a frequency independent term. By employing logarithmic summation techniques, the computation and summation of individual impedance terms can be made graphically through nomograms and design charts. Meanwhile, the blocked pressure can be converted from either far-field sound pressure or near-field sound pressure from a design chart. The acoustic mobility term is determined from existing experimental data which is based on the nature of similarity of the structural response. Finally, the summing of all the terms, as defined for the force response computation, is carried out by the use of a logarithmic computational chart.

Two example problems with different structural configurations were used to demonstrate computation procedures. Satisfactory agreements between analytical predictions and experimental measurements were observed.

RESPONSE OF SEMI-PERIODICALLY STIFFENED PANELS TO CONVECTED RANDOM PRESSURE FIELDS

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A lightweight panel stiffened by a set of stiffeners at regular intervals is used in many structural constructions. Such panels are often subjected to various dynamic disturbances, e. g., random pressure fluctuations due to boundary layer turbulence, noise from jet engines, etc. A wave-propagation method applicable to such spatially periodic structures has been developed during the recent years. The purpose of the present work was to study the effect of any deviations from periodicity in stiffener locations on the panel response; in particular, the object was to see if any such non-periodicity in stiffener locations, either by design or due to difficulty (and cost) of fabricating an exactly periodic structure, has any beneficial or adverse effect on the response and sonic fatigue lift of the structure.

For this purpose the existing wave-propagation method has been combined with the transfer-matrix method to generate a new method which retains the desirable features of both and is also free from some of their limitations. This method is applicable to "semi-periodic" structural configurations consisting of repetition of a basis unit which contains sections of unequal length and/or stiffness properties.

In this study, this method has been applied to predict the response of a structure in which the basic unit is made up of two sections of unequal lengths. The excitation considered is band-limited white noise at normal incidence. The response of this semi-periodic structure has then been compared with that of a periodic structure of equal weight.

The low frequency response of a semi-periodic structure is controlled by two modes of the basic two-span unit. When the span lengths are equal (i.e., the structure is periodic), the higher mode is very strongly excited. The lower mode is antisymmetric and therefore it does not contribute to the total response. As the span lengths are changed, the contribution of lower mode increases and that of the higher mode decreases. The total response passes through a minimum for a configuration in which one span length is about 33% shorter than the other (i. e., for a span ratio of about .67). For this configuration, the maximum r.m.s. stress is reduced by about 18%, compared to the response of the periodic structure of equal weight. The sonic fatigue life is correspondingly increased by a factor of three to five. For various other ratios of span lengths, the response is higher than that of the periodic structure.

Using the proposed configuration, therefore, the sonic fatigue life of a stiffened panel can be improved. Alternatively, for a given level of excitation, the skin thickness can be reduced, resulting in total weight saving.

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ISOLATION AND DAMPING

MATRIX METHODS FOR THE ANALYSIS OF ELASTICALLY SUPPORTED ISOLATION SYSTEMS

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It has been shown[1,2,3] that when a rigid body is supported by a number of linear springs and dampers, the resulting six coupled, linear, second order differential equations can be written in matrix notation. With the availability of matrix routines in most digital computer libraries, this has become a popular representation.

To be presented is a scheme whereby the matrices of coefficients in the equations are developed using six-rank square matrices. The advantage of this technique is that instead of computing four or five three-rank submatrices which must be placed in the proper locations of the six-rank matrix, the six-rank matrix is computed directly. This is accomplished by allowing the translational and rotational coordinates to appear on an equal footing. Because torques or rotations do not have to be "added in," but appear automatically in the equations, a greater unity and simplicity is achieved and the meanings of the terms become more transparent.

The basic problem is to transform vectors, e.g. forces, displacements, small rotations; and matrices, e.g. inertia, stiffness or damping, from one point and orientation in space to another. This is accomplished by applying a rotation transformation and then a position transformation. The transformations are performed, as usual, by matrix multiplication, except that in the scheme presented the vectors have six components, three translational and three rotational, and the matrices are all square and of six rank.

The rotational transformation is developed in detail. Rotations about all three axes are included for generality. The inverse of the displacement transformation is derived and shown to transform vectors in the reverse direction. For completeness, the final matrices are written out in detail.

If the isolators themselves are mounted on a resilient support of negligible inertia and described by its own stiffness and damping matrices, a second set of six coupled differential equations must be solved simultaneously with the first set. It is shown that these twelve equations reduce to six equations but are of third order. For steady-state sinusoidal excitation or free vibration the equations of motion are shown to reduce still further to a form identical to that of a rigid support, but with an equivalent stiffness-damping matrix which has elements that are complex numbers.

The solution of the differential equations are discussed for various types of cases. Free vibration results in an eigenvalue equation. It is shown that the eigenvalues and eigenvectors are complex if the damping is not zero. This gives rise to phase differences and exponentional decays of the modes. A special case of a two degree of freedom system with coupling between the rotational and translational modes is considered and is shown to agree with an earlier work.[4]

For forced sine excitation the single degree of freedom system with a resilient support is also solved algebraically and shown to agree with published results.[5] The differences in the transfer matrix for foundation motion and a force applied directly to the mass are compared and shown to be identical as in the case when the matrices are diagonal, that is, when each degree of freedom is independent of the others. The most general case is that of transient excitation. The differential equations can then be solved using the Runga-Cutta numerical integration technique which is usually found in computer program libraries.

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IMPACT ON COMPLEX MECHANICAL STRUCTURES

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The finite element method appears to be ideally suited to the problem of computing the displacements, velocities, and accelerations at selected points in a complex structure subjected to impulsive loading. To apply this method of analysis one must first prepare or select a conceptual representation of the continuous structure as an assemblage of structural elements interconnected at nodal points. The idealized structure is assumed to be acted on by external equivalent forces and to possess equivalent inertial properties only at the nodal points. Thus the continuous structure is replaced for analytical purposes, by a lumped parameter system. The accuracy of the predicted dynamic response of a structure will depend on how well the structure is represented by the selected lumped-parameter model. It has been reported that even for a very simple beam or uniform plate with boundary conditions that can be exactly expressed in mathematics, errors in the predicted responses can easily be as much as 35 percent. Although no investigation of the validity of lumped parameter models for complex structures, such as vehicles, has been reported in the literature, it may be assumed that discrepancies of even more than 35 percent can be expected for poorly represented structures. The general details of the finite element method have been discussed in the literature. However, the nature of the model which will best represent a structure such as a vehicle, with its varied elements, irregular geometry and discrete masses, is not at all clear. The primary objective in this study has been to select a model and then determine by computation and by experiment how valid the model is. A model for representing a specific vehicle, namely the M-37 military truck cushioned for airdrop, is used to illustrate the procedure.

It is found that the response of a structure such as the M-37 and subject to impact loading is not sensitive to the elastic properties of the interconnecting members. Thus the development of a suitable lumped parameter model is simplified. Special attention must be given, however, to the more important components of the vehicle.

Experimental results show that an appreciable portion of the energy at impact is dissipated in structural damping. Hence damping must be included in the equations of motion.

The most essential factor affecting the dynamic response of the system is the input force. Thus this force must be carefully simulated.

ENERGY ABSORPTION AND PHASE EFFECTS IN SHOCK EXCITED COUPLED SYSTEMS

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The analysis of energy absorption and phase-related uncertainties is based on an idealized model consisting of two simple damped resonators, the second resonator being attached to the mass of the first. If the first resonance frequency is allowed to increase beyond limit, the model becomes a single simple resonator, as a special case. The focus of the analysis is on the peak acceleration of the second mass and the energy absorbed in the second damper. To simplify the mathematics and make the model correspond to a critical case for reliability, it is assumed that the second resonator does not load the first, but the conclusions are not limited to this assumption. The analysis is limited to shock pulses (such as the terminal peak sawtooth) whose dominant feature is a terminal step function, so that we can assume for the two-resonator model a residual response consisting of two transients at the natural frequencies of the two resonators, and these transients become the primary cause of energy dissipation and peak response.

It is shown that the energy dissipated in a single simple resonator is simply related to the undamped residual spectrum. In the two-resonator model, both the peak response of the second mass and the energy dissipated in the second damper are simply related to the undamped residual spectrum except for an uncertainty from an unspecified relative phase for the two transients.

The introduction of energy dissipated as an additional factor in the severity of a shock makes the residual spectrum a more general criterion, less dependent on the assumption of failure modes resembling brittle fracture. If energy increases with peak response, a ductile material that could otherwise survive one extreme strain becomes more likely to fail. Malfunctions that might be insensitive to one peak response become more likely to occur.

In addition, consideration of energy dissipated as well as peak response provides additional reason to focus on residual spectra rather than initial spectra, even if these are interpreted as representative responses rather than descriptions of the excitation. Relatively little energy is dissipated during typical standard pulses.

The phase-related uncertainty results in part from the fact that two transients differing somewhat in frequency can beat against each other so that the peak response does not occur until after some energy has been dissipated. This effect becomes particularly evident if the two transients are similar in duration and initial peak value. In addition, if two transients differ widely in frequency, the peak response can be rather sensitive to the phase of the lower. *Finally, it can be shown that if the phase characteristic* of the excitation is not proportional to frequency, one transient will dissipate some energy before the other is initiated.

It is shown that variability of response energy and peak response can be held within reasonable limits by tolerances placed on a nominal phase versus frequency characteristic as well as a nominal magnitude versus frequency characteristic. The best way to establish the nominal phase characteristic for the simulation of particular shocks would require further analysis.

USE OF RUBBER WEDGES TO REDUCE FUSELAGE STRUCTURE ACOUSTIC RADIATION

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The use of thick, zig-zag shaped foam rubber pads—referred to as "rubber wedges"—to reduce interior acoustic radiation of aircraft fuselage structures has been irregularly under study at Boeing for some time. A fair amount of experimental data dealing with the effect of these devices when applied to structure has been accrued. In certain instances, including an actual flight test, the acoustic attenuation produced by these devices has been impressive, much greater than could be produced by an equal weight of damping tape. It is conjectured that in these instances the wedges functioned primarily as viscoelastic tuned dampers which resonated in a thickness-wise mode. A basic theory, its predicted qualitative impact on the dynamic and acoustic behavior of laboratory and actual aircraft structures is discussed. The observed behavior of actual structure is present and is qualitatively compared to that predicted from theory. A comparison with damping tape type treatment is made.

RESPONSE OF THICK STRUCTURES DAMPED BY VISCOELASTIC MATERIAL WITH APPLICATION TO LAYERED BEAMS AND PLATES

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Addition of viscoelastic material is now often used to damp structures. In this way, the level of vibration can be significantly lowered in a wide range of frequencies.

In the paper, we present, using finite element modelisation, the prediction of the response to harmonic forces of thick layered damped structures with application to thick beams and plates.

These three dimensional structures are described by a very large number of degrees of freedom. In order to reduce this number, the response is calculated as follows:

First, frequencies and mode shapes of the structure are calculated neglecting viscoelastic damping ratio. Then, using these frequencies, mode shapes, and including the damping ratio, a pseudo modal response of the structure is obtained. We call this response pseudo modal because for the whole structure, the damping matrix is only proportional to the stiffness matrix of the viscoelastic material.

Damping ratio and Young's modulus of viscoelastic materials are known to be functions of temperature and frequency. To get numerical values of these characteristics, a thick layered beam is tested and calculated using the method previously described. The results are used to predict the response of the structure damped with thick viscoelastic layers.

The structures studied here are thick plates, but the method can be applied to more complicated devices. Various types of finite elements with consistent mass matrices have been tested; mode shapes and frequencies have been obtained using a simultaneous iterative technique. Computer programs have been written in FORTRAN IV, and experiments have been performed with a Spectral Dynamics SD 1002 system.

The thick beam and plate have been tested under free-free conditions and excited symmetrically. The agreement between calculated and measured driving point impedance modulus is good in a wide range of frequencies.

In conclusion, we think that this method can be used to get dynamical behaviour of various systems and more specially of possible pratical dampers.

CONTROLLING THE DYNAMIC RESPONSE LEVELS OF JET ENGINE COMPONENTS

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Most jet engine components, such as turbine blades, compressor blades, stators, inlet vanes etc., suffer from vibration problems. As engines become ever more complex, and thrust to weight ratios increase, it will become progressively more difficult to design these components so that their natural frequencies do not lie close to some multiple of the blade passage excitation frequency under some operational condition or other. For this reason, the alternative of using damping technology to control peak response levels even at the conjunction of natural and excitation frequencies will eventually become evident to designers for at least the most critical components in each engine.

The purpose of this paper is to (a) discuss the modal response behavior of typical examples of the aforementioned components, (b) show for each case how one or more typical damping techniques can reduce resonant vibration amplitudes significantly, through the application of suitable room temperature damping treatments, (c) demonstrate how the same types of treatment can be used in conjunction with appropriate damping materials, such as enamels, to give the same amplitude reductions at other temperatures even up to 2000°F, and (d) discuss the critical engineering problems to be overcome before each type of treatment can be used successfully under operational conditions.

INVESTIGATION OF THE EFFECT OF DAMPING TREATMENTS ON THE RESPONSE OF HEATED FUSELAGE STRUCTURE

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The Air Force Materials Laboratory, in conjunction with the University of Dayton Research Institute, is in the process of completing an experimental program to develop vibration damping treatments for possible use in the reduction of sonic fatigue in aircraft structures operating at elevated temperatures. Resonant frequencies, mode shapes, and damping of a structural specimen, representative of the aft fuselage of the B-1, were determined for configurations with and without damping treatments. The goal of this program is to determine specific damping treatments that would be effective on this type of structure in the service environment which includes elevated temperatures up to 300°F (from jet plume radiation during take-off operations), exposure to fuel (since materials would be located inside of an integral fuel tank) and long term exposure to moisture. This investigation was divided into essentially four stages.

1. Measurement of natural frequencies and mode shapes in an undamped specimen representative of the full scale structure.

2. Evaluation of effects of elevated temperature, fuel, and water environment on the performance of candidate damping materials.

3. Measurement of the effect of room temperature damping material on the response of the structural specimen.

4. Measurement of the effects of high temperature on the undamped specimen and evaluation of candidate damping materials on the test specimen at high temperature.

This paper will describe the structural response measurement portions of this investigation, including the application of impact testing techniques utilizing digital fast Fourier analysis. Techniques for rapidly determining resonant frequencies, mode shapes and structural damping will be discussed. Frequency response and damping data measured with analog sine sweep methods are compared with similar data obtained with the faster digital techniques. A method for determining modal damping by curve fitting in the complex plane will be discussed along with the limitations of this method. Data will be presented which illustrates transfer functions (showing resonant response peaks), mode shape plots (illustrating in both three dimensions and one dimension the displacements of the structure associated with specific resonant peaks) and Nyquist plots (relating the real and imaginary response vectors in the complex plane) of both damped and undamped structures.

AN ALTERNATIVE SYSTEM FOR MEASURING COMPLEX DYNAMIC MODULI OF DAMPING MATERIALS

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Despite over two decades of progress in developing techniques for measuring complex dynamic moduli of elastomeric, plastic and enamel-like materials, it is still difficult at times to ensure that accurate results are obtained. For example, the vibrating beam type techniques are difficult to use for measuring Young's moduli whose real parts are less than 10,000 Lb/in^2, and most techniques not using beams need expensive instrumentation for reliable use.

In this paper a new variation on these well known themes is examined, namely a compromise between the vibrating beam and nonbeam techniques. The system considered comprises a rigid rod supported horizontally by two beam-like metallic springs at the two ends, and free to move horizontally only. The specimen to be tested is then placed between the center of the rod and a rigid block, either in the form of a shear link or a short beam. The system is excited by a transducer at one end and the response picked up at the other end. The technique can be used for high or low temperature measurements by enclosing the specimen in an environmental chamber or a furnace. In order to establish the advantages and limitations of the technique, tests were conducted on several soft and stiff materials at various temperatures and the results, which will be presented in the paper, show considerable promise.

CLASSIFIED SESSION

COMPARISON OF EXPERIMENTAL AND ANALYTICAL LARGE DEFORMATION SHELL STRAINS ON A MISSILE SUBJECTED TO AN ASYMMETRIC BLAST LOADING

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Several blast tests were performed on an instrumented full-scale missile oriented side-on to the blast in the Sandia THUNDERPIPE blast tube facility. These tests provided measured pressure and strain time history data on the body of the test vehicle at various blast incident overpressure levels. These data are used to evaluate the DEPICS nonlinear shell response code as an adequate analytical tool for predicting shell response of a missile subjected to direct blast pressure loading and undergoing overall bending. The primary instrumented shell section of the missile consisted of a nearly cylindrical shell with three material layers. The inner filament-wound fiberglass layer is the main structural layer and is separated from an outer heat shield by a layer of syntatic foam. Near the center of this shell section the pressures were measured at 22.5° intervals on the outer surface and the axial and circumferential strains were measured at 30° intervals on the structural layer. From these pressure data and other pressure measurements along the body, a pressure model was constructed to describe the spatial and temporal pressure variations on the shell section and the total running normal loads on the missile for determining the overall bending response. A vehicle bending code was used to determine the bending moment time history at the shell section of the missile. Thus, the shell section is considered to be simultaneously loaded by the direct blast pressure and the blast induced bending moment.

The DEPICS shell response code used in this correlation effort is based on the Novozhilov nonlinear strain-displacement relations for large displacements of cylindrical shells and can treat multiple layers of linear, elastic, isotropic or orthotropic material. The Lagrangian equations of motion are developed through an energy formulation and are solved through numerical methods using a modal-type analysis. Both time-dependent lateral pressure loading and end bending moments can be applied to the cylindrical section which is simply supported with or without axial constraints at the ends. The aforementioned pressure model and the analytically determined end bending moments are used with DEPICS to determine analytically the circumferential and axial strain time histories on the shell section for comparison with the corresponding experimental strain responses. Refinements were introduced into DEPICS to investigate the effects of including rotary inertia, nonlinearities in the change of curvature quantities and conical shell geometry for the test condition producing the largest shell response. It was found that these effects were not overly significant, except for the differences in the axial strain between a cylindrical and conical shell representation.

The correlation presents direct comparisons of the major experimental and analytical circumferential and axial strain time histories at $\theta = 0°$ and 180°, where $\theta = 0°$ defines the windward ray, on the outer and inner surfaces of the structural fiberglass wall of the shell section. The tests where performed at various pressure levels that produced geometric linear and nonlinear deformations of the elastic shell section. The DEPICS shell code gave very favorable predictions of the changing strain pattern in the shell section associated with the linear response observed at the low pressure test levels in contrast to the nonlinear response observed at the high pressure test levels. The strain pattern is characterized by the largest circumferential strain occurring at $\theta = 180^\circ$ on the leeward side in the linear response tests and at $\theta = 0^\circ$ on the windward side in the nonlinear response tests. The time of maximum response at $\theta = 0^\circ$ increases significantly as the strain response becomes geometrically nonlinear.

Acceptable agreement was obtained from comparisons of the analytical and experimental major circumferential and axial strain response time histories on the shell section for the various test conditions. The DEPICS shell code demonstrated the capability of predicting strain response in both geometric linear and nonlinear regions. It was found that strain response time histories are very sensitive to the pressure distributions around the body during the engulfment of the blast wave. Thus, the major uncertainties in analytically predicting the strain responses stem from the pressure loading model based on limited experimental data and the axial end boundary conditions on a shell section which is part of a vehicle undergoing overall bending.

THE SHOCK ENVIRONMENT ASSOCIATED WITH THE AIRCRAFT DELIVERY OF A NAVY MINE

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Most Navy mines are designed so that they can be delivered by aircraft. The shocks associated with this aircraft delivery include: (a) bomb-rack ejection, (b) parachute actuation and deployment, and (c) water entry. These shocks are complex in nature, and are not easily described by a single value of peak acceleration or pulse duration.

The accelerations associated with each of these aircraft-delivery shocks were measured during field tests of a new 2000-pound Navy mine. Field tests were conducted over a two-year period using several different types of test vehicles. Accelerations were measured at several locations within the test vehicles and recorded on magnetic tape using a small, self-contained instrumentation system. The heart of the instrumentation system is a rugged, 14-channel tape recorder with FM record electronics. Acceleration is recorded on tape for playback in the laboratory after the test vehicle is recovered. Total recording time is limited to about 40 seconds since tape capacity is small. This recording time is sufficient, however, to record each of the delivery shocks on a single tape record for many delivery conditions. The acceleration records obtained are believed to represent some of the most complete records available on the tactical shock environment associated with the aircraft delivery of a Navy weapon.

Acceleration records obtained were digitized and then analyzed in various forms to assess the effects of these shocks and to establish tests to simulate these effects in the laboratory. Sample acceleration records, along with the velocity-time records obtained by the integration of these acceleration records, are presented. Shock spectrum analyses of records were conducted and were the primary tool used for assessing the results of these tests. Analysis indicates that, for all practical considerations, water entry can be considered to be the most damaging shock associated with the aircraft-delivery environment. This statement is based on shock spectrum analysis between frequencies of 1 and 2500 Hz—analysis at these frequencies indicates that water entry consistently excites the highest responses of the delivery shocks considered. The impact and drag phases associated with the classical water-entry shock pulse are clearly visible on the records obtained. The velocity changes associated with the impact and drag phases of a particular shock are visible on the shock spectrum of that shock. The impact phase becomes insignificant, however, and virtually disappears for low water-entry angles.

This new mine has several large subsystems or components associated with it. Records were obtained under similar delivery conditions for vehicles with and without these components. The components produce secondary impacts at water entry which can be detected on the acceleration-time records and produce a noticeable increase in the computed responses over a wide range of frequencies.

The effects of water-entry shock are simulated at NOL using air guns with two-phase pistons. Impact of the two parts of the piston produces a high acceleration, short duration shock simulating the impact phase of water entry. The two parts of the piston then lock together and accelerate down the barrel of the air gun as a single unit, simulating the low g, long duration drag phase associated with water entry. The shock spectrum produced by this laboratory simulation was computed and compared to the spectrum obtained from field measurements. Initial test levels proved to be overly conservative and new test levels were established. The new tests produce spectra which closely approximate the ones obtained during the field tests. While air guns have been used by NOL to simulate the effects of water entry for many years, test levels have traditionally been selected by comparing the velocity changes of the impact phases and the peak accelerations of the drag phases of the respective shocks. This effort is believed to be one of the most complete and accurate attempts, to date, by NOL to characterize a complex water-entry shock by its shock spectrum and then produce a similar shock spectrum in the laboratory with its air guns.

ELASTIC RESPONSE OF SUBMARINES TO END-ON NUCLEAR ATTACKS

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INTRODUCTION

A knowledge of the shock motions that underwater weapon attacks produce in a submarine is a necessary prerequisite for the design of shock resistant pressure hulls and internal equipment installations. It must be expected that end-on attacks by long-duration nuclear shock waves can produce shock motions throughout the submarine which are comparable in magnitude to those which result from side-on attacks of equivalent severity. The end-on response of a submarine to nuclear attacks has not, however, been well documented, either analytically or experimentally, and a reliable method for predicting a submarine's overall response to such attacks is needed.

In a recent investigation, conducted by the Naval Ship Research and Development Center, the loading environment for submarines subjected to end-on attacks was studied and a simplified method to predict the shock motions in the longitudinal direction was developed. The results of that study suggested that a satisfactory description of the axial response of a submarine may be obtained by mathematically modeling the pressure hull as a one-dimensional, lumped, spring-mass system with all motion restricted to the longitudinal direction; that is, the direction of the propagating wave front.

A recent series of underwater explosion (UNDEX) trials against a stiffened, cylindrical model has provided an excellent opportunity to determine, under well controlled circumstances, the end-on response of a submarine-like model and to fully evaluate the accuracy and suitability of the prediction method.

OBJECTIVE

The objective of this research was to develop a reliable method for predicting the longitudinal response motions of a submarine pressure hull and internal equipment installations which have been subjected to end-on attacks from a nuclear weapon.

SCOPE OF PAPER

The development and application of the lumped-parameter prediction method for end-on nuclear attacks will be presented in two parts. Initially, the loading environment and shell response analyses will be discussed. Also, the simplifications, assumptions and limitations of the lumped-parameter approach will be cited. In the second part of the paper, the UNDEX trials against the cylindrical model will be described and the measured and predicted longitudinal response motions will be compared. A brief description of planned modifications and expansion of the method will also be included.

LUMPED-MASS PREDICTION METHOD

In applying the lumped-mass prediction method, the submarine is modeled mathematically as a discrete, undamped system with unrestrained boundaries. The system is composed of lumped masses, connected by linear springs having a finite number of degrees of freedom. The familiar method of modal analysis is then used to determine the transient response of the system.

It should be pointed out that the spring-mass prediction method is not restricted to a uniform cylindrical shell but can take into account structural and mass variations along the longitudinal axis. Also, the method can readily be extended to include the response of elastically mounted internal equipments.

DISCUSSION

The predicted response motions for the end-on attacks correlated quite well with the experimentally obtained velocity and strain gauge data even though rather significant simplifying assumptions have been introduced.

Most notably the results of this study indicate that for end-on nuclear attacks interaction effects of the impinging shock wave can be minimized and that an accurate description of the response of such a complex structure as a submarine can be obtained using a much simplified lumped, spring-mass approach. Furthermore, it was observed that longitudinal shock motions due to end-on attacks can, in fact, equal or exceed similar motions from side-on attacks of equivalent severity.

PLASTIC DESIGN ANALYSIS OF SHIPBOARD EQUIPMENT SUBJECTED TO SHOCK MOTIONS

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INTRODUCTION

The complexity of modern naval ships and submarines requires constant updating of design methods and procedures. This is particularly true of light high-speed advanced designs. New design methods and design criteria which will result in weight and space savings are essential. In current naval vessels there exist structures which are obviously overdesigned and would allow significant savings in weight and space; foundations in particular.

Shipboard equipment can be divided into three categories of design. One category can be classified as those equipments which always require a close alignment. For these equipments it is necessary that response to operating and shock loads be confined to the elastic range. Equipments such as SINS, turbines, reduction gears, shafting, turbogenerators and similar items fall in this category. A second category which includes a large majority of shipboard structures and equipment may be classified as those items which do not require alignment and can allow small deformations. Piping, resiliently-mounted equipments and many foundations for example can be considered for this category. The enormity of this category in particular warrants a careful consideration of new methods which may lessen costs, weight and space requirements. A third category may be classified as those items which can tolerate large deformations (deformations greater than four elastic limit displacements). Foundations for electronic cabinets, weapon cradles and dollies, decks, stanchions and other support structures can be considered for this category. As can be seen from these examples that "hard-to-harden" and shock sensitive item support structures are major items in this category. Methods which result in significant shock protection are advantageous and necessary for many items in this category.

Little attention has been given to the efficiency of equipment design. Over-designed foundations for example can be improved significantly by taking advantage of the reserve strength in the material above its yield point. This reserve strength combined with the redundancy generally encountered in foundation design can serve to optimize foundation efficiency. To make use of the reserve strength requires plastic design procedures. Currently, plastic design procedures are practically nonexistent for dynamic loads.

Simple one degree-of-freedom beam structures are plastically designed using the limit load procedure (see DDS 9110-7). More complex structures may be designed using an elastic analysis procedure (DDAM) but using reduced inputs. This procedure results in a structure which will deform plastically at the unreduced shock level. The resulting deformation or response of the structure is of course unknown. This limited usage of the plastic design concept does not allow optimization of design efficiency.

PURPOSE

The purpose of this paper is to delineate the plastic design method for shipboard application application and the expected gains.

PLASTIC DESIGN METHODOLOGY

Plastic design is the art of utilizing the reserve strength of ductile materials to optimize the efficiency of structures and to simplify the required analysis. In general this is accomplished by designing a structure for a collapse load greater than the working load. The resulting structure bears the working load with negligible permanent deformation. The formation of a plastic hinge in a structure does not necessarily allow collapse but merely allows a redistribution of stresses. The reserve strength beyond yield allows rotation of the plastic hinge as the load is increased. Redistribution and the increase in load will ultimately result in a collapse mechanism. The collapse mechanism in terms of a static load constitutes catastrophic failure; however, the collapse mechanism also constitutes energy dissipation greater than the elastic strain energy per unit of displacement. For example, the energy dissipated at a displacement twice the elastic limit in a simple structure may be four times that absorbed elastically. At two elastic limit displacements, deflections are not likely to be visible.

CONCLUSION

The inherent reserve strength in structural steels will allow optimization of the structural efficiency of shipboard structures. It is expected that foundation weights can be reduced from that required for elastic design if maximum set displacements are allowed to reach three elastic limit displacements. Similar weight reductions can be expected in structures other than foundations where plastic design is applicable. The weight savings depend on the trade-off allowances for all the factors involved. Considerations for noise, vibration, and maximum set deformations must enter the trade-off allowances.

In addition to the savings in weight which relates to costs, there is also an expected cost reduction due to the simplicity of plastic design analysis methods.

Additional gains are possible employing plastic deformation concepts in shock mitigation.

HIGH PERFORMANCE VIBRATION ISOLATION SYSTEM FOR THE DD 963 GEARS

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Specifications for main reduction gears for the DD 963 Class of destroyers called for structureborne vibration levels markedly lower than vibration levels measured on the similar, though smaller, reduction gears supplied for the DE 1052 Class of destroyer escorts. It was judged that such a large incremental improvement in structureborne noise levels could not be obtained with assurance by any conventional gear design and manufacturing technique particularly in view of weight and cost limitations. Thus, a populsion gear vibration isolation system was proposed. For a number of reasons, a standard Navy rubber mount type of system was not acceptable. An unconventional but much stiffer isolation system featuring damped tension-compression metallic isolators was developed. Although it has not yet been tested at sea, its performance has been determined in extensive shop tests on fully loaded gears.

It was required of the new isolation system that it produce 20 dB of insertion loss in those critical frequency ranges where both first and second reduction mesh frequency occur. This goal has been met. Further, since the basic stiffness of the isolation system is an order of magnitude larger than that of a typical rubber mount system, (vertical natural frequency of 19 cps) care had to be exercised so that neither unbalance vibrations in the gear shafting, nor propeller blade rate excited motions would be unduly magnified by rigid body modes of the gear case. Since the frequency range of interest extends into the high frequency region where no valid structureborne noise data were available, it was deemed prudent to maintain good vibration isolation up to 20 KC. These considerations require that isolator surge frequencies be reasonably damped, and that the isolator high frequency impedance be small compared to the high frequency impedance of both the reduction gear and the ship's foundation.

To demonstrate the performance of such an isolation system in a shop test requires a test foundation whose impedance characteristics approximate those of the ship's foundation on which the isolated gear system is to be mounted. Such a foundation was designed and tested. It is described in this particular paper only in general terms sufficient to permit proper interpretation of given test results.

Since it is not possible for an isolation system to be soft enough to have high isolation performance yet strong enough to carry the maximum values of normal loads plus shock loads, a snubber system had to be designed and carefully integrated with the isolation system. In fact, snubber clearance considerations strongly influence isolator design and determine the cyclic life of the isolators.

The isolator system is described and information on its vibration transmission characteristics is presented. The reasons behind key choices in the design are discussed as are certain limitations of the design and the reasons for them.

Eight 7075 aluminum vertical isolators carry vertical loads and torque reactions. These isolators consist of a central compression member which bolts to the ship's foundation, and pierces the gear flange. At the top of the inner member, an outer tension member 'folds back' and bolts to the top of the gear case flange. This complicated and expensive design was dictated by time and geometry considerations. Both the inner and outer tubes, the flexible members, are damped with a heavy layer of Lord's LD502 damping material. Design features required to use this material successfully are discussed. The horizontal isolators are made from 6AL4V titanium and are simple tension-compression rods. They also are damped by the LD 502 damping material.

Important properties of the isolator such as first surge frequency and Q were determined by calculation and confirmed by test. The shop test vibration performance of the overall system is described and conclusions are drawn. In order to estimate the performance of the isolator system as distinct from the overall performance of the gear and isolation system, special stiff steel connectors were made. These replaced the isolators for a series of tests so that estimates of insertion loss could be made. Reasonable estimates of insertion loss and ansmission loss are made.

DESIGN AND MEASUREMENT OF A HIGH IMPEDANCE FOUNDATION TO 20 KHz AND USE OF DATA IN CORRECTING NOISE MEASUREMENTS

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When structureborne noise measurements are made on a component for shipboard use they are normally conducted in accordance with MIL-STD-740B. Under this normal procedure, the component is mounted on some type of Navy standard mount for the purpose of reducing the impact of extraneous shop background noise and insuring that all of the components of a similar nature are tested in similar environmental conditions. This means that each component is exposed to approximately the same impedance (i.e., the resilient mounts).

This procedure is quite satisfactory when an acceptance criterion is available for these test conditions. Such a criterion might be based on historical data of like or similar components. If such an acceptance criterion is established by means of analytical computation based upon a desired sound pressure level in the water and transfer functions between the hull and the components foundation, then it becomes more desirable to eliminate the isolator as an unknown. This is especially true if the isolator is of a new design and little is known about its insertion loss.

This paper deals with such a program in which a new metal isolator was used to isolate the noise of a shipboard reduction gear and the noise measurements had to be made below the isolators on a foundation whose impedance had to be measured and controlled.

A foundation (10^6 lb) was designed such that the point impedances of each isolator termination point could be controlled from 14 Hz to 20 KHz. The objective was to have each termination impedance approximate a reference impedance and then measure each termination point impedance such that the measured structureborne noise of the gear could be corrected to the reference impedance. This reference impedance is itself an approximation of a typical destroyer engine room foundation and was used as a design objective by the shipbuilder for their engine room foundation.

The design and construction of a large foundation with a controlled impedance as high as 20 KHz presented many technical problems. Similarly, the measurement of the point impedance up to 20 KHz also presented several technical problems.

The paper describes the basic method used to obtain the required impedance. It describes the viso-elastic terminators imbedded in concrete and the associated calculations used to predict the resulting impedance.

The test program that followed the design and construction is also described. This includes the relationship between termination impedance and temperature as well as a complete discussion of the problems associated with obtaining valid impedance measurements at frequencies above 10 KHz. It became virtually impossible to obtain completely valid impedance measurements at frequencies greater than 18 KHz. Therefore, a correction technique was used and it too is described in the paper.

Finally, the paper includes a description of the method used to take and correct structureborne noise on a power average basis. This is a relatively unique method in that all frequencies and all measurement points must be corrected for the difference between a reference impedance and each specific termination impedance. A brief description of the DD963 reduction gear test setup is included to illustrate the application of this structureborne test method.

THE DEVELOPMENT OF A WATER PARTICLE VELOCITY METER

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INTRODUCTION

The design of marine structures which are resistant to the shock of underwater explosions is facilitated when accurate computations of the response of the structures to underwater shock loading can be made. To verify a structural analysis technique or provide information necessary for the further development of the technique, experiments must be performed in which both the loading and structural response are measured. Water particle velocity as well as shock wave pressure is part of the specification of underwater shock wave loading. When the form of the wave propagation (plane, spherical, etc.) is not known or reflections are involved, the relationship between vector particle velocity and scalar pressure is not known well enough to determine the particle velocity loading from the pressure measurement alone. Under these conditions a direct measurement of particle velocity should be made to determine the particle velocity loading.

In September 1973 the Naval Ship Research and Development Center (NSRDC) conducted experiments in the Chesapeake Bay to determine the loading and response of a small scale submarine model subjected to the shock wave of a tapered charge. Because the relationship between water pressure and water particle velocity for tapered charge shock waves is not exactly known, the loading was determined by free field measurements of both pressure and particle velocity. The unavailability of a suitable water particle velocity meter previous to this application necessitated the development of a particle velocity meter at NSRDC especially for this project.

PURPOSE OF PAPER

The purpose of this paper is to give the characteristics of the water particle velocity meter recently developed at NSRDC and to demonstrate its effectiveness in measuring the particle velocity of shock waves resulting from the underwater explosion of compact and tapered charges.

THE PARTICLE VELOCITY METER

The velocity meter normally used by NSRDC to measure structural velocity transients provides the basis for the water particle velocity meter design. This structural velocity meter has a cylindrical shape with an output voltage proportional to the velocity component in the direction of the axis of the cylinder. The light weight of the cylinder allows it to be waterproofed and made neutrally buoyant by the addition of appropriately designed aluminum end caps. For the measurement of water particle velocity, the modified meter is seismically mounted from an underwater structure with the meter axis in the direction of the desired particle velocity component. Because the cylinder is neutrally buoyant, it moves with the surrounding water under the influence of a shock wave. The output of the meter is a voltage proportional to the average water particle velocity along the length of the cylinder.

MEASUREMENTS OF PARTICLE VELOCITY

The particle velocity and corresponding pressure in underwater shock waves produced by the explosion of compact and tapered charges have been measured using the NSRDC particle velocity meter and a commercial tourmaline pressure gage. A computation of particle velocity from measured pressure assuming a spherical wave has been made for comparison with the measured particle velocity. Since the compact charge produces a spherical wave, the comparison of measured particle velocity with that computed from measured pressure provides a check for inconsistency between the particle velocity and pressure measurements. Because the particle velocity measurement is made using an untested meter, the good agreement obtained between computed and measured velocity is taken as a verification of the operation of the particle velocity meter.

Having demonstrated the operation of the particle velocity meter in the spherical shock wave of a compact charge, the meter was used in the tapered charge experiments for which it was developed. The data from the tapered charge experiments show that the spherical wave assumption applies and the comparison of computed and measured particle velocities further strengthens confidence in the water particle velocity meter.

MECHANICAL DESIGN, ANALYSIS, AND TEST OF THE STANDARD ELECTRONICS CABINET AND INTEGRATED MULTIFUNCTION CONSOLE FOR AN/BQQ-5

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The AN/BQQ-5 shipboard system is a sonar signal processor comprised of electronics cabinets whose function is to detect, track, and classify targets in the ocean.

With this system are two basic structural designs which will be discussed, the "standard" electronics cabinet and the Integrated Multifunction Console (IMC). The "standard" electronics cabinet design concept is utilized as the basis for a large percentage of the total system structure while the IMC mechanical design is the most unique and complex within the system.

The units are designed to meet the requirements of MIL-E-16400, which requires qualification to MIL-STD-167 and MIL-S-901 for vibration and shock environments, respectively.

To insure compliance to these requirements, a basic design philosophy was established early in the development stage of design. The governing mechanical design philosophy for shipboard equipment within IBM is twofold:

1. design sufficient structural rigidity, or stiffness to insure that fundamental equipment resonances are substantially greater than the maximum forcing frequency specified in MIL-STD-167. In this manner, the increases in vibratory amplitudes associated with structural resonances are avoided and the corresponding potential fatigue damage precluded, and

2. design sufficient strength in the structural elements to keep stresses resulting from the NAVY Hi-Impact shock testing of MIL-S-901C within allowable material limits.

To accomplish these objectives, computer aided structural analysis techniques were employed. The MECHA (MECHanical Analysis) and STRUDL (STRUctural Design Language) computer programs were used to provide continual monitoring of the mechanical design. The results of the analysis of the final mechanical designs compared very favorable with both empirical data obtained during engineering evaluation testing and with independent computerized analysis performed by the Navy Underwater Sound Laboratory. This established a high level of confidence in the structural integrity of the units.

As a result of the extensive structural analysis and engineering evaluation testing performed upon the units during development, these two units passed the shock and vibration qualification tests with a minimum of mechanical problems or redesign.

The discussion will also include quantitative details of the development and qualification testing which could prove helpful to those engaged in the mechanical design of shipboard equipment.

The standard electronics cabinet is designed with two vertical slide-out chassis which carry most of the electronics. When in the closed position, each slide-out chassis mates with the cabinet housing by means of five bullpins in the rear and nine bullpins around the front latch. In addition, each front latch contains 16 bolts and strikers to positively attach the chassis to the cabinet housing.

The primary load-carrying structure consists of a casting of A357 aluminum alloy both top and bottom. Both sides are extruded 6061-T6 aluminum alloy, which forms an integral wall incorporating corner posts, inner and outer skins, and longitudinal flutes for cooling air. The extrusions are welded to the top and bottom castings. The cabinet is divided into two sections by a vertical partition which is designed to form fluted ducts for cooling air. The rear of the cabinet is a removable 1/8-inch thick panel which attaches by flathead screws.

The two slide-out chassis are each cast in one piece of K01 aluminum alloy. The casting is machined to accept the full length latch on front, slides on top and bottom, and electronic subassemblies on each side.

The IMC is designed with five slide-out chassis—two separate CRT chassis, one combined drum unit and drum power supply chassis, and two NAFI chassis. The structure is designed in two halves to comply with installation size restrictions. Each half is constructed in a similar fashion and they are permanently bolted together to form one integral unit. The desk top then is bolted to the frame.

Each half has an aluminum casting of A357 at the base. The top is a solid plate of 6061-T6 aluminum. The main frame members are generally 6061-T6 aluminym extrusions. The top plate, bottom casting, and extrusions are welded together to form the basic frame. The sides and rear are 1/8-inch thick aluminum panels of 6061-T6 alloy, which are attached by flathead screws. An additional doghouse extends the full height on the rear to form an air duct for cooling the electronics.

The lower half of the IMC contains two similar NAFI slide-out chassis. The entire chassis slides out on two slides located one above the other on the side of the chassis. The slides are attached to the K01 T43 aluminum "L"-shaped single NAFI gate casting. The double NAFI gate is attached by a hinge to the single NAFI frame.

The single NAFI gate, double NAFI gate, hinged door, and IMC frame are integrally locked together, when in the closed position, by bullpins and specially designed wedges and toggle latches.

The upper and lower CRT chassis are constructed on a one piece A356-T6 aluminum casting supported on each side with a slide. When in the closed position, the casting mates with the frame by means of high-strength bullpins, two bullpins at the rear and two bullpins at the front. The chassis is held in the closed position by four thumbscrews.

The drum chassis extends from the IMC frame on two slides located one above the other along the side of the unit. When in the closed position, the unit is supported by four high-strength bullpins, both front and rear. The chassis is held in the closed position by six thumbscrews. The chassis is a cast A356-T6 aluminum frame. The front panel is a separate casting of the same material. The drum itself is mounted in the lower section of the drawer. The upper half contains the drum power supply and the static inverter.

SPECIAL PROBLEMS

EXTENSION OF CONTROL TECHNIQUES FOR DIGITAL CONTROL OF RANDOM VIBRATION TESTS

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In addition to faster test setup, increased spectral resolution, and greater control accuracy, a digital random vibration control system can be easily reprogrammed for new control algorithms that are designed to control more complex test requirements than the single control channel test common on most analog and digital systems. This paper describes three control techniques that have been implemented at the System Environmental Testing Facilities in Area III, Sandia Laboratories, Albuquerque, New Mexico. The random control programs (J. D. Tebbs and N. F. Hunter, "Digital Control of Random Vibration Tests Using a Sigma V Computer," IES 1974 Proceedings, pp 36-43) were modified to include these control techniques.

1. Averaging—The averaging of several input channels in place of a single input can provide a better input definition on larger test items. Input averaging is implemented by sampling each of the inputs for one or more frames during the control loop. The PSD estimates for each input are then summed to provide an estimate of the average PSD. The average PSD estimate is then compared with the stored reference spectrum and the drive signal is modified in the same manner as for a single channel test. Implementation of the technique and an example are discussed.

2. Random-on-Random—This technique provides for super-imposing several swept narrow band random signals on a background stationary random spectrum. This technique is useful for simulating environments where a significant amount of energy is concentrated in a few narrow bands, but where the center frequency of the bands is somewhat uncertain.

Briefly, the technique is implemented by dynamically changing the reference spectrum during the test. The normal control system changes the drive spectrum to match the changing reference spectrum.

Options include the selection of the number of bands, their sweep rates, bandwidths, levels, start frequencies, and stop frequencies. Included in the paper is the implementation method and a discussion of the interactive effects of varying the test parameters. Several examples are discussed.

3. Response Limiting—In many cases, it is desirable to limit the response spectrum of components on a large system during a random vibration test by lowering the input in selected frequency bands. Response limiting was implemented by first estimating the transfer function between the input and the response point by using a transient calibration pulse. Using this transfer function and the reference spectrum of the input, the spectrum of the response is computed. If the response spectrum estimate exceeds the

allowable spectrum, the input reference spectrum is reduced an appropriate amount. The random test is then run, using this modified reference spectrum. The advantages of this implementation include: any number of response channels can be limited without slowing down the control response time, and the technique is relatively easy to implement. The principal disadvantages include the assumption of a linear system and the inability for "real-time" limiting of the response channels. The implementation is discussed and an example is shown.

EXPERIMENTAL DETERMINATION OF MULTIDIRECTIONAL MOBILITY DATA FOR BEAMS

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In order to undertake the vibration analysis of a complex structure it is usually most convenient to consider the structure in terms of its component parts. Each part, or substructure, may be analysed individually by whatever method is best suited to it and a mathematical model of the complete structure may then be built up from the results of these individual analyses using an impedance coupling technique. Clearly, it is important that the data for all substructures should be of the same order of accuracy, and indeed this may generally be achieved with data which are derived from a theoretical analysis. However, there are many cases for which it is necessary to obtain at least some of the required data by experimental measurement, and for these, achieving the necessary degree of accuracy can be very difficult. This paper is concerned with the experimental determination of suitable data for a component by measurement of its mobility or mechanical impedance characteristics.

Previous work has demonstrated the need to include (and thus to obtain data for) vibration in as many directions as the complexity of the component or structure demands. This may involve as many as 6 directions at each point of interest, but is often confined to just 2 or 3 directions, depending upon the degree of symmetry possessed by the structure. However, it is rarely possible to confine interest to just one direction and for this reason, conventional experimental mobility measurement procedures (which only measure translational motion) are generally inadequate for the task of obtaining data required for the analysis of complex structures. There have been few successful attempts to measure the multidirectional mobility data which is required, and both of those reported to date (1) (2) are lengthy and complex.

Following earlier work (1), the present paper describes the development of a more straightforward method for measuring multidirectional mobility data and uses uniform beams (whose properties are known accurately from theory) as test cases. The study is based on the simple concept of (a) measuring the mobility at the end of beam A, (b) measuring similar data at the end of beam B, and then from these two sets of data, (c) predicting the properties of the combined beam A + B formed by joining A and B end to end. This case involves just two coordinates at each point — lateral translation (x) and rotation (θ) — and so the required data are contained in a 2 x 2 mobility matrix for each beam, the elements of which are $(\dot{x}/F_x)$, $(\dot{x}/M_\theta)$, $(\dot{\theta}/F_x)$ and $(\dot{\theta}/M_\theta)$.

Measurements of the individual elements of this matrix generally agree very well with theory, although the $(\dot{\theta}/M_\theta)$ term is less accurate than the others, especially at low frequencies. However, the combined beam (A + B) properties predicted using these data are much less satisfactory. An attempt to 'smooth' out the effects of random experimental errors using a modal identification procedure (3) improved the predictions slightly, but not substantially.

The limiting factor is found to be the relatively low accuracy of the $(\dot{\theta}/M_\theta)$ mobility, which in turn is due to difficulty in measuring the angular motion θ. To overcome this problem, an alternative means of deriving the $(\dot{\theta}/M_\theta)$ mobility term is proposed. The method is based on the fundamental relationship which exists between the contributions to each mobility term from each mode of vibration. Using this relationship, it is shown that all four mobility terms may be derived from measurements of only two of them (the $(\dot{x}/F_x)$ and $(\dot{x}/M_\theta)$ being chosen here as they prove to be the easiest to measure accurately) in conjunction with a straightforward modal identification analysis. Predictions for the combined properties using data obtained by the technique show a marked improvement over the earlier results. Moreover, there is a significant reduction in the quantity, complexity and required accuracy of the experimental measurements which need to be made, so that these may be obtained with standard impedance measurement apparatus.

Although the results presented in the paper are for beams, the technique described may be used for any structure whose multidirectional mobility data need to be measured.

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ANALYTICAL EXTRAPOLATION OF EXPERIMENTALLY DETERMINED MODES OF VIBRATION

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The purpose of this work is to develop a method which can reduce the cost of vibration testing by limiting its use to obtaining a minimum number of the lower modes of a test structure. These lower order modes are then to be used in an analytical procedure to predict the higher order modes. The experimental data required for such a procedure is two modes at all the coordinates of a lumped parameter system; for continuous (distributed parameter) structures, a minimum of one mode is required to be known.

The primary goal of structural dynamic analysis, theoretical or experimental, is to determine the natural frequencies and mode shapes and/or a mathematical model for the structure being considered. This information is essential to predict and control the response of the structure under different types of dynamic loadings. Theoretical analysis is usually based on a set of assumptions which reduce the very complex structure to a set of equations that will presumably represent the behavior of the real structure. Uncertainties in theoretical analysis make vibration testing a necessity.

In spite of advances in modern dynamic testing procedures and technology, a perfect, reliable and economical technique for vibration testing is still far out of reach. One of the most serious limitations in vibration testing is not being able to determine experimentally a number of modes equal to the desired number of measuring stations (degrees of freedom). At the same time a structure in operation might be subjected to exciting forces that contain frequency components beyond those that can be handled by the existing vibration testing methods and equipment.

This paper suggests a method in which a limited number of experimentally determined modes are used to predict analytically the higher order modes of the structure under consideration. As an illustration, the first two modes of a simulated three-degree-of-freedom lumped parameter system were used to predict the system's third mode. The method is also applied to results obtained from a shake test of the 1/10-scale Apollo/Saturn V replica model. In this application, the first experimentally determined mode is used to analytically predict the higher order modes of the model. The analytically predicted second, third and fourth modes are compared to those that were already experimentally determined.

VIBRATION-INDUCED DOPPLER EFFECTS ON AN AIRBORNE SHF COMMUNICATIONS SYSTEM

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and

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The Air Force Flight Dynamics Laboratory has recently been investigating a new vibration problem in airborne communications systems—the Doppler effects on the signal caused by the vibration of the airborne antenna. These effects are proportional to the signal frequency and are becoming more important as higher frequency bands are used, as in communications with airborne command posts.

To develop secure, reliable communications for such airborne command posts, the Air Force Avionics Laboratory has been investigating the use of super-high-frequency (SHF) communications terminals to communicate between aircraft and synchronous satellites. At these frequencies, near 8 GHz, Doppler effects from the aircraft motion may shift the signal frequency by several kilohertz. This shifted signal must be accurately followed by the modem (modulator/demodulator); to do this, the aircraft motions are measured by the inertial navigation system (INS) and the resulting Doppler shift is allowed for by a computer. The additional Doppler effects due to antenna vibration are uncorrected and are thus limiting factors on system performance. Degradation of signal strength or loss of lock can result from excessive Doppler effects due to vibration.

To measure these vibration-induced Doppler effects, the Flight Dynamics Laboratory performed a flight test program on an Avionics Laboratory transport aircraft equipped with an SHF terminal communicating with a synchronous satellite. To completely define the Doppler effects, four parameters of the antenna vibration were needed: the displacement, which causes a phase shift in the signal; the velocity, which causes a frequency shift; the acceleration, which determines the rate of change of the Doppler shift; and the jerk (rate of change of acceleration) which determines the abruptness of the Doppler rate. Each of these parameters affects a feedback loop of the modem; the full four-parameter environment was therefore needed to define all the feedback loop values. In order to provide the greatest accuracy, the antenna line-of-sight acceleration was measured simultaneously with the accelerations of the inertial navigation system. Measurements were recorded for all expected aircraft maneuvers, to ensure reliable communications from before takeoff until after landing.

The data were analyzed initially in terms of acceleration in narrow frequency bands. The jerk, velocity, and displacement were derived from these acceleration spectra by applying the proper factors as functions of frequency. The modem operational criteria were then compared to these results. This comparison indicated the critical frequencies at which difficulties would be expected. A more detailed analysis derived measures of the expected number and duration of exceedances of the criteria for the four parameters of displacement, velocity, acceleration, and jerk. This required the development of new computer programs for time-history analysis of the terminal accelerations. The analysis indicated that for the particular modem the observed flight vibrations created a problem. This was expected, since the modem had originally been designed for a ground base. Two solutions to the problem are being investigated—the reduction of the antenna vibration itself, and the relocation of the INS near the antenna to reduce the uncorrected component of the vibration.

This program has indicated the potential severity of the vibration Doppler problem in SHF communications by airborne terminals. The first step in solving the present problem was to derive improved tracking loop characteristics from the measured vibration parameters. However, the accurate measurement of all four motion parameters is a stringent instrumentation problem. Future systems will need to be designed from inception with vibration in mind. This will become more critical with higher communication frequencies, since the Doppler effects are directly proportional to communication frequency.

FATIGUE DAMAGE EQUIVALENCE OF REAL AND SIMULATED VIBRATIONAL ENVIRONMENTS

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INTRODUCTION

The development of better laboratory test specifications is a continuing goal for the vibrations engineer. His objective must be to simulate a field environment in such a way that high probability of in-service failure in a typical specimen will result in a similar failure in the test environment. This paper describes a method of comparing the damage potential of two such environments when the failure mechanism is fatigue. The method involves testing of an instrumented specimen in both environments, recording and analyzing the specimen responses in each case, and comparing the direct fatigue potential from the two results.

MODEL HARDWARE SPECIMEN

A model hardware specimen (MHS) is designed in the form of a small box that can be fastened into typical avionics equipment racks. The box contains eight cantilever beams having resonances respectively at about 37, 48, 76, 105, 132, and 156 Hz. These frequencies, as well as the spatial orientation of the beams inside the box were selected from random values. Thus, each beam represents some arbitrary mode of a typical hardware specimen. Each beam is instrumented with strain gages so that strain histories can be recorded when the box is subjected to a dynamic environment.

METHOD OF PREDICTING FATIGUE ENVIRONMENT

Strain time histories from a given environment in which the MHS is tested are used to predict fatigue. This is accomplished ultimately through the use of Micro Measurements, Inc. fatigue gages. Two different methods can be employed. If the strain history is sufficiently narrow band, then a peak counting process is employed along with calibration curves for the fatigue gages, to produce the fatigue prediction for each environment. If the strain history is relatively complex, the fatigue gage is used directly to predict the environment by subjecting another beam specimen to the exact strain time history. This beam is instrumented with the fatigue gage so that a direct computation of the fatigue potential is performed. A comparison of the potential of the two environments provides a numerical evaluation of the laboratory test specification in terms of the equivalent number of service hours, missions, etc. This method is particularly useful for comparing complex environments resulting from nonstationary operation of vehicles, aircraft, etc.

SAMPLE RESULTS

The method is applied to evaluation of typical laboratory simulations of helicopter, truck, and tracked vehicle environments. In each case, the numerical fatigue potential is presented. It is shown that the results vary considerably with frequency and vibration level. Thus, no exact simulation can be achieved at all frequencies and levels in any actual hardware, and the resulting limitations on interpretation of results from typical tests must be recognized.

SEISMIC SIMULATOR FOR SILO CONSTRAINED MISSILE GUIDANCE PLATFORM

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The Autonetics Environmental Test Laboratory acquired the responsibility for the design and development of a test facility for simulating silo constrained missile response to ground motion. The test item was to be a 22" diameter inertial guidance platform. The requirements were: accuracy of 1 arc second, 80 db dynamic range, DC to 5 Hz frequency response, simultaneous multiaxis motion, and a motion derived from a complex waveform equation. The requirements dictated a digital approach and with precision equipment. Also, the guidance platform is North oriented, and a change of direction for a possible single linear input would require that the motion actuator change location and that the fixture base rotate. A four degree of freedom test system (3 rotational, 1 linear) with flexibility of linear direction was conceived as being possible to build in the time frame and cost allowed. Utilization of an existing 32K bit computer and analog recorders would be necessary for programming.

The seismic simulation test hardware that was built consists of four major components plus a group of ancillary items. These major components are (1) seismic reaction mass, (2) a linear motion system, (3) a three-axis gimbal system, and (4) an Integrated Control Console.

The Seismic Reaction Mass is a 14' by 14' by 8' deep steel reinforced concrete block. Included in this mass is a two piece granite mounting base. The bottom granite base plate is 5' square and 16" thick and is precision leveled and bonded to the concrete mass. The top granite plate, which is 9½" thick, is designed to rotate on an air bearing and has index pins located at 0°, 45°, and 90°. In the center of the mass is a cavity containing a bi-axial tiltmeter with readability to 1/100 arc second.

The Linear System consists of two major parts. One part is the slip table mounted on four 16" stroke slip bearings which in turn are mounted on the rotatable granite top plate. The second part is the 15" stroke hydraulic actuator which can be mounted on any one of three steel pads that are bonded in the reaction mass. The minimum motion for digital control is .001" and analog is .0002" peak displacement. The table has a dual Linear Variable Differential Transformer (20" and 1") for feedback, a displacement dial with .0005" resolution and a linear accelerometer monitoring from 100 micro g's to 1 g.

The Three Axis Gimbal System utilizes the slip table as its base. The requirement for a large, precision 3-gimbaled structure to sustain side forces and maintain accuracy, required special design. Also, three optical access holes were built into the gimbals to allow guidance platform monitoring. The range of digitally programmed motion for the 3 gimbals is ±3° to ±.0001° with a .0002° mechanical uncertainty (1 arc second programming). Each axis has a 1 rad/Sec^2 maximum angular accelerometer.

The 5-bay Integrated Control Console consists of program control and monitor processing circuits for the input and output signals, hydraulic actuator electronic control, monitoring pen recorder, gimbal control and monitor electronics, and other miscellaneous circuits. The program information (80 db range digital code with 5 motions multiplexed B C D—Serial—Parallel on analog magnetic tape) is processed through demultiplex and arithmetic circuits. Digital and analog signals are processed to recorders for records of test.

This unique equipment has been in service for over one year and has been a successful adjunct to inertial platform improvements. Because of built-in flexibility, numerous special tests beyond the original concept have also been performed.

VIKING DYNAMIC SIMULATOR VIBRATION TESTING AND ANALYSIS MODELING

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The Viking Dynamic Simulator (VDS) was flown from Cape Kennedy on the first Titan Centaur, February 11, 1974. In addition to the value of this flight to verify compatibility of the Titan and Centaur vehicles, the dynamic responses of the VDS are important for complete understanding of the loads and vibrations expected to act on the Viking when it is launched to Mars in 1975.

VIBRATION TEST PROGRAM

The VDS, a product of General Dynamics, was vibration tested in San Diego during the First Quarter, 1973, so that test data would be available for final pre-flight calculations during the summer and fall. This paper includes a description of the vibration testing conducted on the VDS to describe its dynamic characteristics and methods used to separate modes which were close to each other in frequency and also somewhat nonlinear.

Vibration test procedures used are explained together with the rationale used to select test equipment and, in particular, the configuration of the cantilevered boundary.

One objective of the test was to attempt to resolve all modes up to 50 Hz with less than five percent modal coupling. Twelve modes were predicted below 50 Hz so that the 66 independent coupling terms together with the five percent objective was the cause of much concern. It is believed that use of the Spectral Dynamics Corporation "Co/Quad Analyzer" was instrumental in very largely achieving this objective.

Damping constants for the various modes are discussed including a comparison of the damping by means of the free amplitude decay and by means of the steady state (non-decay) method of C. C. Kennedy and C. D. P. Pancu.

Loads for 18 truss members were also experimentally determined for all modes. The data obtained in the above vibration test program were the source of information used in numerous pre-flight calculations required in the normal course of dynamic response and stability studies. In some instances the actual test data have been used directly without alteration, but in general, it is recognized that this manner of use of the data is not entirely correct because of neglect of the attachment flexibility of the structurally redundant interface and other secondary problems related to test artificialities. This paper considers these questions and describes the methods used to generate analytical modal data eliminating nonlinear effects but retaining as much as possible of the nature of the test data. Comments are made on the use of the Nastran Computer Program which was used in the process of optimization of the analysis (or mathematical) model. The criteria used for selection of the optimum fit are considered of possible value to other investigators especially in the area of modal force coefficients for member loads, where large percentage errors are expected in those coefficients having small numerical values.

The final analysis model data are described including figures of modal deflections from test and analysis for comparison.

The final modal data were expanded to include deflections at all locations where flight measurements were to be made. References are included for all reports of test and analysis data published.

AN EVALUATION OF SHOCK RESPONSE TECHNIQUES FOR A SHIPBOARD GAS TURBINE

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When demonstration of combat durability is required, shipboard equipment of greater than 4500 lbs must be tested on the Navy Floating Shock Platform (FSP) as specified by MIL-S-901C. Of particular concern on a gas turbine engine is the integrity of engine structure, engine mounts, accessory mounts and bearings during shock. A detailed shock analysis during development can indicate potential problem areas and through redesign minimize the number of areas requiring post test corrective design. The study herein summarized was made to compare the results of the shock spectrum and transient response normal mode methods as applied to a shipboard gas turbine, to give some indication of the required degree of detail required in modeling and to obtain engine case accelerations, bearing loads and mount loads.

The gas turbine considered in this study consists of a straddle mounted, 470 lb gas generator and a straddle mounted, 404 lb power turbine transmitting shaft power coaxially through the gas generator to a forward gearbox. The total engine weight is about 4700 lbs. In the model the engine was mounted to a rigid substructure which was centrally mounted to the FSP on two beams to simulate a deck mounting. The substructure mounting beams and substructure weight of 15,000 lbs were chosen to give a fundamental frequency of 29.5 cps. The gas generator, power turbine and case were modeled by five, six and seven masses respectively and connected by beams. The total length of the system was 18 feet 4 inches. The rotors were connected to the case by appropriate spring rates. Two models were constructed, one with 98 degrees of freedom and flexible rotors and case and another much simpler model with effectively 22 degrees of freedom and rigid rotors and case. Both models were three dimensional and in general the masses possessed 6 degrees of freedom. Vertical FSP shock on a deck was considered.

Eigenvalues and vectors were obtained for both models. Shock spectrum maximum accelerations, deflections and bearing loads were obtained for the flexible rotor model by three methods of modal summation. The first summation method used was the summation of modal responses, the second method was the square root of the sum of the squares of the modal responses (root square summation) and the third method was the largest modal response plus the square root of the sum of the squares of all other modal responses (modified root square summation). Transient accelerations, deflections and bearing loads were obtained for both models.

Excitation was provided by a square pulse of 0.005 sec. duration and 67.54 gs amplitude. This excitation was chosen since its shock spectrum agrees very well with that of the design shock spectrum for vertical deck mounting on the FSP as given in NRL report 7396. Shock spectrum maximum accelerations, deflections and bearing loads were also obtained for the FSP design shock spectrum and summed by modified root square summation method for comparison with the square pulse shock spectrum results.

When compared to the flexible rotor transient response analysis, the best method of modal summation was the modified root square summation. In general, this method was somewhat conservative, whereas the root square summation was anticonservative and the summation of modal responses was very conservative. Because of the simplicity of analysis, the modified root square summation approach is recommended for shock analysis when maximum values rather than time responses are required.

The shock spectrum of the square pulse agrees very well with that of the design shock spectrum of the FSP. For applications requiring a transient response, this pulse is recommended.

Comparison of the flexible and rigid rotor transient responses indicated that the case and substructure accelerations and deflections agreed very well. Mount loads on the two models also agreed very well. Bearing loads, however, particularly on the more flexible power turbine did not agree as well. As indicated by this model, if the lower modes in the system contain bending, variables associated with the parts in bending require detailed modeling. Experience must be relied on to make a model as simple as possible to take advantage of reduced time for analysis and to minimize the opportunity for undetected mistakes.

EARTHQUAKE RESPONSE OF COMMUNICATIONS EQUIPMENT IN SMALL BUILDINGS

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Telephone communications facilities are so vitally important to public health and safety that special efforts must be made to prevent disruption of these services by a destructive earthquake. For this reason, and in order to protect extremely expensive plant investments, it is important to incorporate earthquake resistant design into telephone facilities. One category of equipment that seems extremely vulnerable to seismic motions consists of wall-braced 9 ft. equipment frames installed in small buildings. If the loadings in the bracing system, at the top of the equipment frames, becomes excessive it is possible for the equipment to lose its entire support structure. The result could be a catastrophic failure with all the equipment frames toppling over.

A finite element dynamic analysis was performed on a wide variety of these small one-story buildings and the equipment within them. The unique part of the analytical study was the inclusion of coupled building and equipment motions. In most of the previous documented earthquake studies of equipment response, it was justifiable to uncouple the building and the equipment because the equipment mass was much less than the building mass and there was no significant feedback. However, it was found that for the building and equipment configurations considered in this study the loads in the bracing system were significantly underestimated when building-equipment coupling was not included.

From the analyses and tests, general recommendations were developed which are applicable to any equipment layout of the category of equipment and buildings being considered. These recommendations consist of guidelines for determining if a particular equipment configuration is properly earthquake braced, and if it is not, the procedures for upgrading.

SEISMIC ANALYSIS OF MOTORS FOR NUCLEAR POWER PLANTS

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Before atomic power became a practical large scale energy source, consideration of seismic design was generally restricted to buildings and other tall, flexible structures. With the recent construction and licensing of nuclear power plants, the potential hazard of an uncontrollable release of excess radio-activity as the result of an earthquake is of great concern. It is of the utmost importance that those components of the plant which are vital to safe operation or shutdown maintain both structural and functional integrity during and after a seismic disturbance. Included in this category are alternating-current induction motors for driving pumps and cooling and recirculating fans.

To determine whether or not a motor meets the requirements of a particular plant or location, it has previously been necessary to subject the motor and driven equipment to a simulated earthquake and measure the appropriate parameters. Special facilities are required and the testing is costly.

Equipment can, however, be qualified for Seismic Class I service through suitable analytical techniques. In general, a dynamic rather than a static analysis will be required. A Mathematical Model of the system to be studied is first developed—it is possible to consider an AC motor as a multi-degree-of-freedom, lumped-mass system with mass-free interconnections. This step in the analysis requires extreme care and good engineering judgement since the accuracy of the results depend strongly on how well the model represents the actual physical system.

Once a satisfactory model has been established, a modal analysis can be performed in conjunction with a response spectrum analysis. The analytical procedure involves writing the equations of motion of the system, solving these equations for the natural frequencies and mode shapes, and evaluating the modal participation factors. Seismic accelerations are then determined from the appropriate floor response spectrum curves. A response spectrum is a plot of the maximum response of a single-degree-of-freedom system at a damping value expressed as a per cent of critical damping, of different natural frequencies, mounted on the surface of interest when that surface is subjected to a given earthquake motion.

For each mode, equivalent static forces are calculated as the products of the mass, coordinate, participation factor and seismic acceleration at each mass point. These forces will result in stresses and deflections which are combined with the normal operating stresses and deflections. If these combined values do not exceed the allowable limits, the equipment has been qualified for Seismic Class I service. Limits are established to insure that the motor will remain functional during the worst anticipated earthquake for the plant site. Some permanent deformation may even be acceptable as long as the motor maintains operability and structural integrity. Redesign or modification are, of course, necessary should any of the stresses or deflections be excessive.

This procedure satisfies the requirements of IEEE No. 344, The Guide for Seismic Qualification of Class I Electrical Equipment for Nuclear Power Generating Stations. IEEE No. 344 is the generally accepted industry standard and is also approved by the governing regulatory bodies for nuclear power plant licensing.

This analytical approach can easily be adapted to suit other rotating equipment and similar devices.

A NEW STUDY OF THE HARMONIC OSCILLATOR WITH NON LINEAR DAMPING

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The damped harmonic oscillator represented by a mass, spring, dashpot system is treated extensively in most applied mathematics and mechanics texts. The retarding force is known as viscous friction and is proportional to the velocity of the mass. This relation, when expressed by an equation of motion, results in a second order linear differential equation which may be solved in closed form.

Comprehensive studies of the solutions result in such familiar terms as viscous damping, under damping, over damping, critical damping and damped frequency. However, no dashpot behaves precisely as linear resistance and, with the exception of electrical analogues, these terms, as defined, exist only in the mathematics. Theoretically, when three dimensional flow is considered, resistance provided by the dashpot is proportional to the fluid density and the square of the piston velocity, and at large velocities, has been considered to be proportional to nearly the cube of the velocity. Non linear damping is elegantly treated by various mathematical techniques in many advanced mechanics and control theory texts. The author presents the results of a new study done with the aid of computer techniques.

We consider the equation,

$$\ddot{x} + \frac{\dot{x}}{|\dot{x}|} 2\gamma |\dot{x}|^{\epsilon} + x = 0$$

The equation was solved by numerical integration for various values of ϵ and the particular value of $\gamma = 1$, (which is the critical damped case in the viscous or linear damped equation).

For the value $\epsilon = 1$ the solution is,

$$x = \ell^{-t}(1 + t).$$

For the value of $\epsilon = 1.2$ the system oscillates with an initial period slightly larger than the period of the natural frequency and is observed to approach the natural frequency as time becomes larger. For the value of $\epsilon = 2$ the initial period after the first overshoot is very nearly but still larger than 2π and for the value of $\epsilon = 3$ it is very nearly but smaller than 2π. In each case it approaches 2π quite rapidly. Other values of γ produce similar results. The system always oscillates except where $\gamma = 1$ and $\epsilon = 1$, the linear critical damped condition. The system even oscillates for some values of $\gamma > 1$ provided $\epsilon > 1$ and in those cases the frequency also converges on the natural frequency as time becomes large.

By trial and error a constant $\epsilon = \epsilon_n$ was found such that after the first overshoot the system always oscillates at precisely its natural frequency for *all* values of γ. Its value, to four significant figures is

$$\epsilon_n = 2.586$$

For values of $\epsilon < 2.586$ the initial period after the first overshoot is always greater than 2π and for values of $\epsilon > 2.586$ it is always less than 2π.

Since it seems reasonable that no damped oscillator would oscillate at a frequency greater than its natural frequency the results of this study strongly suggests that ϵ_n is a natural constant limiting the proportionality of the dependence of resistance on velocity, ν, to the value,

$$\nu^{2.586}$$

It may even suggest that the dependence of *all* fluid friction on velocity is limited to this value.

The conclusions drawn from this study are:

The frequency of all oscillating spring mass systems which rely on fluid damping will approach the natural frequency of the system as time becomes large. The dependence of resistance on velocity is limited by the value of ν^{ϵ} where $1 < \epsilon < 2.856$.